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UNITED STATES NUCLEAR REGULATORY COMMISSION

IN THE MATTER OF:

LONG ISLAND LIGHTING COMPANY

SHOREHAM NUCLEAR POWER STATION

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UNITED STATES OF AMERICA

NUCLEAR REGULATORY COMMISSION

BEFORE THE ATOMIC SAFETY & LICENSING BOARD

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In the matter of:                  :
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SHOREHAM NUCLEAR POWER STATION     : Docket No.50-322-OL
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(Long Island Lighting Company)     :
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State Office Building
Veterans Memorial Highway
Hauppauge, New York

Wednesday, September 12, 1984

Hearing in the above-entitled matter was convened at 9:05 a.m., pursuant to notice.

BEFORE:

JUDGE LAWRENCE BRENNER,
Chairman, Atomic Safety & Licensing Board

JUDGE PETER A. MORRIS,
Member, Atomic Safety & Licensing Board

JUDGE GEORGE A. FERGUSON,
Member, Atomic Safety & Licensing Board

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PROCEEDINGS

JUDGE BRENNER: Good morning. Mr. Dynner.

E X H I B I T S

EXHIBIT	FOR IDENTIFICATION
Suffolk County Diesel No. 69	
(Article from MOTOR SHIP TECHNICAL MAGAZINE entitled Sulzer's Four-Stroke High and Medium Speed Engine Range)	22365
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Suffolk County Diesel No. 71	
(Photo of Piston removed from EDG 103 taken by Anesh Bakshi at June 1984 at SNPS of scuffing)	22421

P R O C E E D I N G S

JUDGE BRENNER: Good morning. Mr. Dynner. If there are no preliminary matters, you may continue your cross-examination. Are there any preliminary matters?

MR. FARLEY: May we approach the bench?

JUDGE BRENNER: I would rather have things on the record unless there is a good reason not to. If you feel strongly about it, we will go off the record.

MR. FARLEY: I would prefer off the record.

JUDGE BRENNER: All right.

(Side bar conference held out of the presence of the public).

JUDGE BRENNER: All right, Mr. Dynner. We discussed yesterday our desire that you finish cross-examination by the lunch break.

MR. FARLEY: Judge Brenner, I think Mr. Johnson is prepared now to talk about the Kodiak thing that we ended up with yesterday, if Mr. Dynner would like to pursue that.

Whereupon,

DAVID O. HARRIS,

DUANE P. JOHNSON,

ROGER L. MC CARTHY

FRANZ F. PISCHINGER,

CRAIG K. SEAMAN,

LEE A. SWANGER,

and

EDWARD J. YOUNGLING

were called as witnesses on behalf of the Applicant and, having been previously duly sworn, were examined and testified as follows:

CONTINUED CROSS-EXAMINATION

BY MR. DYNNER:

JUDGE BRENNER: All right.

DR. JOHNSON: I'd like to clarify the Kodiak results. Two pistons were removed from the Kodiak engine number 4. The two pistons that were removed, and if we refer to the third page of Exhibit 29 --

JUDGE BRENNER: This is LILCO Exhibit P-29?

DR. JOHNSON: Yes, sir. The two pistons that were removed are the ones labeled on that page as 1R and 1L. The stud boss area of both of those pistons was examined by both PT and ET. No indications were detected by either method.

In addition, the rib near the wrist pin boss area was inspected with PT, penetrant S. No linear penetrant indications were observed in the piston which we are calling 1R, a three-quarter inch linear penetrant indication was observed in the piston we are referring to as 1L. The region in the vicinity of the penetrant indication observed in 1L was inspected with eddy current and no linear indication was found.

The 1L piston, that is the piston that had the three-quarter inch linear penetrant indication, was shipped to FaAA laboratory. The other piston, 1R, I understand, was shipped to TD. In the Failure Analysis laboratory we reinspected piston 1L using both penetrant and eddy current technique, and no linear indications were observed with either method. We conclude that there were no relevant indications observed in either piston.

DR. SWANGER: I'd like to add to that answer if I might. The ET that Dr. Johnson referred to is abbreviation for eddy current testing which has been discussed earlier. Also I think part of the pending question was, what was the load on the piston that was removed from these engines. Our basis for evaluating this piston or assigning it a load of 1200 pounds per square inch peak firing pressure is based on the information that we received from the Kodiak Electric Association, that over the 6000 hours of exposure that all of these pistons had had, the engine had operated at an average load of 80 percent of its nameplate rating.

The data shown on this page, taken on March 2, 1982, at 5600 kilowatts, is pressure data taken with the Kiene type gage at 80 percent load. The 5600 kilowatts is 80 percent. We thought given the questions of the accuracy of the Kiene gage and to be conservative in the credit that we were going to take for the exposure of the AE piston in the Kodiak engine, that we did not want to claim that it saw a load of more than 1200 psi, peak firing pressure.

As you can see from the table, based on the Kiene gage, every piston in that engine experienced loads above 1200, from 1240 up to 1340.

Q. Dr. Swanger, to follow up on that, could you explain on the chart we are speaking about, which is the third page of Exhibit P-29, the first part of the chart says "March 31, 1983, 4000 KW, 7200 hours."

Does that reflect the peak pressures of all of the cylinders during the period ended March 31, 1983?

DR. SWANGER: Each of the sets of data on Page 3 of P-29 represents the results of a test at one point in time at one particular load taken by the Kodiak Electric Association. For diagnostic purposes, taken with the Kiene gage. One of the purposes of having both LILCO and FaAA engineers go to Kodiak Alaska, was to gather information from the Kodiak Electric Association on the operation of their engine with the AE pistons in it.

As I had said earlier, Kodiak Electric told us that the engine had been operated at an average load of 80 percent of its nameplate for the 6000 hours preceding the inspection, which is the time that it had AE pistons in it. The hours referred to in these two charts are the total engine hours from the time that it was new.

JUDGE BRENNER: I think that completed the answer.

Q. Dr. Swanger, do you know what the peak firing pressure of the cylinders on the average is for the 80 percent load?

DR. SWANGER: I'm confused because if you look at the page we have been looking at, one set of data for, apparently as you testified, the tests done on March 2, 1982, shows a significantly higher average peak firing pressure than shown at March 31, 1983. Is that because the March 31, 1983 test was taken at 4000 KW?

DR. SWANGER: Yes, it is. There is a direct correlation between the peak firing pressure and the total output of the engine. The data taken on March 2, 1982, at 5600 kilowatts is, at 80 percent of the nameplate output of this, which is 7000 kilowatts, and the peak firing pressures reported there from 1240 to 1340 psi represent the peak firing pressures associated with operation at 80 percent of the nameplate load.

Q. Since the information that you received as a result of the February 17 trip, which is reported in the document in Exhibit P-29, and the data associated with the peak firing pressure, have you done anything to update that information?

DR. SWANGER: I'm not certain that I know what you mean by "update the information", Mr. Dynner. Specifically what are you referring to?

Q. All right. Have you had any

communications with the Kodiak Electric Association since the timeframe of February to March of this year concerning their AE pistons?

MR. SEAMAN: Mr. Dynner, we have spoken to Kodiak since February, and asked them if they had any new information to report on the AE pistons. I don't recall the precise date but they did report that they had had no problems with the AE pistons at that time.

Q. Do you recall the approximate date?

MR. FARLEY: Approximate date of the conversation? The question is not clear.

JUDGE BRENNER: Is that what you meant, Mr. Dynner?

MR. DYNNER: Yes, of course.

MR. SEAMAN: To the best of my recollection, it was the May to June timeframe.

Q. You said that during this conversation you asked them whether they had any problems, or they indicated they had no problems with the AE pistons. Could you further describe the conversation.

MR. SEAMAN: The actual conversation that occurred occurred with a number of my staff, not myself, who reported to me that the pistons were still operating satisfactorily. That's why I don't recall the details of the conversation except that they were still operating satisfactorily.

JUDGE BRENNER: We are going to get a little remote in terms of what this witness will tell us, Mr. Dynner, a conversation with somebody else about a conversation.

MR. DYNNER: Yes, I am moving on.

Q. Gentlemen, if you turn to page 56 of your testimony.

Dr. Johnson, you testified in answer 88 that FaAA inspected two AE pistons from the TD R5 prototype engine after approximately 622 hours of operation at 2000 psi.

Would you identify the type of engine that is designated as the TD R5 prototype engine. R5, two separate things -- excuse me.

Dr. Johnson, this is your testimony in answer to 88. Do you know what the TD R5 engine is?

DR. JOHNSON: The inspection we performed was on two pistons. That information was taken off the box that the two pistons were in. That's what we were told by TD.

JUDGE BRENNER: What was the designation of the engine. You personally don't know what the TD

R5 engine is; is that correct?

DR. JOHNSON: Yes.

Q. Then anyone can please describe it, the TD.

JUDGE BRENNER: Mr. Dynner, I think it would be more efficient if you could have them focus in on any point of similarity or differences you want them to focus in on instead of hearing a recitation by the witnesses on what --

MR. DYNNER: All right. I will rephrase the question then.

Q. Is the TD R5 prototype engine the same type engine as the EDG's at Shoreham?

DR. SWANGER: The TD R5 engine is an evolutionary development of the R4 type engine which is in use at Shoreham. The key dimensions describing the engine, the bore of 17 inches and the stroke of 21 incnes are identical to the engines at Shoreham.

This is borne out by the fact that the same piston, the AE piston can be used interchangeably in the R4 or the R5 engine. The R5 engine has been developed by TD.

JUDGE BRENNER: To provide a higher specific output. I think the nominal rating is 275 psi, brake mean negative pressure compared to the 225 psi brake mean effective pressure in Shoreham. So it is more highly stressed both from a mechanical load standpoint and from a thermal load standpoint.

In addition, the testing that was done by TD with these AE pistons, was accelerated testing, even for the R5, and that for the 622 hours of operation, the brake mean negative pressure was a figure of 304 psi, which is substantially above the 275 psi rating of the R5. The R5 is a large medium speed turbo charged diesel engine of very similar type to those employed at Shoreham.

Q. It is a V16 engine as opposed to a straight 8 engine that we have at Shoreham, is it?

DR. SWANGER: The R5 designation, just like the R4 designation, applies to a family of diesel engines.

Q. I am talking about this particular engine identified in the testimony as the R5 prototype engine. That's a V16 engine rather than a straight 8 engine as we have at Shoreham, isn't it?

DR. SWANGER: No, you are wrong, Mr. Dynner. It happens to be a V12 engine.

Q. All right.

DR. SWANGER: However, as I was trying to explain in the answer, that doesn't make any difference to the testing of the individual components and what's referred to --

Q. I didn't ask you whether --

MR. FARLEY: I object. He is interrupting the witness.

MR. DYNNER: I would like an answer to the question.

JUDGE BRENNER: In this case he answered it. I think what we are getting is fair explanation in addition to the answer.

DR. SWANGER: The reason the testing was significant to FaAA is that the components of the power cylinder, namely the cylinder head, the piston and the liner, the valves, the fuel injection equipment, are identical for any R4 engine. Also they are identical for any R5 engine independent of the number of cylinders in that engine. And, therefore, testing of a component, specifically testing of an AE piston would be independent of whether it was in a 6, an 8, a 16 or even a 20 cylinder engine.

Q. You said the cylinder head is the same in the R5 V12 as in the EDG's?

DR. SWANGER: I did not say that. I said

within the R4 family, the powered cylinder components are the same, and separately within the R5 family the power cylinder components are the same.

Q. Could you describe the differences, if any, between the AE piston skirts you examined in the R5 engine and the AE piston skirts in the EDG's at Shoreham?

DR. SWANGER: The differences between the AE pistons at Kodiak Electric Association which are the same as the AE pistons in the Shoreham EDG's, and the AE pistons that were tested in the R5 are referred to in our direct testimony at question number 89 at page 56. I can expand a little bit on that answer by referring to LILCO's exhibit P-29. On Page 10 of that exhibit there are photographs of the interior of the AE pistons at the Kodiak Electric Association which are the same as the pistons at Shoreham, and on Page 28 of exhibit P-29 there are photographs of the interior of the AE piston that was tested in the R5 development area.

By comparing those two pictures, the differences in the evolution of the AE design can be seen, and they are in two areas: One is in the wrist pin boss area, the area of the casting in which the 6 and three-quarter inch diameter wrist pin goes through the casting, and the difference is that the earlier version of the AE, the one that was tested in the R5, is similar to the AΓ piston in that the wrist pin boss has a reenforcing rib around it, and also has some longitudinal reenforcing ribs.

In the more advanced AE design as shown on Page 10, that wrist pin boss has additional material added to it to stiffen it and strengthen it.

The other difference is that the circumferential rib part way up the skirt of the piston which connects the wrist pin bosses together has been enlarged, tapered, and the radii made more gentle and more blended in the advanced AE design, one at Kodiak and the one at Shoreham.

The R5 engine had the AE piston with a narrower rib which is similar to the ribs in the original AF pistons at Shoreham.

The NRC Staff has addressed these differences in its testimony and has said that in their opinion the R5 piston, the earlier version was improved when the changes were made to generate the AE pistons that were provided for Kodiak and for Shoreham, and that they had no doubt that these changes made an improvement to the AE piston.

We think this makes the experience with

the R5 test engine even more valid because the pistons which are at Shoreham are actually stronger than the AE pistons tested in the R5 engine at 2000 pounds per square inch for essentially ten to the 7 cycles. Therefore, we think that experience with the R5 is very strong and our conclusion that the AE pistons at Shoreham are adequate for their purpose, will not initiate and will not propagate any cracks.

Q. Are the differences that you described the only differences between the R5 AE piston and the AE pistons in Shoreham?

JUDGE BRENNER: While they are conferring, I assume Exhibit P-29 is one of the exhibits where the pages will be numbered for the version that goes as the official exhibit?

MR. FARLEY: Yes, Your Honor.

DR. SWANGER: The differences in design that I just discussed are the only ones I am aware of.

Q. Dr. Swanger, I didn't ask you about differences in design. I said, are those the only differences. So I would like to know, aside from design differences, are there any other differences?

JUDGE BRENNER: Mr. Dynner, is there a reason why under cross-examination you cannot just point to whatever difference you have in mind and say, what about this, isn't that a difference? So we can get quick answers.

MR. DYNNER: If I knew all the answers, that would be the approach I might take, but I don't know all the answers. I have reasons to believe that there may be other differences, but I don't know the answers. That's why I am asking the questions, sir.

JUDGE BRENNER: Not the typical approach to cross-examination.

MR. DYNNER: Sometimes we don't always know the answers to questions that we ask, as you have noted during this examination.

DR. SWANGER: There are some other differences which are contained in the memo by Donald O. Johnson dated February 3, 1984, which is included in the middle of LILCO's exhibit P-29. I'd like to explain those differences, but before I do I think it is important to know while you are listening to the explanation of the differences, that they have no effect on the conclusions drawn from the FaAA report. My explanation of the answer will make it clear why they have no effect on the conclusions, namely that the AE pistons at Shoreham will not initiate or propagate cracks.

The differences that I am referring to are those that were noted by Donald Johnson during his inspection of the R5 pistons. The first one is in the fourth paragraph of the memo, and is as follows: "During the inspection I observed there was a layer of plating on the inside of the skirt and that the casting was very smooth, different from general production runs of cast material. The inside of the skirt was clean and all the flash was removed. The boss area was very smooth as if polished by Cratex and all the ground areas were very carefully polished with smooth radius in the boss."

Then the plating is explained in the following paragraph where it says, "There was evidence of plating on the inside below normal areas. The plating was very thin, approximately .0005 inch to .0001 inch."

I can comment on the measurement of the plating. It was done with the eddy current technique. One of the features of the eddy current technique is that the signal that is developed on the oscilloscope during testing is proportional to the lift-off of the eddy current probe from the cast iron surface being examined, and by calibrating the eddy current probe on known thicknesses of tin plating in the laboratory, we were able to use it to demonstrate that this plating which is between 1/15th and 1/3rd of the nominal plating on the piston probably resulted from some leakage current to the inside of the piston due to this not being a production piston but being a test piston at TD.

The other more important area or the one that might seem more important is the polishing of the boss areas on the inside of the piston. Since this was a test piston at TD, they used techniques which are standard in evaluating the design of a component. In development of components such as pistons, the manufacturer wants to be certain that he separates the effect of the various variables on the performance of a component, and in this case TD was being careful to separate the effect of manufacturing from the effect of design of the component, and took normal precautions which would be taken during a test program to make sure that they were testing features of the design by being sure that the area in the boss area was smooth.

What FaAA has shown with its more detailed and more exhaustive design analysis is that the precautions that TD took in their test actually were unnecessary. We have demonstrated that it is highly unlikely that any fatigue cracks would initiate in that area. We have also demonstrated that fatigue cracks will not propagate. We have been mentioning in these hearings that fatigue cracks up to -- excuse me -- that cracks or preexisting defects up to half an inch deep will not propagate. But I'd like to clarify that by saying -- I'd like to continue by referring to LILCO's Exhibit P-25, which is a plot of the stress intensity factor range versus hypothetical crack depth compared to the threshold stress intensity factor range.

By looking at this chart, it goes out to .5 inches, half an inch on the right, but even at that point the actual delta K working on the defect is well below the delta K threshold. It is just that half an inch was such a ridiculously large feature to presume being in these pistons that we stopped the analysis at that point.

If we had gone on, it might turn out the defects three quarters of an inch, an inch, maybe an inch and a half may be demonstrated to fracture mechanics to be nonpropagating in AE pistons.

Q. Are you through?

DR. SWANGER: I would like to summarize, if I may, that it is because of the experience in the R5 engine at 2000 psi for enough cycles to demonstrate that it was operating below its endurance ratio for a condition which the AE pistons at Shoreham, I don't think, even could reach. That is the 2000 psi peak firing pressure, that we are extremely confident that the AE design as demonstrated by TD's test and by FaAA analysis is very conservative.

Q. Now, Dr. Swanger, let's go back to your initial answers to my question which is about the differences between the AE piston and the R5 engine and that in the Shoreham EDG's as stated in Donald Johnson's memorandum. The first thing it notes there is that there was a layer of plating on the inside of the skirt.

Is there a layer of plating on the inside of the skirt of the AE pistons at Shoreham?

DR. SWANGER: In my own visual inspection of the AE pistons at Shoreham, I have seen some evidence of very minimal amounts of tin due to stray electroplating currents which will get inside the piston and deposit a little bit of tin in some areas. It is extremely innocuous and has no effect on the piston one way or the other DR. MC CARTHY Could we have a moment on this?

JUDGE BRENNER: Is the answer complete? Mr. Dynner.

DR. SWANGER: I don't think I have anything to add to that.

Q. Do you mean to suggest, Dr. Swanger, that any tin plating that was found by FaAA, not just yourself, or LILCO, the inside of the AE skirts at Shoreham was there inadvertently and unintentionally, or is it there by design?

DR. SWANGER: I have discussed the tin plating of pistons with design engineers from TD, and I know that the tin plating is on the outside of the pistons by design, on the inside of the pistons inadvertently as a result of the electroplating process to put one and one half mil of tin on the outside of the piston to protect it during break-in and to protect the outside of the piston during storage.

Q. And the plating that was observed on the inside of the skirts in the two AE pistons from the R5 engine are the ones that were plated very thin as described in the penultimate paragraph in the first page of Dr. Donald Johnson's report, is that correct?

DR. SWANGER: I apologize.

Q. The next to the last paragraph where it says the plating is very thin.

DR. SWANGER: I was going to ask you to define penultimate. Does that mean next to the last?

Q. I was getting even with you. You used a lot of words that I don't know.

JUDGE BRENNER: Why it doesn't mean next to the last? I don't know. But it does mean next to the last.

DR. SWANGER: Thank you, Judge Brenner. Could you rephrase the question for me. I lost the trend of thought.

Q. Is the plating that was observed on the inside of the AE skirts in the R5 engine the tin plating referred to in the last sentence of the next to the last paragraph in Donald Johnson's memorandum, which says, "The plating was very thin, approximately .0005 inch to.0001 inch"?

DR. SWANGER: I still didn't understand that question. I just heard, does the tin, and then you went on to describe it. I didn't detect it. Please help me.

Q. I will say it for the third time. Is the tin plating where it says there was a layer of plating on the inside of the skirt of the AE pistons in the R5 engines, is that layer of tin plating the plating which is described as being .0005 inch to .0001?

MR. FARLEY: I still object to the form of the question. He hasn't said whether by design or inadvertence, which is what we are talking about.

JUDGE BRENNER: He doesn't have to say that in the question.

DR. SWANGER: In the paragraph preceding the penultimate paragraph, there is the statement that, "I observed that there was a layer of plating on the inside of the skirt."

Then in the following paragraph he goes on to quantify that plating by saying, "The plating was very thin, approximately .0005 inch to .0001."

Both sentences refer to the same electroplating on the inside of the piston skirt.

Q. All right. Now, in the same next to the last paragraph, Mr. Johnson also says the plating inside and out. Do you know whether his description of the thickness of the plating also refers to the plating on the outside of the skirt?

DR. SWANGER: Donald Johnson reports directly to Dr. Duane Johnson and in addition to this memo has had conversations with his supervisor.

I believe Dr. Johnson is better able to answer this specific question.

Q. Go ahead.

DR. JOHNSON: The measurement .0005 to .0001 was the measure obtained on the inside of the piston skirt.

Q. Dr. Johnson or anyone, did he take the measurements of the thickness of the tin plating the outside of the skirt as well?

DR. JOHNSON: He checked the plating measurements on the outside and they were equal to the standard which he had set up on, which was a piston containing -- excuse me, the nodular n piston skirt with a normal plating on the outside, which, as I recall, was on the order of 1.5 mils, which is, in materials we have here, 0001.5.

Q. And is the tin plating on the outside of the AE skirts in the R5 engine the same thickness as the tin plating on the outside of the AE skirts in the Shoreham EDG's?

DR. SWANGER: The only data we have for skirts in the R5 engine are the two that we inspected. We don't know what they are in the rest of the R5 engine. But the two that we inspected were similar to the plating thicknesses on the AE skirts at Shoreham.

Q. When you say they are similar, what were the precise differences in the thickness of the tin plating of the AE skirts at Shoreham and the two AE skirts in the R5 engine?

DR. SWANGER: FaAA has made no measurement of thickness of the tin on the outside of the AE piston skirts at Shoreham. However, we have reviewed the engineering drawing for the AE skirt which specifies a thickness of tin on those skirts of.0015 inches. Our measurement of the tin thickness on the R5 AE pistons as discussed by Dr. Johnson showed that its thickness was nominal. That is, it was within the expected range based on calibration standard taken from a nodular iron piston which was made by TD and passed all of the manufacturing and acceptance inspections that would have been given to it by the manufacturer and by LILCO, the customer.

This is significant to us because we think that the successful operation of the normally tin plated AE skirt in the R5 engine demonstrates that certainly there was no adverse effect of the tin plating on the outside of the skirt, and in fact it was probably a benefit of the tin plating the outside of the skirt even in an accelerated high load test in the R5 engine.

Further, we think that the test in the R5 demonstrated that the flash-over or inadvertent amount of tin which may be present on the inside certainly has no effect on the fatigue performance of those pistons because no indications were found in the critical high stress areas of those pistons. The memo we have been discussing, February 3, 1984, by Don Johnson, does in its last paragraph, discuss three eddy current indications --

MR. DYNNER: I move to strike everything he said after what he told me that the tin plating is the same after what is normally called for in the drawings. This witness is not being responsive to my question. He is giving speeches based upon what he thinks I will ask in the future.

JUDGE BRENNER: I will not go back and strike it. The comment is correct. He has the facts on the record. Let's leave it there for a couple of reasons, including efficiency. Let's get the answers more directly to the question and hold the rest of what you might want to say. I am going to become more aggressive and insisting that that's done now. I think we have given you fair leeway and it takes witnesses a while to adjust being in this, what I am sure, is an unusual situation for most people.

I have been a witness myself and it is stressful and unusual even when you are used to proceedings, than to certainly sit in a witness stand. Nevertheless, you have had a few days to acclimate yourself. The object is to answer the question and not just to fill in on all other things that you believe might be of interest within the same or related subject matter.

All right, Mr. Dynner.

Q. Dr. Pischinger, do you know what the purpose was of tin plating the inside of a skirt as in the R5 AE skirts?

DR. PISCHINGER: As was mentioned before, the tin plating on the inside of the skirt was unintentional. That means it was usually a stray of tin plating which is occurring to tin plates of the piston on the outside. That's what I heard.

Q. I am talking, Dr. Pischinger, about the tin plating, the layer of tin plating on the inside of the skirts in the R5 engine that are .0005 to .0001 in thickness. I believe what you are referring to is the inadvertent tin plating on the inside of the

AE skirts at the Shoreham facility, the Shoreham engines.

DR. PISCHINGER: I can only suppose that tin plating on the R5, the AE pistons used on the R5 engine is also to a little higher degree unintentional.

Q. Does anyone on the panel know whether or not the tin plating layer on the inside of the piston skirts, of the two piston skirts in the R5 engine was intentional or unintentional?

DR. SWANGER: Yes. I have had a discussion with engineers from TD and learned from them that the tin plating the inside of the AE skirts in the R5 engine was unintentional.

Q. Did you ask them how they could make that kind of unintentional tin plating in an engine, experimental prototype engine used for testing that particular component?

DR. SWANGER: No, I did not ask them that question.

Q. Thank you.

Dr. Pischinger, can you tell me what the effect would be, if any, of having this thin layer of tin plating on the inside of the AE piston skirt?

DR. PISCHINGER: None.

Q. You also testified that on the R5 AE skirts that we are referring to, Mr. Johnson reported that the casting was very smooth, different from general production runs of cast material.

Was the smooth casting different from the casting on the AE pistons in the Shoreham EDG's?

DR. JOHNSON: Yes. The condition was smoother than the surfaces which we generally observed on the Shoreham pistons on the inside. Of course, that also made it easier to inspect.

Q. Would the smoothness result from a polishing of the boss area of the skirt? I will direct your attention, for your convenience, to the third from the last paragraph of Mr. Johnson's memorandum.

MR. JOHNSON: It could have resulted from polishing, but we don't know that it was the result of polishing.

Q. Has anyone -- Dr. Swanger, when you testified p.eviously, you said that TD was being very careful to be sure that the area and boss is smooth. How did you find that out?

Dr. Swanger, can I have your answer before you confer with your colleagues because I am asking you a question about your prior testimony.

DR. SWANGER: My answer was based on my years as director of product development for Imperial Clevite, Incorporated, Engine Parts Division, and our standard procedures used in development of components. It is typical to use a smooth surface to test for the design of a component as opposed to its manufacture, and to make it easier to detect any indications which might develop on it.

Q. And the polishing and smoothing out of these boss areas and the ground areas would also eliminate or reduce any stress areas in those riser areas, wouldn't it, Dr. Pischinger?

DR. PISCHINGER: Could you please repeat the question.

Q. Yes. The polishing and grinding -- the polishing of the boss areas and the ground areas of the AE skirt would also reduce or eliminate stress risers in the skirt, isn't that so?

DR. PISCHINGER: I wouldn't put it that way. I want to point out that it is very often used and practiced in engine development and development of such parts that if you run such a piston prototype for stress evaluation, you try to get a clean, smooth surface so you are sure you have no crack initiation afterwards, and that it is easier to detect crack initiation.

DR. SWANGER: You asked about --

Q. The effect also is, Dr. Pischinger, an effect would also be that it would reduce or eliminate any potential stress risers in that area; isn't that true?

DR. PISCHINGER: It gives -- such a surface gives evidence of the property of the effect of crack initiation. You want to see if such a piston is prone to crack initiation. Usually in such a run it is not the intention to watch crack propagation of a prefabricated crack. It is to see crack initiation.

DR. SWANGER: I'd like to augment Professor Pischinger's answer.

MR. DYNNER: I'd like Dr. Pischinger to answer the question. I don't think he understood the question.

JUDGE BRENNER: Yes. I think he misunderstood the question.

Q. Let me try to explain once more the question. Dr. Pischinger, do you know what a stress riser is? Do you know what that term means in English?

DR. PISCHINGER: Maybe you think of any

flaw which can develop into a higher crack, a larger crack.

Q. Or an area where the stress concentrations could cause a defect?

JUDGE BRENNER: I haven't heard the term stress riser. Maybe I don't know what it means in English either. In any event, that's not the question you are trying to get at. We don't have to start with overall definitions. I think what he is trying to ask you is whether the process of taking special care to polish the surface would leave the situation such that there would be no areas conducive to flaw or crack initiation. We understand in your answer that it also makes it easier to observe any later indications.

Mr. Dynner is asking you whether or not it would also remove any flaws, if you will, incipient nature, and it would be more conducive to the development of indications during test runs.

DR. PISCHINGER: I think it has to be broken down in two parts. If surface treatment or polishing is going to such an extent that you really change the shape of the region, then, of course, you influence the whole stress fatigue and the result of the experiments with such a piston would not be representative.

If you clean or smooth the surface, you, of course, will not have the effect of a casting flaw -- I hope this is the right expression. In this respect, of course, you remove, if you call it that way, a potential crack initiation, or if you call it that way, a stress riser. But I think there are two different questions to address. One is, will a crack initiate, which has not been there, and the smooth surface gives you better evidence of that.

The other question is, will a crack, a flaw, be the first step of a crack, propagate, and to this purpose, of course, you should not clean the surface.

DR. SWANGER: I think it would be helpful if I explain the concept of stress riser just a little bit.

JUDGE BRENNER: Just a little bit.

DR. SWANGER: Certainly there is a concept of a stress riser, that is a geometric deviation in the surface which causes the tensile stress or the applied stress at that area to increase. There is also the effect of the stress riser, whether or not it would potentially cause a crack or cause a crack to grow. That is directly what the fracture mechanics calculations of Dr. Harris have addressed.

What they have shown is that stress risers, even as deep or deeper than one half inch in the surface, will not propagate. Certainly by polishing the surface, we have not removed stress risers half an inch deep.

Q. It is true, isn't it, that the radius into the boss area of the AE skirt is the area of higher stress in the AE skirt?

DR. HARRIS: To a large extent, Mr. Dynner, your statement is correct. However, the geometry in the stud boss region of the AE piston skirt was quite complex. So it is difficult to put in the words where the maximum stress occurred. In the Failure Analysis Associates piston report which has been entered as County Exhibit 8 --

JUDGE BRENNER: Not yet, but it has been identified. I said that for counsel's benefit. There is some discussion that has to go on with counsel regarding some of these reports.

Go ahead. I'm sorry.

DR. HARRIS: As indicated in at least two places in the report that I just mentioned, there are pictures and photographs of models that indicate where the highest stresses occurred.

On page -- on figure 3-3 of that report there is a photograph of the stud boss region of the AE piston skirt. This is the skirt to which stress coat was applied in order to identify the region of highest stress.

In this figure, if you look carefully, it is possible to see the small cracks running in the high stress region. There is a white circle on the figure that surrounds the region where the cracks were. So this, in a photo, shows the location of the maximum stress.

DR. MC CARTHY: I might add, these were cracks in the stress coat which is a brittle lacker, not the metal itself.

DR. HARRIS: Then on figure 4-5, there is a photograph of the results in the finite element model in the stud boss region, and the different colors in this photograph depict different stress levels. The red, deep red stresses, the deep red areas in the photograph are the regions in which the stresses were the highest.

Q. Gentlemen, on page 57 of your testimony, you state that the R5 has operated successfully, and I underline "successfully" for over 622 hours at

2000 psig. What do you mean by "successfully"?

DR. SWANGER: This statement refers to the fact that the two AE pistons in the R5 engine had operated successfully for the time at the pressure indicated, and by "successfully", we mean that there were no propagating cracks in that piston; that there was no adverse wear in this piston.

In fact from a friction wear and lubrication standpoint, it had operated very successfully. Also, further evidence of that is that in the inspection done by Don Johnson in LILCO's Exhibit 29, it refers to three small eddy current indications that were found.

Q. We are going to get to those. My question, and I am interrupting you because you are going beyond the question.

The question referred to your statement that the R5 is operating successfully, not that the AE piston skirts in the R5 were operating successfully. I asked you what did you mean by "successfully."

JUDGE BRENNER: I think he is answering that question, Mr. Dynner, as long as he is talking about the pistons operating in the R5 engine. Don't you think so?

MR. DYNNER: My question, or the statement in the testimony is not that the --

JUDGE BRENNER: You asked him what he thought the statement?

MR. DYNNER: What he meant by "successfully".

JUDGE BRENNER: He is telling you. Let's not go overboard the other way, either.

DR. SWANGER: Continuing with my answer, what we meant was that the three eddy current indications that were found, which happened to be found in low stress areas of the piston, namely on the lip adjacent to the washer landing opposite from the highly stressed area in the stud boss had no evidence of having been propagating. They were similar in nature to the kinds of manufacturing induced indications that were removed from the AE pistons that Shoreham purchased from TD.

Q. Dr. Swanger, you are referring, I take it, to the pages that are attached to Donald Johnson's memorandum, which is part of exhibit P-29, and referred to piston C-31 as far as the eddy current test is concerned, is that correct? My copy, for the convenience of the parties, are the two pages following Mr. Donald Johnson's memorandum.

JUDGE BRENNER: The February 3rd

memorandum?

MR. FARLEY: Yes.

MR. DYNNER: That's correct.

JUDGE BRENNER: What's your question about that page which contains certain drawings?

MR. DYNNER: Are those the indications that he was referring to in his prior answer?

DR. SWANGER: Yes. The three indications I referred to are discussed in the attachments to the trip report of Donald Johnson dated February 3, 1984.

Q. If we look at the second page, which on the corner of my copy, it says Page 3 of 3, it shows some, I suppose you would call them simplified drawings of a piston skirt. There is a notation on number 2. It shows the line in the lower left hand corner and under that it says, one, it looks like D.

JUDGE BRENNER: V at 1.5, and to the right of that it says 25 percent. Will you explain what that line and what that notation means.

MR. JOHNSON: This page which you are referring to says Page 3 of 3, and these simplified drawings are, of course, two views of the same track area. The comment that says -- and it is there to illustrate with the lines that you will see on 2, 4 and 3, and they are to indicate where the indication was observed, and the comment 1 division at 25 percent, for example, is a notation recording the magnitude of the eddy current indication, and the percent is the percent of the standard signal that we use in this calibration.

Q. Could you tell me, Dr. Johnson -- and we are looking now at the number 2 drawing, labeled number 2. How can you tell what the size of that indication is?

DR. JOHNSON: The number indicates the size.

Q. Which number?

DR. JOHNSON: Either the 1 division had a certain position on the screen or the 25 percent, those two numbers that correspond.

Q. What does the 1.5 mean?

DR. JOHNSON: The eddy current test is done by observing an oscilloscope screen, and the 1.5 refers to -- it is an oscilloscope screen and a two dimensional display of the information. The 1.5 simply is saying where on that screen it is located. The 1 division is indicating the magnitude of the --

Q. What is the magnitude of this indication, if you could put it into inches or fraction of inches or in terms of length and depth?

DR. JOHNSON: Maybe it would be best to express it in terms of the standard we use and then the fact that this signal is a quarter of a signal we get. The standards that we use are such that -- the standard will give a signal not greater than a signal one obtains from a 1/16th inch long by 1/32nd inch deep crack-like defect in the material. This is done one quarter of such a signal.

Q. A 1/32nd inch would be the length or the depth?

DR. JOHNSON: Depth.

Q. What would the length be?

DR. JOHNSON: 1/16th.

Q. One quarter that size? I can't tell from this drawing but what is the precise location of the crack-like indication on number 2?

DR. JOHNSON: Well, there are two views of that indication. So you have to look, also, at the second figure. It is on the same page but you have a circle and once again, the numbers 1, 2, 3, 4 repeated. We are looking at that figure and we are looking down, and in fact I can probably give you an example on the photographs if that will help.

Q. Before we get to the photographs -- that might be helpful if I could decipher mine which are Xerox copies. Perhaps you can describe it with words.

MR. JOHNSON: We are looking down in the figure which is right below the diesel engine piston. You will see number 2. That's looking down on the boss number 2. If we go to the number 2 which is in the upper left hand corner of Page 3 of 3, that's a blow-up or expanded view of that boss area. But now not looking down on it but at right angles.

Q. So this is a crack-like indication in the boss area of the piston?

DR. JOHNSON: A crack-like indication on the washer landing area adjacent to the washer landing area away from the high stress area noted earlier.

Q. I see. This is in an area which is not as high as the highest stress area identified in the report; is that correct?

DR. SWANGER: We can be quantitative about the location and the stress at that location by referring to County's Exhibit 8 and some of the photographs depicted in it. Looking first at photograph of figure 3-3, which is an actual photograph of the inside of an AE piston, we can see --

Q. Excuse me, Dr. Swanger. Just for the record, we can identify that the County's Exhibit 8 is in fact the FaAA piston report.

JUDGE BRENNER: We have done that several times. So I think we have that. We will follow you and let you give your description in the record. We don't have the original photographs in front of us. That brings up two points I wanted to raise.

First of all, I would like for some of these exhibits, before they are put in -- we will have to back up on at least one, I think. Before they are put in, even if it is just marked for Identification, to get the original versions in and the three copies that are going to become part of the official record. It is clear to me already that regardless of how much of what will become County's Exhibit 8 will be in evidence or for Identification, it is going to be marked for Identification. These witnesses have referred to it numerous times already in the nature of identification, namely the photograph as well as some other things. So I'd like the County to, before we put it in, to get original copies in there for 8. If you don't have enough, we will direct LILCO to assist you in getting the three copies.

8 is the example I have in mind. But think ahead and for any others that you are going to put in that have photographs, the witnesses may end up wanting to refer to, and do the same thing. Rebind the books so we have it done easily for the record and for the court reporter.

MR. DYNNER: We never got any copies of the original photographs.

JUDGE BRENNER: It is going to be taken care of now. You got copies. We have the reports along the way.

MR. DYNNER: We have the original photographs here that --

MS. TARLETZ: LILCO would be happy to cooperate with the County. I will renew LILCO's offer to provide the County with the originals if you like.

JUDGE BRENNER: If you can do that now while we keep talking, I would appreciate it. Do you have a copy of the photograph now?

MR. DYNNER: I now have the photograph figure 3-3 that Dr. Swanger alluded to.

JUDGE BRENNER: In terms of backing up, I would like for LILCO exhibit P-29 to be replaced by a version that has the original photographs for purposes of the official record copy. If you could discuss it with the court reporter during the break, I'm sure he can figure it out, some logistical way to do it so it is easiest for the court reporter and note it on the record as to how it was done. It would be acceptable to me, since he has already got a bound version, if you can have just an additional exhibit along with that bound version also labeled P-29. There is no need to use a different number. I don't know if there are any other LILCO exhibits or exhibits that are likely to be referred to. But there are, it strikes me, the exhibits in the beginning, P-1, P-2, perhaps just those also should be similarly supplemented with the original photographs for the exhibit file.

In addition, the Board would like copies of the original photographs marked P-1 and P-2. You don't have to worry about -- and also P-29.

MS. TARLETZ: I believe the copy originally served on the panel did have originals of P-1 and P-2. We will supply another copy with P-29.

JUDGE BRENNER: You think our copy in the office has the originals?

MS. TARLETZ: Yes, and we will supplement originals for P-29.

JUDGE BRENNER: All right. Don't worry about that. I will have that system well in order.

I'm sorry to digress. I had some concern about our ability and the ability of the official record to follow this through Xerox photographs.

DR. SWANGER: Referring to figure 3-3 of the piston report, the location of the indication, the lip adjacent to stud boss number 2 can be referenced to the center of the white circle in that photograph. The location is approximately 7/8ths of an inch to the right of the middle of the white circle, and approximately 3/8ths of an inch below the center of the white circle. It is essentially right at the bottom middle of the stack to have Bellevill washers that's depicted in this photograph.

The Bellevill washers were not there during actual eddy current inspection. So that the entire machine surface can be examined and such machining induced indications as this easily seen.

Also, I might point out that similar small machining induced indications were removed from the AE pistons supplied to LILCO before they left the TD factory.

Dr. Harris then can refer to another photograph to discuss further the position of this indication.

Q. Excuse me. You stuck something in there that I hadn't asked, about machining induced indications. Did you conduct any failure analysis or studies in order to ascertain the cause of the crack-like indications that were found in the AE piston skirt from the R5 engine? Can I have your answer, Dr. Swanger.

JUDGE BRENNER: Why does that have to be just him?

MR. DYNNER: Because he just testified. He is the only one that talked about some kind of machining.

JUDGE BRENNER: I understand. You are asking now whether any testing or evaluation was done of the cause. Why can't you direct it to the entire panel?

MR. DYNNER: Presumably if he didn't know he wouldn't have said it is machining induced. I don't see anything wrong with getting this witness who just made that comment to give the answer. I don't want to argue about it.

JUDGE BRENNER: All right. You may answer.

DR. SWANGER: My knowledge is based on

conversations with Dr. Duane Johnson, whether or not based on his experience is able to recognize such conditions as machining induced. We did not conduct a independent failure analysis. It was not necessary to conduct such an independent failure analysis since no such indications were on the pistons supplied to LILCO.

DR. HARRIS: If I could proceed on.

JUDGE BRENNER: Let me interject. If you could remember, Dr. Harris. We will get right to you. I am confused on the views, and perhaps it is my problem only. I am looking at the drawings in P-29 that we have been discussing. I understand the overhead view but the enlarged views state that these are side views looking out from inside. Yet when I compare that to the photograph, it doesn't appear as if one would be looking from the side. Would you help me a little bit.

DR. SWANGER: What we mean by the inside looking out is if you were right here in the middle of the piston looking toward the outside of the piston, that is looking right in the same direction this photograph was taken. Then view number 2 shows that if indication is off to the left-hand side of the lip, and the way we have located it on figure 3, it is off to the left-hand side of the exact center of the lip if you were looking at exactly from the center of the piston.

JUDGE BRENNER: Thank you. That helps. Dr. Harris, please.

DR. MC CARTHY: In this picture, in our report 3-3, sometimes you see people make ashtrays out of pistons where the head is down and the open bottom of the piston is up. It is exactly that same sort of view. You are standing in the center.

DR. HARRIS: Turning now briefly to the discussion of the stress load in the area of the machine induced indications --

Q. Before you do that, Dr. Harris, if I could just ask a clarifying question. In figure 3-3, the circled area there, is it specifically meant to refer to the crack-like indications in the R5 engine or is it meant to show where the lack of crack or something else is?

DR. SWANGER: Yes. The circle indicates the area where the stress coat cracked, and then I oriented the location of the indication on stud boss number 2 relative to the center of that circle. I indicated that if crack-like indication is at coordinates 7/8ths of an inch to the right of the center of the circle and 3/8ths of an inch below the center of the circle.

Q. But this photograph is not a photograph of the R5 AE skirt, is that correct?

DR. HARRIS: That's correct.

Q. Go ahead. I wanted to clarify that.

JUDGE MORRIS: While we are on that, gentlemen, can you tell me the approximate diameter of that white circle, what it represents?

DR. SWANGER: Do you mean to scale or what it would measure on this photograph?

JUDGE MORRIS: What it would represent in real life.

DR. MC CARTHY: Approximately a half inch. It is a half inch to 5/8.

JUDGE BRENNER: Dr. Harris, you may answer.

DR. HARRIS: Thank you, Judge Brenner.

Turning to figure 4-4 of the piston report, this figure provides a summary of the results of the finite element analysis on AE piston skirt, and the different colors in this finite element model depict different stress levels within the skirt. Backing up for a moment, I should point out that this model is a 1/4th of a complete piston, one quadrant of a complete piston. Due to the symmetry nature of the piston and the symmetry of the loading, you can break the complete piston up into quarter segments and analyze just one quarter. The results for that one quarter, you can determine where the stresses are in the pistons.

The various colors on this photograph provide information on the stress levels. You can see that the colors vary all the way from a fairly dark brown to a very light blue. The stresses are highest at the very dark brown position as indicated by the numbers to the right of the color scale on this photograph. You can see down close to the stud boss region but over where the stud boss meshes into the wall by the wrist pin that the stresses are quite high. This is where the dark brown colors are. That point corresponds to the point at which the stress coat crack that was shown in figure 3-3 that we discussed a moment ago appears.

Hopefully it is apparent where the hole is. There is a hole that goes down through the -- in this case the bottom of the skirt. That's the hole that the stud protrudes through. Then you can see a horizontal landing area around that hole which is the region in which the Bellevill washer is seated upon. You then proceed around the lip of this horizontal surface and you can come around to approximately the point that Dr. Swanger indicated that the indication was, and you find that this corresponds to the very light blue. Looking over on to the color scale on the right-hand side of this figure, you can see the very light blue on the lower stresses of any of the stresses depicted in that color scale.

This shows that that indication was located in a region of relatively very low stress.

Q. Is that true, Dr. Harris, with respect to the crack-like indications that are identified as numbers 3 and 4 as well as number 2?

DR. HARRIS: Yes, Mr. Dynner.

Q. Was this area as depicted before in figure 3-3 of the piston report, is this one of the areas that was highly polished in the R5 skirt?

DR. MC CARTHY: It is our understanding that the general area was polished up. We wouldn't use the term "highly polished ", but it was smooth.

Q. Did you conduct -- and by you I mean FaAA or anyone on the panel, if you know -- did you conduct a die penetrant examination in eddy current test of the R5 AE skirt that we are talking about before it ran for 622 hours?

DR. JOHNSON: No, we didn't run any kind of test, non-destructive examination on these areas before they were run. Neither penetrant nor eddy current.

Q. If you didn't run an eddy current test before the run and if you know that area was highly polished, or polished, I am interested in the basis for your conclusion that these crack-like indications were the result of some machining error or operation rather than the possibility that they were the result of stress or operation of the AE skirt in the R5 engine.

DR. HARRIS: I would like to start out answering that question and quickly pass off to Dr. Johnson. The results of the stress analysis indicate that they were not meeting the crack initiation in that very low stress region based on the stress analysis. We would not expect to see any defects initiated in that region. They would be much more concentrated over from where the high stresses are.

Dr. Johnson, I believe, has some other words he would like to add.

Q. You understand my question. I know you didn't expect to find it there. My question then is how you came to the conclusion that these crack-like indications were the result of some machining or manufacturing operation after they had been polished and not the result of the operation of the engine?

DR. MC CARTHY: I guess I would say I'm extremely confident that these indications were not the result of observation but fabrication because we have seen very similar indications in as-fabricated pistons. What you are looking at is a very thin edge that results from the way the stud boss is machined outboard, and it is not relevant in any way to the strength of the piston, and more important, it is not just that we don't expect cracks to grow in this area. This is, in fact, one of the lowest stressed areas in the whole piston. There is no conceived indication that that could conceivably be.

Q. I understand you said two things there, Dr. McCarthey. You said you have seen similar indications in as-fabricated pistons. Which ones are you talking about?

DR. MC CARTHEY: In the ones that were delivered to the Shoreham Power Station. When we inspected these at TD, one could occasionally see very thin line machine-induced indications in the outboard area of the stud boss. Any such indications we remove are a minimal amount of touching up, and they were gone. They are just not in any conceivably related structural part of the engine. They are just an artifact of manufacture.

Q. How did you remove the indications in the edge fabricated pistons you saw at TD?

DR. JOHNSON: Failure Analysis did not remove those. TD removed them under the supervision of LILCO's QA representative, and they used a grinder, a surface grinder to remove the source of the indication prior to shipment.

Q. You just polished them out?

JUDGE BRENNER: Mr. Dynner, while they are conferring, after the answer, I think it is about time to take a break.

MR. YOUNGLING: TD used a simple pencil grinder to grind the indications out.

Q. A pencil grinder? Do you want to please help me out with that?

MR. YOUNGLING: A small tip grinder to get down in the area.

Q. What was the nature of the abrasive material on the grinder, if it was abrasive?

MR. YOUNGLING: I don't know.

JUDGE BRENNER: We will take a break at this point and come back at 11:10.

(Recess taken)

JUDGE BRENNER: We're back on the record. I'm concerned about the time estimates. It is 11:15. We're going to break at noon for lunch and then come back at 1:30.

Are you ready at -- excessive side thrust yet?

MR. DYNNER: Almost.

JUDGE BRENNER: How long do you think it would take you to finish the excessive side thrust and tin plating which would be sub-parts B and C of Part 4 that you mentioned?

MR. DYNNER: If things go the way they've gone this morning the rest of the day, frankly, I don't want to criticize the witnesses, but as you know there have been enumerable conferences and lengthy periods of time where the answer is given to a question and I realize that the material that we're dealing with is complex and I don't want to criticize the witnesses, but it has taken a lot longer than I ever anticipated.

JUDGE BRENNER: Within reason. That's a fair comment; however, if we're going to be fair there are also times when you take some detail out of an answer either because somebody is handing you a note and you may think it's interesting at the moment or I think it's interesting at the moment, and it turns out to be a nonmaterial point that you've been off on it for 15 minutes when if you had more directly asked the question that you were trying to get to, we would have found out it was nonmaterial.

What I'm saying, Mr. Dynner, you have a better view or are in a better position to exercise a decision as to what's material to the issues before us in controversy as distinguished from a technical person's point of view of something that may be technically interesting or technical inconsistent but is going to really turn out to be a difference without a distinction.

And I don't think we're prepared to give you the rest of the day to complete your cross-examination.

My view of reading the cross plan and what you can more importantly get to is that you should be able to finish it in about two-and-a-half hours, so what we'll do is we'll -- I'm allowing about 15 minutes for you to work up to those two topics because you said you were almost there, so what I would say is we're going to direct you to complete your cross-examination of this panel by two hours after we return from lunch, which would give you a total of two-and-a-half hours, maybe a little more to get there sooner than 15 minutes. We'll re-evaluate the situation. If my present view turns out to be wrong based on the value on efficiency on your part of everything you've asked up until that point, but as of now, that's where we'll stop you so you have to assume that you'll be stopped then. And if there has been a problem with length of answers and so on in the time from now on, we'll take that into account, but as I said, right now we will assume that we will require you to complete the cross-examination of this panel by two hours after our return from lunch.

Why don't you proceed at this point.

BY MR. DYNNER:

Q. Gentlemen, did TDI do an eddy current examination for a liquid die penetrant examination of the AE piston skirts in the R5 engine before they started their 622 hour run?

DR. SWANGER: We don't know whether they did or not.

Q. How many AE piston skirts were in the R5 V12 engine at the time that you selected the two AE skirts for examination?

DR. SWANGER: The pistons were already out of the engine at the time they were made available for our analysis.

I believe that there were only two AE pistons in the R5 engines as part of a development experiment.

Q. Are you aware of an incident with the R5 engine in which a portion of the cylinder liner broke off and fell into the crank case?

MR. FARLEY: Objection. Irrelevant and immaterial.

JUDGE BRENNER: You're going to have to explain that objection to me, Mr. Farley.

MR. FARLEY: I don't think it's relevant to any of the contentions, Your Honor. Or to what's been discussed about the R5 engine.

JUDGE BRENNER: Wasn't it relevant to the contention that the FaAA analysis depends on an ideal situation which is not valid for the actual conditions which may be experienced by the Shoreham diesel?

MR. FARLEY: I think we've demonstrated that wasn't the case.

JUDGE BRENNER: I don't understand. I'm going to overrule the objection.

I'll add that I believe it's apparently relevant, at least at this point, because your witnesses are relying on the experience with the R5 engine to justify the fact that the expected experience in the Shoreham engine will be acceptable and will not be inconsistent with the assumptions in the analyses leading to the predictions by your offices.

You have a question?

DR. SWANGER: If Mr. Dynner would repeat it, it would be helpful.

Q. Are you aware of an incident in which a portion of the cylinder liner in the R5 engine broke off and fell into the crank case?

DR. SWANGER: Yes, I have heard a little bit about that incident.

Q. Could you briefly describe -- retract that.

Did that incident involve in any way the AE pistons in the engine or liners which were in the cylinders that were using AE pistons?

DR. SWANGER: There were only two AE pistons of the type similar to those delivered to LILCO in the R5 engine, and neither of these AE pistons were involved in any incidents involving cylinder liners. Had they been, those pistons would have suffered extreme distress, would not have looked the way they did.

Q. Do you know what kind of piston was involved in that incident?

JUDGE BRENNER: Now, Mr. Dynner, you're going to have to tell me why it's material.

MR. DYNNER: It might have been an AE piston. He said it wasn't only the ones that he looked at.

JUDGE BRENNER: Maybe I can add one and one as well as other people, but he said there were two and he looked at the two.

Maybe I can't add.

MR. DYNNER: But at the time that they selected those two to be tested he said they were to his knowledge two.

I don't know whether it's an AE piston or AF piston or what but if he doesn't know, that may be significant.

JUDGE BRENNER: He said it was not an AE piston and anything further is going to be immaterial.

MR. DYNNER: I don't think he said it wasn't an AE piston, sir. I think his answer was that he doesn't think it was an AE piston because if it had -- the two AE pistons they examined had no damage and they would have had damage if they had been the ones involved.

JUDGE BRENNER: Dr. Swanger, were there ever any other AE pistons ever run in the R5 engine from which you removed pistons or from the engine in which the cylinder liner incident occurred beyond the two that you looked at?

DR. SWANGER: No. Those two pistons were the only AE pistons ever tested in the R5 development engine.

Q. Was any evidence of scuffing of the skirts or fretting of the AE piston skirts from the R5 engine noted by you?

DR. SWANGER: I believe your question referred just to AE pistons. Is there any specific AE pistons you're interested in?

MR DYNNER: Yes.

Q. Yes. I stated in the R5 engines.

DR. SWANGER: I have discussed this point with Donald Johnson who inspected the pistons and I have reviewed the photographs that Don Johnson took of those pistons, and I saw no evidence of scuffing or fretting on the AE pistons from the R5 engine.

JUDGE BRENNER: Dr. Swanger, could you tell me what you mean by the term fretting on the piston?

DR. SWANGER: Fretting is the result of small amounts of relative motion between two metal surfaces which results in the transfer of metal from one metal surface to the other. It's recognizable by a roughened condition of the surface relative to its original appearance.

JUDGE BRENNER: Could you compare that to scuffing -- in my own mind I thought scuffing was a roughened condition of the metal.

DR. SWANGER: Scuffing is the result of large relative motions between two metal surfaces such as when a piston slides up and down 21 inches inside the cylinder.

Fretting would refer to motions on the order of a few thousandths of an inch relative to each other.

Q. Gentlemen, do you know whether DeLaval tested the AE piston before supplying it to customers in the field?

DR. SWANGER: DeLaval conducted at least two engine tests of the AE skirts. These were the tests of two AE skirts in the R5 engine and also the placement of AE piston in the engine at Kodiak Electric was a test of the engines conducted jointly between TDI and Kodiak for the purpose of evaluating the AE piston design.

Q. Is it your testimony that that testing was done by DeLaval before supplying the AE piston to customers in the field?

DR. SWANGER: As I had testified earlier, by supplying the pistons to Kodiak we do not consider supplying pistons to a customer in the field. TDI has a special relationship with the Kodiak Electric Association in which their engines are designated as lead engines for the purpose of gathering test experience for TDI.

Also, it is our information that the AE pistons were put into the R5 engine for test purposes about the same time that AE pistons went into the Kodiak engine and they had been successfully tested and removed from the R5 engine prior to the delivery of pistons to LILCO.

I believe Mr. Youngling can give you further information about delivery of AE piston to customers.

MR DYNNER: I don't want further information. I just want to know whether that's your answer.

JUDGE BRENNER: All right. He's answered the question. I'm beginning to worry about the materiality of this line if I let it go too far, unless it gets tied into something specific. We discussed that --

MR. DYNNER: I have a few questions to ask concerning their testimony regarding the importance of the testing of the operation of the AE piston skirt in the R5 engine and at DeLaval and their testimony regarding it at Kodiak.

JUDGE BRENNER: Well, I know. You've been asking questions about that, and you can go ahead. Maybe I'll ask you on your cross plan --

MR. DYNNER: Right now I just jumped outside of the cross plan for a minute to ask a few questions and I think the pertinence will be quickly obvious. It's too abstract to be helpful. I don't want to hear about that overall testing was done and what testing was first. Ask him about the -- tie it up to the particular point. For example, I don't know if you want to talk about side thrust load with that question or tin plating or Part A of Part 4 of the contention, so you're going to have to be more specific in your questions.

Q. Does the testing that you alluded to give you confidence that the AE piston will last the lifetime -- or it will have unlimited life?

DR. SWANGER: Yes. The test experience with the pistons confirms our conclusion that cracks will not initiate or propagate. It also adds to our opinion that there is no problem associated with the friction wear or lubrication of these skirts and we feel very strongly that this is important evidence and confirmatory evidence that these pistons will fulfill their intended function at the Shoreham Nuclear Power Station.

DR. PISCHINGER: I think I could add to this question.

This AP piston is for the development of the previous AF pistons, modified AF piston, and from a diesel engine engineering man's point of view, this is a minor design modification which had been taken and the design modification is in the -- in all -- in each respect in improving or in strengthening of this piston skirt, and it is clear that in such a case, and. and its general use and its practice in the industry that you rely on the pre-experience with the model from which you derived this piston.

That means that you have to take into account the pre-experience with the AF piston, and that very wide experience with several hundreds of pistons have been delivered and there's a lot of tens of thousands of hours that have been run with this AF which is the DeLaval AF pistons without, to my knowledge, as when I visited the DeLaval in February, without to my knowledge any unfavorable events which can be related to this AF piston.

The only reason why there was a design change was that there could -- cracks could be seen in the stud region, but these cracks didn't lead to any consequences to the engine.

It is in -- it is common use in the diesel engine industries that in such a case you take a further development step, but you rely on all the other experience. The outside of the piston remained completely the same, the ring portions, taking into account what has been investigated by FaAA and the pre-experience with the AF piston that there is additional evidence that this piston will last and do the required -- or fulfills the required functions.

Q. Well, based on that testimony, Dr. Pischinger, evidence of failures of AF pistons would be relevant to your analysis of the quality of the AE piston; isn't that true?

MR. FARLEY: Objection, your Honor. It doesn't necessarily follow.

JUDGE BRENNER: Well, let the witness explain why or why not.

DR. PISCHINGER: Let's say failures which couldn't have been addressed by this design change.

Q. Have you done an analysis of all the failures of the AF pistons in order to determine that they have been effectively solved by the change in the design, Dr. Pischinger?

DR. PISCHINGER: No. I only have knowledge that a number of engines have run for thousand, tens of thousands of hours without piston failure.

Of course, I am aware that this is only information which I got from -- at my visit at TDI, if this is reliable, yes.

Q. Dr. McCarthy, would you turn, please, to page 55 of your testimony for a moment.

I believe you alluded to this earlier in answer to one of my questions, and I'm referring to your answer 87, particularly at the bottom of page 55 continuing to page 56, to where you refer to the fact that, "Information contained in the Iron Castings Handbook by Walton and Aupar, 1981 page 341, Exhibit P-29 shows that the cyclic stress for cracking in ten million cycles is 93 percent of the cyclic stress for cracking in 1.35 million cycles, scatter of seven percent on stress is commonly observed in fatigue data. Therefore, it is likely that cracking indications would be observed in the population of inspected stud bosses if they had been operated for a 1.35 million cycles at stresses above the endurance limit.

Now, does that statement refer to the comment that you made earlier about steel, which you also said applied to nodular iron?

JUDGE BRENNER: Didn't you ask that yesterday and get the answer?

MR DYNNER: No, I didn't. He made a comment about it. I never asked a question about it.

JUDGE BRENNER: No, I mean, did you then follow up and ask him, make sure he was talking about steel, why that referred to this question?

MR. DYNNER: Sure. I want to make sure that this written testimony is what he was referring to yesterday.

JUDGE BRENNER: That's not what you asked, though.

MR. DYNNER: I think I did.

I'll rephrase the question.

JUDGE BRENNER: Why don't you just -- draw it up. I'm sorry, I may have lost your point. You asked him yesterday why, if he was talking about steel, we should be concerned with that and he said if he said steel he didn't mean to say it. He meant to talk about ferritic in general and that his answer referred to metal.

Q. I just want to clarify, does this statement refer to steel?

JUDGE BRENNER: What statement, answer 87?

MR DYNNER: The one that I just read in quotation.

DR. MC CARTHY: The phenomena I described earlier discussing a need or a stress level at which ferritic materials exhibited unlimited life is characteristic of both iron and steel. The specific numbers and referenced exhibit on page 55 -- by the way, it should be Exhibit 28, not Exhibit 29 is for the iron as reflected in the iron castings handbook.

Q. Do you know what the cycles, the millions of cycles approximately would be for steel?

1.35 million cycles for cast iron. What

would be the approximate number of cycles for steel?

DR. MC CARTHY: The difference for all ferritic materials, the difference between the stress level for failure between one million and ten million cycles and infinite life is a few percent. In this particular case, it's seven percent for this iron. It would not be uncommon to see steels range from five to less than ten. It would depend on your exact material, but it's always a few percent.

Q. Then it's true, isn't it, that based upon this testimony, it would have been highly unlikely or unlikely for the crankshaft on diesel 102 to break; isn't that true?

DR. MC CARTHY: Perhaps I missed something in my previous answer. I don't remember discussing the probability; however, that's an excellent point. We ran three crankshafts, one broke, the other two cracked, and they were all within a few percent of their endurance limit. They had enough strength to get them into the one million to ten million cycles, but didn't have enough to get past ten million failed, and it wasn't that one crankshaft failed and the two came out looking cherries. On the contrary, the physical laws applying to crankshaft apply just as well to pistons.

You found one failed crankshaft and when you pulled the other ones out you found crack indications at both of them.

Just a textbook example of how reliable this particular theory is.

Q. Yes. And you have confidence that would take care of that seven percent factor if you ran the AE pistons in the Shoreham engines for ten million cycles, wouldn't you?

DR. MC CARTHY: Once again, maybe I missed something. Not one but all three pistons having been run into this seven percent range -- excuse me, not one but all three crankshafts run into this very narrow seven percent boundary had cracks. Now, we have run 80 fillets, 40 piston bosses into this same range and we got indications on three of three crankshafts.

Now, one can do a probability calculation of what you -- what events would have had to transpire where three of three crankshafts showed cracks, indeed one failed and the other two had cracks and yet turn around and run 40 of 40 piston bosses without a single indication. Of course, the odds are vanishingly small and it's just confirmation that the piston as we've indicated before is not going to crack or propagate.

Q. Would you answer my question which was that if you ran the AE piston skirts in the Shoreham engines for ten million cycles, that would give you the confidence to take care of the seven percent differential in your test that's referred to in your testimony at the bottom of page 55 and the top of page 56?

DR. MC CARTHY: On the contrary because we ran so many, I have high confidence that there's nothing in the seven percent value that needs to be resolved.

If nothing else, the cranks have demonstrated that. They ran --

Q. I'm talking about piston skirts. Sorry. That possibly isn't --

JUDGE BRENNER: Let him finish his answer because you drew an analogy, and I think he's continuing with that thinking.

DR. MC CARTHY: The three cranks were run into exactly this range --

JUDGE BRENNER: Let's say crankshafts.

DR. PISCHINGER: Crankshafts.

DR. MC CARTHY: Yes. I'm sorry. You're correct. The three crankshafts were run into this range. They all showed crack- like indications. In fact, one failed.

Once again demonstrating that if you take a part that is above its endurance limit into this one million to ten million cycle range, you would expect to see some crack indications.

The crank -- once again -- the crankshafts demonstrated this phenomena.

Now, let's turn around, take what we've learned from crankshafts in these engines at Shoreham and apply it to pistons.

We now run 40 piston bosses, 80 stressed areas into exactly the same range and see not a single relevant indication.

Q. Do you know how many millions of cycles the crankshaft on an engine 102 was run before it broke, approximately?

DR. MC CARTHY: It's a few million -- it's a few million -- just one second.

Q. I think if that had help you, Dr. McCarthy, as I recall, and anyone in the panel can correct me, I think it was about 680 hours, as I recall. Mr. Seaman or Mr. Youngling will probably know the exact figure.

DR. SWANGER: The hours that are

significant to the fatigue analysis are the hours at full load where it collects stress cycles at the maximum cyclic stress. EDG 102 had run about 250 hours at full load, so that is the relevant number.

Q. How much is each of the piston skirts run at full load at Shoreham?

JUDGE BRENNER: Maybe I'm wrong. I thought they were going to complete their answer and give you cycles.

Q. I'm sorry. I thought they had completed. Go ahead, please

JUDGE BRENNER: If you still want it.

DR. MC CARTHY: In answer to your question, 3.4 million cycles at full load.

Q. Thank you.

And how many hours has each of the -- or how many hours have been accumulated on the most utilized piston skirt at Shoreham at full load, if you know?

JUDGE BRENNER: On the AE piston skirts.

Q. The AE piston skirts, yes.

DR. MC CARTHY: On EDG 103, the most highly utilized AE piston skirt has gone 1.35 million cycles after which it was inspected. Additionally, and currently operated AE piston skirt in EDG 103, eight of them have gone 1.75 million cycles and are still running, but haven't had a subsequent inspection, but are still performing and service fine.

MR. DYNNER: Judge Brenner, I am ready to go on to successful side thrust if you wish to break a little early for lunch and then we can go straight through if that's convenient for the Board.

JUDGE BRENNER: Let me ask one clarifying question.

Dr. McCarthy, you've referred to the fact that there are eight fillets involved, I guess, on each piston, if I've got it straight. Did you mean welds when you said fillets or did you mean something else?

DR. MC CARTHY: Did I say welds?

JUDGE BRENNER: No. You said fillets. What do you mean by fillets?

DR. MC CARTHY: I'm sorry. In the area where the stud boss blends into the wall of the piston, because of the geometry there, there are two areas that have been termed fillets. They're much less pronounced on the AE design than on the AF design where we started talking about fillets, and thus I've always been -- I've tried to be consistent in distinguishing boss areas and then fill let areas.

JUDGE BRENNER: That's fine. It took me a few days earlier in this case to know what a welder meant by fillets and now I know what you mean by fillets.

Let's break until 1:20.

(Whereupon, at 12:00 p.m., the hearing was recessed, to reconvene at 1:20 p.m., this same day.)

AFTERNOON SESSION

JUDGE BRENNER: Mr. Dynner, I guess you're going to pick up on page 14 of your cross plan, excessive side thrusts.

MR. DYNNER: That's correct.

CONTINUED CROSS-EXAMINATION

BY MR. DYNNER:

Q. Gentlemen, please turn to page 58 of your testimony.

Dr. Pischinger, in your answer to question 92 at the bottom of the page, you state that --

DR. PISCHINGER: Could you give me just a minute?

Q. Certainly. Page 58 of the LILCO direct testimony.

DR. PISCHINGER: Yes.

Q. In your answer to question 92, you state that in current diesel engine design side thrust, the excessive side thrust related by the County is simply not a consideration.

What current diesel engine design were you referring to in that statement?

DR. PISCHINGER: Diesel engine designs at least back-dated to the mid-sixties.

Q. Would you specify the engines, the design of which you were referring to?

DR. PISCHINGER: I do not want to specify a certain engine to which this refers because I know in the state of the art that it refers to all engines.

Q. So is it your testimony that since 1966 no diesel engine design considers side thrust?

DR. PISCHINGER: Yes. Side thrust is no special concern. I can explain to you, if you want, why.

Q. Are you familiar with diesel engines manufactured by Sulzer?

DR. PISCHINGER: Yes.

Q. Is side thrust a consideration in the design of engines manufactured by Sulzer since 1966?

DR. PISCHINGER: I have to add that as we are talking on four strokes, I only referred to four strokes.

Q. Four stroke engines?

DR. PISCHINGER: Four stroke engines.

Q. Are you --

JUDGE BRENNER: Are you trying to tell us that the Sulzer engines are not four stroke engines?

DR. PISCHINGER: They have both types.

Q. Is side thrust a consideration that the design of the four stroke Sulzer diesel engine?

DR. PISCHINGER: As is worked out in the testimony, side thrust is as long no concern as proper lubrication is provided by the design and that is the case with modern design of diesel engines, at least in the state when they are working with the customers.

Q. Is side thrust a consideration in the design of the four stroke Sulzer diesel engine?

DR. PISCHINGER: I am not aware if they have the proper lubrication system.

Q. Isn't Sulzer one of the largest diesel engine manufacturers in the world?

DR. PISCHINGER: It is one of the largest diesel engine manufacturers.

Q. And do you consider Sulzer to be a manufacturer of high reputation and quality?

DR. PISCHINGER: Yes.

Q. Are you at all familiar with the rotating piston which is part of the design of the Sulzer engine?

DR. PISCHINGER: Yes.

Q. That rotating piston is a design element which is specifically directed towards avoiding the distortion of the piston skirt caused by side thrust; isn't it?

DR. PISCHINGER: No. The rotating piston is designed to distribute the wear of the piston skirt equally on the surroundings of the piston, so the rotating piston is a means to prolong the lifetime of a ship engine, to prolong the lifetime of a ship engine. You know, ship engines of the Sulzer design are expected to have the lifetimes of ships and lifetimes of ships are updating from 50,000 to 100,000 hours.

Of course, in such an application, the distribution of wear around the equal -- the equal distribution of wear around the piston is -- and each moving part is wearing and, of course, in this -- in this connection is wearing according to site pressure. This wear is quite normal. It's not dangerous in sho.t running times, and to distribute this wear equally over the skirt, the piston is rotating.

DR. SWANGER: At this point, i would like to add --

MR. DYNNER: Excuse me, if I may. Yes. I'm asking these questions of Dr. Pischinger because he is the sole sponsor of that testimony and I'm about to follow up on this Sulzer engine.

JUDGE BRENNER: I'll let you follow-up but Dr. Swanger's response to 91 which is the start --

MR. DYNNER: Which talks about lubrication and I'm really talking about the diesel engine design issue which was raised by Dr. Pischinger in his answer to 92.

JUDGE BRENNER: I'll let you follow up and then get back to Dr. Swanger.

MR. DYNNER: I'm going to distribute and ask that there be marked for identification Suffolk County Diesel Exhibit 69.

JUDGE BRENNER: I'm sure you're going to use this one before I mark it.

MR. DYNNER: Yes. Which is an article from MOTOR SHIP TECHNICAL MAGAZINE entitled "Sulzer's

Four Stroke High and Medium Speed Engine Range."

JUDGE BRENNER: What did you say this was from?

MR DYNNER: Article from the MOTOR SHIP as identified on the first page, and it's February 1978, and as seen on page 52, which is the first page with text on it, it is entitled, "Sulzer's Four Stroke High and Medium Speed Engine Range."

JUDGE BRENNER: And it runs through page 60?

MR DYNNER: And it runs through page 60, yes, sir.

JUDGE BRENNER: So this will be marked as Suffolk County Diesel Exhibit 69 for identification.

(The document referred to was marked Suffolk County Diesel Exhibit No. 69 for identification.)

Q. Dr. Pischinger, I'd like you to please turn to page 60 and in the left-hand paragraph near the left margin is the following statement.

"The pistons --

DR. PISCHINGER: I didn't find it.

Q. Yes. The last page in the left-hand margin. It states: "The pistons of larger engines are more prone to piston seizure because of the higher deformations involved.

The risk of seizure is aggravated by the customer's demand for low lubricating oil consumption -- and by the requirement to burn low quality heavy fuels.

It goes on to say: "In order to solve these problems and to satisfy the demands connected with high specific output and good reliability, the well-known rotating design piston was adopted for the Z40/48, figure 16 left, as well as for the larger 65/65 engine.

The advantage of such a design is that local overheating is avoided due to the rotary movement."

Now, Dr. Pischinger, does this article refresh your recollection concerning the independent purpose of the design of the rotating piston in the Sulzer engine?

DR. PISCHINGER: It's one point to be stated that engines of the same rating and performing equivalent functions are working with an unrotated piston, though this rotating piston is not in general a requirement.

In this case of an engine as is stated here which has to burn low quality heavy fuel, the well-known increased wear of this heavy fuel coming down the piston is, of course, a concern, and a rotating piston may help in this respect, but this is in no contradiction, when I say, and I remain with it, that side thrust -- side thrust is not addressed here. Side thrust is no concern in modern design diesel engines.

JUDGE BRENNER: Mr. Dynner, maybe I'm totally in the dark. I'll let you read the part and put the question that I assumed you were going to put to him and you didn't disappoint me.

How is anything in what you read inconsistent with what he said and more -- another way of saying that is, how is anything that you read from this exhibit remotely related to side thrust load?

MR DYNNER: I'll explain very succinctly.

As the County's direct testimony states: "Side thrust is a factor which causes the temperature on the piston skirt to be asymmetrical, so that part of the skirt is heated whereas the other side of the skirt is not heated as much.

As the side thrust continues and as the County's testimony states, "The increase temperature on one side of the skirt causes deformation of the skirt which can lead to piston seizure."

It is precisely that issue of side thrust as described in the County's direct testimony which in this article states is the -- an important purpose of the design of the rotating piston in the Sulzer engine. The article as appeared in the paragraph that I read also goes on to state: "The risk of seizure is aggravated by low lubricating oils, but the thrust of the article and statement and testimony goes to the fact that it is a moderate current diesel engine design that is specifically addressed to the issue of side thrust.

JUDGE BRENNER: You've got to get a lot of links in there in order to get there, and what you just read here for identification which you're using for cross-examination doesn't supply many of those links. But as you said, we'll have the County witnesses and Dr. Pischinger's testimony in answer to your question.

DR. PISCHINGER: Judge Brenner, may I add something?

JUDGE BRENNER: Is it in answer to this question? I didn't really have a question of you. My question to Mr. Dynner was for a different purpose. But go ahead and add it.

DR. PISCHINGER: Well, I stated that if proper lubrication is supplied, the side thrust is of no concern, and in this article just given to me, there is written, "With every stroke a fresh oil wetted part of the skirt is turned into load carrying zones substantially reducing the danger of seizure."

That means exactly that also here is one technology for using -- for solving this lubrication problem used, but this is not the only technology in light of the -- that can be seen from the fact that all other engines in the world do not have this rotating piston

JUDGE BRENNER: Incidentally, Mr. Dynner, is this some independent magazine or something published by the Sulzer Company?

MR DYNNER: Independent technical magazine and it is a technical article. The authors are noted as G. Luftgarten and R. Stoffel. The first gentleman according to the asterisk as as having development and design for four stroke engines and I believe that's for Sulzer. The other gentlemen is head of development test beds according to the double asterisk.

DR. PISCHINGER: I know the first

gentleman and had a lot of discussion with him on this business of pistons. He's head of development of Sulzer.

MR. DYNNER: Thank you.

JUDGE BRENNER: Maybe it doesn't matter but the company got quite a plug from the cover of this magazine and it looks to be more like advertising literature rather than trade magazine.

MR. DYNNER: I don't believe that's a fair comment because I think it is a technical article written by the people that presumably best know the engine since it's the head of development design for the company as well as the head of development test beds, so I do think that the information in the article is not subject to attack that it's puffing or that it is written by the advertising department for Sulzer.

MR. FARLEY: Judge Brenner, LILCO has an entirely different position about this particular article.

JUDGE BRENNER: It's not in evidence.

MR. FARLEY: Not now. The extent to which Mr. Dynner has used it now was appropriate, to try to impeach the witness, which he was unable to do.

JUDGE BRENNER: All right. Doctor Swanger, you wanted to jump in before. We'll get back to you now. Only if it's in answer to the question that was pending, and the question was whether side thrust load was a design concern for the Sulzer engines, and I thought Dr. Pischinger answered it.

Do you have something to add?

DR. SWANGER: No. My comment was not going to directly answer that. I was going to put this hundred thousand hours into the context at Shoreham and at the same time correct a misconception I may have given in my earlier testimony.

I testified that the engines at Shoreham would run for three thousand hours. That three thousand hours is the amount of operation expected after the plant goes on line.

I neglected to include in that the pre-operational testing so that the total accumulated hours on the Shoreham engines over their entire life will be about 4,500 hours and the source of this is from the affidavit of John Kaymart (phonetic) which has recently been filed.

MR. DYNNER: May I ask what the witness

is testifying relative to? Because I don't understand what he's talking about.

JUDGE BRENNER: Well, in the first place he said he was correcting a misimpression that he might have given and I think he -- in terms of the number of hours that the diesels were run, and I guess he was concerned that comparing the number of hours of shipment engines as testified to by Dr. Pischinger would make the wrong comparison if we went back to Dr. Swanger's earlier testimony.

MR. DYNNER: That would be appropriate for redirect examination, I think.

JUDGE BRENNER: Well, it would be, but we also give witnesses flexibility to correct something when they may have made an error, which error may be leading to another question on it: As to the rest of what he had to say would have been more appropriate for direct.

DR. MC CARTHY: May I just add --

JUDGE BRENNER: No. Let's wait for another question.

Q. Dr. Pischinger, in connection with your answer to question 92, I'd like to ask you whether the side thrust load is an -- a critical parameter of the Mirrlees KV12 engine. M-i-r-r-l-e-e-s.

DR. PISCHINGER: You mean in light of what has been said now, you mean if I know if the lubrication of the Mirrlees engine piston is done that way that, as usually, side thrust is no concern.

As far as I know, Mirrlees engine -- could you repeat the --

Q. The KV12 I was referring to.

There are thousands of engine abbreviations in the world, so if I remember right what the KV12 is, I think this engine is a good engine.

Q. It's a good engine?

DR. PISCHINGER: Yes.

Q. Who is Mirrlees, are they one of the largest diesel engine manufacturers?

DR. PISCHINGER: Mirrlees is a well-known English -- British diesel engine.

Q. Do they have a high -- good reputation for quality engines?

DR. PISCHINGER: Mirrlees has, to my knowledge I have not been detailing the Mirrlees engines, but to my knowledge Mirrlees has a good reputation.

JUDGE MORRIS: Excuse me, Mr. Dynner, Dr. Pischinger, do you know if this is a ship engine?

DR. PISCHINGER: This is mainly a ship engine, yes.

Q. And it's also used extensively in stationary applications, isn't it, Dr. Pischinger, the Mirrlees diesel engine?

DR. PISCHINGER: Each ship engine could be used if adjusted to stationery application.

MR. DYNNER: I'd like to distribute and have marked for identification Suffolk County Diesel Exhibit 70.

MR. FARLEY: I'll object to that because I don't even think he's allowed -- established the foundation for even using it.

MR. DYNNER: You don't know what it is yet.

JUDGE BRENNER: Why don't we hold off on marking it if there is no foundation. Let's see what you're going to do with it for a little bit first, rather than just marking for identification. We can do that but I don't want to go through the process and find out some of these things go nowhere. Let's establish what it is and we'll be done with it.

You've asked your question about Mirrlees and he's answered it.

Q. Now, Dr. Pischinger, does Mirrlees

consider the maximum thrust pressure on the piston to be a critical parameter?

DR. PISCHINGER: In that very moment when I look at this picture without reading anything, I would think this piston, which is not the very latest design, could have problems besides thrust.

JUDGE BRENNER: Wait a second. You're anticipating them. Let him take a question at a time. He hasn't asked you about the article yet. Although I hope he gets to the question quickly.

You asked him if he's acquainted with Mirrlees. He said yes and in fact in answering that he answered the question you should have asked him more specifically as to lubricating oil and so on. Now I assume by handing up this document you want to follow-up on his answer. So why don't you directly ask whatever it is you want to ask him.

MR. DYNNER: I have a question pending, Judge Brenner, if you want to reread the question.

JUDGE BRENNER: Unless it's my fault. I don't recall what the question is.

MR. DYNNER: I can repeat the question.

Q. Dr. Pischinger, does Mirrlees consider the maximum thrust pressure on the piston to be a critical parameter of their engine?

DR. PISCHINGER: I am not in the thinking of -- or on the brains of the Mirrlees people of 1966 from which this article is stating.

So I cannot answer your question what the Mirrlees people considered at that time.

Q. All right. The County's exhibit, Diesel Exhibit 70, which I've requested to be marked for identification, is an article from the Institute of Marine Engineers, Transactions, January 1966, Volume 78, Number One, and beginning with the first text page, which is page 325, there is an article entitled: "The development of a highly rated medium speed diesel engine of 7,000 to 8,000 horsepower for marine propulsion."

JUDGE BRENNER: 9,000.

MR. DYNNER: Sorry?

JUDGE BRENNER: It's 9,000.

MR. DYNNER: I'm sorry. 7,000 to 9,000 horsepower for marine propulsion.

The authors were shown as J. A. Pope, who is identified as the research and technical director of Mirrlees National Limited and W. Lowe identified as the chief development engineer for Mirrlees National Limited.

MR. FARLEY: Judge Brenner, I will object --

excuse me. I thought you were finished.

MR. DYNNER: I was about --

JUDGE BRENNER: He thought you were finished.

MR. DYNNER: I was about to ask a question on this exhibit.

JUDGE BRENNER: What is your objection?

MR. FARLEY: I'll wait until the question.

JUDGE BRENNER: You're going to ask him, I assume, whether he's familiar with the article or something like that. I let you slide with some of the niceties on the other one because we got some direct answers but from Dr. Pischinger's previous answer if he doesn't know anything about this article I'm not going to proceed very far with taking some excerpt out and asking him what he knows about it and so on, but maybe I'm misguessing as to where you're going with this.

He offered some comment as to the relevance of the fact that it's a 1966 article.

Q. Are you familiar with this article at all, Dr. Pischinger?

DR. PISCHINGER: No.

JUDGE BRENNER: Now. You can go a little bit if you have a particular point and you want to know if that refreshes his recollection as to some --

MR. DYNNER: I was about to ask those questions.

JUDGE BRENNER: Someone he knows in the industry, but --

MR. DYNNER: Yes.

Q. All right, Dr. Pischinger, if you look on page 327, in the left-hand column entitled reliability --

DR. PISCHINGER: I can see.

Q. -- there is a statement from the author's experience of continuous duty diesel engines, the critical parameters to be carefully watched are -- and then there's a table given, and on the left-hand column citing parameter, if one goes down, one finds in the one, two, three, fourth line from the bottom maximum thrust pressure on piston pounds per square inch.

As you see on the Mirrlees engines, those figures are 35.8 for the KV12 engine and then following are figures for 33.5, 34 and 34.8.

Does this information assist your recollection considering whether or not the side thrust pressure on a piston is a critical parameter?

DR. PISCHINGER: Yes. This helps a lot,

and I am hopefully given the time to explain to you.

In the '50s, '40s, '50s and early '60s, it was usual to use an oil scrap ring down the piston skirt. This was partially traditional, partially due to the inferior oil scrape per technology at this time, and partially was taken from two stroke piston, and this aggravates the lubrication, obviously, the lubrication of the piston skirt; and, therefore, manufacturers of modern engine design switched over beginning with -- let's say 1960 -- well, even a little earlier, to move up this piston ring to the upper part of the piston skirt so that the fuel has undisturbed access to the piston skirt.

And this, of course, improved a lot the lubrication which is the main important factor in the sliding of the piston on the thrust, on the anti-thrust side, and since that time I myself was involved in such developments. The lubrication of the piston skirt was so much improved that side thrust figures today are never given with engines, to my knowledge are never given.

But in addition, I can say just to say if you calculate side thrust, you can calculate for each of today's engines, if you calculate side thrust, you can find a lot of engines which have much more excessive, as you call it, side thrust as a TDI engine, for instance, the famous MANL 32, 36 which has 36 percent higher side thrust than the TDI R48 or the MWMD-50 which has 18 percent higher side thrust.

All these engines are in operation on ships for years and are at least as renowned as the Sulzer and today's Mirrlees engine.

Q. Dr. Pischinger, did you personally calculate -- did you personally calculate the side thrust pressure of the MANL 32-36 piston?

DR. PISCHINGER: Yes.

Q. And what was the figure that you arrived at in pounds per square inch for the side thrust for that piston?

DR. PISCHINGER: I unfortunately have here not an explicit figure, but you can arrive at it if you multiply the figure for the side thrust of the TDI engine, which is mentioned in your -- how do you call it, testimony or --

Q. Dr. Pischinger --

DR. PISCHINGER: By 1.36.

Q. Dr. Pischinger, do you have those calculations that you made to calculate the side thrust of the MAN engine with you?

DR. PISCHINGER: Part of it, but certainly I can make it available to you.

Q. When did you make those calculations, Dr. Pischinger, approximately?

DR. PISCHINGER: Was it two or three weeks ago? Just to address this. When I heard the side thrust was -- in advance of writing this opinion involved in this testimony.

Q. And did you also personally make the calculation as to the side thrust in the MWND-50 engine?

DR. PISCHINGER: Yes.

Q. And you did that about the same time?

DR. PISCHINGER: Yes.

Q. Do you have those calculations with you?

DR. PISCHINGER: Well, of course, I didn't -- I don't have it here. I have the results here, but it is also possible to make it available to you.

Q. Could you briefly tell me the MAN engine that we're speaking of, what is the approximate horsepower per cylinder of that engine?

DR. PISCHINGER: The horsepower per cylinder is -- of this engine is 370 kilowatts.

You have to be aware in Europe we are even diesel engines rating in kilowatts.

Q. And what is the RPM speed of that engine?

DR. PISCHINGER: 750 RPM.

Maybe I should mention another MAN engine which is the L 52 which has about the same side thrust as the TDI engine. This is an engine with five 120 millimeter running at rpm's of 500 or also 514, the maximum rating, and the cylinder -- the power per cylinder is 885 kilowatts.

JUDGE BRENNER: Mr. Dynner, are you finished with the Institute of Marine Engineers testimony?

MR DYNNER: Yes, sir.

JUDGE BRENNER: Do you see any reason to mark it for identification given its limited use?

MR DYNNER: Yes. I would like it marked as Exhibit 70, if we may.

MR. FARLEY: LILCO objects, Your Honor.

JUDGE BRENNER: It's only going to be marked for identification.

MR. FARLEY: I don't think he even laid the proper foundation for it to be marked for identification in view of Dr. Pischinger's testimony.

JUDGE BRENNER: You really don't need

much for foundation, in fact you need very little foundation to mark something for identification. The reason I held off was because of the experience of the other day. It was just a mechanical prerogative on my part. I didn't want to start making things and clutter up the record with exhibits marked for identification if it wasn't going to be used at all. As it turned out it was used in a question. I don't know if it needs to be marked for identification given the fact that the questions and answers combined will give the record a picture of what was there but if he still wants to mark it for identification, I always bend to counsel's view and mark it for identification.

MR. FARLEY: I understand.

JUDGE BRENNER: You also get in same rank.

We'll mark that for identification as Suffolk County Diesel Exhibit 70 and it was previously described by Mr. Dynner, I believe.

In any event, it's the article entitled "The Development of a Highly Rated Medium Speed Diesel Engine of 7,000 to 9,000 Horsepower for Marine Propulsion," and it is taken from the Institute of Marine Engineers Transactions, January 1966, consists of pages from that publication 325 through

347.

Since you were so insistent on marking it for identification, and also asking the question you asked about it, Mr. Dynner, of MBI would like to ask Dr. Pischinger one or two questions about the very part that you pointed him to and also talk about the dangers of excerpts from articles with which people are not familiar.

(The document referred to was marked Suffolk County Diesel Exhibit No. 70 for identification.)

BY JUDGE BRENNER:

Q. Dr. Pischinger, do you know whether this table from which you -- from which Mr. Dynner in his question directed you to the figures for maximum thrust pressure for, I believe the same engine, under different power operation in those four columns whether wet those figures are the actual values that you would derive from the engine while it was operating as opposed to parameters that were warned to assure that they were not exceeded, can you tell from that table?

DR. PISCHINGER: I'm not completely sure if I understand your question. You are referring to what table?

JUDGE BRENNER: It's the ending publication involving the Mirrlees engine. It's the table that Mr. Dynner asked you about. And he directed you to the figures for maximum thrust pressure on the piston, pound per square inch.

DR. PISCHINGER: Yes.

JUDGE BRENNER: I guess my first question is whether it's -- let me change my first question, is it clear to you that that means side thrust pressure to determine maximum thrust pressure?

DR. PISCHINGER: Yes. You are right. This is here entitled maximum thrust pressure on piston pounds per square inch. There is, of course, a problem, always how you define and calculate such a figure.

I only at the moment can guess that that is the figure which is meant in the Suffolk County's testimony; but there is no completely general definition, because the real maximum pressure on the side of the piston can only be calculated if you know the oil distribution, the piston distortion, the oil viscosity, the piston movement, and I think at that time when this was written, nobody could, really could calculate this real maximum pressure

JUDGE BRENNER: All right. My question

was a little simpler, can you even tell whether this is meant to be side thrust?

DR. PISCHINGER: I would have -- it would be necessary for me to study the whole article and maybe to make a request by the authors if they mean the same, which we are discussing.

JUDGE BRENNER: Beyond that, there are four columns here. Do you know whether these are measured values for whatever that table means by maximum thrust pressure on the piston as opposed to something else? For example, could it be a parameter that the authors are warning should not be exceeded at that operation as opposed to maximum as opposed to the actual measured parameter?

DR. PISCHINGER: These parameters -- again, I'd have to say, supposedly, are no limitations. Furthermore, they are certainly in the mind of the authors, I think, far below limitations; otherwise, they wouldn't have put it in a scale -- in a table of the engines whether these are critical parameters.

JUDGE BRENNER: Also, I guess it's kind of a follow-up to the question Judge Morris asked you about, whether the Mirrlees engines were ship engines and also Mr. Dynner's follow-up as to that, as to whether they were also used in stationary applications.

The last sentence of the text before that Table 1, and Mr. Dynner had asked you about reads as follows: "From the author's experience of continuous duty diesel engines, the critical parameters to be carefully watched are..."

If you see the term continuous duty diesel engines, what would that mean to you?

DR. PISCHINGER: Continuous duty engine is an engine which is supposed to be continuous on duty. That means that it's most of the lifetime in motion.

JUDGE BRENNER: Would a ship engine be a continuous duty diesel engine?

DR. PISCHINGER: Yes.

JUDGE BRENNER: I suppose from your answer that a stationery application planned to run most of the time could be so described as a --

DR. PISCHINGER: Yes. A power generator plant which continuously has to deliver electricity, of course, is a on continuous duty.

An emergency diesel engine certainly is not on continuous duty.

JUDGE BRENNER: Okay. I guess the point

of my concern, Mr. Dynner, we have no lack of paper or witnesses before us in this case. And it gets difficult, I didn't want to -- I did want to give you some freedom to conduct the cross-examination the way you see fit but it gets difficult when you have an article, certainly more than two or three pages in there that the witness is not familiar with and then to try to get some useful information based on that article, it's true it's only marked for identification and we're not going to rely on this for anything, but the time has pretty much been wasted with it, I believe, in terms of tying it up, the materiality of the point that you -- that the County just could make with respect to side thrust. Certainly at a minimum, something like this should have been emphasized in reference for support in the County testimony, if you believe that it was an important contradiction of the view of LILCO and its witnesses that excessive side thrust load was of no concern, but let's proceed.

MR. DYNNER: I would just like to ask one follow-up to your list of questions on this document.

Dr. Pischinger, in your experience, if you could briefly take a moment and look down the list of what are termed critical parameters, in

Table 1, could you tell me whether the -- whether those critical parameters would be, in your judgment, be equally applicable to the operation of the diesel engine in a nuclear power plant as they would be for diesel engines that are continuous duty engines?

DR. PISCHINGER: I have to reread it.

MR. DYNNER: Yes. Please take your time and, also, if there's some of them that you don't feel are or some aren't, maybe you can quickly identify them. I don't want to take too long on this but given Judge Brenner's questions may be a significant one.

JUDGE BRENNER: I don't understand the significance at all. Tell me again what you're asking to do.

MR. DYNNER: Yes.

JUDGE BRENNER: I don't want to take time either --

MR. DYNNER: Okay. You asked the question about emphasizing, I think, in your question the fact that the author in his introductory statements said these were critical parameters for continuous duty diesel engines and I want to see if he agrees or disagrees they're also critical parameters for nuclear engines which are operating in nuclear power plants to provide on-site electrical power.

JUDGE BRENNER: You mean the actual numbers?

MR DYNNER: The parameters, that is the identification of the parameters in the far left column.

DR. MC CARTHY: Excuse me --

JUDGE BRENNER: I understand your question. That's not going to be significant in terms of my point. I'll be pleased to tell you the conclusion I've reached right now on this minor point and that is that nothing you've asked Dr. Pischinger from the Suffolk County Exhibit 70 for identification to the extent we're able to get anything intelligent out of it, given the lack of knowledge of most of us as to what's in the article including the witness of whom you were inquiring, that Dr. Pischinger's earlier answers to you with respect to the Sulzer engine in terms of -- that you might want to worry about side thrust load, if you wanted to design an engine that would be good for ship-type applications. It's not inconsistent with the kind of applications that apparently these authors have in mind. And I emphasize it apparently because I certainly haven't read the article, but that one sentence there, introducing that table, and the only reason I looked at that table is because you asked all of us to do so, is discussing continuous duty parameters so I didn't mean to imply that you don't have parameters in a diesel engine, but that you look at for all applications.

DR. PISCHINGER: Judge Brenner --

JUDGE BRENNER: If you still want an answer to the question, we'll try it, but I don't want to --

MR. DYNNER: As long as the record shows what you just clarified, I don't need to spend -- have Dr. Pischinger spend his time going through either one of these.

DR. MC CARTHY: Excuse me --

DR. PISCHINGER: Judge Brenner --

JUDGE BRENNER: Wait a second. With all of this we haven't moved one whatever measurement you want to use, mil, angstrom closer towards getting to the merits of whether we should agree with the County or disagree with LILCO with respect to possible concern for excessive side thrust load. That's what I'd like to get some cross-examination on.

MR. DYNNER: Okay. I'm moving on it.

DR. PISCHINGER: Judge Brenner, may I just, it's important for clarification, because I now, while reading a little more in detail, I see we all have been misled and I think it should be said, because critical parameters are not in Table 1. But the critical parameters which are to be watched at the side thrust is not on it as the parameters one to seven on the next column. The text is going on there, and in Table 1 are only the design and performance characteristics, so this comes from quick reading.

JUDGE BRENNER: Okay. I can understand -- let me try to give you a quick lesson. I can understand why you were anxious to jump in and try to assure that we were not misled by that.

I was not going to be misled, and the reason I was not going to be misled is in my words to Mr. Dynner I intended to let him know and maybe I was too subtle that we're not -- I, at least, as one crutch am not going to rely on this article for anything, given the state of direct examination hereof because of lack of familiarity and because of lack of connection to get directly to the merits of the issue. I'm sure there will be a whole bunch of these articles I could misunderstand if I sat back at this hearing and read it but it is not admitted into evidence and one of the important reasons why it's not admitted into evidence is we don't have evidenciary foundation, as your counsel has been quick to point out and properly so, at least with respect to this article.

DR. PISCHINGER: Thank you, Judge.

JUDGE BRENNER: When something is marked for identification, it doesn't mean it's in evidence. Mr. Dynner.

Q. Yes. Dr. Pischinger, in your judgment, does the AE piston skirt have proper lubrication incorporated in its design?

DR. PISCHINGER: Yes. In my opinion, in consideration of the design, and although the -- watching the results of performance, I conclude that the lubrication of the piston skirt of the R45 is in order.

Also, as can be seen from the drawing, the piston rings lubrication, piston rings oil scrap rings are up on this skirt and not one down, as has been in old designs.

Q. Is there a different number --

JUDGE BRENNER: I'm sorry. I didn't hear

you at the end. You said and not down on the skirt --

DR. PISCHINGER: Not down on the skirt as in old designs.

Q. Is there a difference, Dr. Pischinger, in the number of oil leak holes in the design of the AE piston as opposed to the design of the AF piston?

DR. PISCHINGER: I didn't count the leak holes. I just saw that the cross section is sufficient to give enough back flow to the oil, but I do not have the figure with me now.

Q. Dr. Pischinger, you earlier referred to two engines. The MAN and the MWN engine that you said that you had calculated the side thrust on.

Before I forget, I just wanted to ask you two quick questions on the MWN engine.

What is the horsepower per cylinder of that engine?

DR. PISCHINGER: Maximum horsepower, 370 kilowatt.

Q. That's the same as the MAN?

DR. PISCHINGER: It's practically the same.

Q. And what is the speed in rpm's of that nature?

DR. PISCHINGER: Ranging from 600 to for

special application, 750.

Q. Thank you.

Now, Dr. Pischinger, you testified, I think, that the MANL 32-36 engine had side thrust load 36 percent higher than that calculated for the DeLaval AE piston skirt.

Did you make an independent calculation of the side thrust of the AE piston skirt?

DR. PISCHINGER: Yes.

Q. And did your calculation agree with the calculation made by Professor Christiansen, which is set forth in the County's direct testimony?

JUDGE BRENNER: Can you point the doctor to the page reference, if you can.

Also this is the record of Dr. Pischinger.

MR. DYNNER: That would be Suffolk County Diesel Exhibit 18.

DR. PISCHINGER: Maybe you could help me a little, where in this text side thrusts final result is defined, to speed up the situation?

MR DYNNER: Yes. If you turn to the last page in Exhibit 18, you'll see a computer readout, be on the far left, the first grouping of figures under the column one, you'll see the figure 123.44.

DR. PISCHINGER: I am still on the --

seeking -- well, I, again, have to ask, what is the unit of this -- of these figures?

MR DYNNER: It's explained on page 48 to 49 of the County's direct testimony, Dr. Pischinger.

It is 123 pounds per square inch.

JUDGE MORRIS: Excuse me, Mr. Dynner, I wasn't sure what your question was, whether he agreed with the result or whether he agreed with the methodology or perhaps both.

Q. My question was, did he agree with the results of the calculation of Professor Christianson on the side thrust.

If you look on page 49 of the Suffolk County testimony, it says the calculated mean units side thrust of the AE Piston is 123 psi, exceeded the upper value by 44 percent.

There's a reference on the previous Page 2 to Exhibit 18. Exhibit 18 contains the calculation. The last page of Exhibit 18 shows in the last column, that is, in the first paragraph of numbers, in the first column, number one, you go down to the last number. It says 123.44 unit thrust -- unital thrust, so that's his conclusion.

DR. PISCHINGER: I -- yes. I have to say that my calculation gives a lower side thrust than this which is given in the -- in your reference.

The calculated side thrust with TDI is 78.7 psi. The calculation is done by computer program using the gas pressure, using the gas pressure and the mass forces in the point of maximum side thrust. The mass forces counteract the gas forces, and the maximum value which is calculated as 78.7 psi, which means I calculated it in bar, of course, 5.42 bars, which is, by the way, less than the limit of 85 psi which you give in your testimony as a standard design value.

Q. Yes. So your figure of 78.7 is the maximum unital side thrust; is that correct?

DR. PISCHINGER: Yes, yes.

Q. So the numbers that you gave for the MAN engine is 36 percent higher. Was that 36 percent higher than your number of 78.7?

DR. PISCHINGER: Yes, it is, yes.

Q. And it's not 36 percent higher than Professor Christiansen's number of 123 psi; isn't that true?

DR. PISCHINGER: Yes.

Q. And the same is true with the number you gave for the MWN engine, that would be 18 percent higher than your calculated side thrust of 78.7; is that correct?

DR. PISCHINGER: Yes.

MR DYNNER: I would like in view of this testimony, if possible, to get a copy of your calculations which show this difference of opinion with Professor Christiansen's calculations, if that's possible.

If you could state the methodology of it.

JUDGE BRENNER: Why don't you ask him how he did it. He's here to give testimony.

BY MR. DYNNER:

Q. What methodology did you use in calculating the maximum side thrust on the AE piston?

DR. PISCHINGER: The method is to use the gas pressure diagram versus crank angle to calculate the mass forces out of the acceleration of the piston to combine these two forces, and then to calculate the side site force for each crank angle or certain distances of crank angles which can be done geometrically by the angle of the connecting rod for each position, and to use this force to calculate a pressure and a nominal pressure which is related to the projected area of the piston skirt, the dimensions of the piston skirt are known and by the side thrust. It's a unital pressure that can be calculated.

Q. Dr. Pischinger, in looking at Professor Christiansen's calculations in Exhibit 18, if you could take -- I think it might be worthwhile in this case, if you could take a minute or two and perhaps tell me whether you have, in your calculations, any significant disagreement with the figures that Professor Christiansen sets forth in his calculations.

Dr. Harris, did you want to add something?

DR. HARRIS: If I'm permitted, I'd like to confer with my fellow panel member.

JUDGE BRENNER: Let me see if I understand what you want to do, Mr. Dynner. In effect you want them to critique with the proposed County Exhibit 18 which are the calculations by you to see if they agree or, more to the point, where they might disagree with figures used and the methodology.

MR. DYNNER: I wouldn't say critique was the word. What I was looking for was to see whether there was any -- since these are not -- whether there was any obvious or significant error or difference in the figures which would result in such a significant difference in the result. I don't know how long it will take. If it will take too long, we obviously can't sit here and use up hearing time to do it but if there's something that jumped out at him that could account for the difference in result, I'd like that to be identified.

JUDGE BRENNER: We can try that, if there's nothing that jumps out at them, we could let you come back to just that one point and then take a break to give them time to look at it although I don't know if a break is sufficient time to do that.

How much time would you need? I don't know if they're prepared for something like this already or not.

Dr. Pischinger, if you -- or in conjunction with the people on the panel, if you go through these calculations and the County Exhibit 18 the point is if we give you a little of time you can tell us if you disagree with what was done. We know you disagree with the result but whether you can identify why you do.

DR. PISCHINGER: Yes. Of course, I can try to follow up. I didn't do it until now, to follow up the way which this was done and -- but I cannot say how long it was. It's a question how obvious. There must be a disagreement how obvious it is and how long it takes to find.

JUDGE BRENNER: All right. Let's pass it for now and come back to the -- and we'll come back to that.

Q. Dr. Pischinger, we'll move on to something else and then perhaps during the break you'll be able to examine that.

BY MR. DYNNER:

Q. Dr. Pischinger, in your judgment, would proper lubrication be capable of handling any side thrust load regardless of its magnitude?

DR. PISCHINGER: If this question is a general engineering question, I say no.

If this question is related to standard design, diesel engines, I say yes.

Q. Well, if the side thrust in the AE piston turned out to be a 123 psi instead of 78 psi, in your judgment would the lubrication in the AE piston skirt be adequate to eliminate any possible adverse results from that side thrust load of 123 psi?

A. No.

DR. PISCHINGER: No concern absolutely.

JUDGE BRENNER: I guess someone will remember to ask Professor Christiansen the same question about Dr. Pischinger's figure, just in case we can't put the two together later. I will if nobody else does which is why I like to try a subject together with all party witnesses but we can't always do what we'd like to do in this life. Go ahead.

Q. Dr. Pischinger, if you will please turn to -- if you're not already there, to page 59, in your question, answer to question 93, if, in fact, a temperature distorted the piston skirt so that it rubbed on the liner, wouldn't the friction of the skirt rubbing on the liner destroy the lubrication at that point?

DR. PISCHINGER: The piston in this operation cannot distort to such an amount, so I find your question theoretical.

Q. All right. Well, let me be more precise.

The first sentence of your answer is that with an adequately lubricated piston, side thrust will not create a dramatic temperature differential.

Isn't it true that given enough side thrust pressure there will be a significant temperature differential from one portion of the skirt to another?

DR. PISCHINGER: The friction work of a piston is so low, so small compared with all the other thermal effects that there will be no increased temperature. This has been measured in numerous pistons -- piston designs -- also, of similar size, and; therefore, again, I cannot see where this distortion should come from if the temperature distribution is quite uniform.

Q. All right. I'm trying to get at the point, Dr. Pischinger, in your answer where you say that an adequately lubricated piston side thrust will not create a dramatic temperature differential.

Is that statement true even if the side thrust is 400 pounds per square inch?

JUDGE BRENNER: Let me interrupt, Dr. Pischinger.

Mr. Dynner, within reason, it's your time, but you may have exceeded reason with that question.

How is that ever going to be material, even taking the highest number believed to be accurate by your witness? We're not talking about side thrust --

MR. DYNNER: Obviously what we're trying to get to is there is a relationship between side thrust and the effects of side thrust which are not always handled by adequate lubrication, and I think that the witness testified -- I asked him the question specifically as to generally, now I'm giving an illustration.

JUDGE BRENNER: It's just never going to be material in my view. The right question to ask him in that area you already asked, what if the pressures turn out out to be as high as Professor Christiansen believes, and you asked him that question. You got the answer. And then you asked the general question and he gave you the answer as to why he thought that was theoretical or words to that effect. And it's just not going to be --

MR. DYNNER: It's two entirely different answers. If his answer to the theoretical question is one which indicates that he does not have a grasp of the technical underpinnings of how side thrust affects it, then I think that that is a significant issue to bring out in cross-examination.

If he's saying -- even if you had a thousand pounds of side thrust, it still wouldn't affect the temperature of this skirt, that might be something this Board would be interested in. I don't know. Why would I be interested in it?

MR DYNNER: I think it would be --

JUDGE BRENNER: Are you going to put in evidence that the side thrust at the Shoreham pistons is a thousand pounds per square inch?

MR DYNNER: No.

JUDGE BRENNER: So why would I be interested in it? I want to be interested in things that are interesting.

MR. DYNNER: All right. We have testimony on the record which states that there is a relationship and that the side thrust at a certain point will create a temperature differential which will create the distortion, which will create the rubbing on the side which will create a destruction of lubrication which will lead to piston seizure. That's on the record and if Professor Pischinger is testifying that isn't the case at all from a technical and scientific matter that it can happen, I believe that's useful testimony on cross-examination.

We have other expert witnesses on the Staff and others that -- and I should think that the Board might be interested in whether everybody disagrees with that approach.

But I will move on if the Board doesn't think that's a pertinent question.

JUDGE BRENNER: I think you're way out there in terms of anything that's going to be important to the findings.

I understand how part of your argument would relate to a view of Dr. Pischinger's expertise, although I thought we were past that on Monday. Nevertheless, certainly the content of the substantive questions and answers can in turn be related back to some of the prerequisites for believing testimony by his expertise and so on, but -- and then I understand, like a good lawyer, you've made an argument that shows there is some arguable relevance. I'll give you that much.

The scale of what's relevant, way off in the distance of the solar system of relevance, well, you understand the difference in degrees between things that are remotely relevant and when you're on a time limit, a time limit I thought was reasonable if you had stayed with that which I thought was important and productive.

MR. DYNNER: I'll move on.

JUDGE BRENNER: Give me just one moment.

Q. I've made my point.

Judge Morris on his own saw that you were going to make the argument you made. He doesn't disagree with my comment as to its importance to the record, but it's your privilege to put it in the record if you want to, and I'll let you proceed with it and then we will change our mind when we look at it later.

Q. All right, Dr. Pischinger, taking your statement, the first line of your answer to question 93, would it make any difference to your conclusion if the side thrust were 400 psi or a thousand psi?

DR. PISCHINGER: In an internal combustion engine, the side thrust cannot exceed a certain percentage of the gas pressure, and, therefore, there are natural limitations to side thrust, so, again, I have to say this question is totally hypothetical.

If you want to build an engine which with ever increasing side thrust, then this engine could not run because of too short connecting rod or the piston would be no longer a piston but a disc, Within the design, within today's design limitations of a diesel engine, I am well aware of it, the side load, the side thrust which is within this limitation will not create any dramatic effect given proper lubrication, but maybe I could make you available some of my -- I shouldn't say that.

DR. SWANGER: As a co-sponsor to answer number 93, I do have some things that I would like to add --

JUDGE BRENNER: Even as a co-sponsor you still have to direct your comments to answer the question.

DR. SWANGER: Yes. Question was would -- as I recall, and Mr. Dynner can correct me, if the side thrust were 400 or a thousand psi, would that make a difference in the analysis.

MR DYNNER: In the first sentence of your answer, Dr. Swanger, that with an adequately lubricated piston side thrust will not create a dramatic temperature differential.

DR. SWANGER: Following up the answer that Professor Pischinger gave that there are natural limits to the side thrust, taking into account the geometry of the components in the TDI engine, the natural limit is that the side thrust cannot exceed 22 percent of the gas pressure.

22 percent of the maximum gas pressure of one thousand -- of one 1670 psi times the 227 square inches of piston.

DR. MC CARTHY: Which parenthetically don't occur at the same time, so they would be indeed the extreme end and completely unobtainable in conjunction with the real engine.

DR. SWANGER: When we do that, we get a net -- an upper limit of side thrust in pounds of 81 thousand pounds of force, which is about four times the number that Professor Christiansen computed.

FaAA did do independent computations of the side thrust taking into account the gas pressure, and we agree with Professor Christiansen that the true maximum side thrust total in pounds in the TDI engine is about 22 thousand pounds.

Dividing that number by just the projected area of the skirt, which is 17 inches in diameter, times 18 and a half inches in height or 314.5 square inches, dividing that number into 22 thousand pounds gives us a unit side thrust of 71 pounds per square inch.

Our number agrees very closely with Professor Pischinger's number, and since it uses a side thrust total load at the same magnitude that Professor Christiansen used, we feel that the error in Professor Christiansen's calculations must have been in the calculation of the projected area of the skirt or the use of that area in his further computation?

JUDGE BRENNER: You can look at that some more during the break.

MR. DYNNER: Gentlemen, earlier we talked about the -- a bit about FaAA's current work on the circumferential rib as it relates to side thrust, and I recall that we had decided that the appropriate time to ask you without that would be during the discussion of the excessive side thrust.

Would you briefly describe the work that you're doing on the circumferential rib of the AE piston at FaAA as it relates to side thrust issues.

DR. HARRIS: In recent effort -- in recent days, Failure Analysis Associates has been involved in some additional experimentation regarding the strain levels in the ribs of AE skirts. This was done under top dead center loading.

We've also done some finite element calculations, stress in the ribs in the AE skirt under side loading.

We have found that the stresses in the circumferential rib between the wrist pin boss in the AE skirt are lower when a maximum side load of 22,500 pounds is applied, and the corresponding stresses at top dead center when the full pressure loading is applied. This leads to the conclusion the maximum stresses that are -- that the AE circumferential rib is subjected to and controlled by the peak firing pressures rather than any side thrust and is further evidence of the lack of influence of side thrust on cyclic stresses in the AE skirt.

We're now convinced that this is also a conclusion that can be applied to the ribs themselves.

Additionally, some recent inspections have been performed on the circumferential rib, the AF skirts that were removed from the diesel engines at Shoreham. As far as I can recall, I have been informed that there were no relevant indications in the region of any of the AF skirts that were inspected at Shoreham.

JUDGE BRENNER: Excuse me, there was a term that I didn't understand. You said this was evidence of a lack of influence of side thrust on some type of stresses.

DR. HARRIS: Cyclic stresses.

JUDGE BRENNER: Cyclic, all right.

MR. DYNNER: The cyclic stress you're referring to is the stress with which might cause cracks to initiate in the boss area that you had studied earlier; isn't that true?

DR. HARRIS: No. My answer was addressed

to the ribs themselves, which would have to do with the crack initiation in the ribs.

Q. But the County has not alleged that there's cracking in the ribs, have they?

DR. HARRIS: Not that I am aware of. However, you were asking about our studies in regards to AE ribs.

JUDGE BRENNER: Again, Mr. Dynner, I was going to jump in, but I didn't want to.

MR. DYNNER: It took only a second.

JUDGE BRENNER: Not on that question. I was going to to jump in when you asked your opening question. It was a very broad question, and then you wanted to know why he gave you the answer given that question.

I think you could have been more precise when you started to ask about the work on the circumferential rib. Just take that into account in your future questions.

BY MR. DYNNER:

Q. Gentlemen, Dr. Pischinger, and Dr. Swanger, turning for a moment to page 60 of your testimony, you point out in that testimony that modern materials have a higher tensile strength than those which may have been available in the source authority that was listed in the County's testimony.

What would be the effect of materials in the piston skirt having a higher tensile strength on the issue of side thrust leading to distortion of piston skirt, if any?

DR. SWANGER: The change in materials going from cast iron with the modules of elasticity of about 15 million pounds per square inch to nodular iron with a module elasticity of approximately 23 to 24 million pounds per square inch would by itself increase the stiffness of the piston and its resistance to distortion.

Q. Just to clarify, Dr. Swanger, I was speaking about thermal distortion.

Would your answer be the same with respect to the effects of thermal distortion?

DR. PISCHINGER: Higher strength material gives the possibility to use minor -- or smaller dimensions for the walls of any part of the piston skirt, and smaller dimensions mean lower temperature differences as is well-known.

The piston is cooled from the inside with splashed oil. The piston is cooled from the outside with -- by contact with a cooled cylinder liner of the oil film, and the thinner the wall the more --

the less temperature differences between the inside of the piston and the outside of the piston skirt, but, of course, this temperature difference is in this design of piston, this two part design was -- with oil cooling of the crown and which by that preventing of a lot of in-flow from the combustion chamber into the crown for such piston temperature difference as has been already mentioned yesterday, I think, is very small in the skirt, and with higher strength materials this can even be further degree in proof.

MR. DYNNER: Dr. Swanger, I'm sorry, thank you, Dr. Pischinger. My question was directed to your testimony, is it true -- your testimony was true with respect to thermal distortion.

DR. SWANGER: It would be true for thermal distortion for the reasons that Professor Pischinger gave.

MR. DYNNER: Thank you.

(There is a discussion off the record)

JUDGE BRENNER: We're going to go back on the record now.

BY MR. DYNNER:

Q. Dr. Pischinger, let me go back just for one second.

The two engine skirts and the engines you referred to, the MANL 32-36 engine; what is that piston skirt made out of?

DR. PISCHINGER: If I recall right, it is aluminum alloy.

Q. And the MWND-5, is that aluminum also?

DR. PISCHINGER: Aluminum piston skirt. It has no affect on the calculation, of course.

DR. SWANGER: Judge Brenner, the one minute break we had was enough to absolutely resolve the difference between our calculations and Professor Christiansen's calculations, if you would like to know the origin of discrepancy.

JUDGE BRENNER: Is that okay with you, Mr. Dynner?

MR DYNNER: I'd be delighted to know it.

DR. SWANGER: Referring to County Exhibit 18, on the first page of that exhibit, point number five, it is written: "Effective thrust area on skirt equals skirt height times cylinder bore divided by the square root of two."

In my experience, eight years with the engine parts division of Imperial Clevite Manufacturing sleeve bearings for such engines, I have dealt extensively with the concept of unital pressures, and the standard definition for unital pressures would be strictly a height times a diameter. The factor of square root of two is non-standard in this type of calculation.

FaAA's calculations, Professor Pischinger's calculations are done with the standard definition of unital pressures, not with what we feel is the non-standard definition in the County's exhibit. That factor accounts for the difference in the numbers reported.

DR. PISCHINGER: May I add that figures given for the other comparable engines or engines in comparison are, of course, all calculated according to the same definition. That means still are comparable, that means that steel engine with at least 36 higher side thrusts are in operation.

JUDGE BRENNER: So it would be -- I want to make sure that we've got all the terms equated, Dr. Swanger, what you had calculated would be multiplied. The way you would caluclate it would be to multiply the skirt height times the cylinder bore.

DR. SWANGER: Yes. That would give you the projected area of the skirt. Its diameter times its height and that is the standard technique for computing unital pressures.

BY MR. DYNNER:

Q. Is there some textural authority that you can give us for your stating is the standard technique?

DR. SWANGER: Since we just discovered this discrepancy in a one minute break, I can't give you the textural reference now, but I'm sure if you look in ASME Wear Control Handbook, for instance, in --

JUDGE BRENNER: I'm sorry, what was the title?

DR. SWANGER: ASME, American Society of Mechanical Engineers. Also --

JUDGE BRENNER: Yes. I knew that much. But we're --

DR. SWANGER: Wear Control Handbook. Also publications in the Society of Automotive Engineers Literature about calculation of journal bearing pressures would all use the same unit pressure definition.

JUDGE BRENNER: Do you want to -- did you include Dr. Pischinger in your question, too, as to whether he had a textural reference?

MR DYNNER: Yes. Do you have a textural reference -- I've included any one of them.

DR. PISCHINGER: So numerous that I have

really difficulty now in naming them. Of course, it's German literature, but there are a lot of textbooks which the unital pressure on -- bearing whatsoever slightly on the surface is the projected -- the force divided by the projected area.

Q. Dr. Swanger, doesn't your statement assume that the bearing -- that the journal sits in a bearing and contacts the surface over 180 degrees?

DR. SWANGER: As Professor Pischinger had said, knowing the exact distribution of pressure around the skirt, it's a very difficult problem in elastohydrodynamics to solve; therefore, for rule of thumb or design guideline calculations such as this, simplifications have to be used.

The one that is in use in the diesel engine industry is to divide unital -- is to define unital pressures as projected areas, diameters times lengths with no other correction factor.

MR. DYNNER: Judge Brenner, if you're ready, we have --

JUDGE BRENNER: I'm ready.

MR. DYNNER: We have unfortunately only a limited number of photographs. We have two copies of the photograph, we tried to get extra copies, but, unfortunately, the photographic store has not cooperated and I don't know where the extra copies are now but they will be furnished later.

We would like to distribute at least one copy, show the copies to counsel and the staff, and have a chance to ask the witnesses a couple of questions.

The photographs in question, just for the Board's information are in the first place a photograph taken by Mr. Bacchi of Ocean Fleets and during the June 1984 inspection of the AE cylinder -- of the AE piston skirts at Shoreham. It is a picture of the scuffing on EDG 103 piston skirt which is referred to in the County's direct testimony.

And the other photograph is of a piston removed from EDG 103 during the same time, which shows the scoring that is alluded to.

We do not have these photographs available in time to attach them to the direct testimony, but I think that they would aid and assist the Board in understanding that testimony as well as in eliciting some cross-examination testimony from this panel about what these marks mean.

JUDGE BRENNER: All right. Well, I'll

let you get them marked for identification, we're going to have to work out the logistics of looking at it here, and we'll let you go with it subject to your representation that you're going to be able to tie them up as being what you represented them to be, presumably, through your witnesses.

When they get on the stand it may be you don't have to do that because it may be these witnesses can do that for you, but if they can't, you'll have to remember to close the loop in order for us to use any of this.

We have only two copies.

MR. DYNNER: I'm sorry.

JUDGE BRENNER: Why don't you let us look at it for a moment and then we'll give it back to you, somebody else.

You have two photos.

MR. DYNNER: Yes. I want to give you the other one, we're going to try two keep these separate, we're going to mark them for identification. It might be less confusing to do them one at a time.

JUDGE BRENNER: Can you get it for the reporter tomorrow, the additional copies?

MR DYNNER: We've got people trying to

bring them up by car somewhere. We had a foul-up. See if you can get them as soon as possible so we can use them in as close in time when we use it.

MR. DYNNER: The first -- I have to read it, Joe.

JUDGE BRENNER: I'll do it. It refers -- the first photo, Suffolk County Exhibit 71 for identification has been labeled by the County, a photo of a piston removed from EDG 103 taken by Anesh Bakshi at June 1984 at SNPS.

JUDGE BRENNER: Unfortunately, as Judge Morris points out, you've entitled both photographs, we didn't get to the second one yet, exactly the same. Can we call the first one something and the second one something else?

What was the point of the first one, again? One was scoring and one was scuffing.

MR. DYNNER: Yes. I think the first one is scuffing. Mr. Brigati will help you out there.

JUDGE BRENNER: Add to the title scuffing after SNPS.

(Exhibit Diesel-71 is marked for identification.)

JUDGE BRENNER: I hope we can get all the important descriptive things, both the question and the answer in words on the transcript rather than reli ·c· on the photos and follow it as we're doing it now since we don't have a photo in front of us, although we have looked at the photo prior to the question.

Whenever you're ready, Mr. Dynner.

Thank you.

BY MR. DYNNER:

Q. Gentlemen, have you had a chance to examine the photograph of the County Diesel Exhibit 71 which purports to show scuffing of a piston skirt, you any of you seen an AE piston skirt removed from EDG 103 that had this appearance which, if I can desc·ibe it vaguely, is an U-shaped pattern within the sort of V or U-shaped pattern, there is a darker color material than what appears to be the lighter colored material in the other portions of the skirt.

Do any of you recognize this piston skirt?

JUDGE BRENNER: Mr. Brigati, while they're doing that, you can show this -- you can look at show it to the staff.

So that the question is -- reminds me of some other cases that I've been at but the question is have you ever seen this piston?

DR. PISCHINGER: Yes. That was the

question.

JUDGE BRENNER: All right.

One entrance remark, the picture is out of focus which makes it very difficult to say anything definite.

JUDGE BRENNER: That was my -- it was my opinion that the picture appeared to be out of focus also. As to the second part, you'll have to testify to that, whether you can tell anything or not.

MR. YOUNGLING: Judge Brenner, none of us have looked at the pistons in the 103 engine after they were removed --

DR. PISCHINGER: Of the present.

MR. YOUNGLING: After they were removed to replace the engine block; however, the pistons were all inspected at part of the DRQR program during that repair process and found to be acceptable.

Perhaps Dr. Pischinger could comment on the photo, if he feels he can.

JUDGE MORRIS: Before he does, you said they were inspected, Mr. Youngling, by whom?

MR. SEAMAN: The pistons were inspected by DRQR personnel, quality control personnel associated with the Owners Group.

JUDGE BRENNER: Well, some of them may or may not have been local people. DRQR is a conglomerate or conglomeration. Did any LILCO people look at it?

MR. YOUNGLING: Yes. Test Engineers in the start-up organization who were supervising rebuilding the engine looked at them, in addition there were TDI personnel on site supervising the rebuilding effort who also looked at them.

JUDGE BRENNER: I wasn't very clear, I'm sorry. I understand that anybody would have used them in the work may have seen them. I meant people looking at them for the purposes of inspecting it, were there any LILCO personnel --

MR. YOUNGLING: Standard practice when you take an engine apart as that engine.

JUDGE BRENNER: Dr. Pischinger, you were invited by one of your colleagues on the panel to comment further, if you thought you could.

DR. PISCHINGER: Yes. Of course. I am willing to say something, but, of course, with very much precaution of the out-of-focus picture.

May I say this, it reminds me, I know an expert who had been shown a cross-section of a sausage as a metallographic structure and he identified it as nodular cast iron.

JUDGE BRENNER: Your point is well taken as stated. Stated by you much better than my poor attempt to remind Mr. Dynner that he was having to tie up the pedigree of that photograph with his own witnesses if the witnesses here could not and he'll keep that in mind.

We'll also not try to qualify his expert witness as an expert.

DR. PISCHINGER: But, of course, I will do my best with all precautions.

You have to be aware, it looks to me as if the tin plating of this piston partially is worn in the black or dark colored area.

This would be not so unusual, because with all this tin plated pistons, the tin plate is used for breaking purposes, during breaking to give further safety, and it usually wears after short or longer time depending on the thickness of the layer, and the breaking condition obtained; so the only thing I can say, there is a difference in color, one color, let's say, silver shining in the photograph seems to be thin layer which is still there or at least partial layer and the darker colored area seems to be a thin layer which is already worn that the normal way.

As far as 1 can see by this focus, there is no sign of scuffing nor scoring.

Q. When you testified, Dr. Pischinger, it would be worn in the normal way, worn after how many hours of operation, that certainly would be a consideration, wouldn't it?

DR. PISCHINGER: Certainly. It depends upon the -- again, as I say, on the thickness of the thin layer of the way of operation of the hours, of the lubrication oil, a lot of influences are there.

It is not unusual after severe operation -- severe I mean parts of over load, that means higher than the usual one hundred percent load that thin layer is worn out earlier than if you have continuous operation at lower load, below the highest load.

Q. Can anyone on the panel tell me how many hours the EDG 103 ran with the new AE pistons in them before the engine was disassembled for the inspection in June?

JUDGE BRENNER: That's the question. Just before that, I want to make sure I'm hearing you correctly, Dr. Pischinger.

Are you saying tin layer rather than thin layer?

DR. PISCHINGER: Tin, the metal tin.

JUDGE BRENNER: That's what it sounded like. I just wanted to make sure. Not a thin but a tin. Maybe a thin tin layer.

JUDGE BRENNER: Mr. Dynner's other question is pending.

MR. YOUNGLING: Diesel engine 103, operated approximately 530 hours prior to the time that the pistons were removed in June, and I should also point out that after the block rebuild, these pistons were put back in the engine and have operated successfully for an additional 250 hours approximately.

Q. Mr. Youngling, of the 530 hours, you say the AE pistons were run prior to this inspection in EDG 103. Can you tell me how many of those hours were at or above a hundred percent load?

MR. YOUNGLING: Between 125 hours.

Q. Dr. Pischinger, in your experience, would you think that the type of markings shown in this photograph on this piston skirt would be expected on a skirt that was run for about 530 hours, 100 to 125 hours of which were at full or above full load of 3,500 kw?

DR. PISCHINGER: That's not unusual.

May I, perhaps for understanding it, those engine people who are only familiar with coated pistons of other types where the coating is of a dark color, they are usually surprised by the pictures because if this dark colored layer wears, then there is no big difference in color, and it doesn't look so interesting.

In this case, you have the difference of the colors of thin and the cast iron surface which is the first moment gives you an opinion of what have we here, but all source of wear, within several hundred hours of operation part of it being full load.

Q. Gentlemen, you have responded in part on page 61 in your answers 96 and 97 to the County's direct testimony about their interpretation of the scuffing on this particular piston. I just want to be sure that I understood. Is it correct that none of you gentlemen individually inspected the scuffed piston to which the County has referred and which this photograph represents, as will be established later on?

MR. YOUNGLING: As Dr. Pischinger has testified, this not a scuffed piston, and, yes, you're accurate that we have testified that we did not personally inspect them in June.

Q. All right. Did you inspect this -- any of the pistons which, according to your own exhibit 32 -- if you'll turn to Exhibit P-32 for a moment, you will see the last seven pages of that cover inspection reports on EDG 103 and relate to what is called here scuffing of pistons five, seven and eight on EDG 103 during inspections that took place apparently in March of 1984.

Now, were these piston skirts that are identified as five, seven and eight that were inspected and then dealt with in the -- these DRQR reports or reports that made part of the DRQR report, would any of these -- the piston skirts that were made available from the disassembled EDG 103 for the County to inspect in June of 1984?

JUDGE BRENNER: While they're considering their answer, it's 3:30, we're going to break after this answer. I hope time we have set for you is over.

What did you have left to get to?

MR DYNNER: The few quick questions on the other photograph and I suppose maybe twenty minutes on the balance of the cross-examination plan.

JUDGE BRENNER: For the life of me, I don't understand with why you decided to waste your first half hour this afternoon on looking at those articles.

There are other engines that the witnesses -- the articles at least with which the witnesses were not familiar and then wait until your time is up to ask particular questions about the Shoreham pistons. That's the problem you have.

MR. DYNNER: Judge Brenner, we're going to forget this other picture.

JUDGE BRENNER: They're still preparing the answer. This may be a good time to break. We'll give you fifteen more minutes after the break. You can include the other picture or not. It's up to you. It's going to be your fifteen minutes.

MR. DYNNER: I think I've heard this before in other contexts and I will do my best to try to speed things up as much as I can.

JUDGE BRENNER: I think you may know it's uncharacteristic of us to -- I don't think you've merited the other fifteen minutes. I want you to know that in the past there for one or two times where we gave you additional time where we thought it was merited. I don't think it was merited here because while you may still have a few things to get to we'd like to hear, you went through other things first which weren't necessary to go through to get there.

I also took some of your time in my questions of that article, just to, I thought, emphasize to you the problems with using articles like that when witnesses are not familiar with them, and I take that into account, too. I probably could have refrained from doing that, protected your time for that ten minutes. We'll come back at 3:50.

(Recess)

JUDGE BRENNER: All right.

Mr. Dynner, you may complete your cross-examination now. It's 3:50. Keep an eye on the clock and come to a logical conclusion before five.

BY MR. DYNNER:

Q. Yes. There was a pending question, gentlemen, as you'll recall, whether any of the piston skirts that are shown on Engine 103 in Exhibit P-32 are the same as the skirt with these marks which was inspected in June by the County. Do you have an answer to that question?

MR. YOUNGLING: Yes, Mr. Dynner.

As part of DRQR program, TER number Q-500, a visual inspection of all eight of the pistons was performed during the block replacements outage. That would have included this piston.

The results of that inspection showed, and I'm reading from the document, no unusual scuffing or scratching was observed on the outboard portions of the piston and piston skirts.

Q. Is that document in the group of documents in Exhibit-32?

MR. YOUNGLING: Yes, it is. You should go from the back of the exhibit, and if you go forward, one, two pages.

Q. Mr. Youngling, can you identify for me who made the decision that this -- that the markings on these -- on the pistons shown in the photograph that we've given you to look at has no unusual scuffing or scratching on it?

MR. YOUNGLING: These inspections were performed by members of the DRQR program in the quality assurance arm of that program.

MR. FARLEY: Mr. Dynner, I think there's a failure of communication. Either you're waiting for an answer or the panel is waiting for a question.

MR. DYNNER: I'm sorry. I was waiting

for you to identify the individuals who made this decision.

MR. YOUNGLING: I'm sorry, Mr. Dynner.

MR. DYNNER: Thank you, Mr. Farley.

MR. YOUNGLING: If you look at the exhibit on the bottom, there are sign-offs of the signature of the individual who prepared the inspection report, and a review by signature.

These were both Stone Webster employees working in the DRQR effort.

Q. So it's your testimony, Mr. Youngling, that the subject piston shown in the photograph is, in fact, not covered by one of the inspection reports in the earlier pages which referred to pistons five, seven and eight; is that correct?

MR. YOUNGLING: No, that isn't correct.

Since I don't know what piston that is, I can't make a determination as to whether it was end of number five, seven or eight cylinder.

JUDGE BRENNER: So the sign-offs here that you're talking about are the ones on the second to the last page.

MR. YOUNGLING: Yes, judge.

JUDGE BRENNER: Sign-offs on the last page of Exhibit 2, is that part of the same inspection report?

MR. YOUNGLING: Yes, it is. It's a follow on document to take care of the second paragraph of the -- the concerns in the second paragraph of the previous page dealing with the carbon problem.

JUDGE BRENNER: Those sign-offs have LILCO people as well as S & W people according to the printed boxes. I don't know if the signatures are --

MR. YOUNGLING: Yes, there are Stone Webster people on there. There are start-up personnel, and there are LILCO quality assurance personnel.

If you'd like, I can identify the LILCO employees.

JUDGE BRENNER: I don't need it. It's up to Mr. Dynner.

Q. Mr. Youngling, I just want to be sure that I understand your testimony, because in the first couple of pages on EDG 103, there is a document called Q 159 followed by LDR number 2198 that do identify scuffing on piston skirts for numbers five, seven and eight pistons, and attached to that, on the back, there is a document that appears to be from DeLaval dealing with those pistons, number five, seven and eight, that appear to have scuffing. I just want to be sure that you're certain, if you can be, that the piston skirt shown in the photograph is not, in fact, either five, seven or eight.

MR. YOUNGLING: Mr. Dynner, I am certain what I testified to.

TER-159 was an inspection performed after the pre-operational program, but the approximately 100 hours on the pistons at greater than or equal to 3,500 kw.

At that time the engine was taken down and disassembled.

Three of the pistons were looked at under the program, TERQ 159 was generated.

Then in May -- I'm sorry, in April of 1984 -- as part of the repair of the engine to replace the engine block, we dismantled the engine again and pulled out all eight of the pistons and performed the inspection covered by TERQ 500.

Now, since I don't know what piston this is, I cannot identify whether it is five, seven or eight. So I can't tell if it relates to the TER 159.

MR. FARLEY: And this, for the record, is

County Exhibit 71 for identification.

Q. Gentlemen, if you turn for a minute to page 69, if the answer to question 110, it stated that in 1983 the Shoreham EDG's had Kompers piston rings and were experiencing an excessive amount of carbon buildup on the piston crown as a result of a recommendation of the DRQR program, those rings have been replaced; however, with Muskegan piston rings.

When were those Muskegan piston rings installed approximately?

JUDGE BRENNER: Mr. Dynner, if your interest is in relation to some other time, maybe they can answer that, rather than find a particular date. Or was it a particular date that you had to have?

MR DYNNER: An approximate day, if I could.

JUDGE BRENNER: What I mean is the only significance would be whether it was before or after something else. Why don't you ask them the question in that way.

MR. DYNNER: Maybe they have the answer. Do you have the answer?

MR. YOUNGLING: Yes. We replaced the piston rings on each of the engines at different times, and as I remember, the last engine was done by March of 1984.

Q. Was an analysis made to find out the source of the carbon that had become built up on the piston crown?

MR. YOUNGLING: As part of the DRQR program and the LILCO effort to review the engines, we saw this carbon buildup, and both FaAA and Dr. Pischinger, looked at the buildup and resulted in the recommendations that we talk about in our answer to the question 110 in our testimony. I'll ask Dr. Pischinger to comment on his observations.

DR. PISCHINGER: Well, carbon buildup is not unusual for -- it's usual in an engine of such type.

What we were concerned with was that carbon buildup behind the piston rings and near the piston rings on the crown was a little more than usual, which at least could lead to engine wear in a shorter time than usual, and we, therefore, recommended to use in combination with the recommended and now used Muskegan piston rings which have a shape unsymmetrical bell-shaped face on the first ring, which used in combination with this rings, a high detergent oil which is, in general, beneficial in reducing such coat buildup.

This detergent oils which are widely used also in marine engines help to dissolve this carbon products.

It is not usual to analyze such carbon products in this region, because it's completely clear where it's coming from, it's formed partially out of soot stemming from the combustion, from the combustion chamber together with products or -- coat products of the lubrication oil, and knowledge of any composition is of no help in assessing of what to do.

In combination with this high detergent oil, it was decided to use fuel injection tip with one hundred CERT 135 degrees which means sprays are not so much directed to the cylinder walls, is a good experience with thes injectors, that it also reduces carbon buildup.

There is experience at the Catawba nuclear power station with the engine that a higher grade detergent oil really sufficiently works and prevents this carbon buildup.

Q. You said that the carbon production -- production of carbon buildup was associated with the higher temperatures, with the high temperatures; is that correct?

DR. PISCHINGER: No. I said -- I couldn't remember. Did I say that?

Q. Is it associated with high temperatures in the piston crown or skirt?

DR. PISCHINGER: The usual environment in a piston crown is such that has always carbon formed with diesel engines.

Q. Were all three of these changes made at approximately the same time, that is to say, the changeover in the piston rings and the use of the higher detergent oil in the new fuel injection tips?

MR. YOUNGLING: No, Mr. Dynner.

Of the three recommendations, the Muskegan ring recommendation has been fully implemented. The change over to the 135 degree tips has been fully implemented, and we are beginning now to change the oil charge out of the engine and replace it with the higher detergent oil.

Q. When were the tips changed, approximately?

MR. YOUNGLING: That was accomplished by March of 1984 also.

DR. PISCHINGER: May I add something?

MR DYNNER: Certainly.

DR. PISCHINGER: Of course, one could

question why not do these changes all at the same time. Of course, this problem is --

MR. DYNNER: Actually I don't have that pressure.

DR. PISCHINGER: It's not urgent.

JUDGE BRENNER: He's not concerned with that.

DR. PISCHINGER: Because it's a long range wear problem and you could pursue it in connection with the usual inspection of the engines.

JUDGE BRENNER: It's almost 4:10, Mr. Dynner, why don't you ask your last question.

MR. DYNNER: Yes, one more. I have to think about that for one second.

JUDGE BRENNER: Is that a coincidence I asked you to do it just when you had one more?

MR. DYNNER: Oh, no. You know me better than that.

BY MR. DYNNER:

Q. Did any of you examine the possibility that the carbon and/or coat buildup that was noted in the piston crown might have been associated with the clearance between the piston and the liner?

DR. PISCHINGER: Just to make this question clear, what part of the piston do you mean?

Because adherence varies over the whole --

Q. Yes. The piston crown in the liner which you may know, if you read depositions of Mr. Lowry testified to at one point was a problem which causes a similar situation.

DR. PISCHINGER: We are aware that such engine as with TDI engines, when using a marine diesel fuel, that means a heavy oil fuel, heavy fuel oil, where the coat buildup is still more of a problem, it could be convenient to have a large adherence on the piston crown.

We do not think that this is necessary for Shoreham where number two grade diesel fuel, that means a very good diesel fuel is used.

JUDGE BRENNER: Okay. Interesting that you chose your last question a question that at this moment I don't see as being within the contention, but maybe in the final analysis will show me otherwise.

MR. GODDARD: I believe this question is best addressed to Dr. Johnson.

Would you please refer to LILCO Exhibit P-29.

Within that, I'd like you to turn to Page 2 of Donald Johnson's trip report on Kodiak, Page 2, the final paragraph thereof.

JUDGE BRENNER: Which trip report?

MR. GODDARD: I'm sorry, February 17th, Page 2 of the Exhibit 12.

BY MR. GODDARD:

Q. The final paragraph thereof references a three quarter inch indication which was found by penetrant which did not appear to be a crack like indication upon inspection with eddy current.

Could you tell us what the results of your investigation of that indication indicated it to be?

DR. JOHNSON: First of all, that indication was not down in the boss area. It was up in the area of the rib. We brought that piston back to Palo Alto, investigated it in the laboratory, that area, very carefully.

That penetrant indication was not reproduceable, that is, we never were able to get a penetrant indication on that area when we returned it.

I believe that it's due to the fact that it's awkward geometry to be working up at Kodiak. It was cold where they were working, about 38 degrees -- 38 to 45 degrees. It was cold.

And I think they simply did not wipe off all the penetrant in the process of doing the initial penetrant inspection.

Q. Did you at any time have occasion to discuss that indication with Donald Johnson?

DR. JOHNSON: Yes, we discussed it.

Q. Did he concur in your evaluation there was, in fact, a failure to properly remove the penetrant?

DR. JOHNSON: Yes, he did.

Q. Thank you.

The remainder of the staff's questions concern the issue of tin plating will be directed primarily to you, Dr. Pischinger, and Dr. Swanger.

Do you know whether the electroplating process which resulted in the tin plating of the AE skirts for Shoreham station were done by TDI facility in Oakland?

DR. SWANGER: We don't know that.

Q. Is there anyone else on the panel that can answer that question as to the source of that plating?

DR. HARRIS: In my discussions with TDI personnel, my impression is that the tin plating is not done at the TDI plant in Oakland but is done somewhere outside. I've never been informed as to who outside does the tin plating.

Q. Thank you, Dr. Harris.

Dr. Pischinger, the NRC testimony at page 54 indicates that a plating thickness for a piston skirt in an engine of this type, meaning a medium size diesel operating on a good grade number two diesel fuel would be a thickness of one to one and a half mils is acceptable.

Do you concur in that evaluation?

DR. PISCHINGER: Roughly, I would say within this range.

If it's a lot more tin on it, you get this tin migration which is not so favorable.

Q. You used the term migration. Would it be fair to call that smear or balling up of the tin?

DR. PISCHINGER: Yes. But with this thickness of thin tin plating, which is done here in this piston, this is a very favorable procedure. It's not the cheapest procedure to treat the piston, but it's very good.

DR. SWANGER: I might add in our visual inspections of the pistons, we have never seen evidence of the tin migration or tin smearing that was referenced.

Q. How was the thickness of the application of the tin plating controlled in the electroplating process?

DR. SWANGER: I don't know the specifics of how the subcontractor at TDI controls it, but if you wish, I can address the general principles for controlling tin plating thickness.

Q. Electroplating is a fairly common industrial application; is it not?

DR. SWANGER: Yes, it is.

Q. Fine. Proceed to do that.

DR. SWANGER: Tin plating involves the electro dissolution of tin from tin anodes and aqueous electroplating bath and the cathodic electro deposition of that tin on to the article being plated.

The thickness of plating will be directly proportional to the current density over the surface of the item being plated where the current density is higher for a given amount of time, the thickness of tin being deposited will also be higher.

In areas where you don't want any tin at all, they can be masked off with an insulator such as tape or wax burning the essential zero density at that point and no deposition; therefore, control or the electric -- of the current density is what controls the thickness.

The methods of controlling the current density are primarily geometric and chemical. The geometric methods are to place anodes around the part being plated such that every portion of the part being plated is about equidistant from a source of the tin, from the anodes, for instance.

The chemical means of controlling the thickness is referred to as the throwing power of an electroplating bath and is basically proportional to the conductivity of the aqueous electroplating bath itself. The higher the conductivity of the bath, the more even the iso potentials within the bath are, so the more even the current density is.

With tin, especially, it is easy to get highly loaded high conductivity baths and tin is known as being one of the plating metals with the highest stroke power, meaning that electroplated tin is about the most uniform metal which can be deposited by electroplating.

Q. Thank you.

If I understood you correctly, please correct me if I did not, the descriptions you gave would indicate how you would expect to get an even or equal thickness of deposition of the tin; is that correct.

DR. SWANGER: Going past the point where I discussed masking, which is intended to keep tin off of the part, yes, the principals that I talked about could be used to put down an even layer of tin. A high conductivity bath and an even spacing of electrodes would all be aimed at achieving a uniform thickness of tin on an electroplated part.

Q. Right. Dr. Swanger, as opposed to the uniformity of the thickness of the plating, how would you determine, during the plating process, when you have achieved the desired thickness on a uniform basis; in other words, I'm concerned with the uniformity question as much as I am with the overall thickness, the addition to the OD of the piston skirt.

DR. SWANGER: In my experience, the best way to do that is to monitor the overall plating current being used and to calibrate that against measurements of current versus time.

If you have the correct integrated value of the current times the time, you will have through Coulomb's law deposited the proper amount of tin on the part.

Q. Thank you.

I take it from your description of how you would apply the tin in a uniform manner that this would then operate the control concentricity of the application?

DR. SWANGER: Directing the discussion now to the tin plating of AE piston skirts, yes. This is how one would achieve a concentric layer of tin on the OD of the piston skirt.

Q. Thank you.

If these processes such as you've described for monitoring the application were not followed during the electroplating process itself, I would assume that in an easy way or an accurate way to determine the thickness of the application would be by measurement of the other outer diameter of the piston before and after the application of the plating process; is that correct.

DR. SWANGER: In the case of pistons where the diameter changed from unplated to plated is three thousandths of an inch on a 17 inch nominal diameter part that is possible in theory, but I think it's a -- it's not the way that would be most efficient for maintaining it.

DR. SWANGER: One of the facts of life of tin plating is that tin is expensive. It costs about seven or eight dollars a pound, and electro platers take special care not to put too much of this valuable material on their parts.

Q. Especially if the part is the size of an AE piston skirt.

DR. SWANGER: Yes.

Q. Mr. Youngling, to the best of your knowledge, did LILCO in their recent inspections of these pistons take any steps to measure the thickness of the tin plating on those pistons?

MR. YOUNGLING: No, we did not take any steps to measure the thickness of the tin plating; however, as part of our inspection of the pistons prior to release for shipment from TDI, we performed measurements of the pistons including measurements of the OD of the pistons to insure that they conformed to the design specifications.

DR. PISCHINGER: May I add, shortly --

Q. Please, Dr. Pischinger.

DR. PISCHINGER: In addition, it should be noted that no tin migration or how do you call it, smearing problem, is known, number one.

Number two, to my experience, a little too thin tin plating is usually no problem.

Q. You say a little too thin tin plating is no problem. Meaning that you would not, in fact, have to do any tin plating at all; is that correct?

DR. PISCHINGER: Yes. These pistons usually run with no tin plating, tin plating being some additional comfort.

Q. Mr. Youngling, in the process of the recent inspections, do you know whether LILCO reviewed any process documents which dealt with the tin plating of these piston skirts?

MR. YOUNGLING: As part of the inspection done by LILCO prior to releasing the pistons from the factory, one of the attributes that Stone and Webster inspectors had to look at was a review of the routing sheets which are used by TDI in the manufacture of the piston.

I don't have those documents here, but as part of those routing sheets, as long as there was a sign-off for the tin plating, there would have been a review that the tin plating had been done.

In addition, the inspector had to perform a visual inspection of the piston in conformance with the design documents, and he would have noticed if the tin plating had not been done.

Q. Well, visually, I guess we all would have noted if the tin plating had not been done. As Dr. Pischinger put it out you've got a dark colored nodular iron cuffed with a very light colored tin, so if it hadn't been done it would be somewhat readily apparent.

What I'm asking is whether there were any process documents reviewed by LILCO upon taking receipt of these pistons which indicates the appropriate thickness of the plating.

MR. YOUNGLING: We've reviewed some information here on the routing sheets and there is a check-off point that the tin plating had to be put in place. It's indicated as being bought out, meaning it's done outside on a subcontractor basis.

That is the degree of documentation that we have reviewed to insure that it was done properly.

In addition, I'm sure that TDI has specifications as part of their drawings to insure that the proper thickness is specified.

Q. Does that complete your answer?

MR. YOUNGLING: Yes, it does.

Q. You indicated that upon receipt you mentioned the outer diameter of the pistons and found them to be in conformance; however, as Dr. Swanger testified, measuring for the existence of one to one-and-a-half mils on each radius of a 17 inch piston does not really comport with real world tin plating, I believe.

In regard to those inspections, was that the only inspection done by LILCO which might evidence the thickness of the tin plating?

DR. SWANGER: If I may, I might just clarify my comment.

In order to know the thickness of the tin plating, you'd also have to know the preexisting as machined diameter of the piston.

There's certainly going to be a manufacturing tolerance on that, and something this size is probably one to two mils and that would add that much to the uncertainty of a -- an imputed thickness of tin.

Without knowledge of what that preexisting diameter is, that's why I said it was difficult to determine the tin plating thickness by direct measurement of the OD.

Q. Thank you.

Then in view of what you just said, it would be impossible to draw any conclusions to the thickness of the tin plating by virtue of the receipt inspection measuring the ODs; is that correct?

DR. SWANGER: No, I don't think that's true either. Because there is a specification for the machine OD of the piston prior to plating and by computing the stack-up of the tolerances, you can draw a conclusion as to the limits of what the tin plating thickness would be.

Q. Dr. Swanger, do you know what the tolerance is on the manufacturing OD of the piston prior to plating?

DR. PISCHINGER: Yes, we can tell you. Just a moment.

JUDGE BRENNER: While they're looking, give me just one moment, Mr. Goddard.

DR. SWANGER: In answer to your question, yes, I do know what the tolerance is on the diameter before plating.

Q. And that is?

DR. SWANGER: It's a range of four thousandths of an inch.

Q. Then the measurement of the OD after plating could, in fact, indicate a tin thickness of eight mils possible; is that correct?

DR. SWANGER: Assuming a minimum size manufactured piston and a maximum size after plating, the drawing indicates that it is possible to have three-and-a-half mils of tin thickness on the piston.

Q. That's three-and-a-half on the radius, seven on the diameter.

DR. SWANGER: That's correct. I testified to the thickness of it.

Q. I just wanted to make sure that we're together on that.

DR. SWANGER: I can point out the danger of this kind of thinking by looking at the other extreme in tolerances.

It's possible to have a negative one thousandths of an inch of tin on the diameter.

Q. I think not.

JUDGE BRENNER: At some point, Mr. Goddard, I'm going to ask you it might -- in how many mils difference in terms of materiality is the issue before us?

One thing, I assume all these answers were without regard to the measuring capability of the measuring equipment and on and on and on.

MR. GODDARD: That is correct. We are concerned about thicknesses, let's say, substantially in excess of the one to one and a half mil range based on the application and the greater fuel for this type engine.

JUDGE BRENNER: Well, I don't know what you mean by substantially in excess, and we're talking here about one and a half mils, three mils, four mils.

MR. GODDARD: The staff would be concerned if it were in the three, three-and-a-half mil range as indicated which would be the maximum possible under the manufacturing tolerances as testified to by Dr. Swanger.

JUDGE BRENNER: All right. If that's the case, I guess that's why you've asked your questions.

DR. PISCHINGER: May I --

JUDGE BRENNER: Let's -- no, there is not a pending question, I don't believe.

MR. GODDARD: The staff has no further questions for this panel.

JUDGE FERGUSON: I'd like to ask the panel a few questions at this time. Before I do, let me ask whether or not you have benefit of Tuesday's transcript, do you have that available?

BY JUDGE FERGUSON:

Q. What I'd like to focus on first is some concerns I've had about our discussion of crack initiation and propagation, so we're going to re-visit that for a short while, and the exhibit that I want to use in that discussion is Exhibit P-9; so if you could have that in front of you, it would be helpful.

I want to ask just briefly about the concern that I have regarding the engine cylinder pressure logs.

Those engine cylinder pressure logs are divided into two general categories. One is pre-crank shaft failure category and one a post-crank shaft replacement category.

Is the panel following?

DR. PISCHINGER: Yes.

Q. Now, I'd like to very briefly look at the first set of logs that have to do with the pre-crank shaft failure.

I see they're logs for EDG 101, 102 and 103, and just looking at the numbers in those logs, and these are pressure values, I believe, we can sort of eyeball those numbers on EDG 101 to get an average, perhaps, of around 1550 for the pressure, and they're all about that number, and if we do the same kind of quick averaging for EDG 102, it's slightly higher, maybe 1625 or so, and then when we look at 103, again, a rough eyeball average might be a cylinder pressure of maybe 1525 or 30.

Keeping those numbers in mind, and turning forward to the post-crank shaft replacement logs, we note immediately that for 101, the average is somewhat higher.

If we look at the log for 102, it may be slightly higher than what we found in the pre-crank shaft replacement logs.

And that same comment as to 103.

Now, the first question that I'd like to ask is what is it -- or what can we ascribe the differences between, say, the average of the cylinder pressures pre-crank shaft failure to post-crank shaft replacement.

MR. YOUNGLING: Judge Ferguson, let me point out to you one possible difference.

If you look at the procedures that implement these testing requirements, these are base -- what we call base line data.

What we're trying to do is prior to releasing the engine to plant staff of permanent operation, we just take the engine to approximate full load condition and take a set of base line data so they know for approximately full load, so there is a possibility of a difference in load of a couple of, maybe, 25 kw, so forth.

In fact, if you look at the post-crank shaft data, you'll see for 101 it was 3,500, 102, 3528 and 103 3595 kilowatts.

In addition, the post -- I'm sorry, the pre-crank shaft data was taken in August time frame, the summer conditions, while the data after crankshaft replacement was taken in the April and March time frame, different temperature situations.

But the last contributor and which is probably the most significant contributor was the fact that after the crankshaft failure, we disassembled the entire machine and had to reset the engine up, retime the engine.

And as a result of that retiming, it is entirely possible that you would see the differences in the firing pressures.

And perhaps Dr. Pischinger could add some more to that.

DR. PISCHINGER: Of course, in disassembling the engine including the crankshaft, the whole gear to the injection pumps have to be re-adjusted, and though it is necessary as has been told that injection has to be retimed, and in my experience, this sometimes happened, when you have an engine that had time factory set timing and after a certain time you reset according to the handbook, you get a difference, and I think this is what happened here.

Q. It's easier to use perhaps a different setting of timing.

DR. PISCHINGER: Yes. I think this is a probable cause.

Q. Let's stick with EDG 101 for the time being, and this is the group that have to do with the post-crank shaft replacement.

Do you have that in front of you? This is the one that we spent a good deal of time on before, but I think there's one or two things that we should discuss briefly to help clarify the record.

As we look at EDG, the report on EDG 101, we see that the numbers in that particular table range from a low, perhaps, of 1640 to a high of 1720.

Would you think that that range, different pressures and in those instances might be due to the causes that you just described?

MR. YOUNGLING: The spread of the numbers that you're seeing there is quite typical of our balancing of the engine.

We have a roadway requirement in the TDI manual that we have to have no more than 200 psi between the maximum values.

We generally have been able to time -- to set the engines up to around a hundred pounds difference. Sometimes it's 120, sometimes it's 80, but generally we run in the hundred pound range, about half of the TDI limit. That's just the amount to balance the engine out.

We're certainly well within the TDI specification for balance.

Q. There's nothing peculiar about 101 that would make a difference say, from 102; is that correct? It just seems to me you did a much better job on 102 than you did on 101.

MR. YOUNGLING: Just like taking three cars to the same mechanic. One comes out running a little differently than the other.

Basically, we have three engines that are within specification, a few pounds difference, but still in satisfactory specification.

Also, when you set these engines up, the way you balance them is by inserting shims under the fuel pumps, and the shim stock is only of so much thickness, and you can only buy a certain amount of balance, so that's the kind of procedure and these are the kind of results that you get.

I think we do pretty well, actually.

Q. Okay. Well, let me proceed.

The reason I was focusing on those numbers is that in the failure analysis that was done, apparently these numbers were used as a guide, I think the number 1670 psi was tested by, too, as sort of being an average of a group of numbers; isn't that correct?

DR. MC CARTHY: Close. An average peak pressure that was conservative, that is, a value that the actual numbers fell below, as I indicated in my testimony, that we had not measured that.

In fact, here is a plot of peak pressure for 200 seconds --

Q. The record can't see that.

DR. MC CARTHY: This is, we will put it in the record then, this is just a plot --

JUDGE BRENNER: Why don't you try to describe it. Let your counsel decide whether he wants to put it in.

DR. MC CARTHY: When this question came up, I got a telecopy of our original data from our engine test where what I've got plotted is peak pressures for 250 seconds of running time, four minutes for every peak pressure of every combustion cycle was plotted, and what we saw was that the pressures running at full load ranged from about 1550 psi gage to the single highest point reading we got was below 1668 psi gage with an average for the 200 and 240 seconds about 1604 psi gage.

We took a number of cycles and, of course, this is a large number of data points, none of which got up to the 1670.

Now, these are done with a quartz transducer that's highly accurate. A Kiene gage is accurate to a percent, and so using the 1670, which was a value we didn't actually measure as high up, we called a conservative average peak value for fatigue damage purposes.

Q. The group of data that you just described are all lower than the group that we're looking at; is that correct?

DR. MC CARTHY: No. As I indicated, they range from 1550 to a high of 1668, which falls in the range of the Kiene gage, but the key Kiene gage measurements -- but, remember, the Kiene gage has a check valve to keep its high reading, and doesn't do the same operation.

Q. Okay.

One of the points in asking this line of --
introducing this line of questions is that there was a number -- you say 1670, was that number that you used in the failure analysis?

DR. MC CARTHY: That is correct.

Q. Let's stick with that number.

DR. MC CARTHY: I'm sorry. That is not -- we used that in the p ston analysis on -- for the earlier crankshaft analysis before we had measured data of our own, we used a value of 1680 supplied to us by TDI, but we actually measured the pressure and we used 1670 for the pistons. 1780 is a peak pressure for bending in the crankshaft analysis.

The peak pressure only affects the bending stresses at the crank shaft, not the torsional stresses.

Q. Let me get to really the thing that I'm interested in. And that is if you, in fact, measure pressures as indicated in this exhibit that we just described that are higher than the number that you have used in your failure analysis, what effect have you in your analysis looked -- have you in your analysis looked at the effect of the extreme values?

Is the question clear?

DR. MC CARTHY: I believe, your Honor, I

understand your question in that let's take as a point of argument, and let's say the peak pressure were 1720 for a supposition to answer your question, which is the highest value of measure here, we did did take the fracture mechanics analysis, in fact, beyond 1720 to 2,000 to see if that not only would -- the sensitivity of our analysis of the piston have to our assumption of peak pressure under normal operating conditions, but, of course, the additional effect the overload conditions that the engine is sometimes required to run at and that has to be considered as well.

As we've testified to previously, our analysis and testing indicates that the cracks -- a crack would not grow even if the average peak pressure were 2,000 psi, and, therefore, our analysis -- our conclusions are completely insensitive to a pressure difference of this small an amount.

DR. HARRIS: If I could just interject here for a moment to expand somewhat on Dr. McCarthy's testimony, we took the analysis to two 2,200 psig, not 2,000. Very minute point.

Q. All right. That's been helpful to me.

Let me ask you now to turn to Page 22227.

This question is going to be directed to Dr. McCarthy and Dr. Swanger. Do you have that front of you now?

I'm going to read the question that was asked.

And just to save me the reading of the lengthy answer, Dr. McCarthy, I'd like for you to read your answer, if you would be kind enough to do it and then, Dr. Swanger, I'm going to ask you after I ask Dr. McCarthy a question to read your shorter answer.

I'm reading the question now. Dr. Harris and Dr. McCarthy, on page 44 in the response to question 69 you referred to the use of engineering fracture mechanics in modern design and analysis in structures such as aircraft, spacecraft, pipelines and turbines, et cetera.

"You mean to suggest that fracture mechanisms are used in the design of these various structures in order to insure that if there are defects or crack like indications that they won't propagate to dangerous levels?"

Dr. McCarthy, would you be good enough to read your answer ought out loud so that all of us can hear that answer.

DR. MC CARTHY: "In a nutshell, yes. A lot of the work we do at Failure Analysis is just making those analysis for people of critical boss size and what kind of critical flaw can exist in your structure" and I guess it should say to correct that, not just -- there are two things that you could do. One is a critical flaw size to know your structure will not fail in an overload and then there's a second question. "Does a critical flaw size determine how much you have to go back and look at your" -- it should be structure "because not only do we deal with the analysis of when a crack will initiate but indeed how fast it will propagate and at what size you will begin to affect the critical nature of your structure, in effect, the engine, the aircraft engine problem here referenced there was not an assumption of flaw size problem as much as a problem to analyze the rate at which cracks could grow in," I guess it should be "in the field and how often such parts have to be inspected so that you can catch any growing crack at the appropriate time. You don't have to postulate an initial flaw. These are expensive parts that are extensively inspected" that should be "but come out with no real measurable flaws but, in fact, operate in the initiation range and, in fact, a crack" should be "will initiate and grow and this was to establish their inspection interval."

DR. MC CARTHY: You see my motive for having to read that, but Dr. Swanger, would you read your answer and then I'll ask my question.

DR. SWANGER: "I might put this -- might put this into the context of AE pistons at Shoreham and that is that our analysis says that no cracks are possible to propagate in these pistons; therefore, they do not need any reinspection." It should be "An initial inspection upon manufacture is sufficient to show that there are no cracks and we have demonstrated through fracture mechanics that no further operational inspections are required.

Q. That's the point I'd like to focus on very, very briefly, if I may.

My understanding of these words you had just recently read is that an initial inspection of the piston is made after it has been manufactured, and I'm going on to assume that the manufactured part is installed. Let me go back. The manufacture is inspected and installed. Does this last statement indicate that you're recommending that no further operational inspections are required at all at any time.

DR. SWANGER: We, of course, would recommend that LILCO follow the recommendations of TDI as far as routine inspections of the piston.

Q. Do you know what they are?

DR. SWANGER: I don't know what they are right now.

Q. Is there anyone on the panel that briefly can tell me whether or not there's a routine reinspection after this has been installed?

DR. MC CARTHY: With regard to cracks in the stud boss area, though, the recommendation is exactly as Dr. Swanger stated.

After an initial inspection at the time of manufacture and the finding of no critical flaws, that is, flaws of a size bigger than we predict will grow, there's no need to inspect the stud boss region for cracks with eddy current or die penetrant at later phases in the operation, at least for the operating stresses that we've stressed in our finite element model and have testified about.

Q. That's the point I really wanted to get to.

I envision at some date the machine running many cycles and unless you tell me that there is some routine reinspection, I'm not so -- I'm not convinced at the moment that that initial inspection together with your analysis will verify that nothing, in fact, could happen.

I understand all of the testimony about your predictions, if there are no cracks, none will initiate or grow. I understand that. But it seems to me that -- I'd like to know whether or not there are any routine inspections planned.

Now, if you have it before you, I'd like to have it today, but if not, maybe at a convenient time to break and we'll pick up with this tomorrow.

MR. YOUNGLING: Judge Ferguson, we don't have the DRQR matrix which was developed. It's out in our anteroom. Perhaps we could come back to you tomorrow morning.

Q. Tomorrow simply tell me what the routine inspection is.

MR. YOUNGLING: Yes, we will.

JUDGE BRENNER: We're prepared to recess and come back at nine o'clock tomorrow morning.

Did you want to say something, Mr. Farley?

MR. FARLEY: Please, Judge Brenner.

The Board will have mailed to it a letter that was delivered today to Mr. Denton, and also delivered to Mr. Berlinger and a copy was afforded to counsel for the County.

It deals with a proposal that's been under discussion regarding the crankshafts and the cam gallery area of the block, and in the interest of professional candor, I would like to hand a copy of this letter to each member of the Board.

JUDGE BRENNER: Okay. That's the first I've heard about it. We'll take the letter and read it.

MR. DYNNER: If I would just add, Judge Brenner, we were given a copy of this letter at the lunch break.

It involves some proposals by LILCO that are based upon just a very cursory review of the letter that seem to appear to be a significant revision of the SFAR to derate the diesel engines and also contains discussicns about or some discussion about discussions that were apparently going on between the staff and LILCO concerning additional testing of the engines.

I just will say as a preliminary matter that it seems to us that these matters discussed in this letter could have potentially a very significant bearing upon these hearings and as the

Board reads the letter, it will quickly become apparent that it involves things such as load factors for the engines upon which both our testimony and LILCO testimony involving the crankshafts were based as well as other related matters, so I'm not going to go any further except to say that I think that this material may have a significant bearing on the hearings.

JUDGE BRENNER: Well, I haven't read it and I don't know what's in the letter. There have been discussions of different load factors going back quite some months, and back then I said if anybody -- have any argument that there was something material in there, presuming they would present it to us.

We sit here in an adjudicatory proceeding. I'm not worried about the routine correspondence or even non routine correspondence that goes on between the staff and LILCO.

We'll get copies so we've apprised of the situations, but we don't make any fact findings based on that type of material and nobody brought anything to us in the proceedings with respect to the different load factors, that there's -- that there have been any contention that loads assumed for use in the analysis of what the diesels would have to run in an emergency situation are incorrect and that's where that stood.

Now if there's anything new or different in there, presumably, we'll hear about it from somebody.

MR. FARLEY: I just felt obligated to bring it to your attention as soon as it had gone out.

JUDGE BRENNER: But I assume at some point you're going to discuss among the parties, if anybody is going to bring something before us and you can tell me -- if LILCO can point out whether they think it's material or not material to anything before us and the other parties can do the same.

MR. FARLEY: Our present plan now, your Honor, is to proceed with the crank shafts on the file testimony.

JUDGE BRENNER: We'll recess until nine o'clock tomorrow morning.

(Whereupon, at 5:05 p.m., the hearing was adjourned, to reconvene at 9:00 a.m., Thursday, September 13, 1984)

CERTIFICATE OF OFFICIAL REPORTER

This is to certify that the attached proceedings before the UNITED STATES NUCLEAR REGULATORY COMMISSION in the matter of:

NAME OF PROCEEDING:

SHOREHAM NUCLEAR POWER STATION

Long Island Lighting Company

DOCKET NO.: 50-322-OL

PLACE: Hauppauge, New York

DATE: September 11, 1984

were held as herein appears, and that this is the original transcript therecf for the file of the United States Nuclear Regulatory Commission.

(Sigt)

(TYPED) HELEN DOHOGNE

Helen S. Dohogne

Official Reporter

Reporter's Affiliation

MOTOR SHIP

SULZER

The 1st choice for marine diesel engines all over the world

NUCLEAR REGULATORY COMMISSION

Docket No. ________ Official Exh. No. Diesel 69
Suffolk County

In the matter of ________

Staff ________ IDENTIFIED ________

Applicant ________ RECEIVED ________

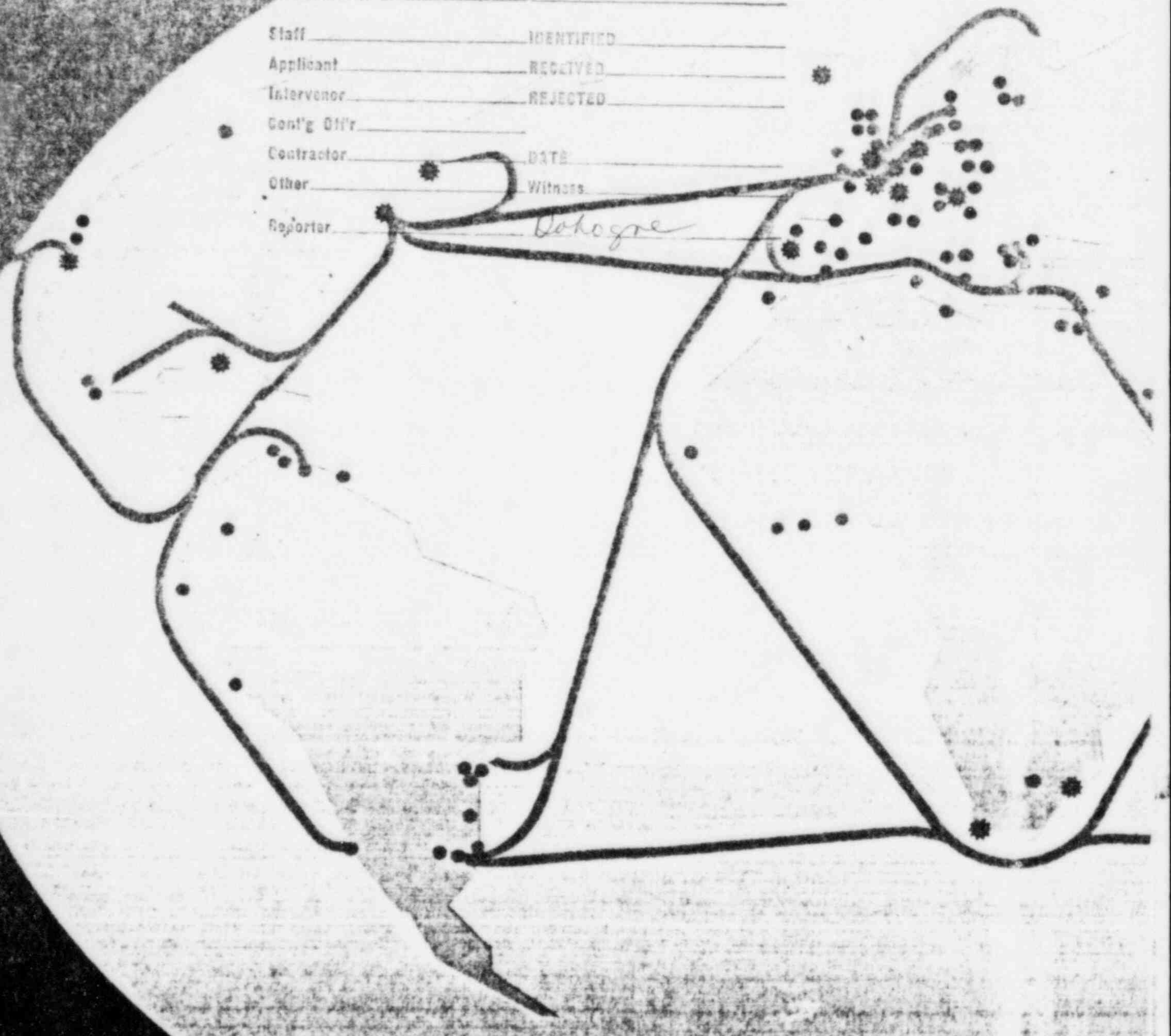
Intervenor ________ REJECTED ________

Cont'g Off'r ________

Contractor ________ DATE ________

Other ________ Witness ________

Reporter ________ Dohogne



Sulzer's four-stroke high-and medium-speed engine range

by G. Lustgarten* and R. Stoffel**

IN RECENT YEARS the four-stroke diesel engine has consolidated its position in the marine field. Besides its application for auxiliary power generation, the four-stroke engine is an interesting alternative to the two-stroke crosshead diesel where limited space is available for the propulsion unit, for example in roll-on/roll-off ships, passenger vessels and ferries. Sulzer has systematically worked to offer an engine programme to meet such demands.

The power range between 620 kW and 15 900 kW is covered by three engine types:

- A-engines for outputs up to 3 600 kW (4 860 bhp)
- Z-engines for outputs up to 9 600 kW (13 050 bhp).
- 65/65 engine for outputs up to 23 850 kW (32 400 bhp).

*Head of development and design, four-stroke engines

**Head of development in ...beds

The first two Sulzer-designed types have now reached an advanced stage with wide service and testbed experience gained over the past ten years. The 65/65 engine—a joint development of Sulzer and M.A.N.—had completed an intensive development programme by the end of 1977.

The AS25/30 and AL20/24 engines

The AS 25/30 design is built in six, eight and 10 cylinder in-line and 12,16 and 18V versions, Figs. 1, 2 and 3. The main features of the engine are a simple and robust design, good accessibility to the parts subject to wear and easy maintenance.

The original cylinder output of the two-valve A 25/30 engine was 135 kW/cyl (184 bhp/cyl). During subsequent development stages, however, the AS 25/30 engine was provided with four valves and the cylinder output increased to 200 kW/cyl (270 bhp/cyl). In the course of developing this originally Sulzer-designed engine, the AS 25/30 was incorporated into the common engine programme with M.A.N., which has also taken up its production. Extensive endurance tests at 220 kW/cyl (300 bhp/cyl) have already been carried out with a view to further uprating. By the middle of 1977 over 1 350 A 25/30 and AS 25/30 engines were in service and the maximum running hours had exceeded 35 000 hours.

When first introduced, the A 25/30 engine was used mainly for auxiliary power generation. Due to the increase in output as well as the subsequent development of the larger V-cylinder units, the AS 25/30 range has also gradually made its mark as a propulsion engine. The smaller AL 20/24 engine (Fig. 4), based on a similar design, is being manufactured in six- and eight-cylinder in-line versions for auxiliary power generation, the propulsion of small ships and for power packages (as illustrated by the 500 kW container unit in Fig. 5).

The Z 40/48 engine

The Z 40/48 engine is offered in six- and eight-cylinder in-line form and 10, 12, 14, 16 and 18 cylinder V-versions (Fig. 6, 7 and

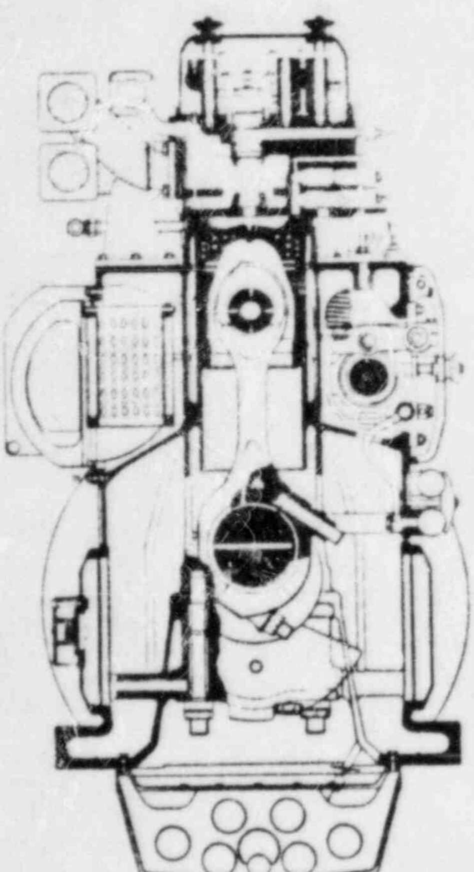


Fig. 1 Cross-section of the ASL 25/30 engine

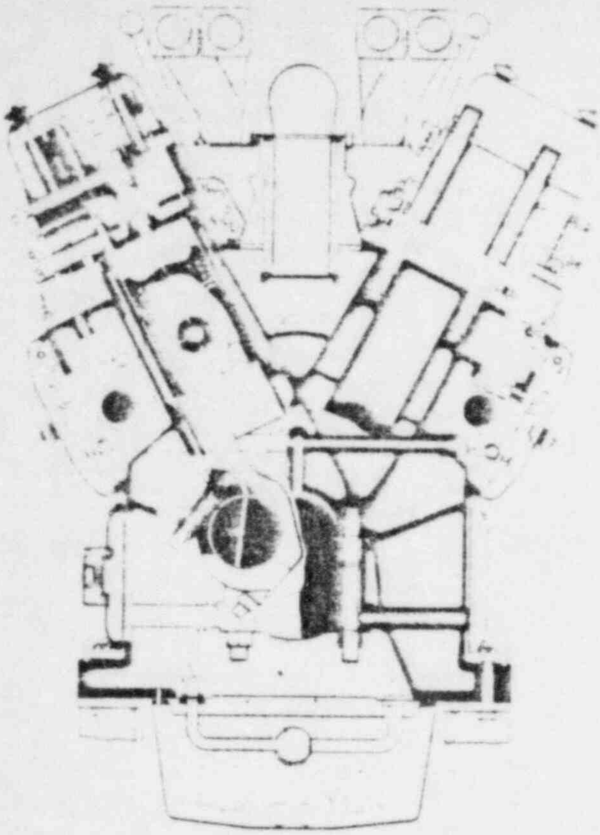


Fig. 2 Cross-section of the ASV 25/30 engine

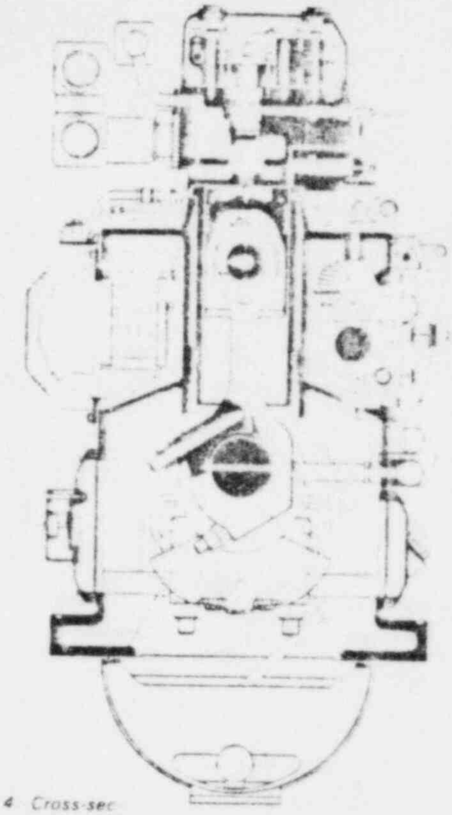


Fig. 4 Cross-section of the AL 20/24 engine

8). This engine, which is the main representative of our medium speed engine programme, was initially introduced as a two-stroke version with 330 kW/cyl. (450 bhp/cyl.). As a four-stroke design, it now has a maximum cylinder output of 550 kW/cyl at 560 rev/min. (For stationary applications only, the maximum nominal speed is 600 rev/min.)

Design has been concentrated on solutions to meet the requirements of high reliability in heavy fuel service. Special measures have been adopted to reduce thermal stresses and achieve low wear rates as well as extremely low lubricating oil consumption: the rotating piston is the most outstanding feature of the Z 40/48 engine.

The number of engines in service by the end of 1977 amounted to some 180 with maximum running periods of 52 000 hours for the two-stroke versions and 24 000 hours for the four-stroke models.

The 65/65 engine

The thorough development tests with the 12-cylinder prototype 65/65 (Fig. 10) by Sulzer in Winterthur as well as with a four-cylinder test engine by M.A.N. in Hamburg were completed by the end of 1977. Many technical solutions, which proved to be very reliable in operation with the Z 40/48-engine, have been adopted for the 65/65 engine. The larger dimensions, however, made it necessary to adopt different solutions for certain components; for example, a welded crankcase with bolted-on cast iron cylinder blocks and constant pressure turbocharging were specified.

Fig. 3 A 16-cylinder ASV 25/30 engine

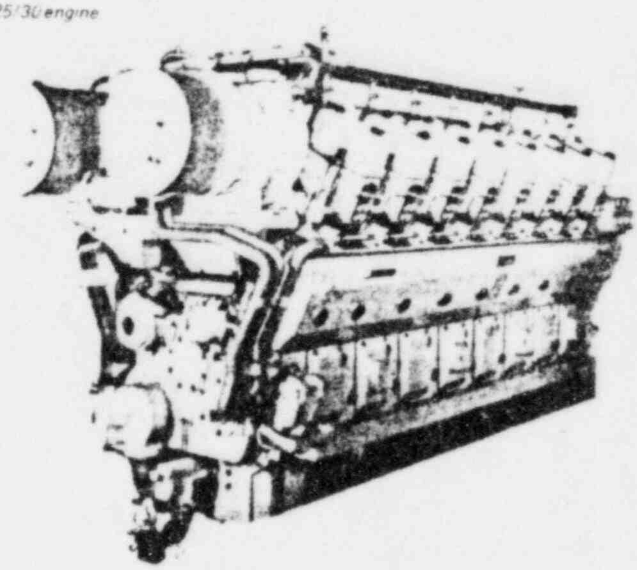
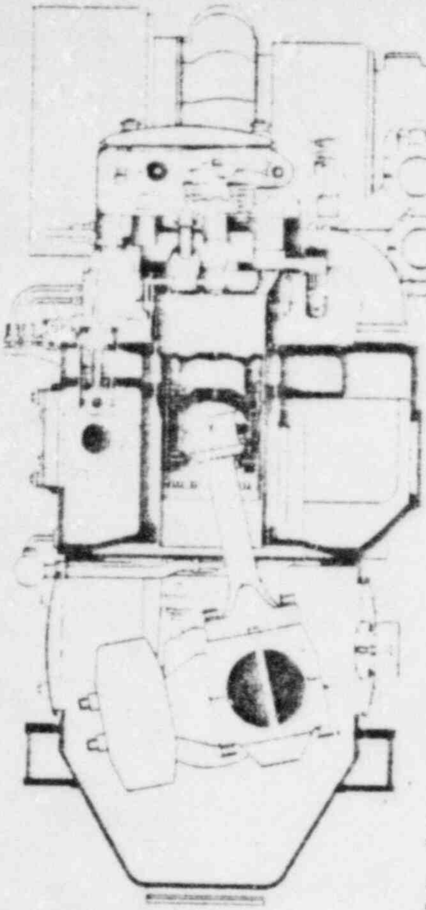


Table 1: The Sulzer medium speed engine range.

Engine Type	Bore/Stroke (mm)	Speed rev/min	b.m.e.p. (bar)	Cyl. output (kW)
AS 25/30	250/300	1000	16·29	200
		900	16·75	185
		750	17·38	160
AL 20/24	200/240	1000	16·31	102·5
Z 40/48	400/480	560*	19·54	550
		530*	20·02	533
65/65	650/650	400	18·43	1 325

** Maximum engine speed for stationary applications is 600 rev/min.*



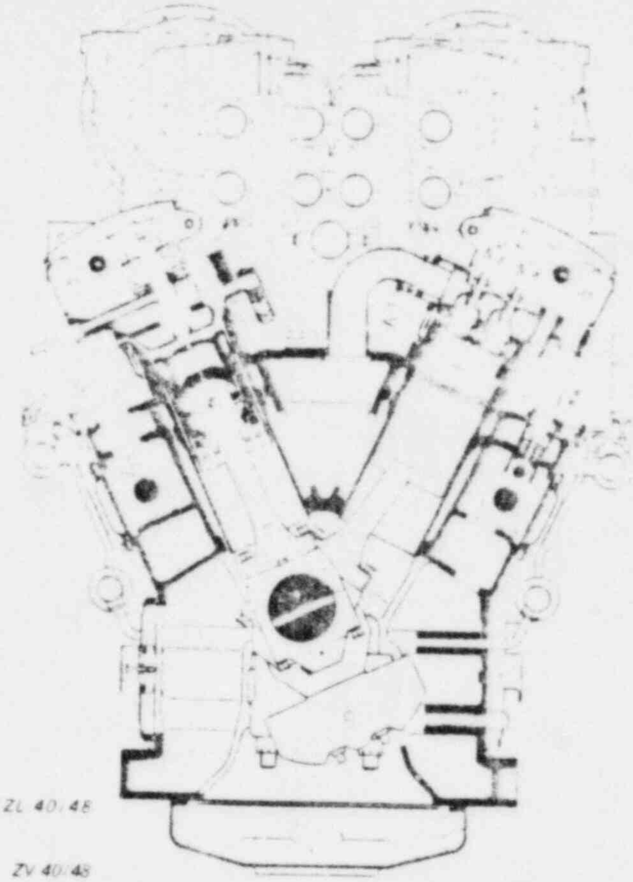
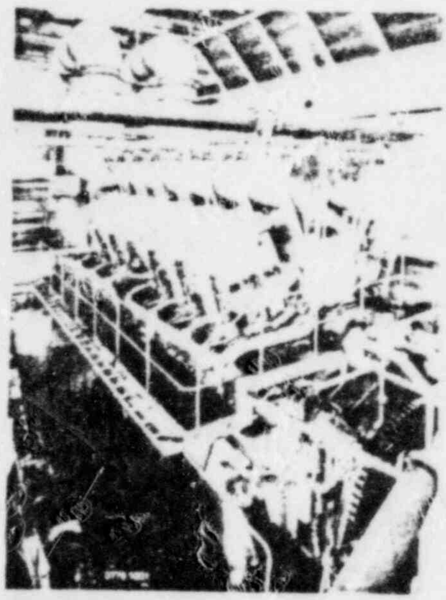
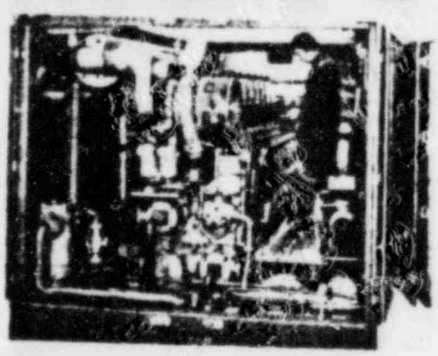


Fig. 6. (Left) Cross-section of ZL 40/48 engine.

Fig. 7. (Right) Cross-section of ZV 40/48 engine.





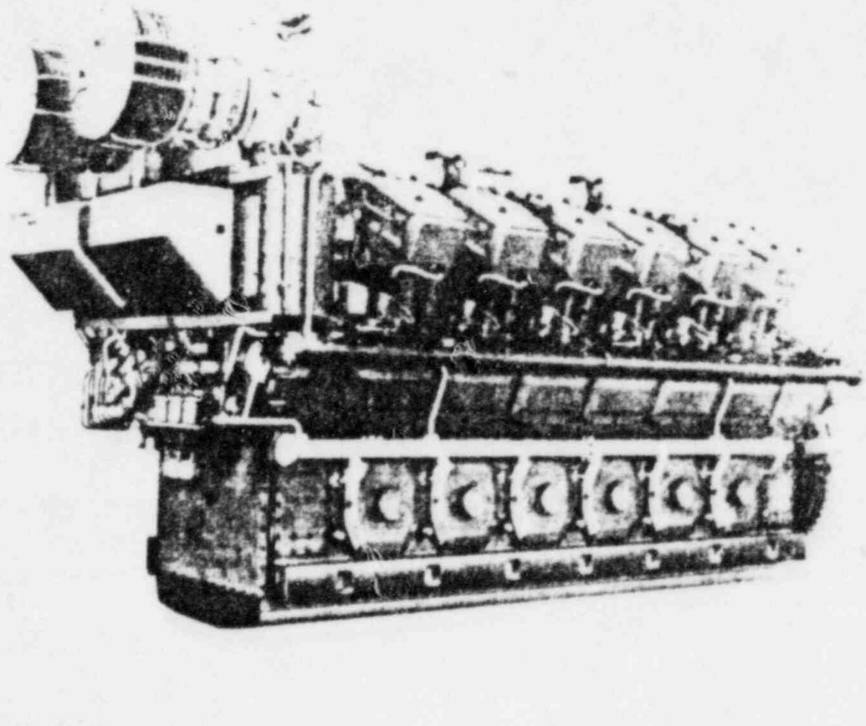


Fig. 10. The 12-cylinder V65/65 engine on the test bed (upper left).

Fig. 8. A 12-cylinder ZV 40/48 engine (above).

Fig. 5. A container fitted out with a six-cylinder ASL 20/24 engine to provide a 500 kW power package (left).

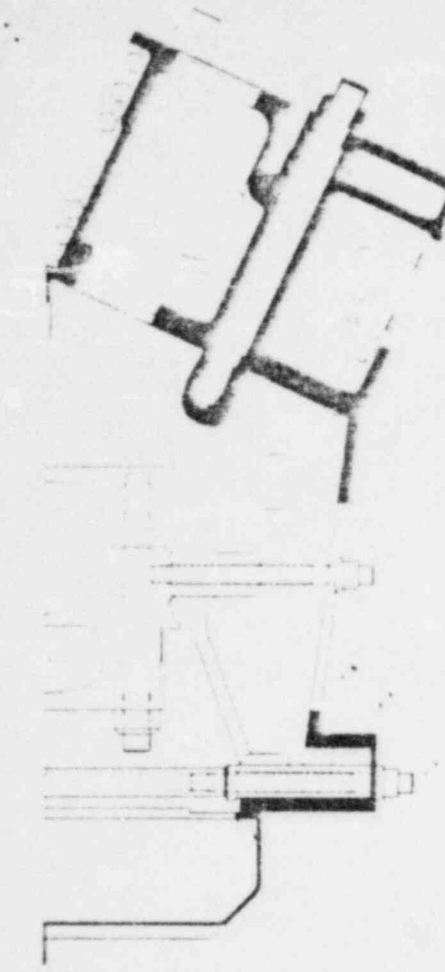


Fig. 12 Engine casing design of the V65/65 engine

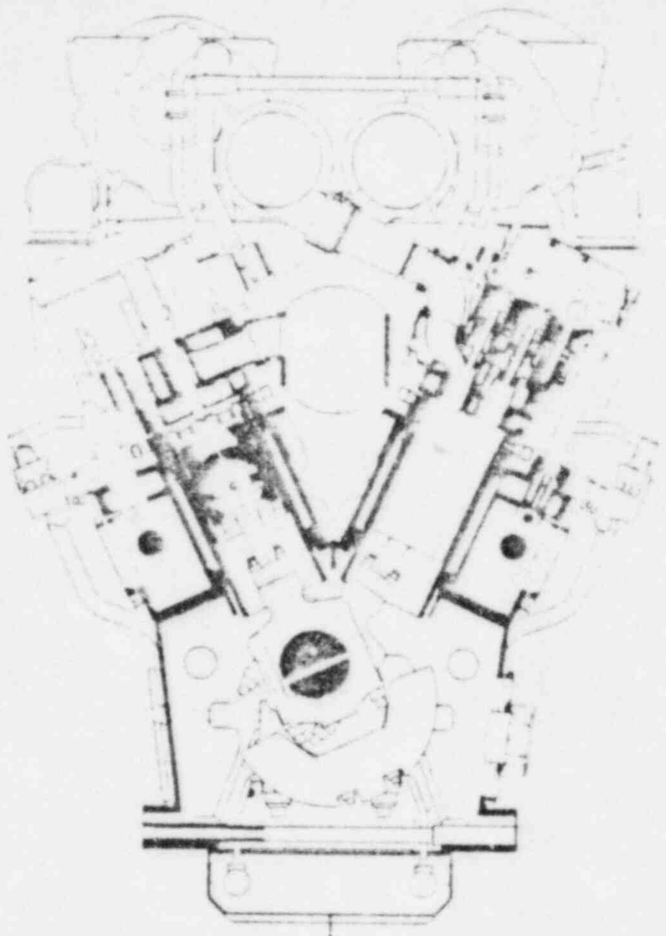


Fig. 9 Cross-section of V65/65 engine

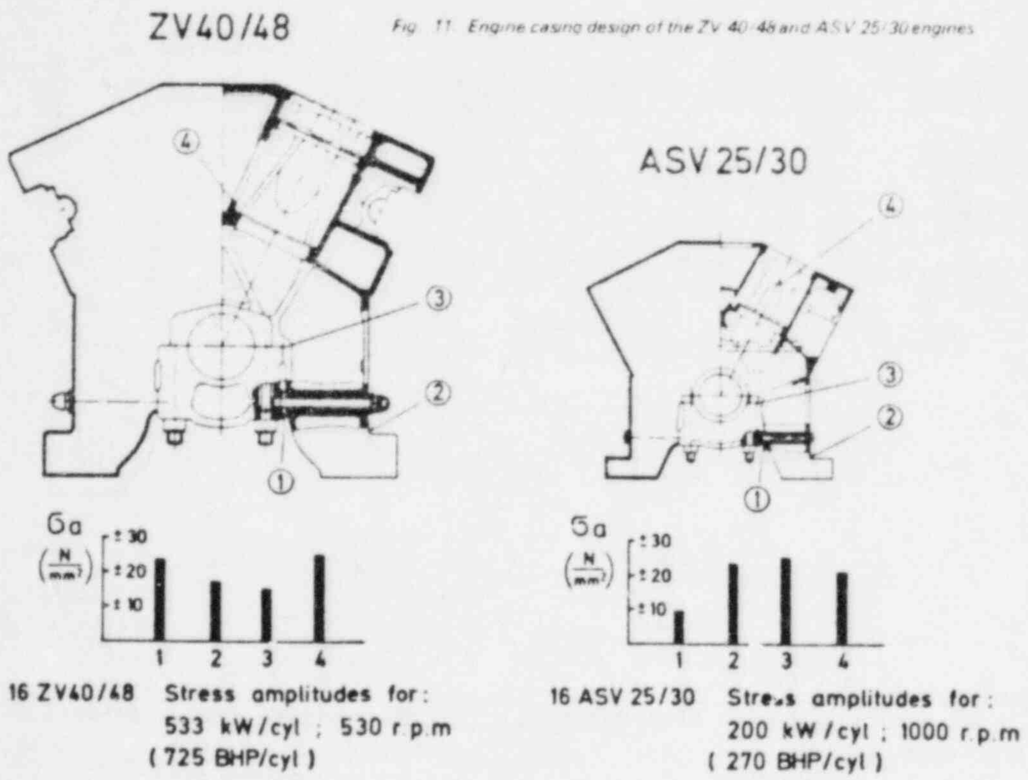


Fig. 11 Engine casing design of the ZV 40/48 and ASV 25/30 engines

The following discussion presents some of the specific features of the Sulzer four-stroke engine*. Special emphasis will be placed on the Z 40/48 engine, which is the "backbone" of our four-stroke programme.

Low level of mechanical stresses

The incorporation of a high power reserve for future development allows the engine to enter production at a reasonably safe level. Furthermore, the attainment of a higher cylinder output in the course of the final development stages does not negatively affect the operational safety, but represents only the full utilisation of the reliability designed into the engine at an early stage. A low stress level in the major engine components is one of the necessary requirements.

Engine casings: Fig. 11 shows the casings of the ASV 25/30 and ZV 40/48 engines which are, in their basic design, very similar. The underslung crankshaft allows the forces induced into the bearings to be transmitted directly to the cylinder head. Tie rods transmit the horizontal forces of the bearing loads from the bearing caps into the frame. The low stress level is reflected in the maximum amplitudes of less than ±24 N/mm^2 (± 2·5 kp/mm^2) at full load. Even for b.m.e.p.'s of 24·5 bar (2·5 kp/mm^2), which corresponds to a cylinder output of 735 kW (1 000 bhp) in the case of the Z 40/48 engine, the safety factor will remain above 2·0. This explains why no mechanical failure of the casings were experienced during service by a Z 40/48 or AS 25/30 engine.

The stress level in the casing of the 65/65 engine is also at a similarly low level.

A new solution was sought for the connecting rod of the 65/65 in order to simplify overhaul work. The shaft is extremely short, allowing the dismantling height of the piston to be reduced to a minimum. The big end is bolted to the shaft by means of eight hydraulically tightened bolts, which are readily accessible.

Low thermal loading

Z 40/48 wall temperatures: an engine with high specific output intended to burn low quality heavy fuels must be provided with adequately-cooled combustion chamber walls in order to avoid thermal cracks and high temperature corrosion. Fig. 14 shows the combustion chamber temperatures of the Z 40/48 engine. The cylinder liner temperatures are kept at an adequate level due to the application of the bore-cooling principle developed at Sulzer many years ago.

The design principle of the double bottom has been applied to the cylinder head and the thin flame plate is intensively cooled. The mechanical loads due to gas pressure are carried by the massive intermediate supporting deck.

Z 40/48 valve seat design: the exhaust valve is one of the most critical parts of the cylinder head, if not of the entire medium speed engine. For reliable performance, a perfect sealing of the seat as well as efficient cooling are necessary. In the case of the Z 40/48, a thorough investigation was carried out in order to find the best design for an engine with a high reliability and capable of burning heavy fuels up to 3 500 sec. Red. I with high sodium and vanadium contents.

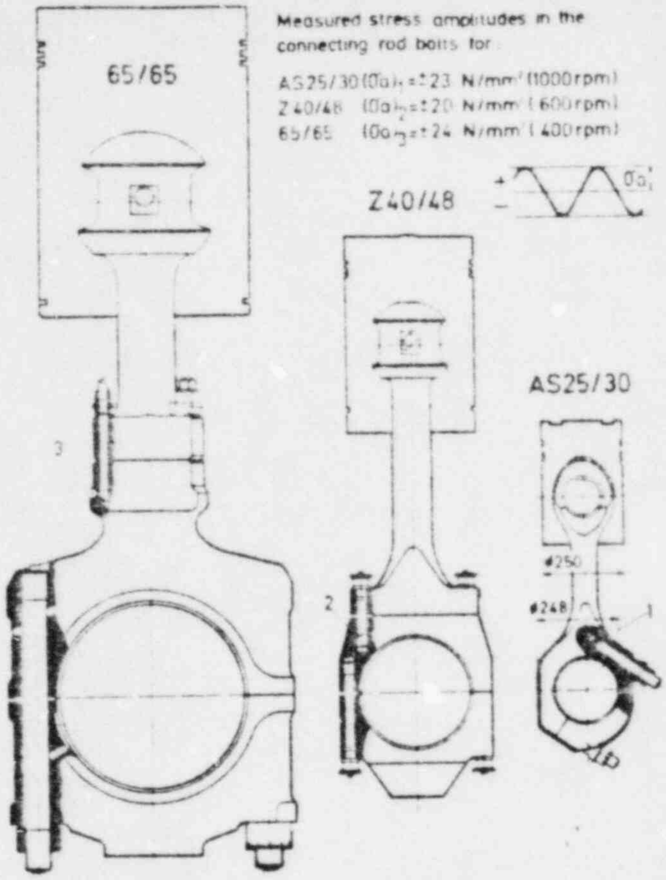


Fig. 13. Comparison of the 65/55, Z40/48 and AS25/30 engine connecting rods.

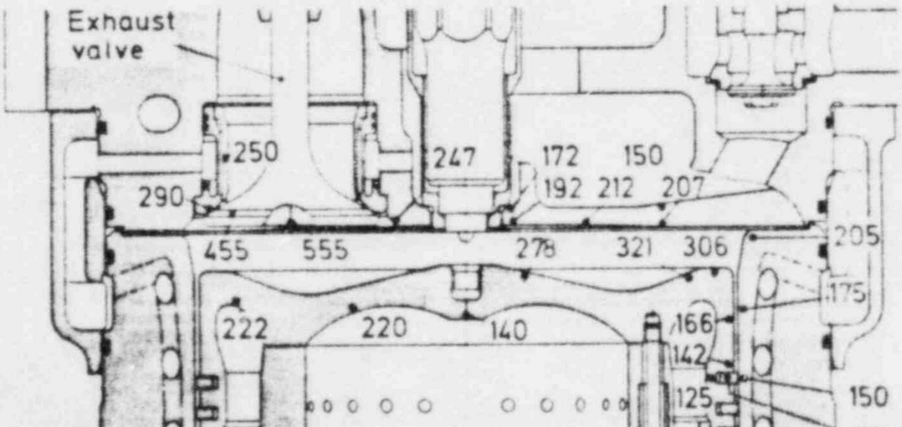


Fig. 14. Measured temperatures (°C) in the combustion chamber area of the Z40/48 engine at operating conditions of 530 rev/min, b.m.e.p. 20 bar, 533 kW/cyl.

At an early design stage, it was realised that the usual design practice, incorporating valve cages, was not the best solution for an engine of the size of the Z 40/48. The cylinder head was therefore provided with press-fitted valve seat inserts (Fig. 15), a design which offers the following advantages:

- The valve seat is fully symmetrical. Therefore, under preloading or gas load, full symmetry of deformation is achieved.
- The cooling of the seat area is very efficient. The temperature distribution around the seat (Fig. 15) is uniform and leads to good sealing properties.
- The water passing through the valve seat inserts flows directly to the centre of the flame plate, ensuring effective cooling of this area.
- Finally, higher gas passage areas are possible in comparison with a cylinder head having valve cages.

The press-fitted valve seat contains an additional inner sleeve in order to prevent sulphur corrosion of the lateral seat surface. This is achieved by the insulating air gap between the overcooled region of the seat and the gas passage. The reliability of the valve for heavy fuel operation was considerably increased by the application of a specially-developed and patented plasma coating for the valve seat. In addition, the Z 40/48 engine is supplied with specially-designed tools for rapid dismantling of the cylinder head during regular inspections.

The application of the above mentioned features made it possible to extend the overhaul intervals of the exhaust valves. On the basis of service experience gained with overhaul intervals initially fixed at 6 000 hours, it was decided by the end of 1976 to extend the recommended valve overhaul periods to between 12 000 and 18 000 hours. This period is also customary for pulling the pistons. This eliminates altogether basic need of valve cages. Subsequent service experience with the exhaust valves on the Z 40/48 engine has generally confirmed the reliability of this design. The few valve failures experienced were explained by faults during the manufacturing process or because non-approved valve makes were used, which did not meet Sulzer requirements.

65/65 wall temperatures:

Because similar principles adopted in the Z 40/48 were applied to the 65/65 engine, the maximum temperatures are remarkably low for an engine of this size. Unlike the Z 40/48 design, however, valve cages are provided on the 65/65 engine in order to keep the fitting and dismantling intervals of the valves acceptably low (the weight of the cylinder head is considerably higher).

The rotating piston

In view of the large power range covered by Sulzer medium speed engines, two different piston designs have been adopted:

- **The requirements of the AS 25/30 engine led to the use of a conventional light alloy gudgeon pin piston (Fig. 16, right), whereby the piston skirt is lubricated by splash oil. The lubricating oil consumption, which ranges between 1·3 to 2·0 g/kWh (1·0 to 1·5 g/bhph), is controlled by the ring package placed above the pin.**

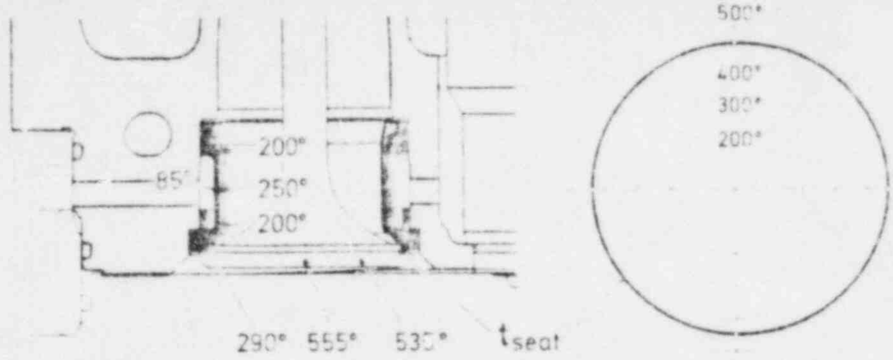


Fig. 15. Z40/48 engine exhaust valve seat temperatures (°C), engine running at 530 rev/min, b.m.e.p. 21 bar

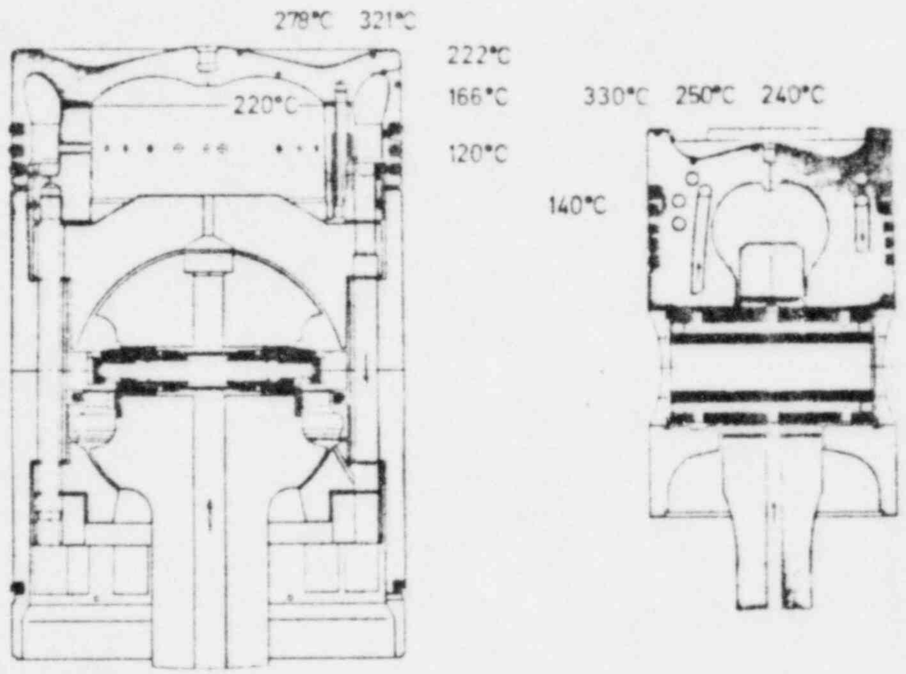


Fig. 16. The rotating piston design of the Z40/48 (left) indicating the temperatures (°C) measured at operating conditions of 530 rev/min, 533 kW/cyl (725 bhp/cyl). Piston with gudgeon pin of the AS 25/30 (right) with temperatures measured at 1 000 rev/min, 200 kW/cyl (270 bhp/cyl)

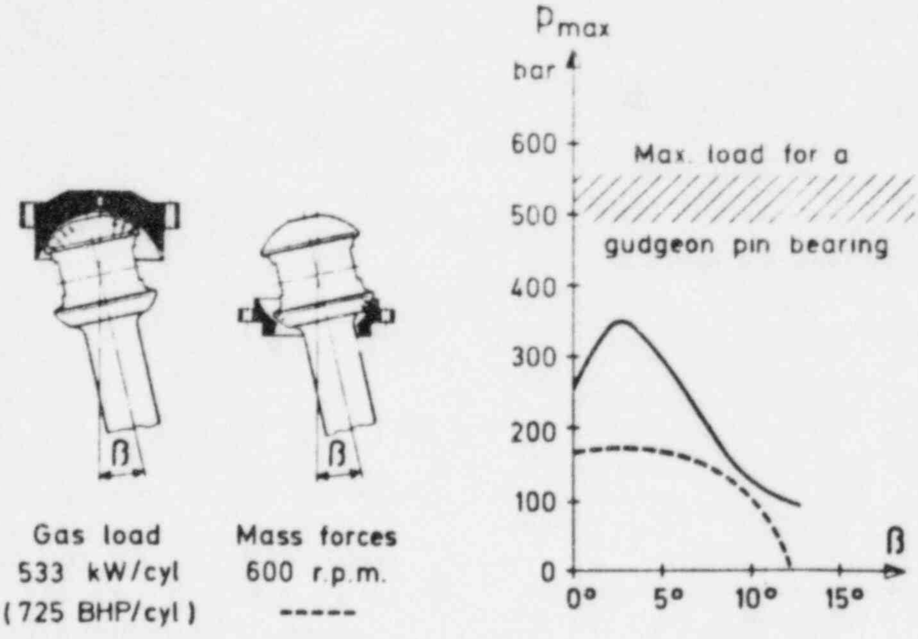


Fig. 17. Specific load of the spherical bearing of the Z40/48 engine.

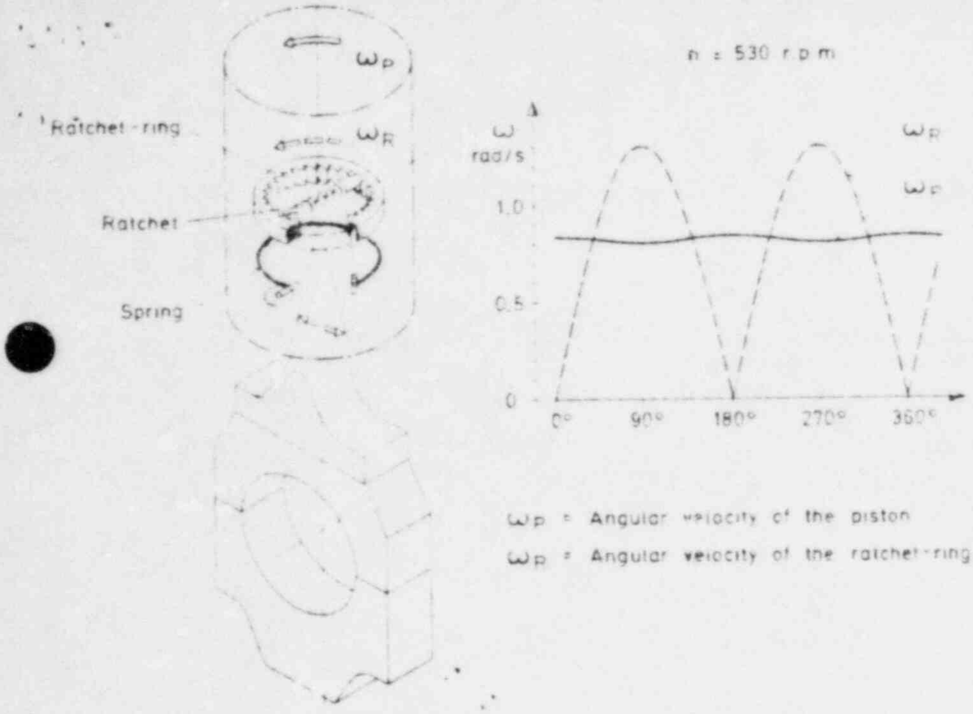


Fig. 18. Rotating mechanism of the Z40/48 piston (left) and a schematic representation showing its very low loading

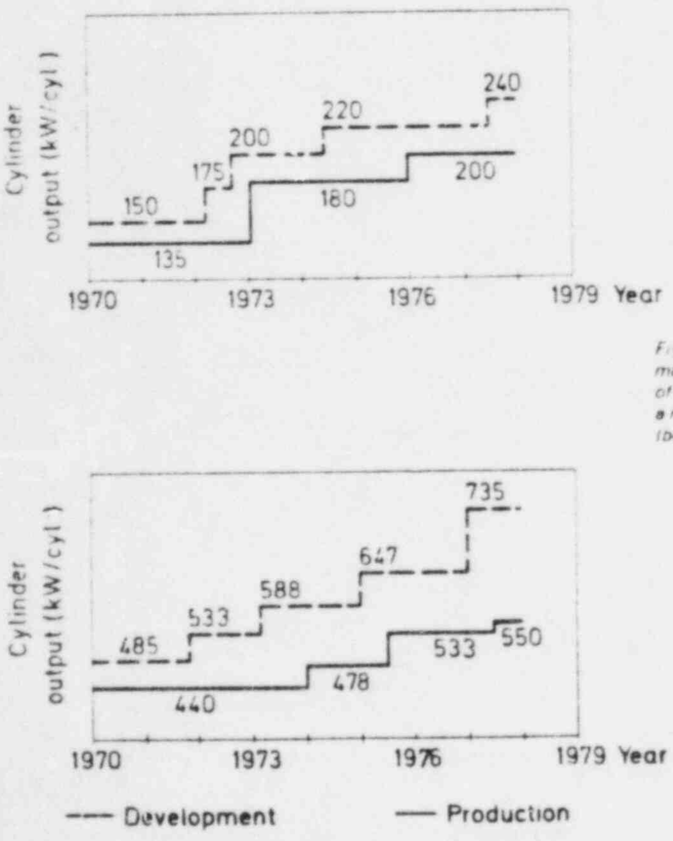


Fig. 19. Evolution of maximum cylinder output of the AS 25 engine (top) and Z V 40/48 engine (bottom).

- The pistons of larger engines are more prone to piston seizure because of the higher deformations involved. The risk of seizure is aggravated by the customer's demand for low lubricating oil consumption and by the requirement to burn low quality heavy fuels.

In order to solve these problems and to satisfy the demands connected with high specific output and good reliability, the well-known rotating piston design was adopted for the Z 40/48 (Fig. 16, left) as well as for the larger 65/65 engine. The advantages of such a design are that local overheating is avoided, due to the rotary movement. Temperatures are symmetrically distributed and thermal deformations are also symmetrical. Thanks to the spherical bearing and the symmetrical form of the skirt, deformations due to gas pressure are symmetrical as well. With every stroke, a fresh oil-wetted part of the skirt is turned into the load-carrying zone, substantially reducing the danger of seizure. Furthermore, piston clearance can be reduced to a minimum due to the symmetrical shape of the skirt.

An improved mechanical reliability is also incorporated in the rotating piston, due to the following characteristics:

- The maximum specific load of the spherical bearing under gas forces is about 30 to 40 per cent lower than in the case of the small end bearing of a conventional gudgeon pin. It is also more evenly distributed (Fig. 17).
- The loading of the rotating mechanism is very low as explained schematically in Fig. 18: the oscillating motion of the connecting rod shaft imposes an intermittent rotation to the ratchet ring. On the other hand, a periodic acceleration and deceleration of the heavy piston skirt should be avoided in order to prevent damage of the rotating mechanism due to high inertia forces. For this purpose, the intermittent motion of the ratchet ring is transmitted to the piston skirt over a circular spring. The spring accumulates the energy transmitted periodically by the rotating mechanism and delivers it smoothly to the piston skirt. According to Fig. 18, the intermittent rotation of the ratchet ring (ω_R) is transformed into an almost steady rotation movement (ω_P) of the piston skirt around its axis. Due to the elimination of inertial forces, the forces induced at the ratchet ring periphery are low.

Lubricating oil consumption: this consideration is one of the most exacting problems which have to be solved during the development of a trunk-piston engine. The unique features of the rotating piston have permitted a much better control of lubricating oil consumption because of the following factors:

- The absolutely symmetrical conditions of the piston skirt as well as the small running clearances result in a reduced piston slap, which can be dampened with comparatively smaller amounts of oil.
- The reduced danger of seizure due to the rotating movement permits a low lubricating oil rate.

In addition, the lubricating oil consumption is exactly controlled by a separate cylinder lubrication system: the rate is automatically adjusted, as a function of the engine load. As a result, the total lubricating oil consumption of the Z 40/48 engine remains below 1·3 g/kWh (1·0 g/bhph). This performance is fully confirmed by long-term service experience with the engine.

Testbed results

As already mentioned, a high power reserve was incorporated in Sulzer engines at the design stage, a principle which allowed the engine to be introduced at a reasonably safe output level in order to gain experience for further power increases. At the same time, tests with much higher ratings were carried out on the testbed. Fig. 19 illustrates the evolution of testbed performance against the production ratings of the AS 25/30 and Z 40/48 engines. The even, increased lead time basically results in successive improve-

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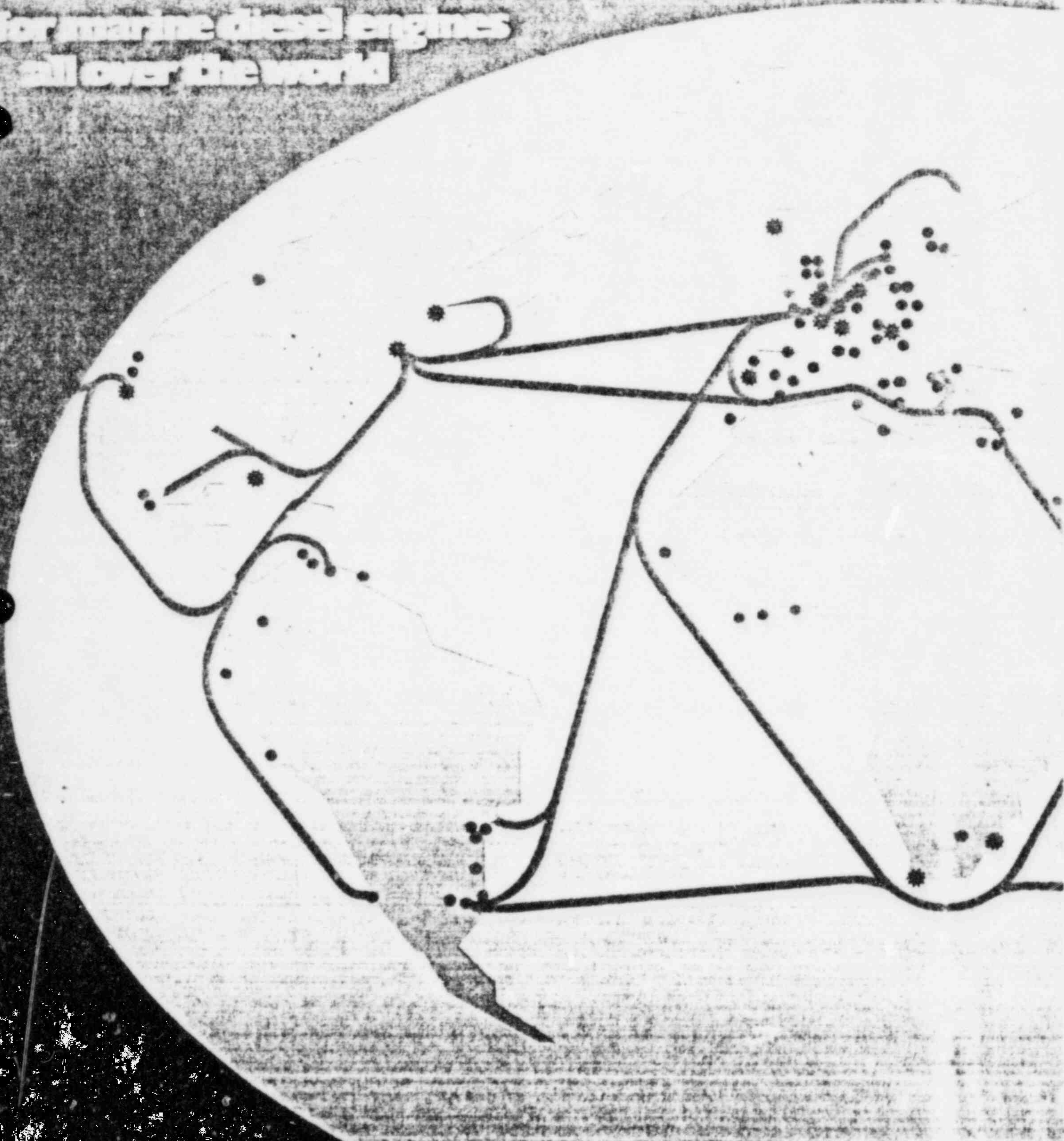

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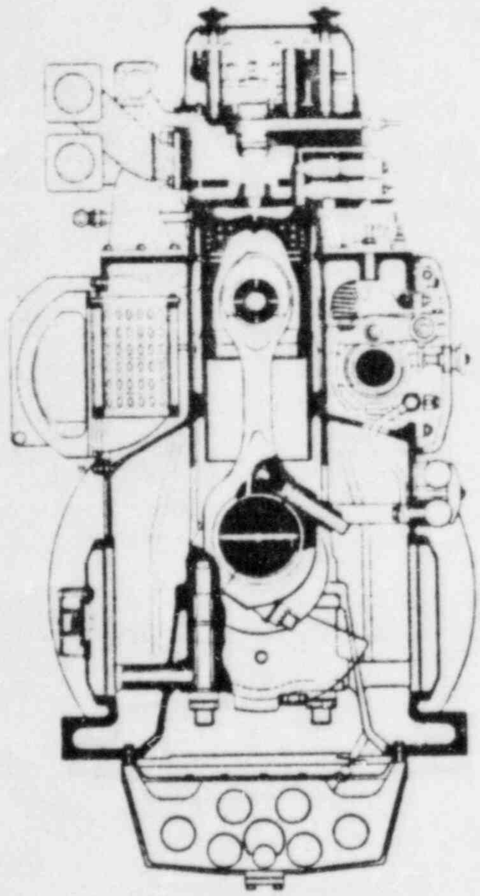

Fig. 1 Cross-section of the ASL 25/30 engine

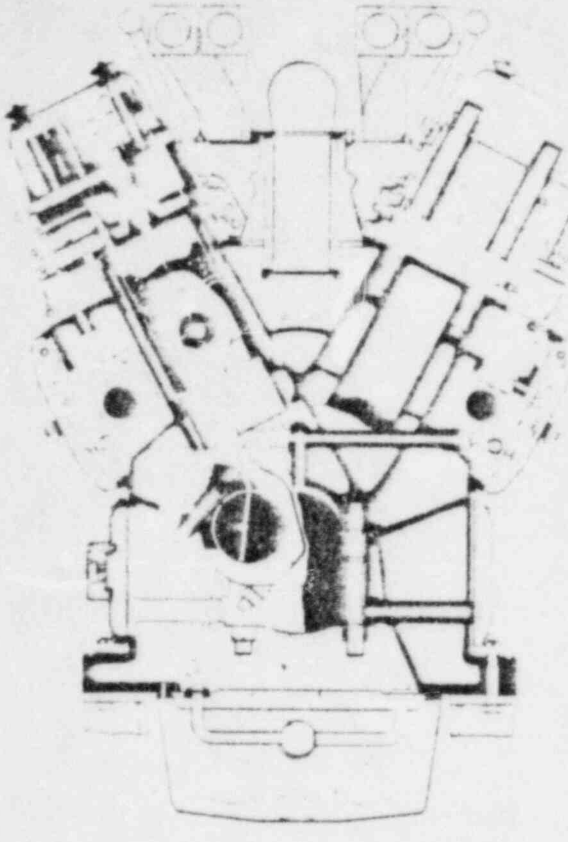

Fig. 2. Cross-section of the ASV 25/30 engine

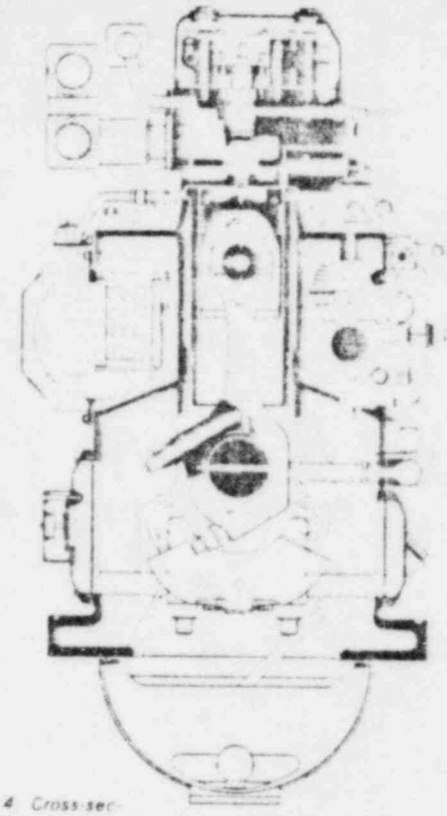

Fig. 4. Cross-section of the AL 20/24 engine.

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Fig. 3. A 16-cylinder ASV 25/30 engine

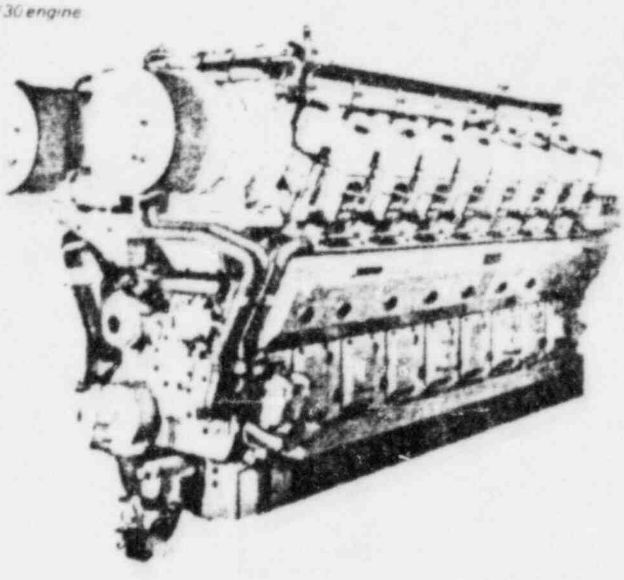

Table 1: The Sulzer medium speed engine range.

Engine Type	Bore/Stroke (mm)	Speed rev/min	b.m.e.p. (bar)	Cyl. output (kW)
		1000	16·29	200
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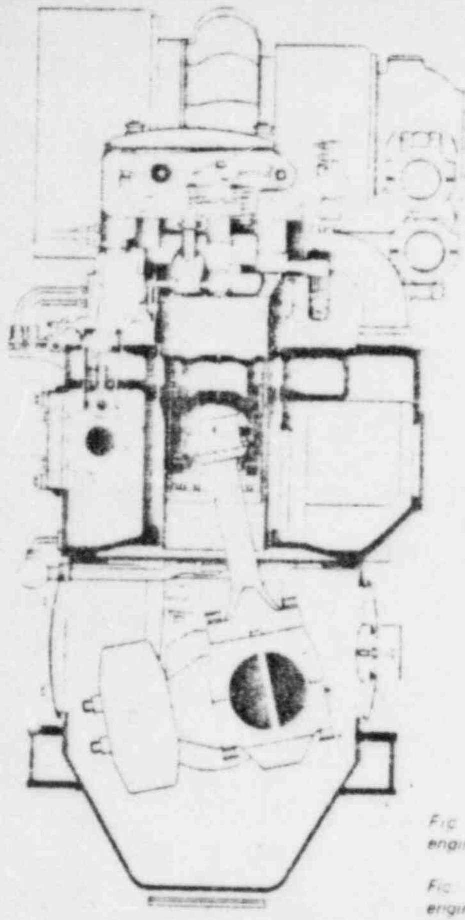
		750	17·38	160
AL 20/24	200/240	1000	16·31	102·5
Z 40/48	400/480	560*	19·54	550
		530*	20·02	533
65/65	650/650	400	18·43	1 325

* Maximum engine speed for stationary applications is 600 rev/min.



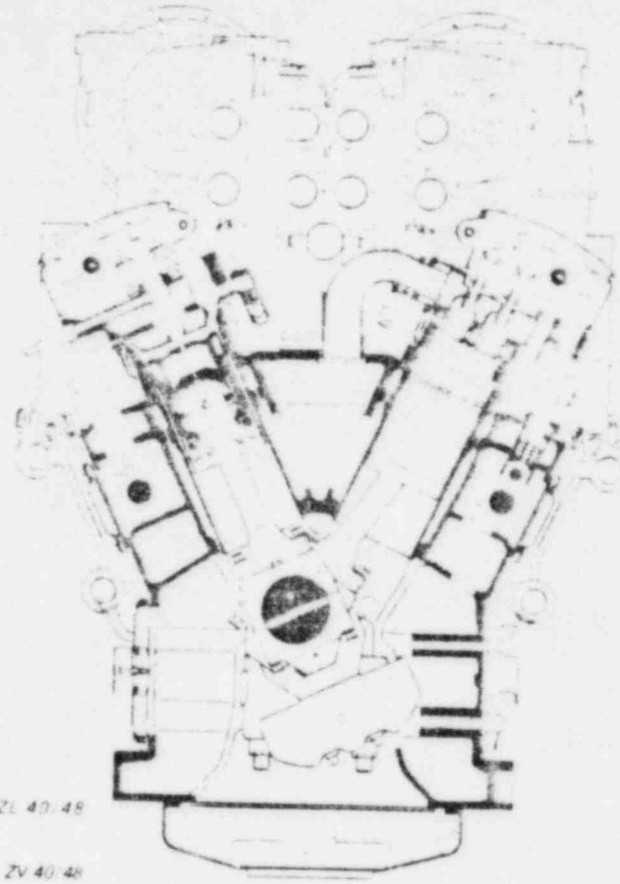
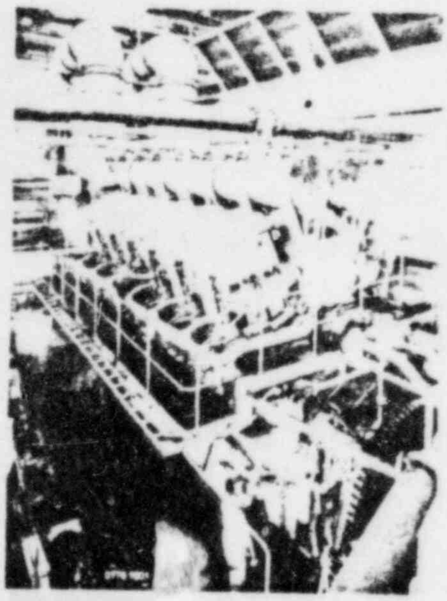
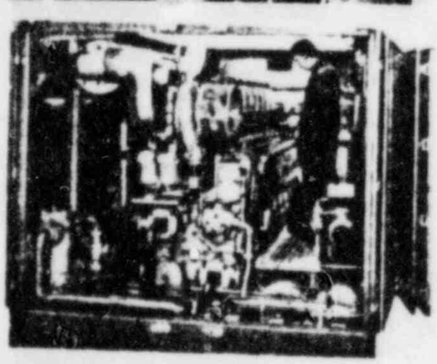
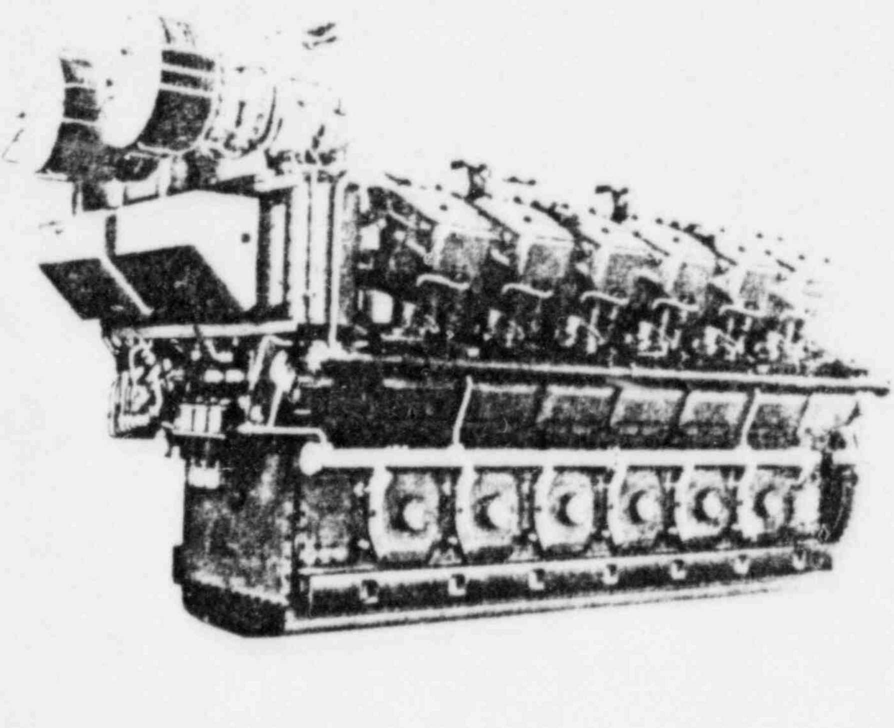


Fig. 6. (Left) Cross-section of ZL 40/48 engine

Fig. 7. (Right) Cross-section of ZV 40/48 engine







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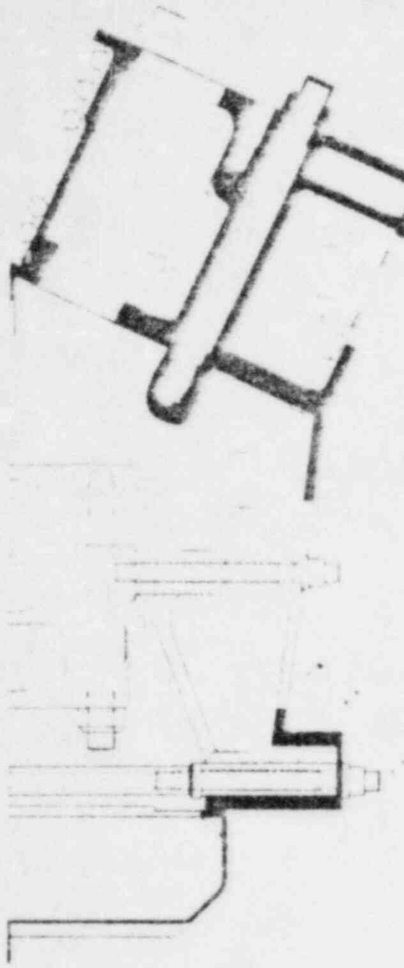


Fig. 12 Engine casing design of the V65/65 engine

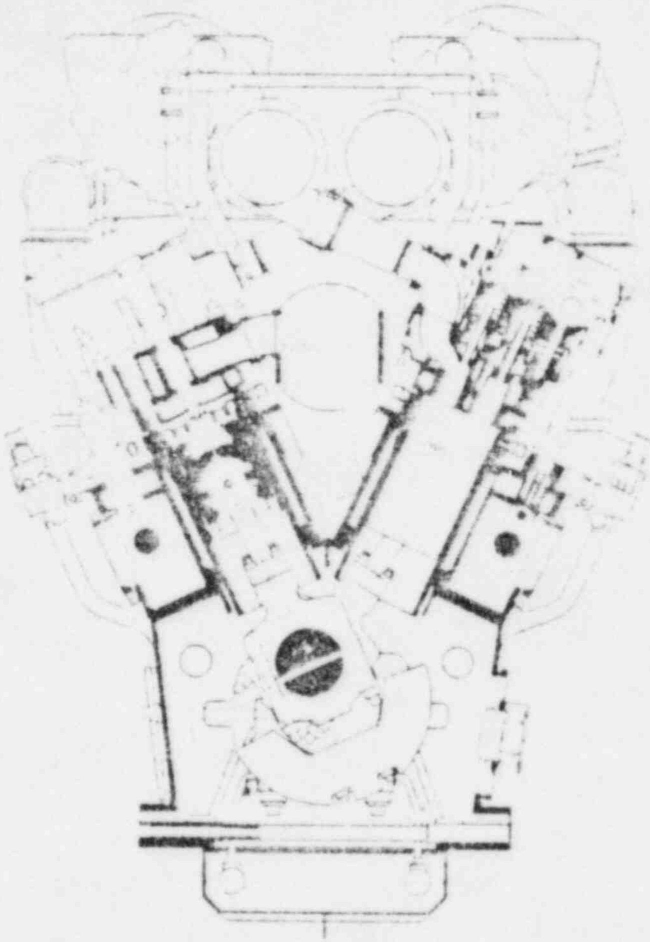
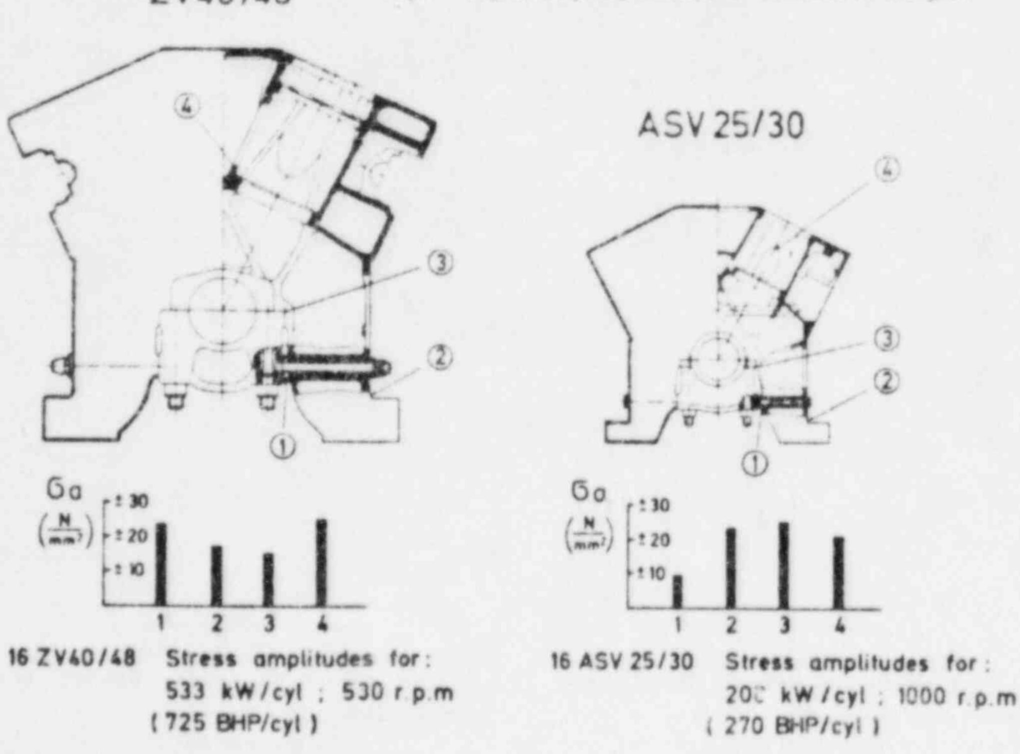


Fig. 9 Cross-section of V65/65 engine

Fig. 11 Engine casing design of the ZV 40/48 and ASV 25/30 engines

ZV40/48



The following discussion presents some of the specific features of the Sulzer four-stroke engines. Special emphasis will be placed on the Z 40/48 engine, which is the "backbone" of our four-stroke programme.

Low level of mechanical stresses

The incorporation of a high power reserve for future development allows the engine to enter production at a reasonably safe level. Furthermore, the attainment of a higher cylinder output in the course of the final development stages does not negatively affect the operational safety, but represents only the full utilisation of the reliability designed into the engine at an early stage. A low stress level in the major engine components is one of the necessary requirements.

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The stress level in the casing of the 65/65 engine is also at a similarly low level.

A new solution was sought for the connecting rod of the 65/65 in order to simplify overhaul work. The shaft is extremely short, allowing the dismantling height of the piston to be reduced to a minimum. The big end is bolted to the shaft by means of eight hydraulically tightened bolts, which are readily accessible.

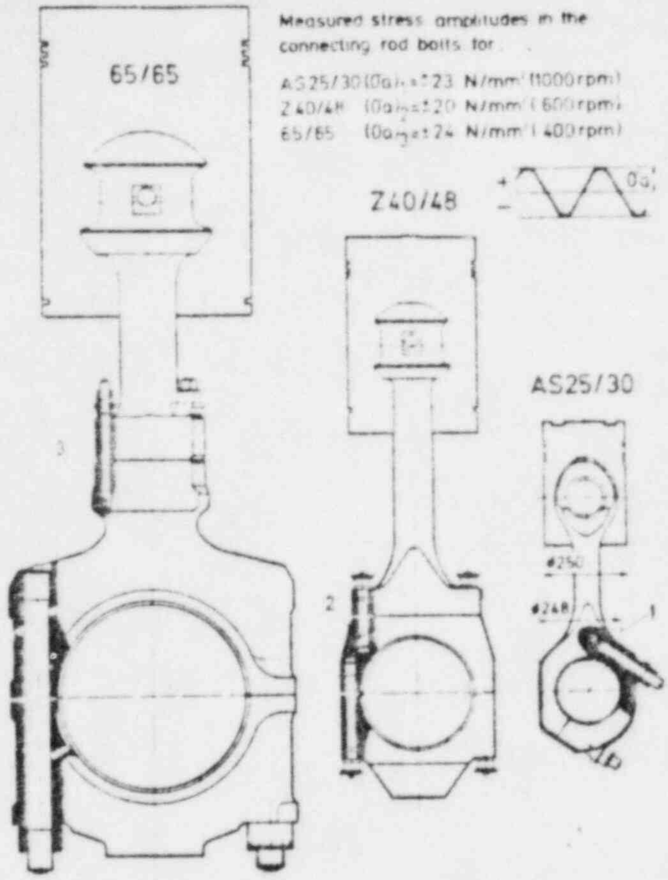


Fig. 13. Comparison of the 65/65, Z40/48 and AS25/30 engine connecting rods

Low thermal loading

Z 40/48 wall temperatures: an engine with high specific output intended to burn low quality heavy fuels must be provided with adequately-cooled combustion chamber walls in order to avoid thermal cracks and high temperature corrosion. Fig. 14 shows the combustion chamber temperatures of the Z 40/48 engine. The cylinder liner temperatures are kept at an adequate level due to the application of the bore-cooling principle developed at Sulzer many years ago.

The design principle of the double bottom has been applied to the cylinder head and the thin flame plate is intensively cooled. The mechanical loads due to gas pressure are carried by the massive intermediate supporting deck.

Z 40/48 valve seat design: the exhaust valve is one of the most critical parts of the cylinder head, if not of the entire medium speed engine. For reliable performance, a perfect sealing of the seat as well as efficient cooling are necessary. In the case of the Z 40/48, a thorough investigation was carried out in order to find the best design for an engine with a high reliability and capable of burning heavy fuels up to 3 500 sec. Red. I with high sodium and vanadium contents.

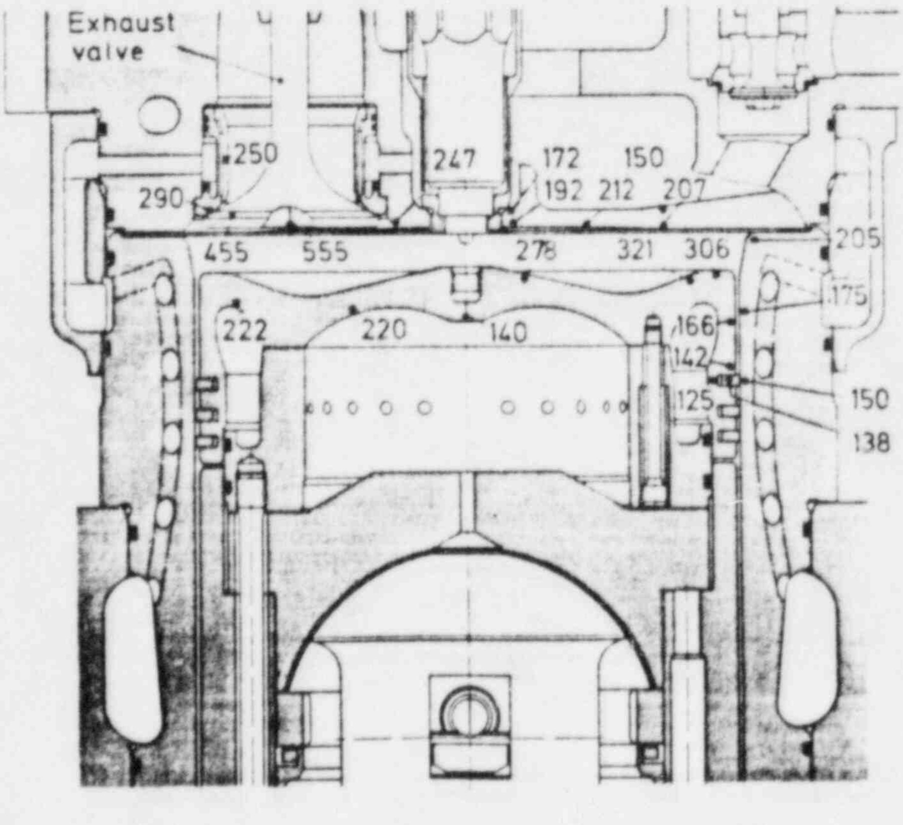


Fig. 14. Measured temperatures (°C) in the combustion chamber area of the Z40/48 engine at operating conditions of 530 rev/min, b.m.e.p. 20 bar, 533 kW/cyl).

At an early design stage, it was realised that the usual design practice, incorporating valve cages, was not the best solution for an engine of the size of the Z 40/48. The cylinder head was therefore provided with press-fitted valve seat inserts (Fig. 15), a design which offers the following advantages.

- The valve seat is fully symmetrical. Therefore, under preloading or gas load, full symmetry of deformation is achieved.
- The cooling of the seat area is very efficient. The temperature distribution around the seat (Fig. 15) is uniform and leads to good sealing properties.
- The water passing through the valve seat inserts flows directly to the centre of the flame plate, ensuring effective cooling of this area.
- Finally, higher gas passage areas are possible in comparison with a cylinder head having valve cages.

The press-fitted valve seat contains an additional inner sleeve in order to prevent sulphur corrosion of the lateral seat surface. This is achieved by the insulating air gap between the overcooled region of the seat and the gas passage. The reliability of the valve for heavy fuel operation was considerably increased by the application of a specially-developed and patented plasma coating for the valve seat. In addition, the Z 40/48 engine is supplied with specially-designed tools for rapid dismantling of the cylinder head during regular inspections.

The application of the above mentioned features made it possible to extend the overhaul intervals of the exhaust valves. On the basis of service experience gained with overhaul intervals initially fixed at 6 000 hours, it was decided by the end of 1976 to extend the recommended valve overhaul periods to between 12 000 and 18 000 hours. This period is also customary for pulling the pistons. This eliminates altogether basic need of valve cages. Subsequent service experience with the exhaust valves on the Z 40/48 engine has generally confirmed the reliability of this design. The few valve failures experienced were explained by faults during the manufacturing process or because non-approved valve makes were used, which did not meet Sulzer requirements.

65/65 wall temperatures:

Because similar principles adopted in the Z 40/48 were applied to the 65/65 engine, the maximum temperatures are remarkably low for an engine of this size. Unlike the Z 40/48 design, however, valve cages are provided on the 65/65 engine in order to keep the fitting and dismantling intervals of the valves acceptably low (the weight of the cylinder head is considerably higher).

The rotating piston

In view of the large power range covered by Sulzer medium speed engines, two different piston designs have been adopted:

- The requirements of the AS 25/30 engine led to the use of a conventional light alloy gudgeon pin piston (Fig. 16, right), whereby the piston skirt is lubricated by splash oil. The lubricating oil consumption, which ranges between 1·3 to 2·0 g/kWh (1·0 to 1·5 g/bhph), is controlled by the ring package placed above the pin.

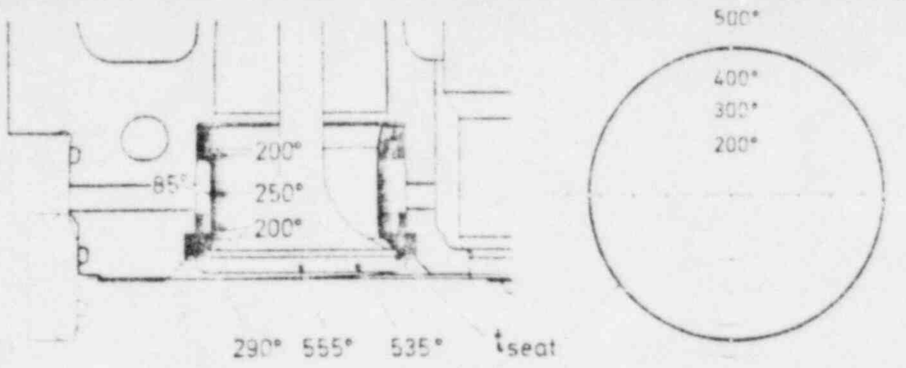


Fig. 15. Z40/48 engine exhaust valve seat temperatures (°C): engine running at 530 rev/min, b.m.e.p. 20 bar

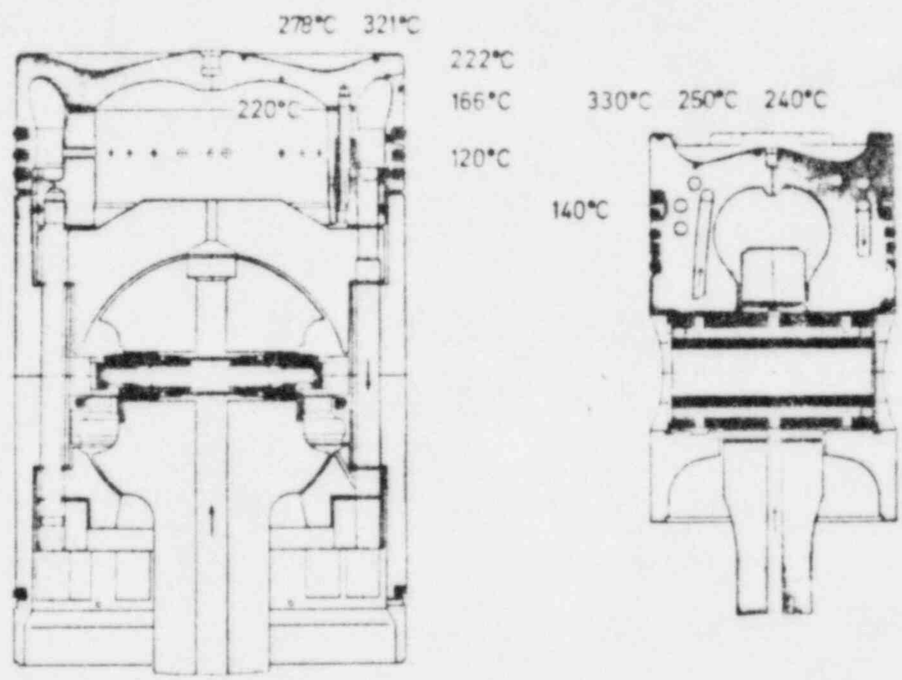


Fig. 16. The rotating piston design of the Z40/48 (left) indicating the temperatures (°C) measured at operating conditions of 530 rev/min, 533 kW/cyl (725 bhp/cyl). Piston with gudgeon pin of the AS 25/30 (right) with temperatures measured at 1 000 rev/min, 200 kW/cyl (270 bhp/cyl).

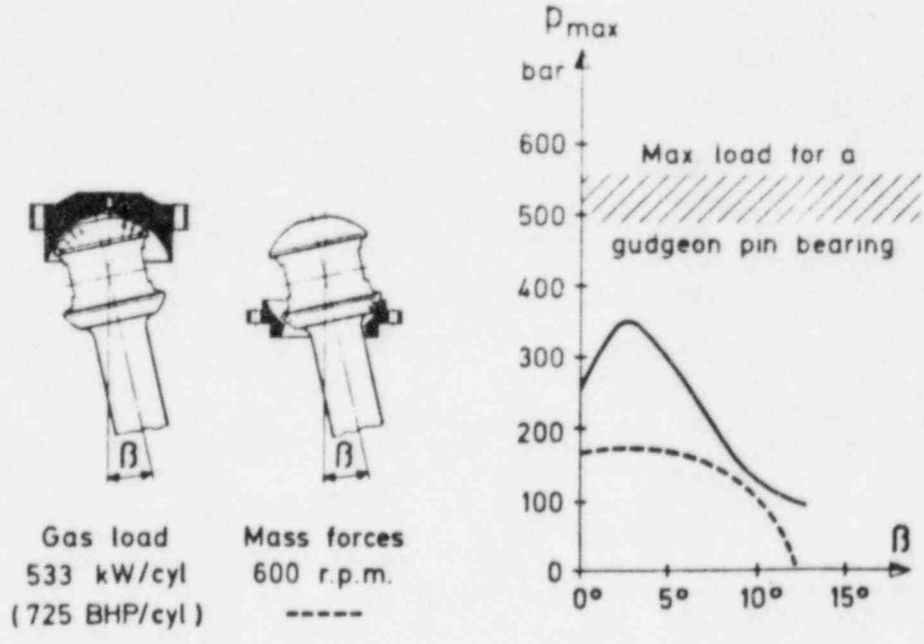


Fig. 17. Specific load of the spherical bearing of the Z40/48 engine.

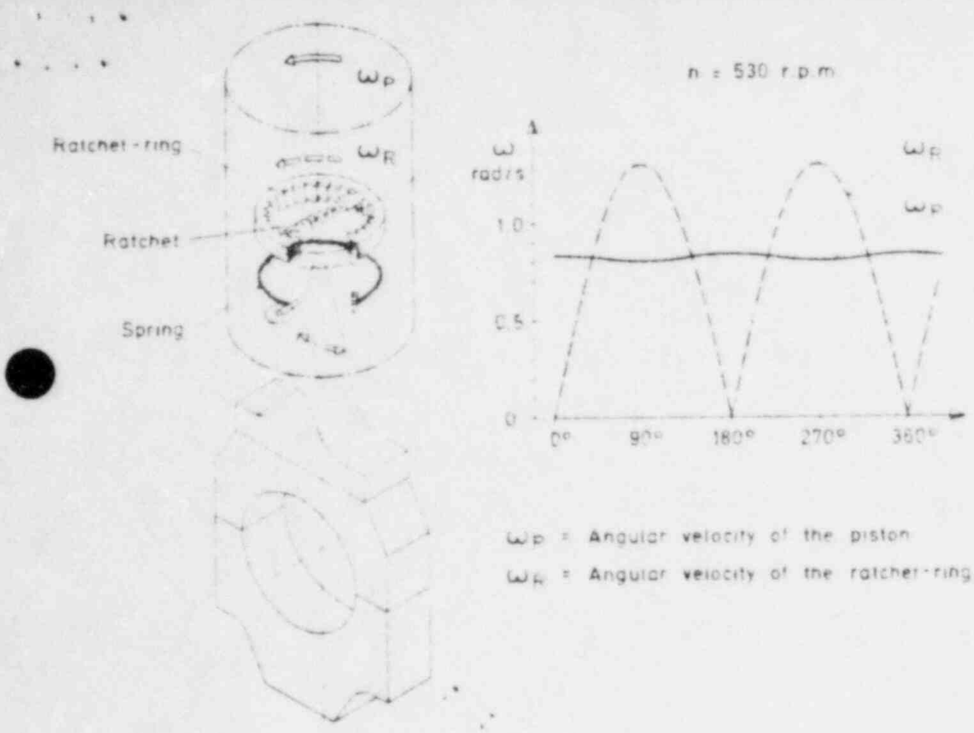


Fig. 18. Rotating mechanism of the Z40/48 piston (left) and a schematic representation showing its very low loading

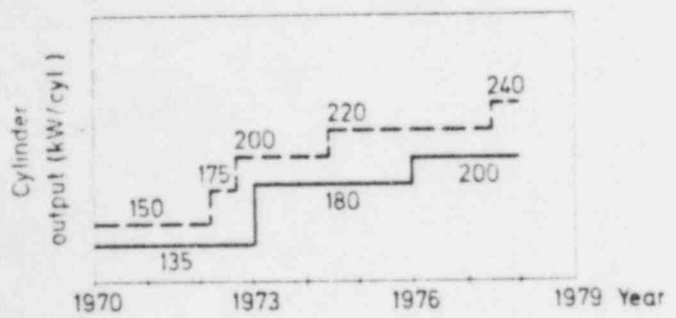
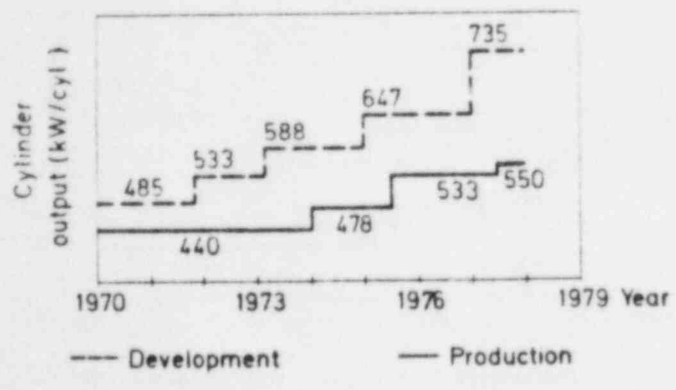


Fig. 19. Evolution of maximum cylinder output of the AS 25 engine (top) and ZV 40/48 engine (bottom)



- The pistons of larger engines are more prone to piston seizure because of the higher deformations involved. The risk of seizure is aggravated by the customer's demand for low lubricating oil consumption and by the requirement to burn low quality heavy fuels.

In order to solve these problems and to satisfy the demands connected with high specific output and good reliability, the well-known rotating piston design was adopted for the Z 40/48 (Fig. 16, left) as well as for the larger 65/65 engine. The advantages of such a design are that local overheating is avoided, due to the rotary movement. Temperatures are symmetrically distributed and thermal deformations are also symmetrical. Thanks to the spherical bearing and the symmetrical form of the skirt, deformations due to gas pressure are symmetrical as well. With every stroke, a fresh oil-wetted part of the skirt is turned into the load-carrying zone, substantially reducing the danger of seizure. Furthermore, piston clearance can be reduced to a minimum due to the symmetrical shape of the skirt.

An improved mechanical reliability is also incorporated in the rotating piston, due to the following characteristics:

- The maximum specific load of the spherical bearing under gas forces is about 30 to 40 per cent lower than in the case of the small end bearing of a conventional gudgeon pin. It is also more evenly distributed (Fig. 17).
- The loading of the rotating mechanism is very low as explained schematically in Fig. 18: the oscillating motion of the connecting rod shaft imposes an intermittent rotation to the ratchet ring. On the other hand, a periodic acceleration and deceleration of the heavy piston skirt should be avoided in order to prevent damage of the rotating mechanism due to high inertia forces. For this purpose, the intermittent motion of the ratchet ring is transmitted to the piston skirt over a circular spring. The spring accumulates the energy transmitted periodically by the rotating mechanism and delivers it smoothly to the piston skirt. According to Fig. 18, the intermittent rotation of the ratchet ring (ω_R) is transformed into an almost steady rotation movement (ω_P) of the piston skirt around its axis. Due to the elimination of inertial forces, the forces induced at the ratchet ring periphery are low.

Lubricating oil consumption: this consideration is one of the most exacting problems which have to be solved during the development of a trunk-piston engine. The unique features of the rotating piston have permitted a much better control of lubricating oil consumption because of the following factors:

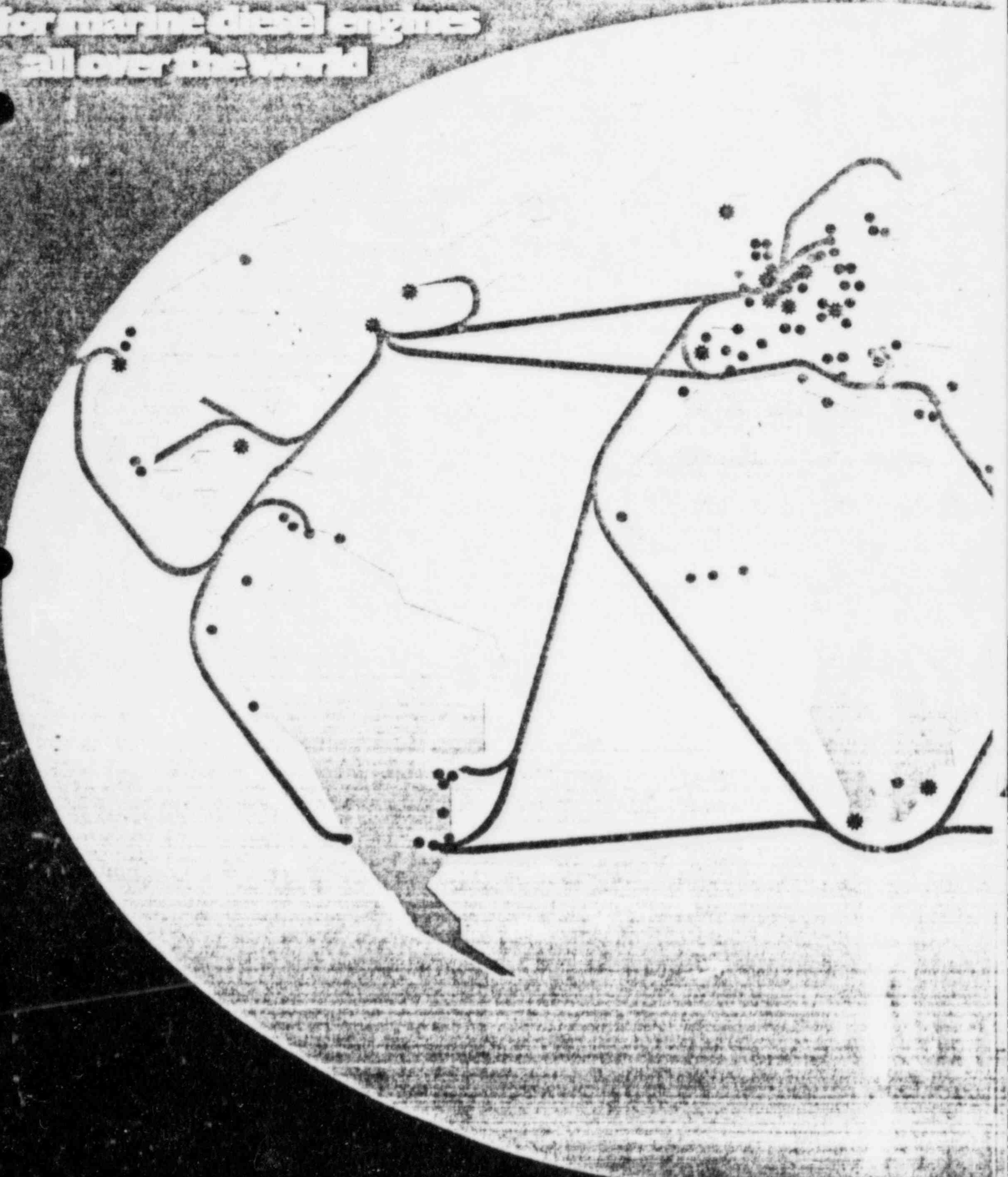
- The absolutely symmetrical conditions of the piston skirt as well as the small running clearances result in a reduced piston slap, which can be dampened with comparatively smaller amounts of oil.
- The reduced danger of seizure due to the rotating movement permits a low lubricating oil rate.

In addition, the lubricating oil consumption is exactly controlled by a separate cylinder lubrication system: the rate is automatically adjusted as a function of the engine load. As a result, the total lubricating oil consumption of the Z 40/48 engine remains below 1·3 g/kWh (1·0 g/bhph). This performance is fully confirmed by long-term service experience with the engine.

Testbed results

As already mentioned, a high power reserve was incorporated in Sulzer engines at the design stage, a principle which allowed the engine to be introduced at a reasonably safe output level in order to gain experience for further power increases. At the same time, tests with much higher ratings were carried out on the testbed. Fig. 19 illustrates the evolution of testbed performance against the production ratings of the AS 25/30 and Z 40/48 engines. The even, increased lead time basically results in successive improve-

MOTOR SHIP



Sulzer's four-stroke high-and medium-speed engine range

by G. Lustgarten* and R. Stoffel**

IN RECENT YEARS the four-stroke diesel engine has consolidated its position in the marine field. Besides its application for auxiliary power generation, the four-stroke engine is an interesting alternative to the two-stroke crosshead diesel where limited space is available for the propulsion unit, for example in roll-on roll-off ships, passenger vessels and ferries. Sulzer has systematically worked to offer an engine programme to meet such demands.

The power range between 620 kW and 15 900 kW is covered by three engine types.

- A-engines for outputs up to 3 600 kW (4 860 bhp)
- Z-engines for outputs up to 9 600 kW (13 050 bhp)
- 65/65 engine for outputs up to 23 850 kW (32 400 bhp).

* Head of development and design, four-stroke engines

** Head of development test beds

The first two Sulzer-designed types have now reached an advanced stage with wide service and testbed experience earned over the past ten years. The 65/65 engine—a joint development of Sulzer and M.A.N.—had completed an intensive development programme by the end of 1977.

The AS25/30 and AL20/24 engines

The AS 25/30 design is built in six, eight and 10 cylinder in-line and 12,16 and 18V versions, Figs. 1, 2 and 3. The main features of the engine are a simple and robust design, good accessibility to the parts subject to wear and easy maintenance.

The original cylinder output of the two-valve A 25/30 engine was 135 kW/cyl (184 bhp/cyl). During subsequent development stages, however, the AS 25/30 engine was provided with four valves and the cylinder output increased to 200 kW/cyl (270 bhp/cyl). In the course of developing this originally Sulzer-designed engine, the AS 25/30 was incorporated into the common engine programme with M.A.N., which has also taken up its production. Extensive endurance tests at 220 kW/cyl (300 bhp/cyl) have already been carried out with a view to further uprating. By the middle of 1977 over 1 350 A 25/30 and AS 25/30 engines were in service and the maximum running hours had exceeded 35 000 hours.

When first introduced, the A 25/30 engine was used mainly for auxiliary power generation. Due to the increase in output as well as the subsequent development of the larger V-cylinder units, the AS 25/30 range has also gradually made its mark as a propulsion engine. The smaller AL 20/24 engine (Fig. 4), based on a similar design, is being manufactured in six- and eight-cylinder in-line versions for auxiliary power generation, the propulsion of small ships and for power packages (as illustrated by the 500 kW container unit in Fig. 5).

The Z 40/48 engine

The Z 40/48 engine is offered in six- and eight-cylinder in-line form and 10, 12, 14, 16 and 18 cylinder V-versions (Fig. 6, 7 and

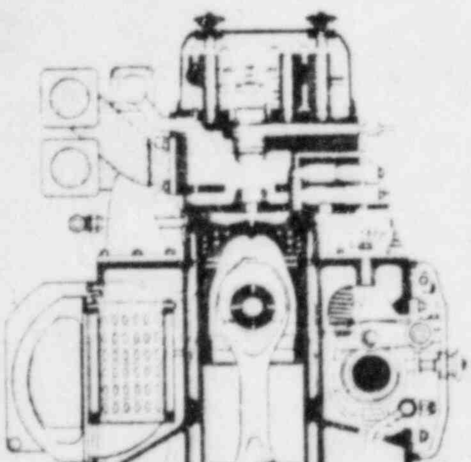


Fig. 1. Cross-section of the ASL 25/30 engine

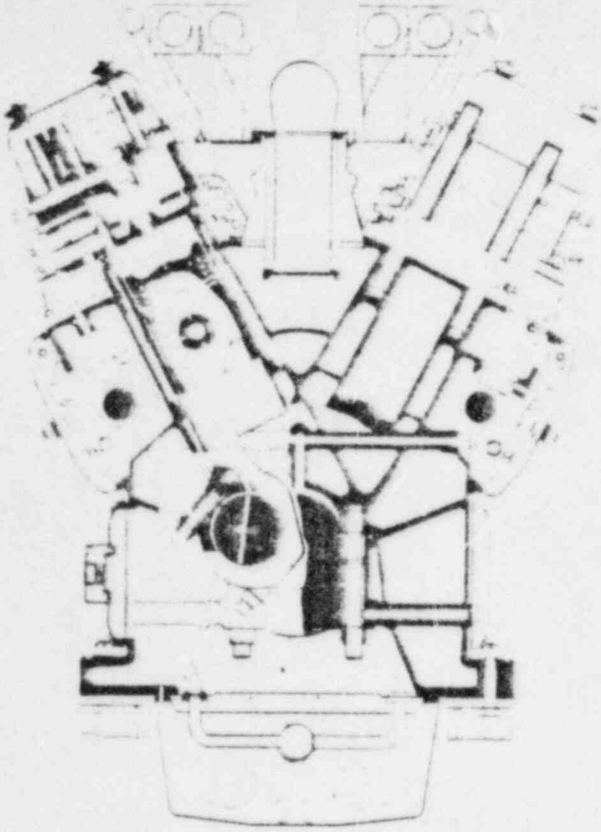
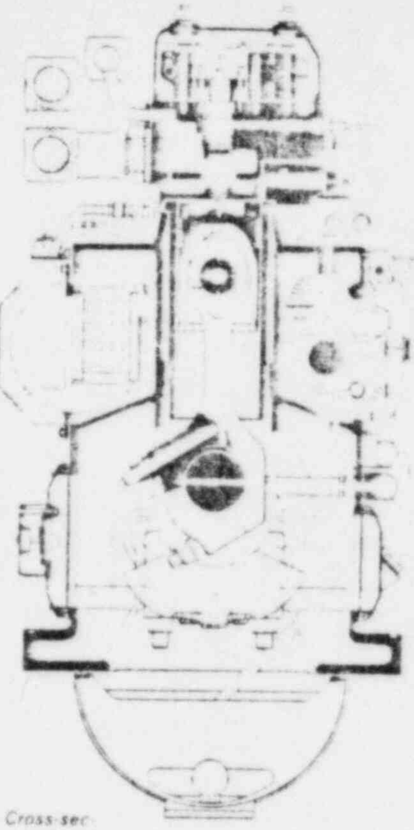


Fig. 2. Cross-section of the ASV 25/30 engine



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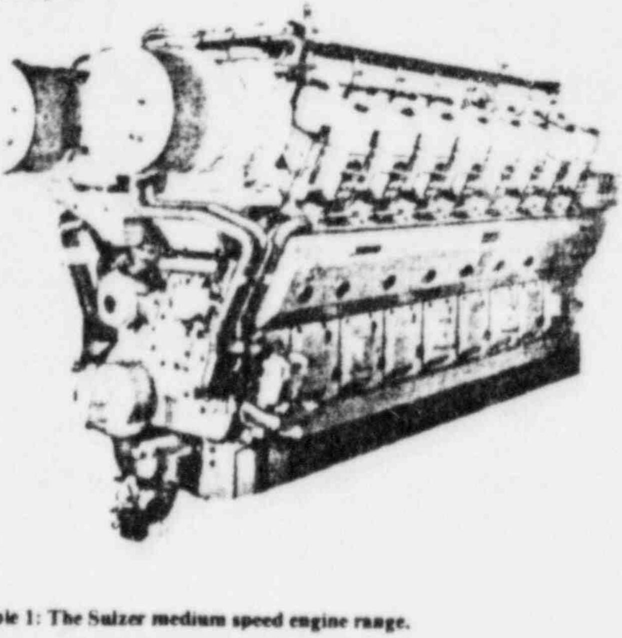

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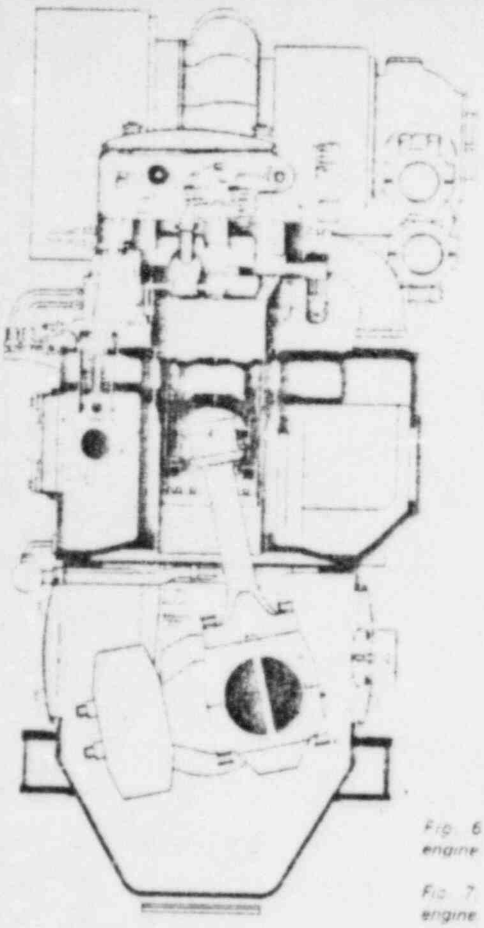

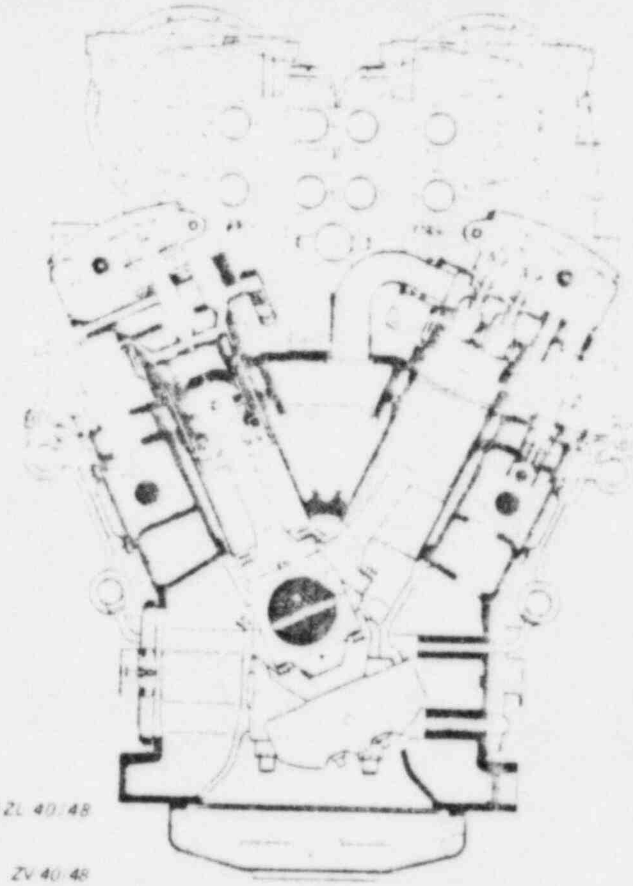

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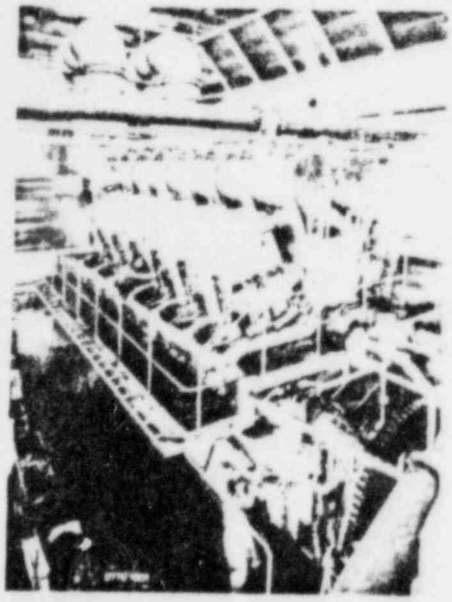

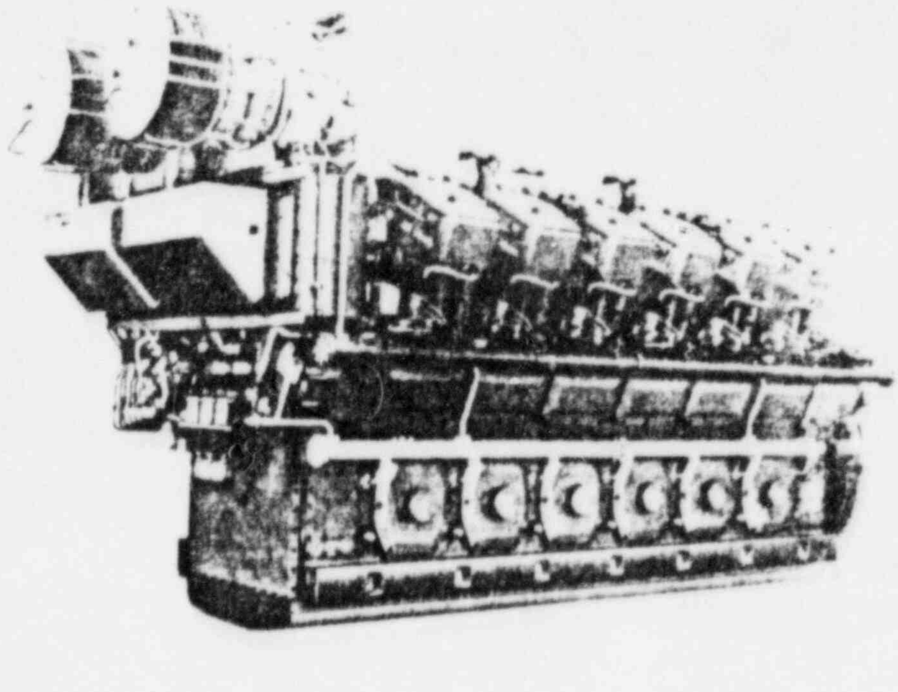

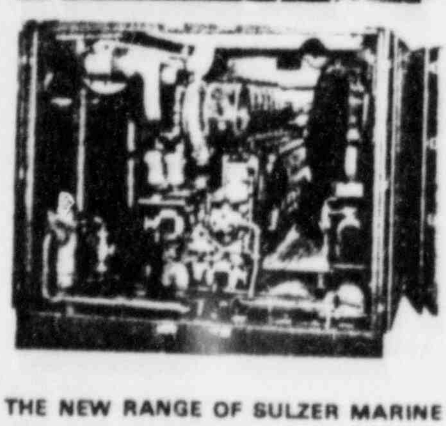

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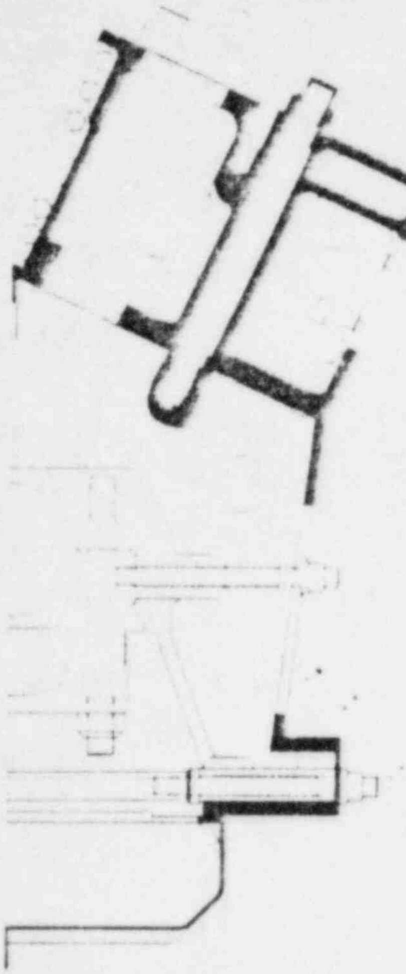

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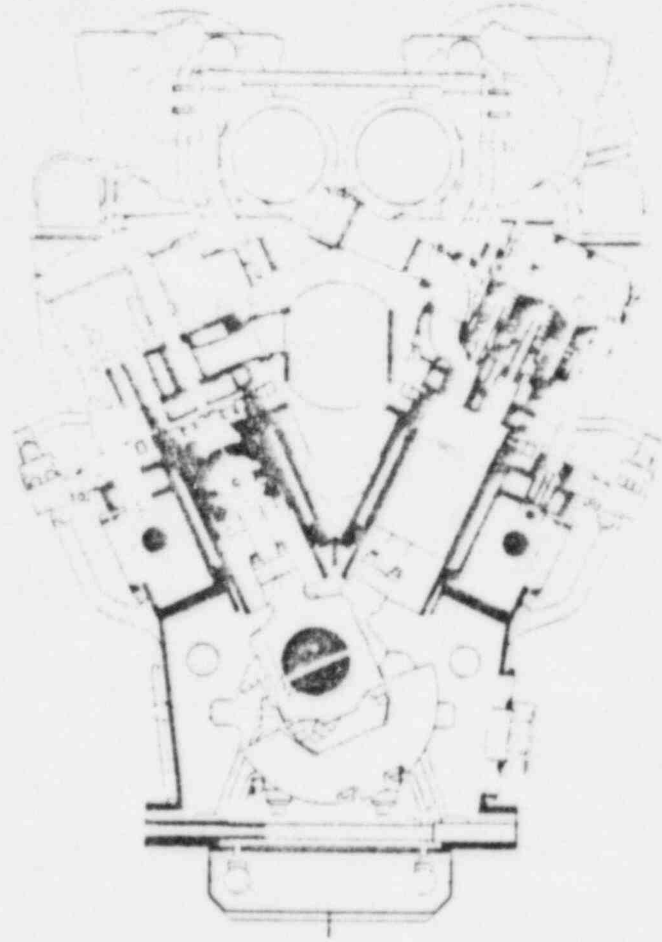

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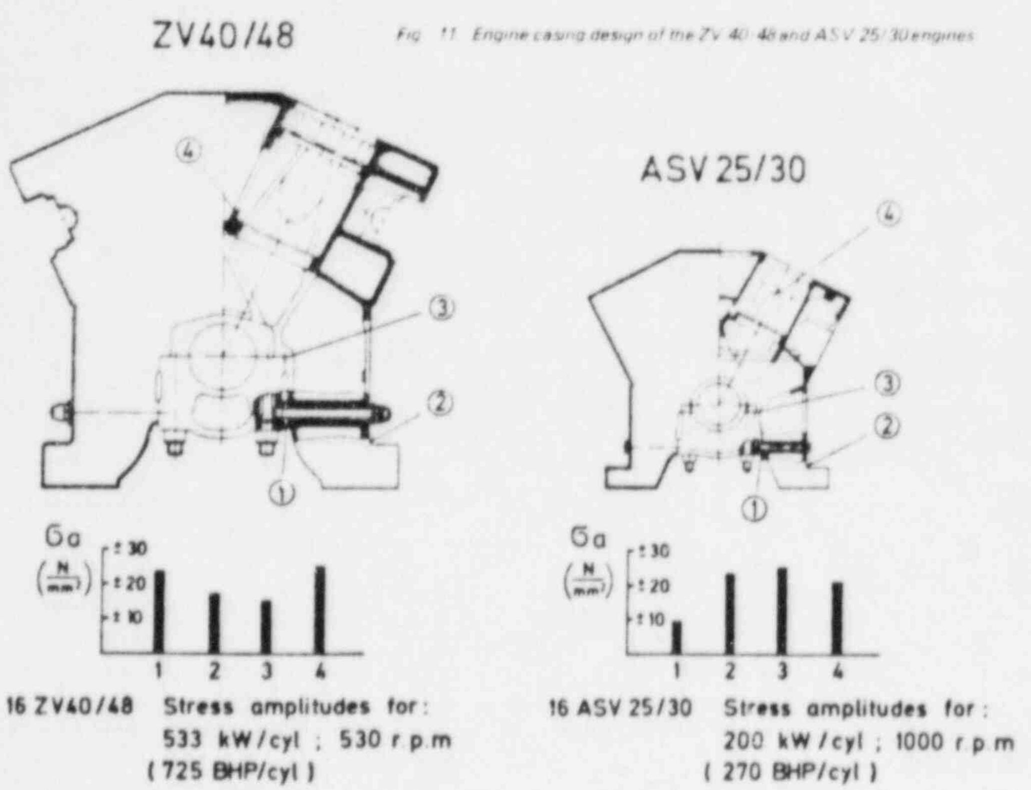

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Low thermal loading

Z 40/48 wall temperatures: an engine with high specific output intended to burn low quality heavy fuels must be provided with adequately-cooled combustion chamber walls in order to avoid thermal cracks and high temperature corrosion. Fig. 14 shows the combustion chamber temperatures of the Z 40/48 engine. The cylinder liner temperatures are kept at an adequate level due to the application of the bore-cooling principle developed at Sulzer many years ago.

The design principle of the double bottom has been applied to the cylinder head and the thin flame plate is intensively cooled. The mechanical loads due to gas pressure are carried by the massive intermediate supporting deck.

Z 40/48 valve seat design: the exhaust valve is one of the most critical parts of the cylinder head, if not of the entire medium speed engine. For reliable performance, a perfect sealing of the seat as well as efficient cooling are necessary. In the case of the Z 40/48, a thorough investigation was carried out in order to find the best design for an engine with a high reliability and capable of burning heavy fuels up to 3 500 sec. Red. 1 with high sodium and vanadium contents.

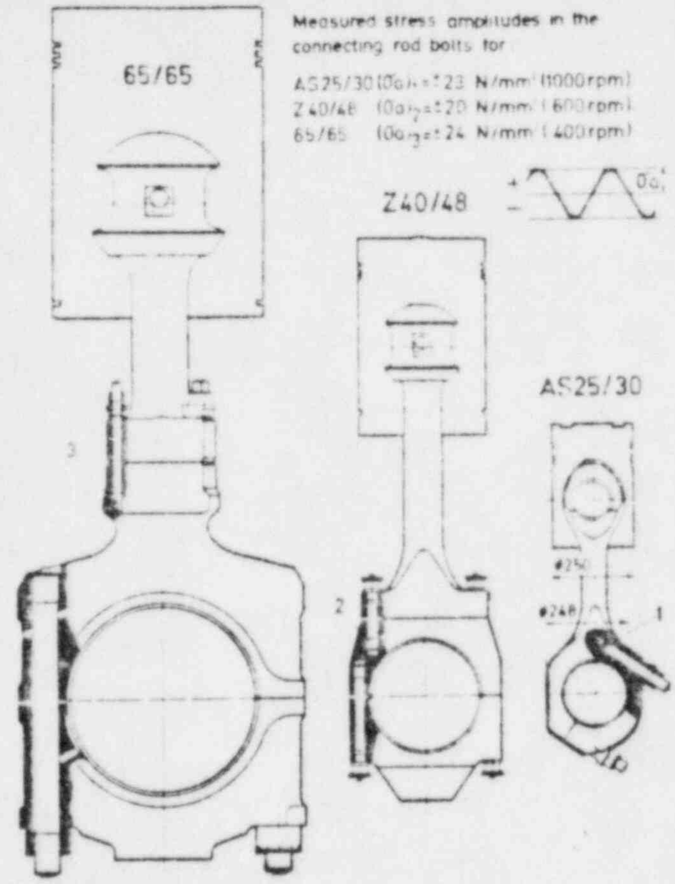

Fig. 13. Comparison of the 65/65, Z40/48 and AS25/30 engine connecting rods

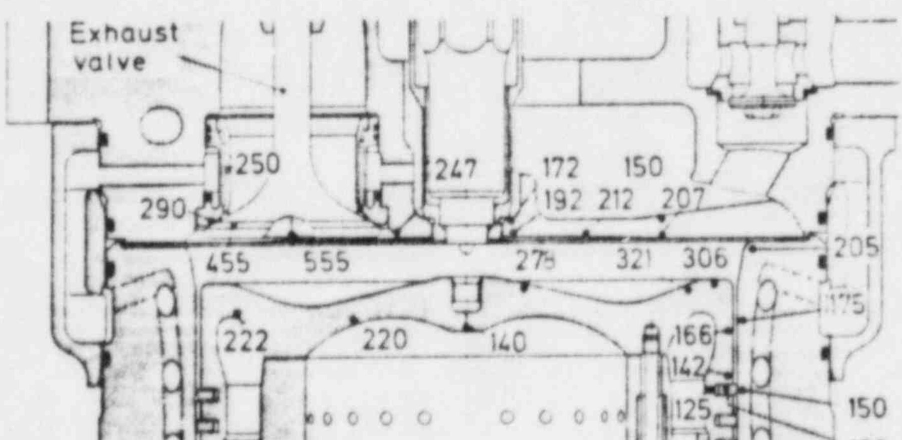

Fig. 14. Measured temperatures (°C) in the combustion chamber area of the Z40/48 engine at operating conditions of 530 rev/min, b.m.e.p. 20 bar, 533 kW/cyl.

At an early design stage, it was realised that the usual design practice, incorporating valve cages, was not the best solution for an engine of the size of the Z 40/48. The cylinder head was therefore provided with press-fitted valve seat inserts (Fig. 15), a design which offers the following advantages:

- The valve seat is fully symmetrical. Therefore, under preloading or gas load, full symmetry of deformation is achieved.
- The cooling of the seat area is very efficient. The temperature distribution around the seat (Fig. 15) is uniform and leads to good sealing properties.
- The water passing through the valve seat inserts flows directly to the centre of the flame plate, ensuring effective cooling of this area.
- Finally, higher gas passage areas are possible in comparison with a cylinder head having valve cages.

The press-fitted valve seat contains an additional inner sleeve in order to prevent sulphur corrosion of the lateral seat surface. This is achieved by the insulating air gap between the overcooled region of the seat and the gas passage. The reliability of the valve for heavy fuel operation was considerably increased by the application of a specially-developed and patented plasma coating for the valve seat. In addition, the Z 40/48 engine is supplied with specially-designed tools for rapid dismantling of the cylinder head during regular inspections.

The application of the above mentioned features made it possible to extend the overhaul intervals of the exhaust valves. On the basis of service experience gained with overhaul intervals initially fixed at 6 000 hours, it was decided by the end of 1976 to extend the recommended valve overhaul periods to between 12 000 and 18 000 hours. This period is also customary for pulling the pistons. This eliminates altogether basic need of valve cages. Subsequent service experience with the exhaust valves on the Z 40/48 engine has generally confirmed the reliability of this design. The few valve failures experienced were explained by faults during the manufacturing process or because non-approved valve makes were used, which did not meet Sulzer requirements.

65/65 wall temperatures:

Because similar principles adopted in the Z 40/48 were applied to the 65/65 engine, the maximum temperatures are remarkably low for an engine of this size. Unlike the Z 40/48 design, however, valve cages are provided on the 65/65 engine in order to keep the fitting and dismantling intervals of the valves acceptably low (the weight of the cylinder head is considerably higher).

The rotating piston

In view of the large power range covered by Sulzer medium speed engines, two different piston designs have been adopted:

- **The requirements of the AS 25/30 engine led to the use of a conventional light alloy gudgeon pin piston (Fig. 16, right), whereby the piston skirt is lubricated by splash oil. The lubricating oil consumption, which ranges between 1·3 to 2·0 g/kWh (1·0 to 1·5 g/bhph), is controlled by the ring package placed above the pin.**

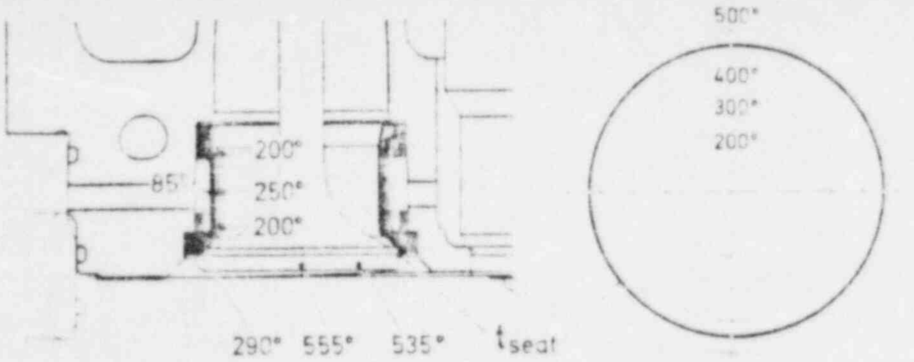

Fig. 15. Z40/48 engine exhaust valve seat temperatures (°C) engine running at 530 rev/min, [illegible]

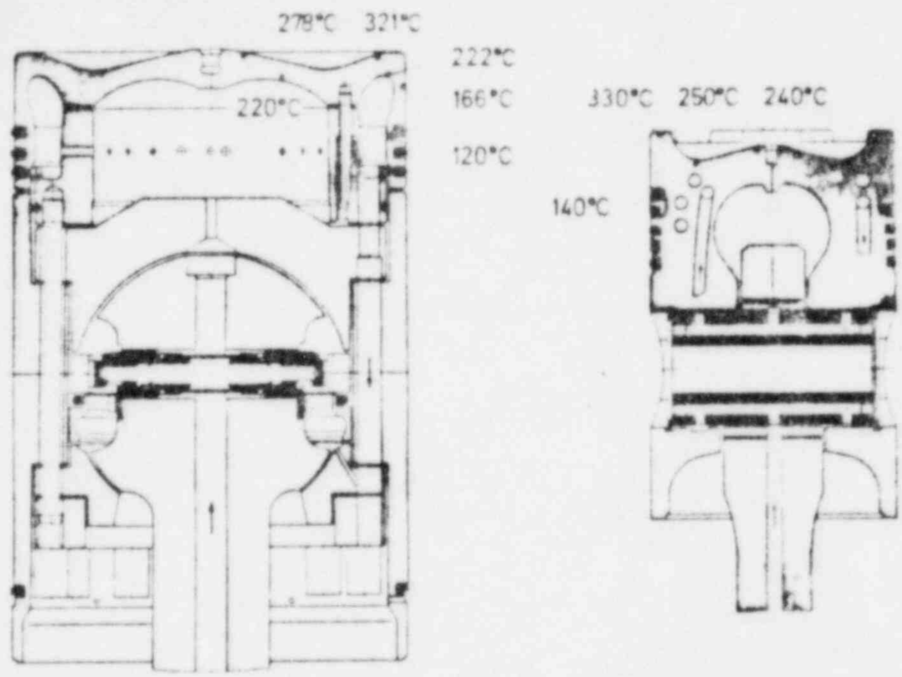

Fig. 16. The rotating piston design of the Z40/48 (left) indicating the temperatures (°C) measured at operating conditions of 530 rev/min, 533 kW/cyl (725 bhp/cyl). Piston with gudgeon pin of the AS 25/30 (right) with temperatures measured at 1 000 rev/min, 200 kW/cyl (270 bhp/cyl).

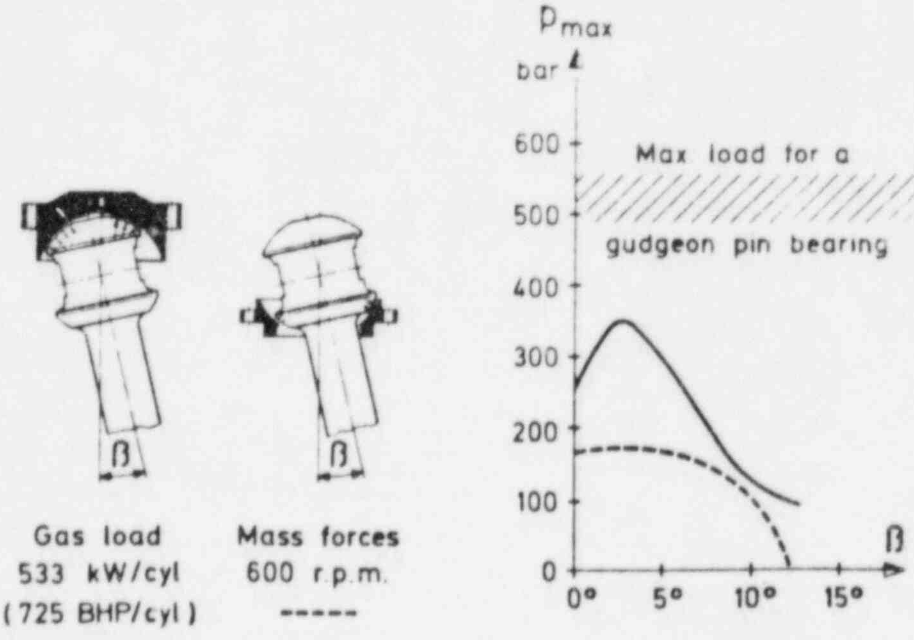

Fig. 17. Specific load of the spherical bearing of the Z40/48 engine.

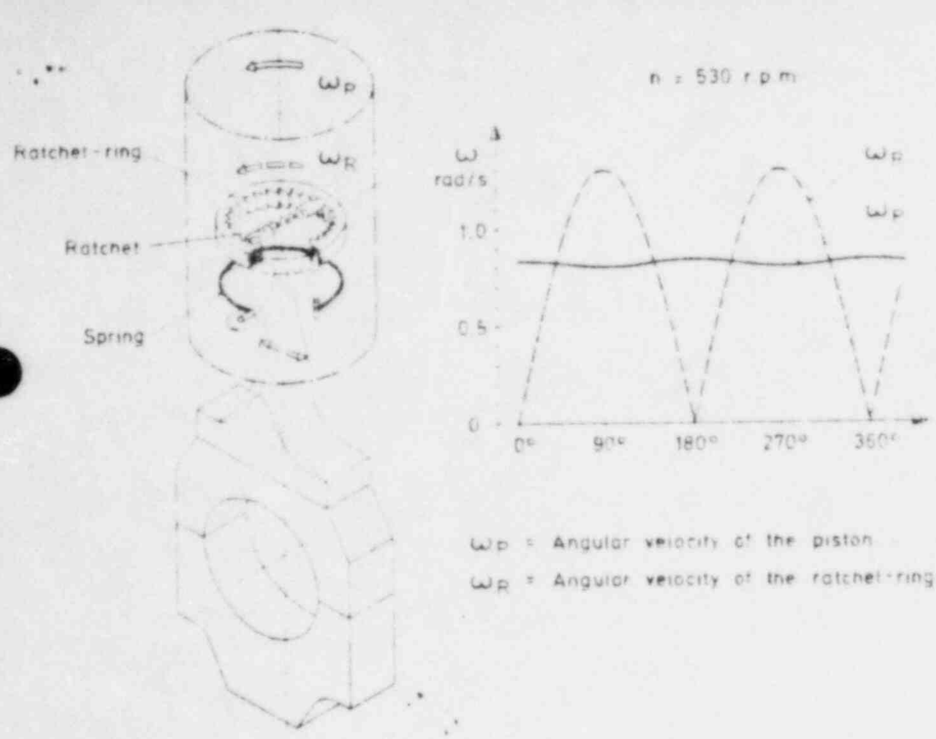

Fig. 18. Rotating mechanism of the Z40/48 piston (left) and a schematic representation showing its very low loading.

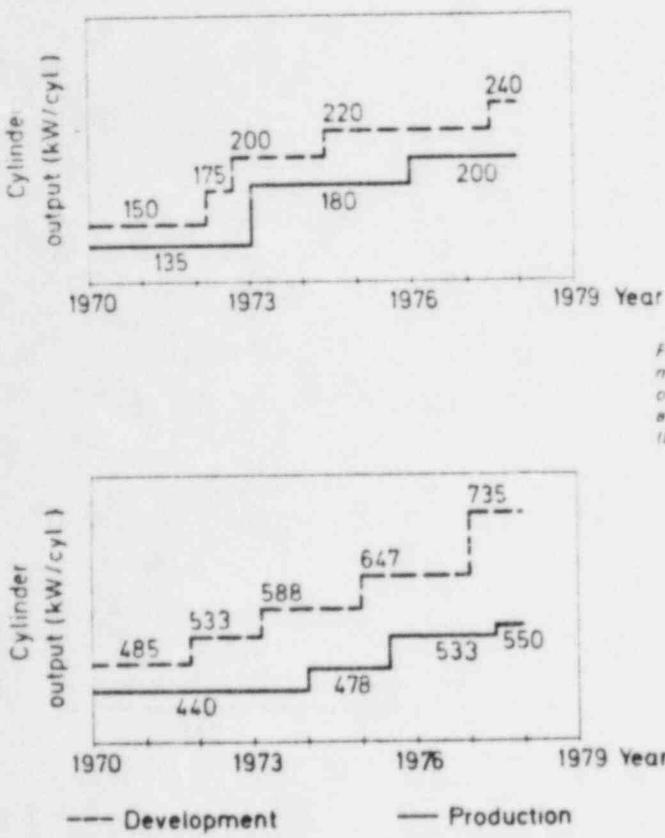

Fig. 19. Evolution of maximum cylinder output of the AS 25 engine (top) and ZV 40/48 engine (bottom).

- The pistons of larger engines are more prone to piston seizure because of the higher deformations involved. The risk of seizure is aggravated by the customer's demand for low lubricating oil consumption and by the requirement to burn low quality heavy fuels.

In order to solve these problems and to satisfy the demands connected with high specific output and good reliability, the well-known rotating piston design was adopted for the Z 40/48 (Fig. 16, left) as well as for the larger 65/65 engine. The advantages of such a design are that local overheating is avoided, due to the rotary movement. Temperatures are symmetrically distributed and thermal deformations are also symmetrical. Thanks to the spherical bearing and the symmetrical form of the skirt, deformations due to gas pressure are symmetrical as well. With every stroke, a fresh oil-wetted part of the skirt is turned into the load-carrying zone, substantially reducing the danger of seizure. Furthermore, piston clearance can be reduced to a minimum due to the symmetrical shape of the skirt.

An improved mechanical reliability is also incorporated in the rotating piston, due to the following characteristics:

- The maximum specific load of the spherical bearing under gas force is about 30 to 40 per cent lower than in the case of the small end bearing of a conventional gudgeon pin. It is also more evenly distributed (Fig. 17).
- The loading of the rotating mechanism is very low as explained schematically in Fig. 18: the oscillating motion of the connecting rod shaft imposes an intermittent rotation to the ratchet ring. On the other hand, a periodic acceleration and deceleration of the heavy piston skirt should be avoided in order to prevent damage of the rotating mechanism due to high inertia forces. For this purpose, the intermittent motion of the ratchet ring is transmitted to the piston skirt over a circular spring. The spring accumulates the energy transmitted periodically by the rotating mechanism and delivers it smoothly to the piston skirt. According to Fig. 18, the intermittent rotation of the ratchet ring (ω_R) is transformed into an almost steady rotation movement (ω_P) of the piston skirt around its axis. Due to the elimination of inertial forces, the forces induced at the ratchet ring periphery are low.

Lubricating oil consumption: this consideration is one of the most exacting problems which have to be solved during the development of a trunk-piston engine. The unique features of the rotating piston have permitted a much better control of lubricating oil consumption because of the following factors:

- The absolutely symmetrical conditions of the piston skirt as well as the small running clearances result in a reduced piston slap, which can be dampened with comparatively smaller amounts of oil.
- The reduced danger of seizure due to the rotating movement permits a low lubricating oil rate.

In addition, the lubricating oil consumption is exactly controlled by a separate cylinder lubrication system; the rate is automatically adjusted, as a function of the engine load. As a result, the total lubricating oil consumption of the Z 40/48 engine remains below 1·3 g/kWh (1·0 g/bhph). This performance is fully confirmed by long-term service experience with the engine.

Testbed results

As already mentioned, a high power reserve was incorporated in Sulzer engines at the design stage, a principle which allowed the engine to be introduced at a reasonably safe output level in order to gain experience for further power increases. At the same time, tests with much higher ratings were carried out on the testbed. Fig. 19 illustrates the evolution of testbed performance against the production ratings of the AS 25/30 and Z 40/48 engines. The even, increased lead time basically results in successive improve-

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The Development of a Highly-rated Medium-speed Diesel Engine of 7,000—9,000 Horsepower for Marine Propulsion

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The design considerations and development tests are described which have resulted in the production of the Mirrlees-National K Major engine, which has a current commercial rating of 3,000 to 7,500 b.h.p. in 6 to 18-cylinder units, and a projected future rating of 9,000 b.h.p. in 18 cylinders.

The Mirrlees K engine has been well established for over 12 years, some 980 engines now being in service for power generation and marine propulsion. Of these engines, 250 are operating on heavy fuels with viscosities ranging from 200 to 4,600 seconds Redwood I, representing over 650,000 horsepower. The objective in the design and development of the K Major engine has been to increase the specific power output by 50 per cent and at the same time to maintain or to increase the safety factors possessed by the original K engine. These factors, which determine the ability of the engine to operate on residual fuels with low maintenance and high availability, are discussed and the achievement of the objective is illustrated.

Component parts of the engine are described in turn, with details of the methods of measurement of pressure and temperature levels, air flow and wear rates in test rigs and in a prototype three-cylinder engine which was equipped with special features, such as a camshaft with variable timing, to facilitate development work.

The test results obtained on the first 12-cylinder KV Major engine are shown to confirm the performance expected from the rig and prototype engine tests.

INTRODUCTION

In general, the requirements of a marine propulsion engine are:

a) reliability;
b) low fuel consumption;
c) the ability to burn heavy fuels obtained in any part of the world;
d) low lubricating oil consumption;
e) low maintenance requirements;
f) minimum space and weight in keeping with a) and e).

These requirements are obvious but can only be achieved if certain basic principles in design are followed. The paper is divided into sections, each dealing with one aspect of design which affects these overall qualities.

However, before detailing these definite sections, some observations must be made on the application for which the engine is to be used and its suitability for that application. An engine developing 7,000 to 9,000 b.h.p. in 18 cylinders would be ideal for medium-speed marine propulsion since the power range available would be from 3,000 h.p., in a single six-cylinder engine, to 18,000 h.p. with twin 18-cylinder engines. This power range covers a large section of the marine market, as illustrated in Fig. 1, so that if conditions a) to f) can be achieved a worthwhile market should exist for such an engine.

The initial design study showed that the dimensions of the Mirrlees-National K engine (15-in. bore × 18-in. stroke) would fit this power range very well, if the new design embodied the modern features resulting from research and development which would enable high specific outputs to be obtained whilst retaining economy and reliability. At 500 r.p.m., and 200 lb./sq. in., b.m.e.p., this size of engine would give 402 h.p./cylinder, while at 525 r.p.m., and 220 lb./sq. in., b.m.e.p., 465 h.p./cylinder would be developed. The design of the K Major engine was based on a continuous rating of 250 lb./sq. in., b.m.e.p., at 525 r.p.m., giving 528 h.p./cylinder, and the development programme was planned to achieve this rating, using heavy fuel, in the three stages mentioned.

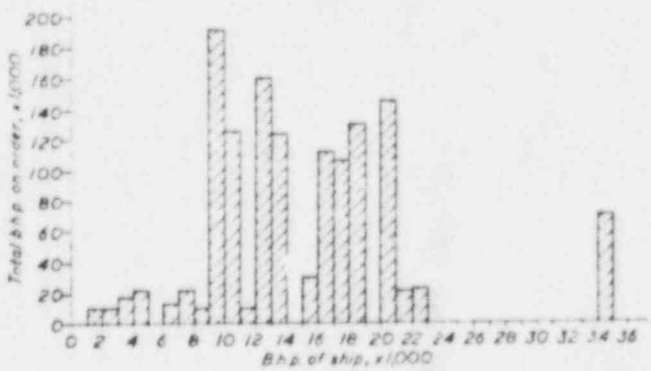

FIG. 1—*Distribution of horsepower for ships over 2,000 d.w.t. on order in Great Britain in March 1965*

At the present time, the K Major is released for the commercial market at a rating of 200 lb./sq. in., b.m.e.p., at 500 r.p.m., and development testing for the second stage of 220 lb./sq. in., b.m.e.p., at 525 r.p.m. is well advanced.

A cross-section of the engine, showing its general construction, is shown in Fig. 2 and the details of the design will be dealt with in the following sections of the paper under the headings a) to f) already given.

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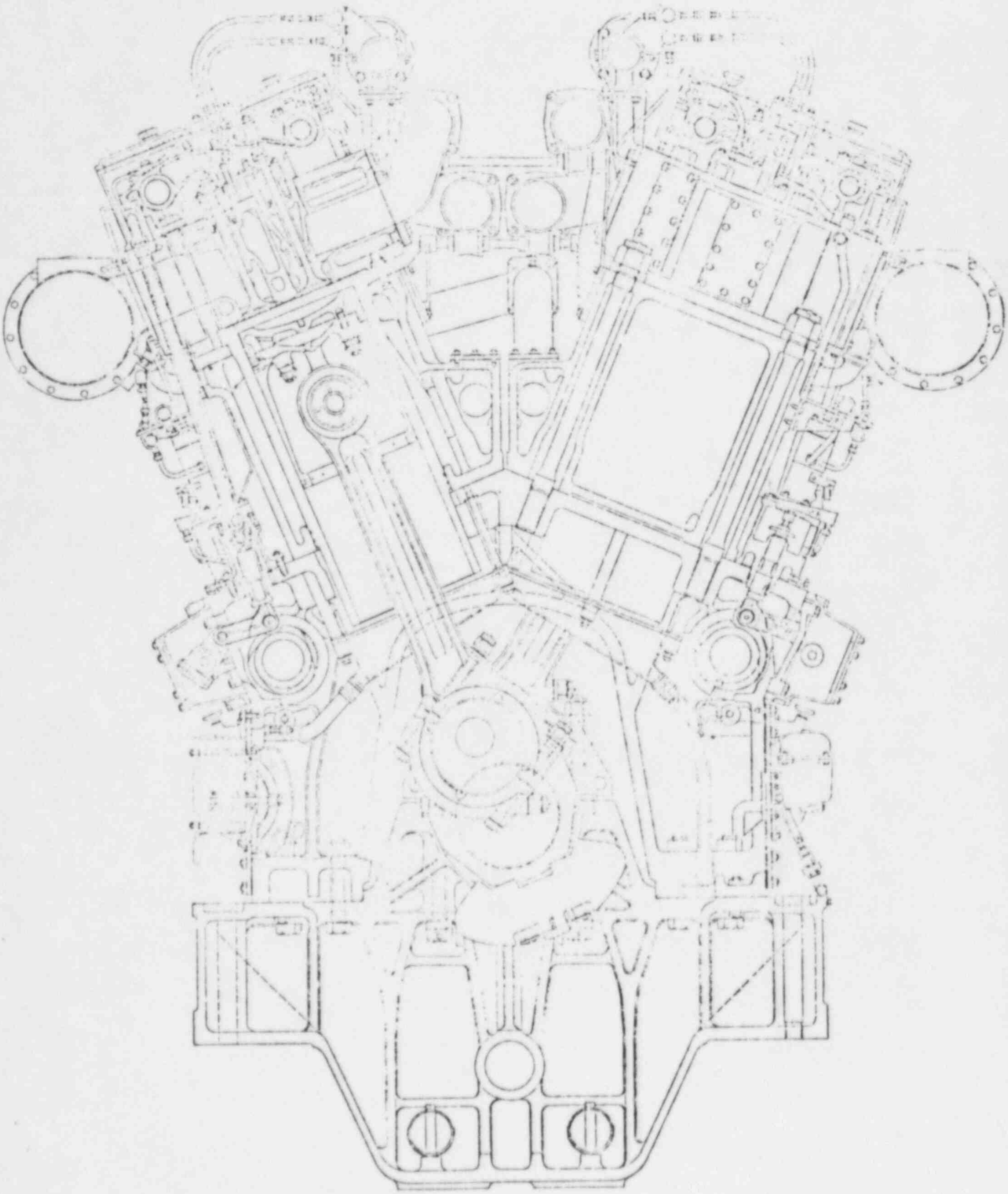


FIG. 2—*KVD Major engine—Cross-section*

a) RELIABILITY

General Considerations

Experience in engineering has shown that one of the surest methods of producing intrinsic reliability in a complex piece of machinery, such as a Diesel engine, is to proceed by a process of logical evolution from one successful design to the next, taking care that the critical parameters, proved in the original design, are maintained in the new design. From the authors' experience of continuous-duty Diesel engines, the critical parameters to be carefully watched are:

TABLE I

Parameter	Mirrlees KV12 engine	Mirrlees KV12 Major engine		
	154 lb./sq. in. b.m.e.p. 450 r.p.m.	200 lb./sq. in. b.m.e.p. 500 r.p.m.	220 lb./sq. in. b.m.e.p. 525 r.p.m.	250 lb./sq. in. b.m.e.p. 525 r.p.m.
Moment of inertia of bedplate (in.[4])	18,650	31,430	31,430	31,430
Maximum internal couple, tons. ft.	210	264	290	290
Ratio, maximum couple ÷ moment of inertia	0·0113	0·0084	0·0092	0·0092
Relative stress in crankshaft	1	0·82	0·91	0·96
Maximum cylinder pressure, lb./sq. in.	1,080	1,350	1,350	1,400*
Cylinder head stud stress, tons/sq. in.	8·3	10	10·1	10·4*
Fatigue strength of threads, tons/sq. in.	17	27	27	27
Ratio, stress ÷ fatigue strength	0·49	0·37	0·375	0·385*
He[illegible]n stress in fuel [illegible], lb./sq. in.	262,000	205,000	215,000	225,000*
Main bearing load, lb./sq. in.	814	1,234	1,275	1,330*
Maximum permissible bearing load, lb./sq. in.	1,500	2,500	2,500	2,500
Ratio load/permissible load	0·54	0·49	0·51	0·53*
Large end bearing load, lb./sq. in.	2,400	2,800	2,900	3,050*
Maximum permissible bearing load, lb./sq. in.	2,700	5,000	5,000	5,000
Ratio load/permissible load	0·89	0·56	0·58	0·61*
Maximum stress in piston ÷ U.T.S. of material	0·44	0·33	0·35	0·36*
Top piston ring groove temperature, deg. C.	220	165	185	205*
Exhaust temperature at cylinders, deg. F.	800	810	850	890*
Air flow, lb./b.h.p.-hr.	13·3	13·8	13·9	13·7*
Exhaust valve seat temperature, deg. C.	540	460	490	520*
Specific fuel consumption, lb./b.h.p.-hr.	0·336	0·335	0·336	0·340*
Lubricating oil consumption, lb./b.h.p.-hr. at full load	0·0030	0·0020	0·0020	0·0018*
Weight of engine, lb./b.h.p.	44	30	26	23
Th[illegible] area of piston, [illegible]	138	162	162	162
Maximum thrust pressure on piston, lb./sq. in.	35·8	33·5	34·0	34·8*
Depth of cylinder head, in.	11·75	13·5	13·5	13·5
Injector nozzle tip temperature (fuel at 200 deg. F.). deg. C.	177	127	130	136*
Inlet valve seat wear factor	192	156	156	162*

*Extrapolated values

1) exhaust temperature after the valves should not exceed 820 deg. F. (440 deg. C.) with uncooled valve seats and 930 deg. F. (500 deg. C.) with cooled seats;
2) top piston ring groove temperature should not exceed 430 deg. F. (230 deg. C.);
3) injector nozzle tip temperature should not exceed 350 deg. F. (180 deg. C.);
4) exhaust valve seat temperature should not exceed 1,020 deg. F. (550 deg. C.);
5) lubricating oil consumption should not exceed one per cent of fuel consumption at full load and if possible should approach 0·5 per cent of full load fuel consumption;
6) all bearings should be well within their load-carrying capacity;
7) the stressing of all components, both in fatigue and static loading conditions, should be such that an adequate factor of safety exists.

Some of the more important design and performance characteristics of the K Major engine are compared in Table I with those of the earlier and successful K engine, the comparison showing that safe values of the critical parameters have been maintained and, in many cases, improved.

Piston Design

The control of top piston ring groove temperatures by cooling the underside of the crown of the conventional single-piece cast iron piston, used in the K engine, is acceptable up to a rating of about 180 lb./sq. in., b.m.e.p., using a cast iron having a U.T.S. in the ring belt of 17 tons/sq. in., but, above this load, high tensile thermal stresses are produced on the inside wall of the piston behind the ring grooves[1]. For the K Major engine, a two-piece construction has been developed, as illustrated in Fig. 3, which has a high-tensile steel crown and a "Meehanite" skirt. This design incorporates an inner load-carrying boss, so that no pressure load is taken on the outer wall which carries the rings, and the latter may be quite thin, thus reducing the heat-flow path to the piston rings and giving efficient oil cooling of the ring belt, as well as ensuring that the roots of the piston

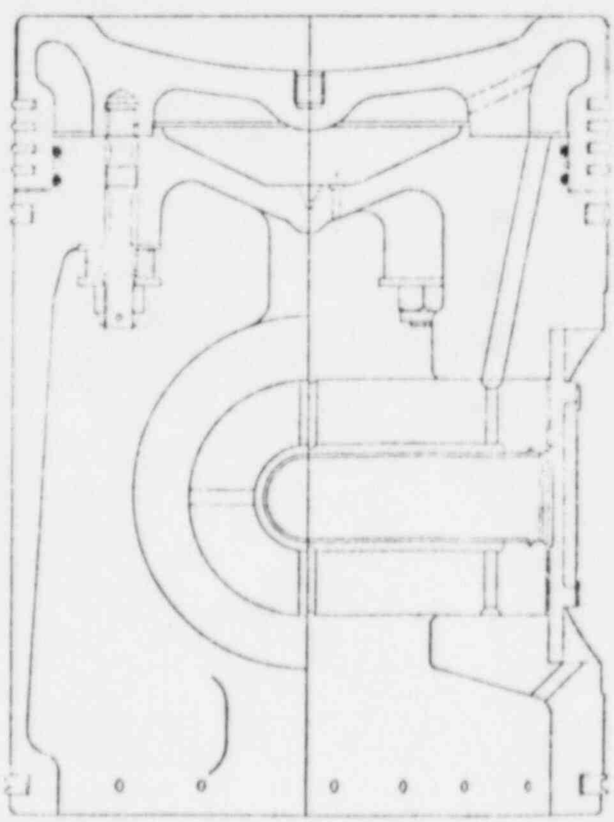


FIG. 3—*Assembly of two-piece oil-cooled piston*

ring grooves are stress-free. The piston crown is retained by four high-tensile studs which have rolled threads to give maximum fatigue strength, and heat-resisting "Helicoil" inserts are used to carry the studs, thus further improving the fatigue strength of the assembly and also acting as a heat barrier for the studs. Disc springs are fitted under the castle nuts to increase the resilience of the assembly and to provide an accurate method of checking the correct pre-load of the studs, this being achieved by measurement of the gap between the two retaining plates for the springs. Lubricating oil is fed, via a drilling in the connecting rod, through the piston pin and to the annulus chamber behind the ring grooves, through which it circulates at high velocity before meeting a transfer drilling to the inner chamber below the piston crown, from where it finally passes down an integral drain drilling in the piston skirt. The returning oil is collected in a cast aluminium tray, supported from the engine column, and is fed through a flexible connexion to a sight-flow and temperature indicator mounted adjacent to the crankcase door.

The thermally-induced and pressure-induced stresses have been thoroughly investigated in test rigs prior to tests in the prototype engine. Fig. 4 is a diagram of the thermal stress rig which is used to simulate the heat flow through the piston which occurs in the engine, heat being supplied by electric immersion heaters using solder as the medium for transferring the heat to the piston crown. Heat transfer through the piston rings is achieved by water-cooling the standard engine liner and oil-cooling of the piston internally is arranged in the same way as in the engine. Thermal stresses are measured by Budd self-temperature compensated strain gauges, having an overall size of ¼in. × ¼in., so that the effect of the gauges on the heat transfer conditions is extremely small. Fig. 5 shows the variation in thermal stress in the piston wall and also the temperature in the region of the top ring groove as a function of brake mean effective pressure for both the original single-piece piston and for the K Major two-piece piston; Fig. 6 illustrates the temperature and stress distri-

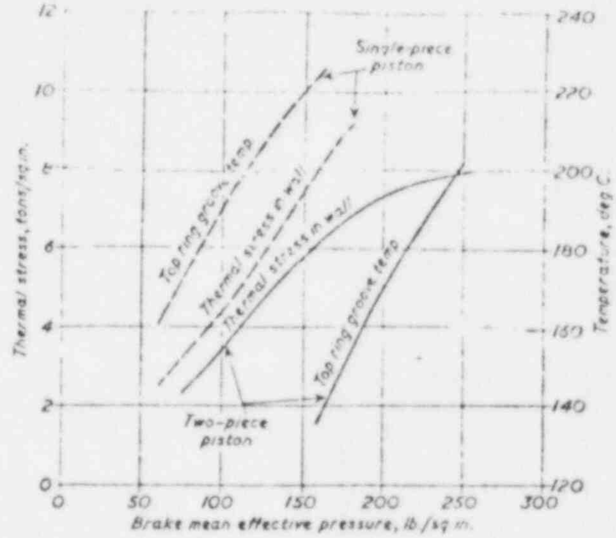


FIG. 5—*Comparison of temperatures and stresses in single-piece and two-piece piston designs*

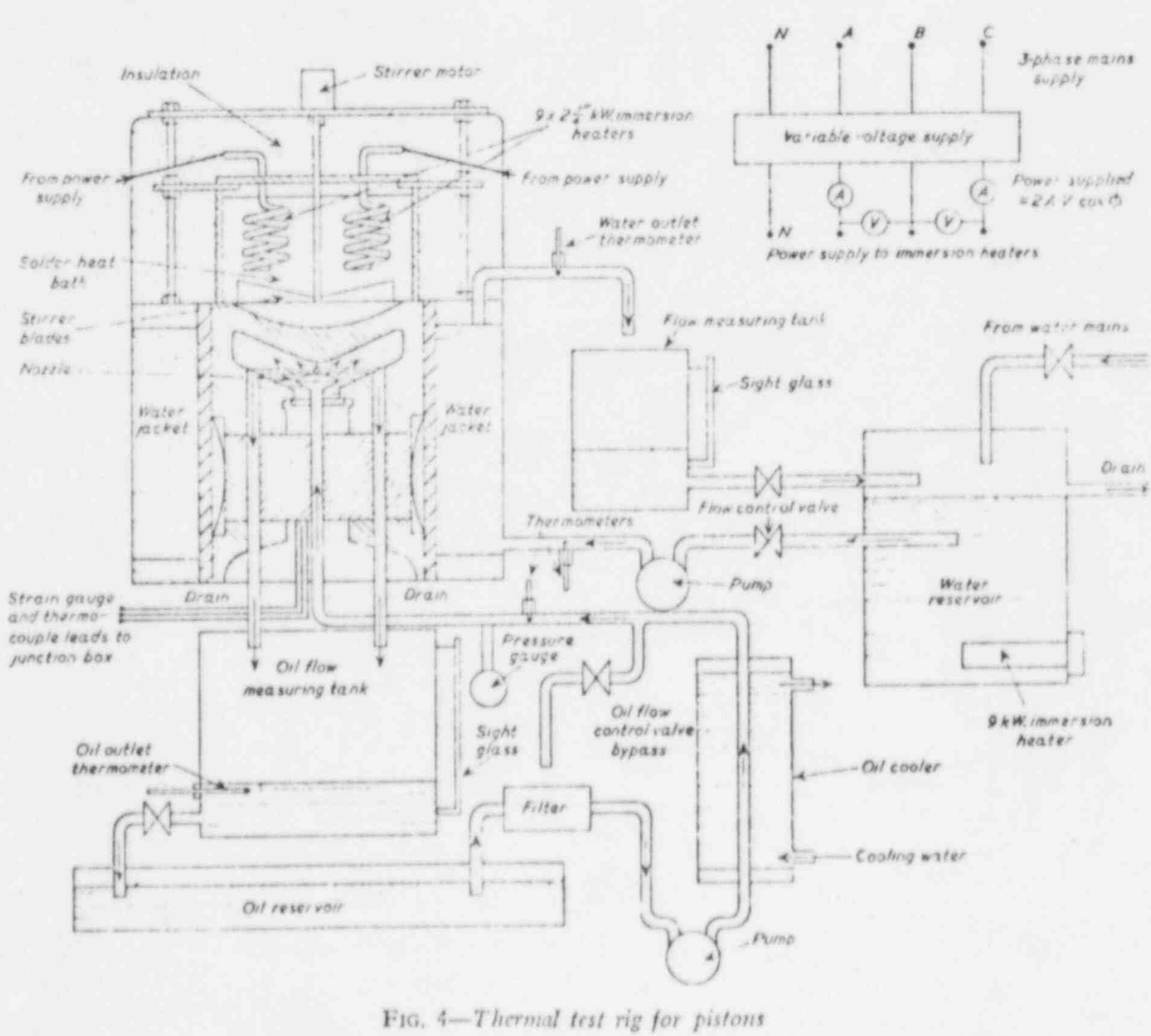


FIG. 4—*Thermal test rig for pistons*

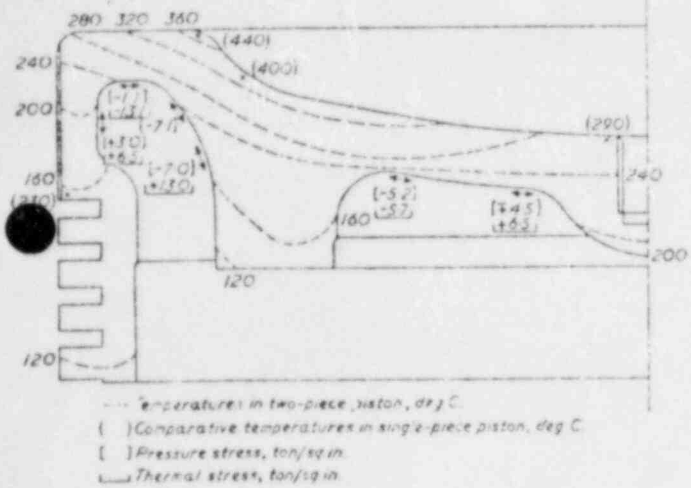
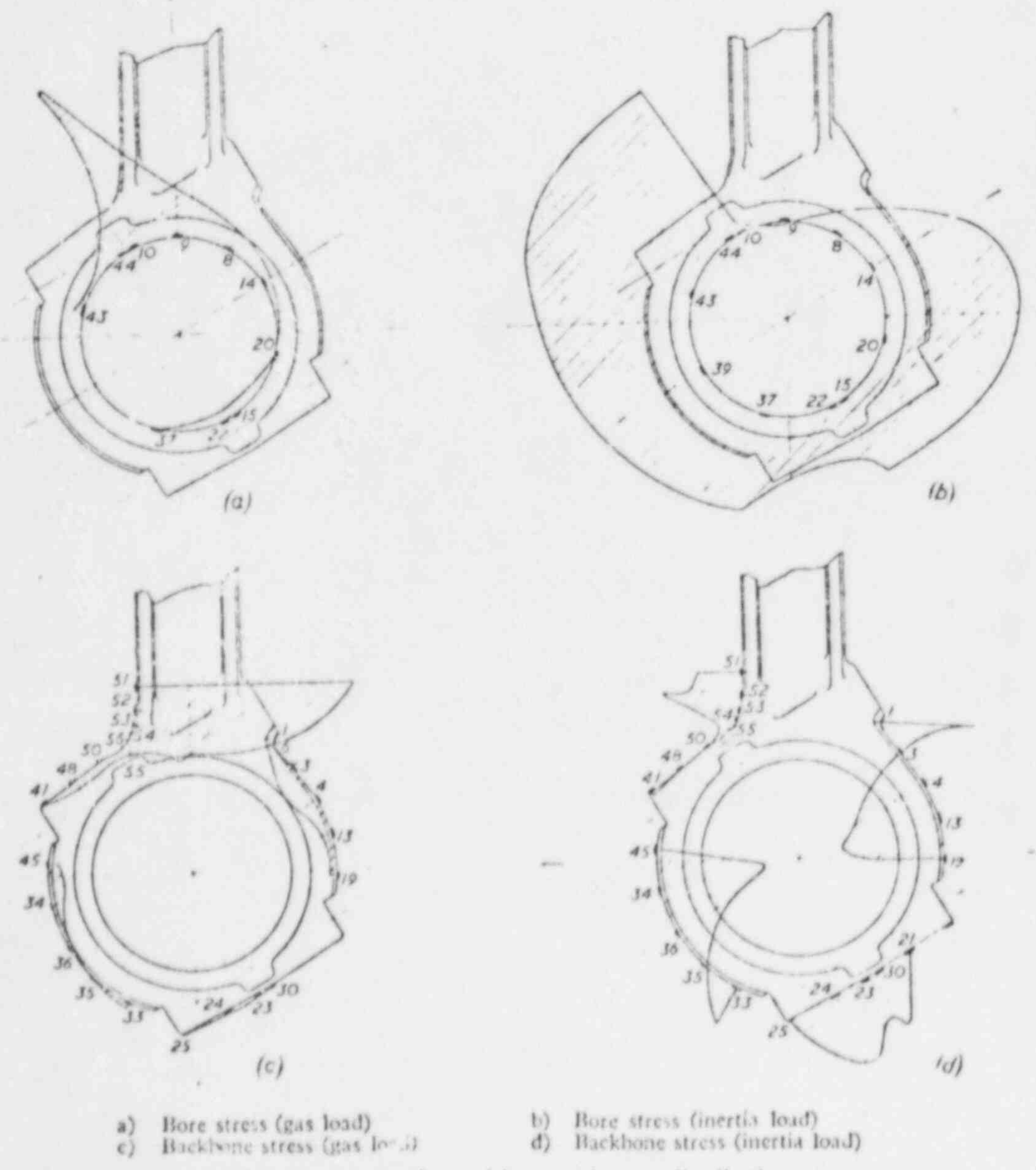


FIG. 6—*Two-piece piston crown temperatures and stresses*

bution in the crown of the two-piece piston. The reduction in top ring groove temperature by some 126 deg. F. (70 deg. C.), achieved by the new design, has made available a wide potential for increase in rating in the future, before any limitation due to lubricating oil break-down is reached. The thermal and pressure stresses are quite modest for the steel crown, which has a U.T.S. of 60 tons/sq. in., at room temperature, so that the factors of safety are much increased over the original single-piece cast iron design.

Connecting Rod

The connecting rods are one-piece stampings with the large-end bearing housing obliquely split at 30 degrees to the rod axis, and carry thin-wall tin-aluminium bearings. This construction permits a crank pin of maximum diameter, consistent with the withdrawal of the connecting rod through the cylinder bore. The optimization of the connecting rod proportions has been assisted by rig tests in a full scale static rig, in which gas loads and inertia loads are simulated by hydraulic pressure and the resulting stresses measured by strain gauges attached to the connecting rod. It was thus possible to reduce the weight of the connecting rod by 15 per cent from that of the original K rod so that, even at



a) Bore stress (gas load)
b) Bore stress (inertia load)
c) Backbone stress (gas load)
d) Backbone stress (inertia load)

FIG. 7—*Connecting rod large-end stress distribution*

the increased speed and load, the connecting rod stresses are lower than in the original design. Fig. 7 shows the stresses in the large-end of the connecting rod under firing pressure and inertia loading, and, in Table II, the safety factors at the most highly stressed points have been listed. In determining these values, allowance has been made for factors which would affect the fatigue strength of the material, such as specimen size effect and surface decarburization where it exists, so that the resulting values indicate the worst conditions and show that the rod design has a large margin of safety.

TABLE II

Gauge No.	Position	Factor of safety
1	Bolt platform radius	3·8
19	Supporting rib	6·8
33	Supporting rib	5·7
51	Base of shank	3·4
55	Shank radius	3·7
59	Bolt platform radius	4·4

Bearings

Main and large-end bearings are thin-wall steel shells lined with tin-aluminium, the increase in bearing loads from the K to the K Major being more than compensated by the improvement in fatigue strength of the bearing material. The actual and permissible bearing loads given in Table I illustrate the increased factor of safety in the new engine, the figures given being the conventional pressures obtained by dividing the maximum bearing load by the projected area of the bearing so that a simple comparison can be made. In the design of the K Major the more accurate methods of calculation, which have been made possible by the use of computers, have been used to assess oil film thickness over the range of speeds and loads so that the true factor of safety is even higher than the simple comparison suggests.

A positive displacement lubricating oil pump is driven from the free end of the engine by a flexible drive and delivers oil through a 15-micron full-flow filter to the main oil gallery cast in the bedplate. An oil-pressure regulating valve is fitted at the engine gallery to ensure that engine oil pressure remains constant, regardless of the degree of contamination of the filter, and a pressure-safety valve at the pump delivery protects the pump in the event of a complete blockage of the system. In addition to the full-flow filter, about five per cent of the flow is bypassed and filtered by small centrifuges mounted at the engine. This dual filtration ensures that carbon and water particles are removed from the lubricating oil and prevents the formation of sludge in the main filters. Tests have shown that a considerable increase in filter life is achieved by this system.

The quantity of lubricating oil circulated through the engine has been determined after thorough development tests to investigate the distribution of oil to main bearings, large-end bearings, piston cooling and other requirements, and the oil quantity has been chosen not only to lubricate but also to cool the main and large-end bearings, thus ensuring that the fatigue strength of the bearing material is maintained at its maximum value.

b) LOW FUEL CONSUMPTION

The importance of adequate air flow in a high-powered Diesel engine cannot be over-emphasized, the air delivered by the turbocharger having to perform the duties of scavenging the cylinder from the products of combustion and of cooling the components in the combustion space region, as well as providing a high mass of trapped air for the combustion process. In recent years the efforts of specialist turbocharger manufacturers to improve turbine and compressor efficiencies have made a substantial contribution to the success of the highly-rated Diesel engine, and the engine manufacturer can play his part by ensuring the maximum utilization of exhaust gas energy and by minimizing flow losses in the porting and ducting.

Air Ports

Air flow tests on the K cylinder head showed that the pressure drop in the inlet passages was made up as follows:

Inlet passage up to valve	11 per cent
Velocity change round valve seat	38 per cent
Loss of velocity head at outlet	35 per cent
Interaction between valves and cylinder wall	11 per cent

The large percentage loss around the valve seat indicated that optimization of the valve head profile and inlet passage shape in this region would be worth while, and the tests also showed that a greater effective flow area could be made available by increasing the valve lift beyond the value of a quarter of valve diameter at which the minimum geometric area becomes constant. Fig. 8 shows the increase in coefficient of discharge beyond the normal L/D ratio of 0·25 and the K Major valve lift was chosen

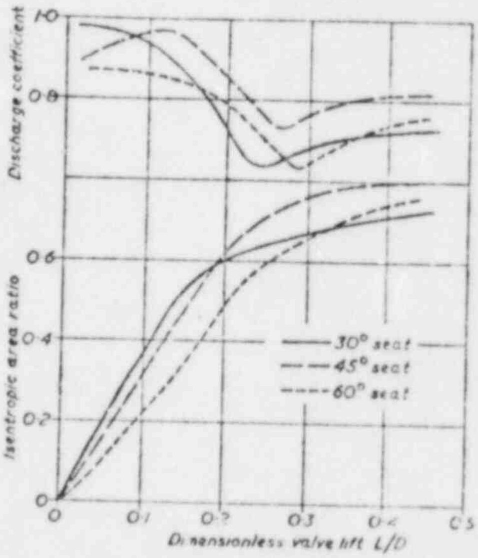


FIG. 8—*Flow characteristics of valves with 30-degree, 45-degree and 60-degree seats*

to be 0·3 of the valve diameter, giving an increase in maximum effective area of just over five per cent. This improvement is quite significant when it is remembered that it is effective over a large valve opening period.

The effect of varying valve seat angle on flow characteristics was also examined and Fig. 8 shows the characteristics of valves of the same throat area with seats at 30, 45 and 60 degrees to the face of the valve. The 60-degree seat valve is clearly inferior to the other two, and the 30-degree seat is the best at small valve openings, whereas the 45-degree seat is best at large valve openings, while the advantage to be gained by increasing the valve lift beyond $L/D = 0·25$ is valid for all values of seat angle.

There are other factors to be considered in choosing seat angle which determine the relative merits of 30 and 45-degree seats, of which the most important is that of useful seat life in service. Here, the conditions for inlet and exhaust valves are quite different, and will be considered separately.

The inlet valve operates in a relatively unlubricated condition at the seat so that seat wear, due to the relative movement of the valve seating face against the face in the cylinder head, as a result of the gas pressure, may be quite appreciable. A "wear factor" was derived theoretically and its validity confirmed by rig and engine tests from which the K Major inlet valve head profile was determined to give the minimum practicable relative movement and hence minimum wear[2]. The "wear factor" is defined as:

$$F_W = \frac{P_m^2 . N . \mu . D^2}{E . B . t . v^2 . b . \cos\theta}$$

where μ = coefficient of friction;
P_m = maximum cylinder pressure;
N = engine speed;
D = valve disc diameter;
θ = seat angle;
E = Young's modulus;
B = wear resistance factor (hardness number);
b = seat width;
t = distance from valve disc face to top of seat;
v = height of valve disc cone.

It can be seen that a decrease in θ, or increase in t and v have the effect of reducing the "wear factor" and the K Major inlet valve head profile was designed from these considerations with a 30-degree seat angle and a stiff valve head. From experience on other engines a wear factor o[illegible] 250 gives unsatisfactory life in service and a value of 200 is satisfactory. It will be seen from Table I that the original K engine has a satisfactory value, which is confirmed by service experience, and the K Major has an even bigger safety margin.

The criteria for the seat of the exhaust valve are quite different and will be discussed later in the paper under the heading of "Heavy Fuel Operation".

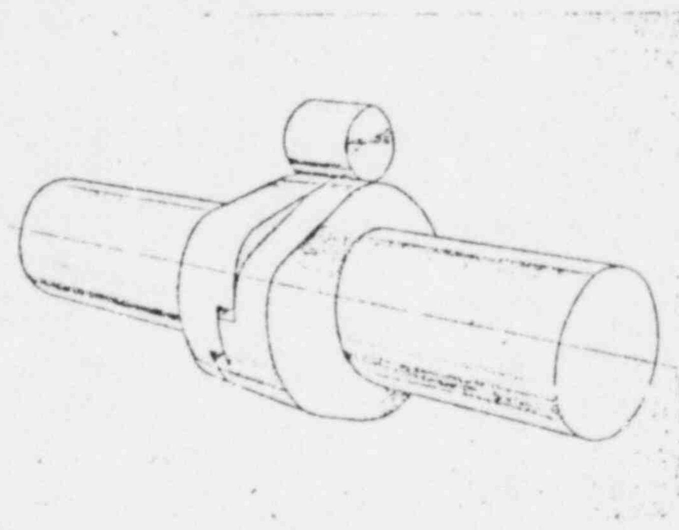


FIG. 9—*Construction of variable timing cam*

Valve Timing

The influence of valve timing on the exhaust, scavenge and charging processes has been examined experimentally on a three-cylinder engine, which was fitted with a special camshaft, in which the timing of both opening and closing of the air and exhaust valves, and of fuel injection were widely variable. Fig. 9 is a pictorial sketch of one of the variable timing cams showing the method by which the valve period is adjusted. Each cam is made in two pieces which are able to rotate relative to each other when hydraulic pressure is applied between the cams and the shaft from a hand pump. Release of the pressure then shrinks the cam on to the shaft to give an interference fit and the two parts of each cam are interlocked to form a bridge over which the cam follower roller can run without any discontinuity of profile. This method of hydraulic mounting allows the whole composite cam to be rotated to any desired position, as well as permitting the opening and closing flanks to be rotated relative to each other. Engine tests have been carried out over a wide range of valve timings, recording overall engine performance and pressure diagrams in the air inlet passages, engine cylinder and exhaust passages, from which optimum cam timings can be determined for any engine speed and load condition.

It will be appreciated that the optimization of valve timing is a complex operation and for a given set of timings it is necessary to match the injection equipment and the turbocharger performance for the engine speed and load range being considered. Some results have been selected from the range of tests carried out on the three-cylinder engine to illustrate the way in which changes in timing can affect the power range over which minimum fuel consumption is achieved.

In Fig. 10, the performance of the three-cylinder engine is shown with all valve timings held constant except the point of exhaust valve opening, the turbocharger match being changed to give the same total air flow. The valve timings were:

	E.V.O. before B.D.C.	E.V.C. after T.D.C.	A.V.O. before T.D.C.	A.V.C. after B.D.C.
Timing A	43 degrees	62 degrees	73 degrees	32 degrees
Timing B	65 degrees	62 degrees	73 degrees	32 degrees
Timing C	75 degrees	62 degrees	73 degrees	32 degrees

The left-hand curves of Fig. 10 show the performance at 450 r.p.m., and since the engine did not have the improved air flow already described in the previous sub-section, the optimum fuel consumption occurs close to the original K rating of 150 lb./sq. in., b.m.e.p. As the exhaust valve opening point is advanced, the position of minimum consumption moves further up the b.m.e.p. scale. This point is more strikingly illustrated in the right-hand curves of Fig. 10 where fuel consumption is plotted against exhaust valve opening point. At the lower rating of 140

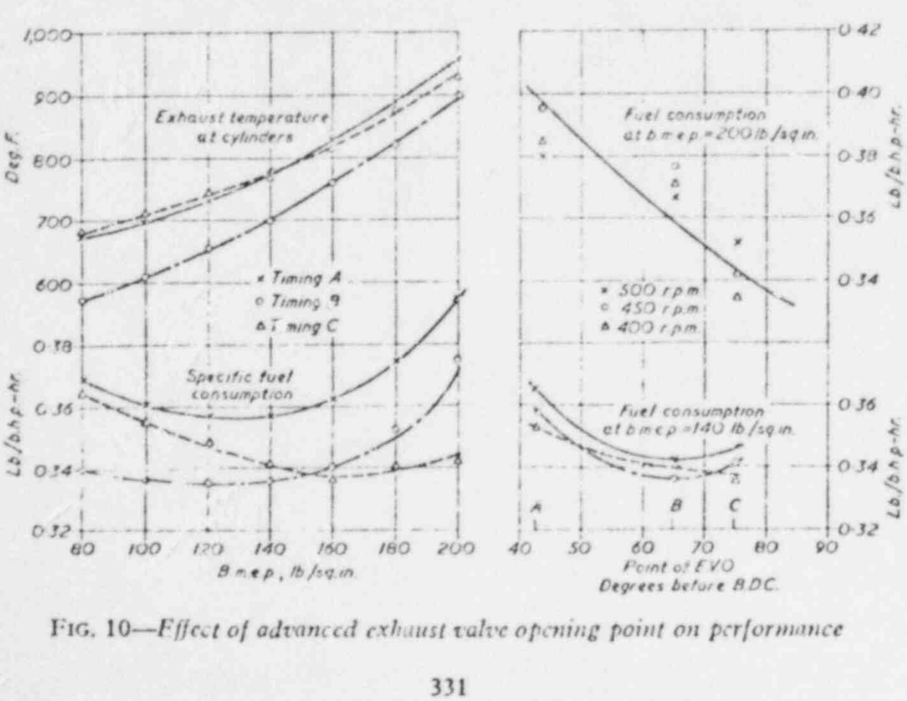


FIG. 10—*Effect of advanced exhaust valve opening point on performance*

'sq. in., b.m.e.p., the change in fuel consumption is small but, at 200 lb./sq. in., b.m.e.p., there is a marked reduction in fuel consumption as exhaust valve opening is advanced. It must be emphasized that this illustration is intended to be indicative only of the beneficial effects of early exhaust valve openings. It is important that, at high b.m.e.p. ratings, good thermal efficiency is maintained and in the 12-cylinder K Major engine the minimum fuel consumption occurs at about 200 lb./sq. in., b.m.e.p., as illustrated later in Fig. 14, this result being achieved by improvement in air flow and fuel injection. Fig. 11 shows a low-pressure cylinder and manifold diagram for the 12-cylinder engine at 240 lb./sq. in., b.m.e.p., and 500 r.p.m., and demonstrates the good scavenging and adequate charging of the cylinder which has been obtained. As the development of the engine continues to even higher ratings it will be necessary to move the specific fuel consumption loop still further and the indications from the three-cylinder engine tests are that the advantages of earlier exhaust valve opening will be realized at this stage.

Cam Design

The increase in speed and loading, accompanied by faster opening and closing rates of the air and exhaust valves and the increased lift already described, would be expected to make much

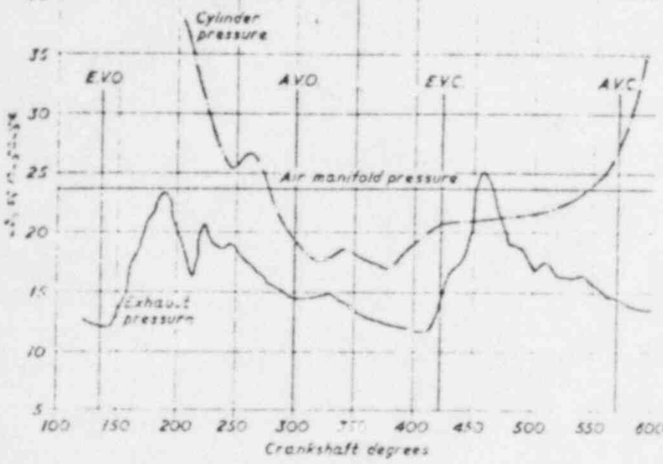


FIG. 11—*Cylinder and manifold low pressure diagrams at 240lb./sq. in., b.m.e.p., and 500 r.p.m.*

greater demands on the air and exhaust cams and follower gear. However, the design of cam profile, to optimize on rates of opening without exceeding established acceleration levels, has been considerably facilitated by the use of computer calculation techniques. The K Major air and exhaust cams are of polynomial profile, the mathematical analysis of the profile by computer calculations making selection of the most desirable curve a relatively simple procedure. The behaviour of the valve gear mechanism under running conditions, to determine the degree and frequency of vibration, has also been programmed and vibration calculations confirmed by a very simple technique of attaching strain gauges to the engine push rods. Fig. 12 is a typical push-rod strain trace which clearly indicates the natur frequency of the valve gear system and confirms that there is n tendency for separation of the valve train to occur.

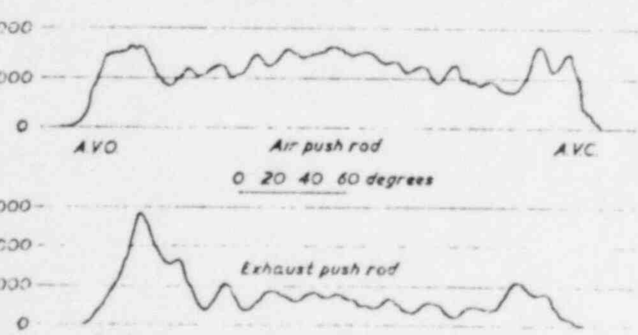


FIG. 12—*Push-rod strain at 200lb/sq. in., b.m.e.p., and 500 r.p.m.*

Fuel Injection

To obtain a maximum rate of injection, the fuel cams ar of a profile which gives a constant plunger velocity during th injection period, and the correct matching of the injection equip ment was facilitated by the use of test rigs which enabled th injection characteristics to be determined and the design of th injection equipment to be very nearly finalized before engine tes were started, only the confirmation of nozzle spray angle and th number and diameter of nozzle holes of a predetermined ar remaining for final decision from the performance of the engin

These test rigs enable a large number of permutations of fu cam, pump plunger diameter, delivery valve design, nozzl design, etc., to be tested quickly and cheaply, using convention methods of electronic indication of needle lift, fuel line pressur and nozzle sac pressure. The latter has proved to be of conside able importance in ensuring long life of injector nozzles by ex plaining the reason for over-rapid deterioration of nozzles in th K engine under certain service conditions. This phenomeno was a difficult one to explain until, as a result of calculation and rig tests carried out by the fuel injection manufacture it was realized that a particular combination of load and spee resulted in a hydrodynamic system in which there was a sudde reduction in fuel pressure in the nozzle sac just before th needle closed, the time interval between the two events bein of the order of a quarter of a millisecond. This resulted in penetration of gas from the cylinder into the nozzle sac durin the combustion process, the hot gases impinging on the botto of the needle and eventually impairing its performance. In Fig 13 this condition can be seen at (a) on the left, where the pre sure in the nozzle sac has fallen down to a low level at a poin 16½ degrees after spill closure and there is a period of o degree during which the needle is still off its seat and gas ca blow past it into the sac. The rig tests now ensure that th seating of the needle occurs before the sac pressure falls, a illustrated at (b) on the right of Fig. 13. The value of th preliminary rig work was confirmed by the performance pro duced in the 12-cylinder engine at a very early stage in it development running, many hours of "cut and try" tests t optimize injection equipment being saved. Fig. 14 show

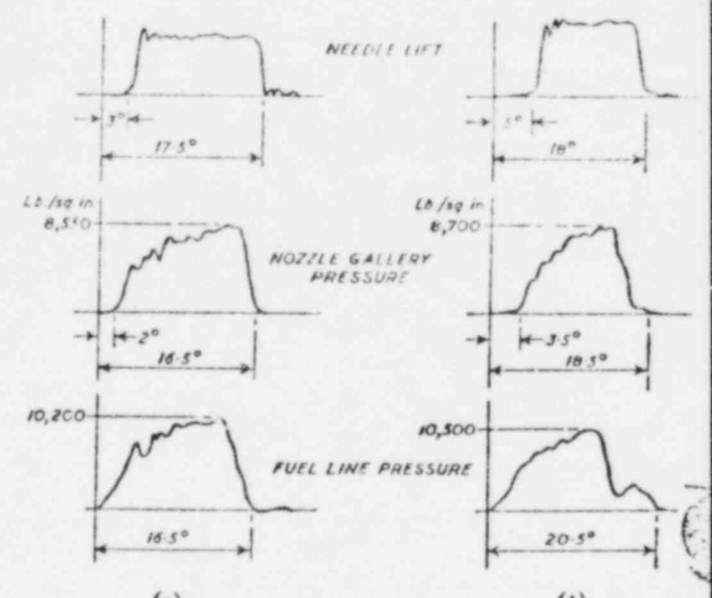


FIG. 13—*Injector needle lift, nozzle gallery pressure and fuel line pressure diagrams*

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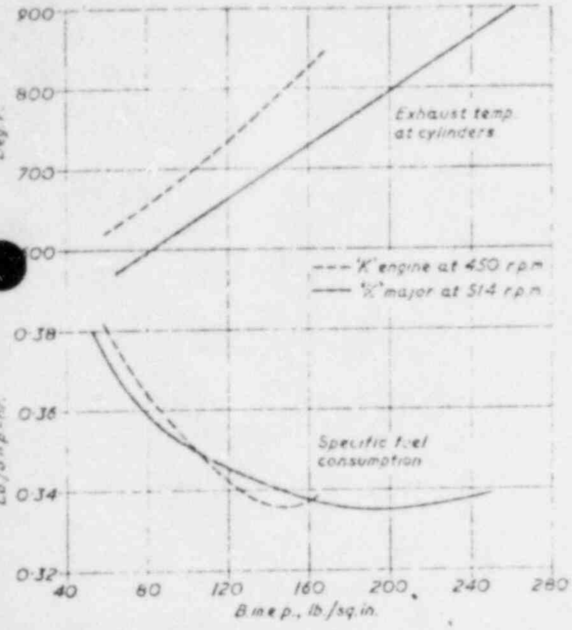

FIG. 14—*K and K Major performance comparison*

the performance of the engine as compared with that of the original K engine, from which it can be seen that the specific fuel consumption of the K Major engine is below 0·34 lb./b.h.p.-hr. over a very wide range of power, i.e., from 140 lb./sq. in. to 250 lb./sq. in., b.m.e.p. The curve also shows that, in spite of the increase in speed, from 450 to 514 r.p.m., and an increase in brake mean effective pressure, from 150 lb./sq. in. to 200 lb./sq. in. (i.e., a power increase of 56 per cent) the same exhaust temperature as in the K engine has been maintained.

c) HEAVY FUEL OPERATION

The operation of a Diesel engine on heavy fuel, the two items which normally deteriorate most rapidly are the injector nozzles and the exhaust valves, and the frequency of servicing of these two items is of predominating importance. In both cases there is a "threshold" of temperature of the critical parts of the components so that, as ratings increase, the design of the component must be improved to maintain safe operating temperature levels.

Exhaust Valves

Exhaust valve life with residual fuels is usually limited by the formation of deposits on the valve seat, resulting from the incombustible constituents of the fuel and largely from the combination of the sodium and vanadium salts present. As the seat deposits build up, they prevent the valve from making full contact on its seat, thus reducing the degree of heat transfer and eventually allowing tracking across the seat between the gaps in the deposits. The left-hand picture of Fig. 15 shows such a condition for an uncooled valve after 600 hours operation at 180 lb./sq. in., b.m.e.p., on a blended fuel of 300 seconds Redwood 1 viscosity with a three per cent sulphur content, 85 p.p.m. sodium and 100 p.p.m. vanadium. The beginning of erosion across the seat face between the deposits can be clearly seen.

Although the chemistry of the formation of these deposits is a most complex study, and is beyond the scope of this paper, field experience and engine tests have shown quite clearly that the presence of sodium and vanadium is of great significance and a practical assessment of the temperature range in which deposits are likely to adhere to the seating face of the valve can be made. Table III gives the melting point of possible deposit constituents which are in the temperature range which may appertain in the seat region of an exhaust valve, as is shown in Fig. 16, where the left-hand valve is of the normal uncooled design corresponding to the left-hand illustration of Fig. 15.

TABLE III

Compound	Melting point (deg. C.)
Nickel vanadate $N_iO.V_2O_5$	900
Sodium sulphate Na_2SO_4	880
Sodium orthovanadate Na_3VO_4	850
Vanadium pentoxide V_2O_5	675
Sodium pyrovanadate $2Na_2O.V_2O_5$	640
Sodium metavanadate $NaVO_3$	630
Sodium vanadyl vanadate (1.1.5) $Na_2O.V_2O_4.5V_2O_5$	625
Sodium vanadyl vanadate (5.1.11) $5Na_2O.V_2O_4.11V_2O_5$	535

From the Diesel engine designer's point of view, it is sufficient to accept that if the valve seat temperature can be kept below about 1,020 deg. F. (550 deg. C.), adhesion of any of these components will not occur to any appreciable extent so that rapid build-up of the deposits will not be possible. The problem is thus quite different to that of the gas turbine engineer, who has to consider the corrosive effect which occurs at higher temperatures. However, the achievement of low valve seat temperatures at high outputs is not easy and calls for careful attention to details of design and patient development engine testing to achieve the desired result. The three-cylinder prototype engine, shown in Fig. 17, has been used for continuous testing on heavy fuel in the research laboratory for the past three years and more

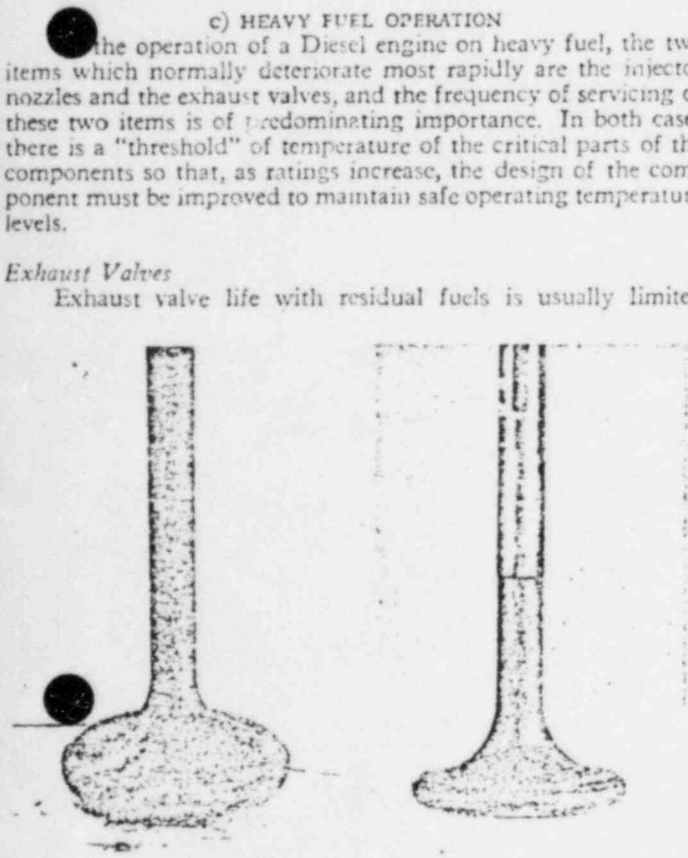

FIG. 15—*Comparison of exhaust valve condition after operation on heavy fuel*

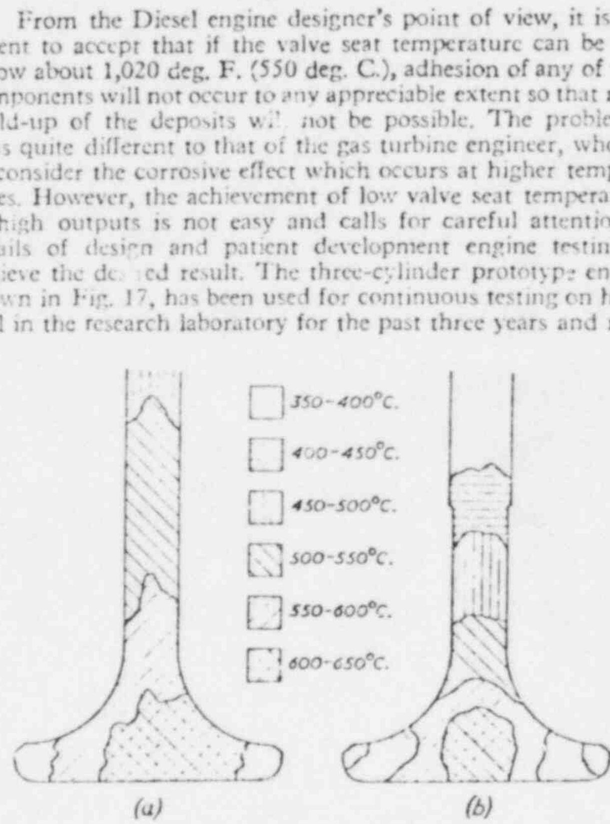

FIG. 16—*Temperature distribution in exhaust valves with uncooled and cooled cages*

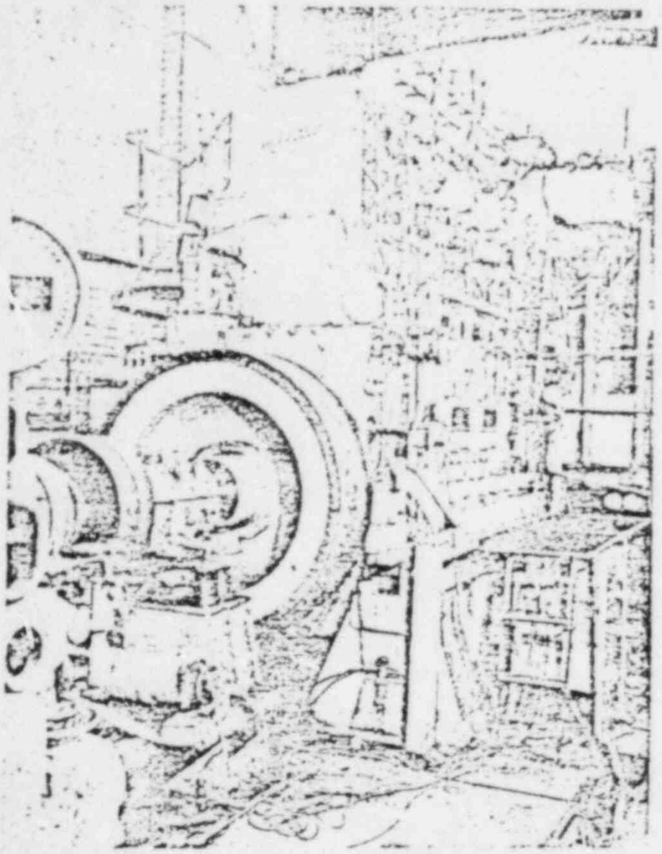

FIG. 17—*Prototype three-cylinder development engine*

than 60 exhaust valve and cage design combinations have been tested, of duration between 300 and 1,300 hours each to determine the effect of different factors in the design. A basic test duration of 500 hours at 200 lb./sq. in., b.m.e.p., loading was chosen and valve seat condition as the main parameter, together with other features such as valve guide wear, was compared with a reference design which was maintained throughout. In many cases the test

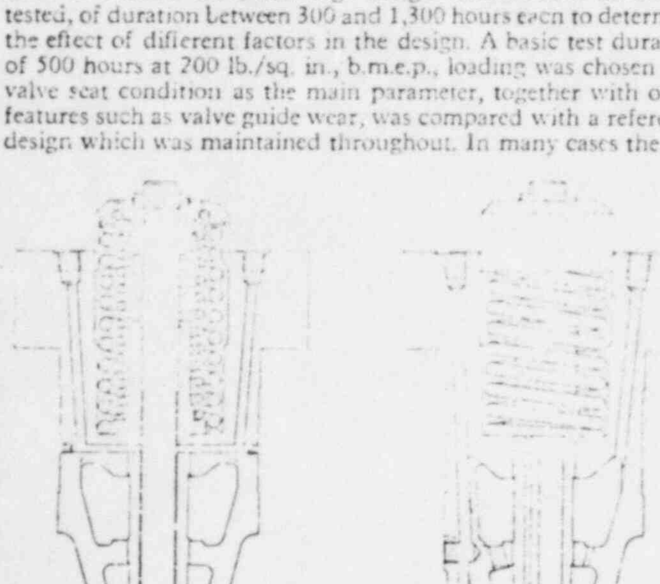

FIG. 18—*Assembly of exhaust valve and water-cooled cage*

was prematurely stopped at about the halfway stage where the valve condition was not satisfactory and, in the later stages, valve assemblies were replaced in the engine without re-grinding for second and third runs. Space does not permit a detailed report of the individual tests, but the resulting K Major exhaust valve and cage design will be described in detail to illustrate the factors which were found to be important.

Fig. 18 shows the valve and cage and it can be seen that a cooling passage is provided in the cage close to the valve seat. The seat is made of a single piece of Stellite 6, grooved to form the lower part of the cooling passage, and the upper portion is machined in the valve cage which is a three per cent Cr-Mo steel casting, the two being welded together by electron beam welding. The Stellite portion thus provides the facility for simple rebuilding of the seat by oxy-gas deposition of Stellite after a long period of time in service. The valve stem is of increased diameter and the valve of high-conductivity Cr-Ni-Si steel, to assist heat transfer from the head of the valve through the stem, and the valve guide is also surrounded by a water-cooled space, the cooling water passing along a drilled passage from the top of the cage direct to the annulus around the seat, then through a drilling to the space around the guide and via another drilled hole to the outlet at the top of the cage. The heat transfer from the valve to the cooled guide is assisted by the close fit between the valve stem and guide, the previously mentioned development tests having shown that a diametral clearance of 0·012in. resulted in an uncooled guide temperature of 450 deg. F. (232 deg. C.) in its middle position, with rapid deterioration of the lubricant and the formation of hard carbon. Halving the clearance reduced the temperature to 380 deg. F. (193 deg. C.) and, halving it again together with stem lubrication and guide cooling, brought the temperature down to 170 deg. F. (77 deg. C.), i.e., about 10 deg. F. (6 deg. C.) higher than the cooling water temperature, with no deterioration of lubricant and a guide wear rate of less than 0·0005in. in the first thousand hours.

The valve is fitted with a rotator in the top spring carrier which helps mechanically to prevent build-up of seat deposits, but its most important function is to ensure an even temperature distribution around the valve so that there is no local high-temperature region. The flow lubrication of the valve guide takes advantage of this rotation by using the valve itself as a timing device. Two flats are provided on the valve spindle which periodically line up with oil inlet and outlet drillings on the guide as the valve rotates. The linear positioning of the slots only allows the oil to pass while the valve is open and thus the oil space around the valve is only pressurized when the exhaust pulse pressure is present in the valve cage gas passage, the oil acting as a seal against gas penetration up the stem and the exhaust pressure preventing leakage of oil from the guide. It was at first feared that a continuous oil supply to the guide might result in excessive leakage of oil from the bottom of the guide, but this has not proved to be the case and, in fact, the tendency is for the leakage to be upwards as the retardation when the valve meets its seat is greater than the acceleration during opening and the inertia of the oil carries it upwards. This intermittent pressure lubrication of the valve stem makes it possible to use a very small stem/guide bore clearance without any risk of valve sticking, and this helps the heat transfer from the valve to the water-cooled guide. In addition, the danger of stem or guide bore corrosion at low load running conditions is avoided.

Since the stem to guide clearance is important in the heat transfer process, the reduction of guide wear helps to maintain low valve seat temperatures over a long period in service, and many of the development tests were concerned with valve guide material and valve rocker lever geometry to this end. The long guide and the small overhang of the valve head beyond the guide will be noticed in the illustration and were found to be important factors in reducing guide wear, as was the composition of the special "Mechanite" iron which was finally used for the guide material.

Sodium-cooled and water-cooled valves were tested among the many combinations but were found to offer no advantage over the design finally adopted, mainly, it is thought, because of the difficulty, with an internally-cooled valve, of providing cool-

ing passages close enough to the actual seat of the valve. The usual methods of drilling down the centre of the valve stem, although successfully cooling the centre of the head, still leave a fairly high temperature at the seat, and in the case of the internally water-cooled valve, the water connexions to the valve are a difficult problem.

The right-hand valve of Fig. 15 shows the results of this development, the valve having run for 900 hours at 200 lb./sq. in., b.m.e.p., on the same type of fuel as before. The good condition of the seating face shows that no re-grinding is necessary and the valve can operate for a much longer period without attention. The corresponding temperature distribution in the valve head is shown in the right-hand illustration of Fig. 16, and the effect of rating on exhaust valve seat temperature is given in Table I.

The valve development tests also included investigations into the effect of fuel treatment on exhaust valve life and while one fuel additive showed promise, in that the nature of the valve seat deposits was altered, it was not effective enough to justify its adoption. The principle of this additive was that other chemicals were added to the fuel so that the compounds, which were formed during combustion, would have higher melting points than those listed in Table III. It seems likely that, with further development work by the additive manufacturers, there may be some advantage to be gained in the future from this type of additive. Water washing of the fuel, to remove the sodium content, was found to be quite effective and the sodium could be reduced from 90 p.p.m. to about half of this value without difficulty, engine tests showing that the washing had quite an appreciable beneficial effect on the exhaust valve seat condition. As can be quickly calculated from Table III the critical sodium/vanadium ratios in the important temperature zone range from 1:0·74 to 1:13·3, the lower melting point compounds being associated with the latter end of the range, so that a reduction in sodium content may tend to produce the compounds with the lower melting points and, with particular fuel compositions, have an undesirable effect. Thus, with the wide variation in constituents in fuel from different parts of the world, it is difficult to make a clear case for water washing of the fuel.

Injectors

Fuel injection nozzles, when operated at high temperatures, tend to form carbon around the holes in the nozzles, known as "trumpeting", which may interfere with the injection spray pattern and reduce combustion efficiency, thus aggravating the temperature problem. For a time, the carbon formation develops until the "trumpets" become detached from the nozzle and a periodic rise and fall of exhaust temperatures can often be seen as this occurs. The general trend of temperature, however, is upwards and conditions eventually level out at the top end of the exhaust temperature cyclic range. In more extreme cases of high temperature, the needle seat may lose its hardness and the needle rapidly hammers its way into the seat. The temperature at the nozzle tip can be measured by thermocouple and a temperature of about 356 deg. F. (180 deg. C) is considered to be the limit

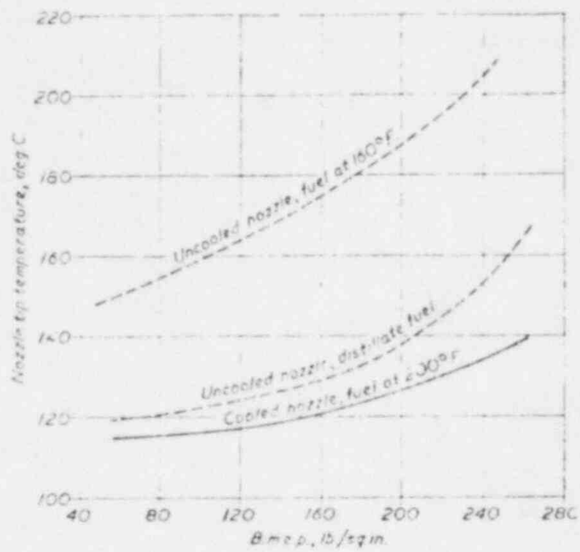

FIG. 19—*Injector nozzle tip temperatures with cooled and uncooled injectors*

for satisfactory operation. In Fig. 19, the middle curve shows the variation of nozzle tip temperature with load for the K Major engine, using an uncooled nozzle and distillate fuel, where the fuel itself has a considerable cooling effect, and there would be no difficulty in operating an uncooled nozzle on this type of fuel up to a load of about 280 lb./sq. in., b.m.e.p. In the upper curve, however, blended fuel of 300 seconds Redwood I viscosity was used, with a fuel temperature of 160 deg. F. (71 deg. C.) and it can be seen that the loss in cooling effect from the fuel has limited the acceptable load level to about 180 lb./sq. in., b.m.e.p., and with heavier, and hence hotter, fuels the load limit would be much lower. A water-cooled nozzle is therefore necessary for high ratings on heavy fuel, and the lower curve shows the tip temperature for a cooled nozzle using 1,000 seconds fuel at 200 deg. F. (93 deg. C.) with cooling water at 150 deg. F. (66 deg. C.). It is important that the nozzle should not be over-cooled as cold corrosion can occur at temperatures below 230 deg. F. (110 deg. C.), but this is controlled by the water-circulation system which is separate from that of the engine-cooling water. Fig. 20 shows the cooling system which is a closed circuit serving the injectors and water-cooled seat exhaust valve cages with a thermostatically controlled bypass around the heat exchanger and minimum volume in the system to ensure that correct operating temperatures are reached quickly.

d) LUBRICATING OIL CONSUMPTION

The consumption of lubricating oil in a Diesel engine is an important factor in maintenance costs and it is not always realized that, at a reasonable consumption rate of one per cent of

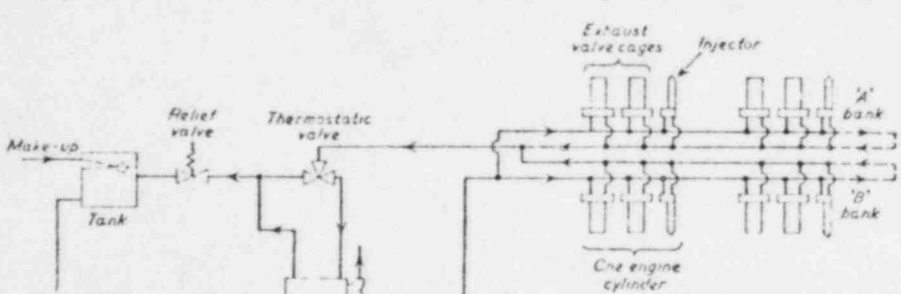

FIG. 20—*Arrangement of injector and valve cage cooling system*

fuel consumption, a 4,000 h.p. engine will burn a quantity of lubricating oil equivalent to its sump capacity in a period of the order of 300 hours. Emphasis is often laid on long periods between oil changes which are extended by an engine with a high oil consumption, whereas the relative importance of oil consumption to oil change period is around 50 to 1. The cost of modern high-duty detergent oils is quite appreciable, so that an oil consumption of one per cent of the fuel consumption represents something like ten per cent of the fuel bill. Not all of this could be saved, of course, but a reduction of 50 per cent in lubricating oil consumption is equivalent to a five per cent saving on the fuel bill, and would be well worth having from the point of view of running costs.

Piston Ring Design

To carry out lubricating oil consumption tests in the relatively short running periods of 500 hours or so in the research laboratory, it was necessary to develop an accurate method of measuring top-up rate and a system was devised, and has proved very successful, whereby consumption can be measured consistently over successive two hour periods and plotted consecutively. The running-in period and the levelling-out to a steady consumption can now be followed and it has been possible to obtain steady state results after a total test period of only 300 hours, which allows much more latitude for testing variations on a ring pack than was previously the case. There is a large number of detail points to be considered such as liner finish, roundness, drainage in the piston, etc., but the basic concept which has been established is to provide a parallel-faced chrome-plated top compression ring, three taper-faced plain compression rings, a relatively mild scraper ring below the gudgeon pin, and a more severe scraper ring above the pin. This ensures that adequate lubrication is available around the body of the piston but that the minimum of oil is allowed to pass up into the combustion space. The consistency of oil consumption measurement has enabled some interesting facts to emerge, and Fig. 21 illustrates one of these—the effect of the wall pressure of the scraper ring above the pin. The left-hand curve is from the three-cylinder, 15-in. bore prototype engine, and the right-hand curve from a

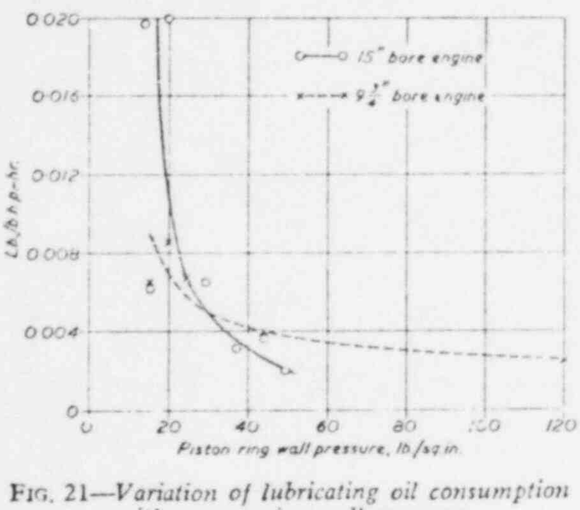

FIG. 21—*Variation of lubricating oil consumption with scraper ring wall pressure*

completely different high-speed engine of 9¾-in. bore, the points marked being the stable lubricating oil consumption achieved after running periods of about 300 hours in each case. Both curves show the same trend of reducing oil consumption with increased ring wall pressure and the tendency for the curves to level out at higher values of wall pressure. The value of wall pressure necessary to achieve a satisfactory consumption can be seen to be much higher for the smaller high-speed engine than for the K Major engine, and in the case of the smaller engine it was necessary to use a spring-loaded conformable scraper ring to achieve the desired consumption. A conventional type of slotted scraper ring was adequate to provide the 50 lb./sq. in. pressure needed for the K Major engine, the resulting consumption, of less than 0·002 lb./b.h.p.-hr., being confirmed in the 12-cylinder engine during development running.

Piston Ring Quality

Consistent oil consumption and low wear rates are largely dependent on the quality of the piston rings, from the point of view of metallurgical structure as well as accuracy of manufacture. Accuracy and good finish in manufacture can be assured by conventional inspection methods, and such methods can easily be extended to give some indication of material quality, such as by measuring the permanent set of the ring at a given load value above that required to close the gap. A simple sample checking method on metallurgical structure was devised in which a small piece of ring is clamped with its working face subjected to a given load and resting on the surface of a ring of liner iron. The ring is then rotated at a standard speed for a fixed time without lubrication and the weight loss of the piece of ring measured. Weight loss is used as a measure of the relative wear resistance of the material and, although "rough and ready", is found to co-relate well with the differences in micro-structure of the ring material. Some typical results are given in Table IV, and illustrated in Fig. 22, and show that with the same Brinell hardness, increasing amounts of free ferrite give progressively worse results and these are not improved by increase in phosphorus content within the amounts to comply with mechanical strength requirements[4].

TABLE IV

Sample No.	Structure	Hardness, HB	Weight loss, gm.
A	Greatly undercooled graphite, considerable free ferrite (centricast) 3·15 per cent. T.C., 0·83 per cent P.	210	0·404
B	Some undercooled graphite, a little free ferrite (centricast) 3·20 per cent T.C., 0·40 per cent P.	210	0·185
C	Random uniform medium flake graphite, fully pearlitic (sandcast) 3·45 per cent T.C., 0·55 per cent P.	210	0·017

e) MAINTENANCE

Engine running times between overhauls are dependent upon the load, duty and running conditions, and the preceding sections have indicated the attention that has been paid to the components which operate under the most arduous conditions. By reducing the critical temperatures of injector nozzles and exhaust valve seats, so that when operating on heavy fuels these temperatures are below the "threshold" values at which deterioration becomes rapid, it has been the aim to achieve periods of 2,000 to 3,000 hours before servicing of injectors or exhaust valves is necessary. Experience on the prototype engine has indicated that this ambition is by no means unreasonable but, of course, true confirmation of success will only come from the accumulation of service experience. Maintenance of other components would not be different from that established over many years, e.g., piston removal annually, complete overhaul every two years, the periods generally being dictated to suit the convenience of the operator rather than by the demands of the engine.

f) SPACE AND WEIGHT

In achieving high engine ratings reliably, the weight per horsepower, and space per horsepower, are naturally reduced and the emphasis on reliability for commercial marine work necessitates a different approach from that which would be appropriate for naval work where light-weight constructions become necessary but short life may be permitted. Sight should not be lost of the importance of low fuel consumption in the considera-

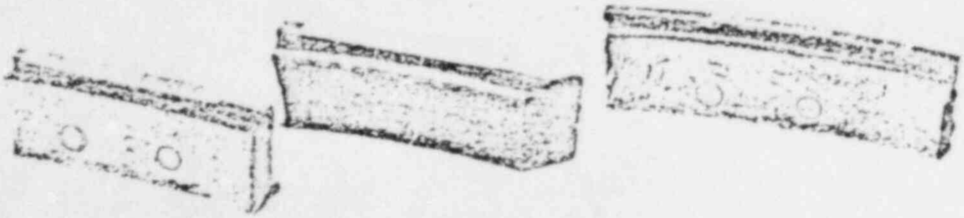

FIG. 22—*Comparison of piston rings after wear rig tests*

tion of weight. A ship refuelling every 3,000 miles, for example, at an average speed of 15 knots, and having engines weighing 34 lb./b.h.p., and a specific fuel consumption of 0·34 lb./b.h.p.-hr., would re-bunker an amount of fuel equivalent to twice the weight of the engines. Thus a five per cent reduction in fuel consumption would be equivalent to a ten per cent reduction in engine weight in addition to the saving in fuel cost.

In the design of the K Major engine, cast iron has been used as the main structural material and, in the authors' experience, has many advantages over fabricated steel designs. Few fabricated structures are able to avoid fillet welds in load-carrying regions and the fatigue strength of such a weld is as low as ± 1·2 tons/sq. in. Even butt welds must allow for discontinuity so that their fatigue strength is only ± 3·8 tons/sq. in., these values being for good quality welds, the strength of an imperfect weld being, of course, very low indeed. A good quality cast iron has a fatigue strength of over 5 tons/sq. in., and as well as freedom from the notch sensitivity, which so drastically reduces the fatigue strength of a steel structure, cast iron has good internal damping properties and also possesses the useful property of a diminishing E value with increased stress so that stress concentrations are considerably reduced and the material tends to relieve itself of any excessive stresses.

In keeping with the philosophy of designing for maximum

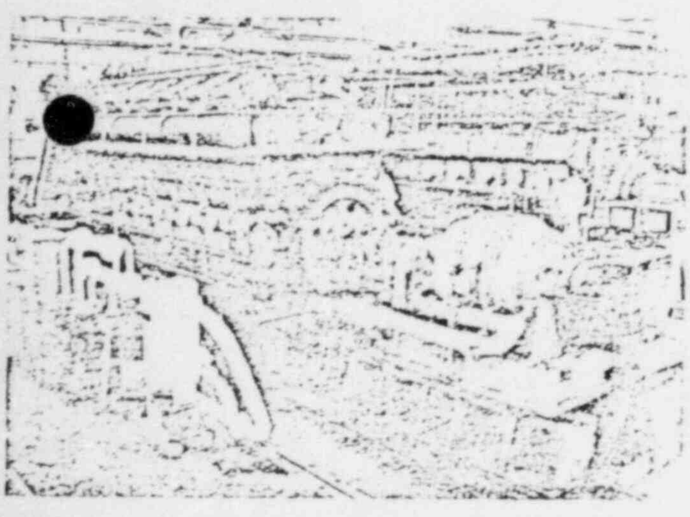

FIG. 23—*Prototype KV Major 12-cylinder engine*

TABLE V

Cylinder bore	15in.
Stroke	18in.
Compression ratio	11.35:1
Maximum r.p.m.	525
Minimum working r.p.m.	125
Continuous rated b.m.e.p.	200 lb./sq. in.
Maximum continuous b.h.p./cylinder	420 b.h.p. (426 cv)
Lubricating oil inlet temperature	150 deg. F. (65 deg. C.)
Lubricating oil outlet temperature	165 deg. F. (74 deg. C.)
Lubricating oil drain tank capacity	650 gal. (2,960 litres)
Fresh and salt water flow rates	5.5 gal./b.h.p.-hr at 50ft. head (25 litres/cv-hr.)
Engine cooling water inlet temperature	155 deg. F. (68 deg. C.)
Engine cooling water outlet temperature	170 deg. F. (77 deg. C.)
Exhaust temperature after turbocharger	800 deg. F. (427 deg. C.)
Starting air pressure	400 lb./sq. in.
Specific fuel consumption	0·335 lb./b.h.p.-hr. (l.c.v. of 18,400 B.t.u/lb.)
Thermal efficiency	42 per cent

TABLE VI—POWER RANGE

	B.m.e.p. lb./sq. in.	No. of cylinders					
		6	8	9	12	16	18
B.h.p. output at 250 r.p.m.	200	1,200	1,600	1,800	2,400	3,200	3,600
	250	1,500	2,000	2,250	3,000	4,000	4,500
B.h.p. output at 350 r.p.m.	200	1,680	2,240	2,520	3,360	4,480	5,040
	250	2,100	2,800	3,150	4,200	5,600	6,300
B.h.p. output at 450 r.p.m.	200	2,160	2,880	3,240	4,320	5,760	6,480
	250	2,700	3,600	4,050	5,400	7,200	8,100
B.h.p. output at 525 r.p.m.	200	2,520	3,360	3,780	5,040	6,720	7,560
	250	3,150	4,200	4,720	6,300	8,400	9,450
Overall length of engine		20ft. 0in.	24ft. 0in.	26ft. 0in.	24ft. 3in.	29ft. 10in.	32ft. 7in.
Overall width of engine		7ft. 6in.	7ft. 6in.	8ft. 2in.	11ft. 7in.	11ft. 11in.	11ft. 11in.
Overall height of engine		11ft. 6in.	11ft. 6in.	11ft. 6in.	11ft. 11in.	11ft. 11in.	11ft. 11in.
Height above crankshaft C.L.		8ft. 9in.	8ft. 9in.	8ft. 9in.	8ft. 2in.	8ft. 2in.	8ft. 2in.
Engine weight (dry) tons		38	44	48	65	85	95

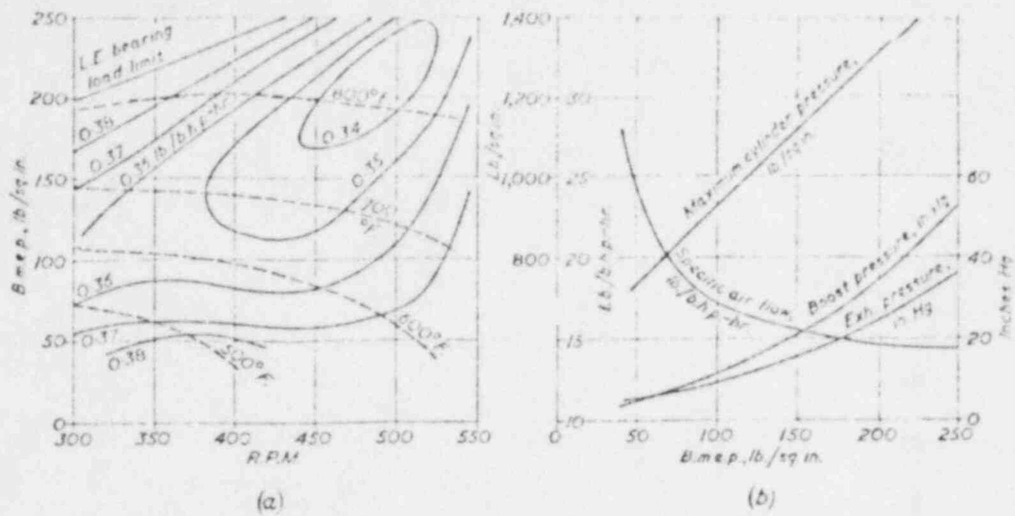

FIG. 24—*KV Major engine performance characteristics*

reliability and easy maintenance, the bedplate type of construction has been retained for the K Major engine and a useful facility has been added by the inclusion of a machined strip on the top surface of the bed so that alignment of the engine can be quickly and accurately checked, and crankshaft deflexion measurements more easily interpreted.

Fig. 23 shows the 12-cylinder KV Major on the test bed, from which the general construction and appearance of the engine can be seen, and Tables V and VI give the specification for the range of engines available at the current commercial rating of 200 lb./sq. in., b.m.e.p., and the future rating of 250 lb./sq. in., b.m.e.p., under normal temperature and pressure conditions with sea water up to 75 deg. F. (24 deg. C.) to the charge air cooler.

CONCLUSIONS

The design and development of a highly-rated medium-speed Diesel engine, to operate economically and reliably on heavy fuels, has been described and it has been shown that, for the K Major engine, the critical parts of the engine, which determine its reliability, have adequate safety margins for its current rating of 200 lb./sq. in., b.m.e.p., and have potential for a substantial increase in rating, to 250 lb./sq. in., b.m.e.p., in the future. The performance of the 12-cylinder prototype engine, beyond the current commercial rating, is illustrated in Fig. 24, curve (a) showing the performance at variable speed and curve (b) the performance at a constant speed of 514 r.p.m., and these, in conjunction with Fig. 14, show clearly the enormous strides which are being made in the Diesel engine industry towards higher specific outputs without exceeding the temperature and pressure levels which past experience has shown to give reliable and trouble-free operation. There is little doubt that in the marine propulsion field there is considerable interest in the use of medium-speed Diesel engines for higher powers than have hitherto been possible, and that within the next few years engines of this type will be available to cover almost the whole range of power demands of British shipping.

ACKNOWLEDGMENTS

The authors wish to express their thanks to the Board of Mirrlees National Limited for permission to publish the information contained in this paper and acknowledge the assistance given by Bryce Berger Ltd. and by their colleagues in the compilation of data.

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Discussion

MR. R. COOK, M.Sc. (Member of Council) said that, at the present time, the manufacturers of medium-speed engines in Great Britain were making very strenuous efforts to extend their share of the marine market in propulsion machinery. The paper was, therefore, timely and few who had read it would have failed to be impressed by the manner in which the authors and their colleagues were applying the latest knowledge and research techniques to the solution of the problems which arose when such machinery was developed to operate at high ratings on residual fuels. One could hardly doubt that success would attend their efforts, although he suspected that the large direct-drive Diesel would be about for quite a few years to come.

The histogram shown in Fig. 1 was interesting. It would be noted that by far the largest horsepower on order at the present time was between 9,000 and 21,000 s.h.p. per ship. With the machinery described in the paper this implied the use of some 18 to 36 cylinders of 15 in. diameter, each with two exhaust valves, two inlet valves, together with the injection and starting equipment. He said that he could not help wondering whether the modern seagoing engineer, who was perhaps not quite so amenable to long and arduous hard work as his forebears, would take kindly to the never-ending task of top-overhauling such a formidable number of cylinders.

Another point for thought was the effect which such maintenance requirements would have upon the reduction in engine-room staff now being achieved with direct-drive machinery by the application of an increased degree of automatic control. He hoped that some superintendents would comment on these aspects later in the discussion.

He said that some years ago Dr. Pope had made a very thorough theoretical and experimental investigation for the British Shipbuilding Research Association into the causes of failure of pistons, liners and cylinder heads in marine oil engines, with results which had since been published in the Transactions of another Institution. He was not surprised, therefore, to see the attention which the authors had given to thermal and pressure-induced stresses in the design of the two-piece, oil-cooled piston. Presumably, the piston temperatures shown in Fig. 6 were measured on the actual engines and the rig used to check the thermal stresses calculated from these temperature measurements. If so, he wondered whether good correlation was achieved. It would be interesting to know how the temperature distribution in the stationary-rig piston correlated with that on the engine.

He said that he was interested to observe the use of rolled threads on the high-tensile steel studs used for securing the piston crown. Work by B.S.R.A. on the rolling of threads of large mild-steel bolts such as those used in the dynamically-loaded components of direct-drive Diesels, had shown very strong improvements in fatigue strength. Reference to a paper* appearing in the Transactions of the Institute four years ago would show that form-rolling increased the fatigue strength of large forged bolts made from mild steel some 2½ to 3 times when compared with cut-thread specimens. Rolling of thread roots gave almost as great an increase. The degree of rolling had been found to be not very critical, but his Association was at present investigating more fully the optimum degree for various sizes and pitches. He imagined that with the high-tensile steel material used by the authors, the gain in fatigue strength would not be so great as in the case of mild steel, but it would be interesting if the authors would quote some figures. Form-rolling could be a very cheap method of bolt production, particularly in small sizes.

The means adopted to ensure correct pre-loading of studs was to be commended since there was no doubt that the majority of failures of dynamically-loaded bolts were due to fatigue caused by inadequate tightening. It was not always appreciated that, with a properly-designed bolted connexion, fatigue failure was virtually impossible if the bolt was adequately pre-loaded.

The section in the paper dealing with heavy fuel operation was, of course, of the greatest possible interest, since a solution of the difficult problems involved was essential if the medium-speed engine was to be able to compete with the direct-drive engine, which was so much less fastidious as to its diet. Here again, the authors had given evidence of a careful scientific approach which should go a long way to ensure success. They had commented on the possible use of fuel additives. One could imagine this approach being successful where fuel supplies of constant composition were available, but this was seldom possible in marine practice and the chances of obtaining a cheap additive, which was effective with a wide variety of fuels, seemed somewhat remote. The authors' approach, by tackling the design, was certainly the right one. Their remarks on the drawbacks of water washing were also worth noting.

No reference had been made in this section to sump-oil contamination when using heavy fuels. Presumably this must occur to some degree in this trunk-piston design, and it would be useful if the authors were to give some information on the procedure involved in maintaining the lubricating oil in a suitable condition.

Dr. Pope had, over the years, made many investigations into the properties of cast iron. Few were, therefore, more familiar with its strength and frailties. Sir Harry Ricardo had once referred to cast iron as "the material which served our forefathers so well for lamp posts and kitchen ranges", but he was sure that Sir Harry would be the first to acknowledge the advantages which the authors had enumerated. Its use as the main structural material in the K Major engine had much to commend it, since weight was rarely of paramount importance in merchant ships.

On the subject of cast iron, he said that it might be inferred from the data given in Table IV that centri-cast piston rings were inferior to sand-cast rings. He felt sure that this would not be the authors' intention. Centri-cast rings had been widely employed with success. He took it that the authors' purpose had been simply to show that, with this type of material, under-cooling and consequent presence of free ferrite was most undesirable.

The paper had touched in an interesting manner on so many aspects of Diesel design that to point to ommissions might seem somewhat churlish. He wished, however, that the authors had found it possible to touch on the subject of turbo-

* Cook, R., and McClimont, W. 1961. "The Influence of Screw Forming Methods on the Fatigue Strength of Large Bolts". Trans. I.Mar.E., Vol. 73, p. 417.

charging. Perhaps they might, at some later date, find it possible to give a paper on their experiences in turbocharging up to the 250 lb./sq.in. in b.m.e.p. which was involved in the third stage of the development of the engine.

COMMANDER E. TYRRELL, R.N. (Member), in a contribution read by Mr. T. P. Everett, referred to the successful introduction, by the authors' company, of an engine which he considered met both industrial and marine needs which so far had only been filled by Britain's foreign competitors. As the authors had so rightly said, reliability was the main requirement of a marine propulsion engine, and nobody who had read this paper could fail to be impressed by the systematic way in which Mirrless National had carried out the research and development work necessary to ensure that this engine would operate satisfactorily at the ratings envisaged.

If the medium-speed geared Diesel engine was to compete with the slow-runing direct-coupled engine, it was essential that it should operate satisfactorily while burning heavy fuel, and it was evident that the authors had taken very considerable trouble and had expended a comparatively large sum of money in trying to ensure that this would be the case. There was, however, one point which he felt should have been mentioned in this respect, and that was the effect of various grades and compositions of lubricating oil on the problems associated with the burning of heavy fuel. There was little doubt that trunk-piston engines called for careful selection of the lubricating oil if satisfactory operation with heavy fuel was to be achieved. New types of lubricating oil and testing for quality and make-up by the addition of detergents, anti-oxidants, and alkalis could have a profound effect on the satisfactory operation of trunk-piston engines while burning this type of fuel. The wrong type of lubricant, or one which had been allowed to deteriorate unduly, could give rise to ring-sticking and crankshaft corrosion. The operating temperature of the oil was also important if these defects were to be avoided. Perhaps the authors would like to remark on the type of lubricating oil and its optimum operating temperature for this type of engine burning heavy fuel.

He thought that the title of this paper was slightly misleading. He could not agree that the Mirrlees K Major engine should be regarded as highly rated when operating at the conditions given in the paper. In his opinion there was considerable scope for further advances in b.m.e.p. These were important as they should give worthwhile reductions in the cost per horsepower. Introduction of a reliable engine operating at these higher brake mean effective pressures would do much to increase the competitive power of this type of engine against its competitors abroad and other types of prime mover. In this technically-competitive world, the main object of any Diesel engine manufacturer must be consistently to uprate his engines in order to give better value for money. He must at the same time retain reliability.

Many of those present would be aware that the Ministry of Technology had recently placed a contract with the Yarrow-Admiralty Research Department to investigate the use of medium-speed geared Diesel engines as propulsion units for ocean-going merchant ships. This survey was now almost complete. He thought that it was true to say that the results of this survey would give encouragement to those manufacturers of medium-speed Diesel engines who thought that the medium-speed Diesel had a future and could compete in many ships with the slow-running direct-coupled engine. The report showed that every type of ship and trade must be treated on its merits, but that a shipowner who failed to carry out a detailed economic survey into the possible use of medium-speed Diesel engines, as an alternative to the slow-running direct-coupled engine, did so at his peril.

MR. S. H. HENSHALL, B.Sc. (Member) said that, as an engine builder of medium-speed engines, he found that he was on the side of Dr. Pope in a lot of the things he had said. The paper, however, had been very stimulating and he would like to ask several questions about it.

With regard to Table I, the specific air flow for the 250 lb./sq.in. b.m.e.p. showed a drop compared with lower b.m.e.p., and, although this drop was only a small one, he would have thought it desirable to go on increasing the specific air flow.

Turning to the piston design, he said that the features of it were, in many ways, those with which he agreed, but it was mentioned that it was a steel crown and a cast iron junk. The steel crown probably had a higher coefficient of linear expansion as compared with the cast iron. This meant that there were some problems in its connexion, for instance, it must make life a little difficult for the sealing rings between the two portions. Cold clearance between piston crown and liner must be increased.

Figs. 2, 3 and 6 showed a double line round the junk. He wondered what this signified and whether it was some device to overcome the cold clearance problem.

With regard to exhaust valves and operation on heavy fuels, the importance of losing heat via the stems of the valves was certainly to be considered. In the paper the clearance was mentioned as being of importance. The wear rate was obviously kept down by the ingenious device of continuous lubrication, and he said that he would be interested to know what was the maximum allowable clearance of the stem to guide and what sort of life the valve had in this respect. Also, it appeared that the guide could be renewed, although he was not sure whether it was intended to be renewed.

He suggested that there was an argument for not water washing fuel, in that the deposits also occurred on turbocharger blades, and turbochargers could be water washed more easily if the sodium was allowed to remain in the fuel.

On the question of cast iron or steel as the main structural material, he said that steel had its own advantages, and structures could be designed with low stresses where welds occurred. Modern techniques of manufacture and inspection could ensure good quality welds.

He said that surely greater reliability resulted from a design in which the major loads did not have to pass through a joint between the crankcase and bedplate, and easy maintenance was not confined to the bedplate type of construction. The principle of a machined strip used for checking alignment of the engine quickly was also used on engines of fabricated design having underslung crankshafts and light sumps instead of baseplates.

IR. E. R. GRÖSCHEL said that he proposed to limit his observations and comments to the fuel injection side of the paper, departing only for a moment in order to fully endorse the authors' statement under "a) Reliability: *General Considerations*", where they said that the surest method of producing intrinsic reliability was to proceed by a process of logical evolution from one successful design to the next, taking care that the critical parameters proved in the original design were maintained in the new design. This statement deserved thunderous applause from both engine manufacturers and engine users, particularly marine engine users.

The use of test rigs for fuel-injection equipment development was, of course, fully appreciated and valuable data regarding performance and life of the equipment might be gained. However, care should be taken, when applying results obtained from injection-equipment test rigs to engine conditions, unless injection into a pressurized medium was strictly simulated. He wondered if this was done in the case where nozzle gallery pressure was found to be lower than the prevailing gas pressure with a needle still open which permitted gas entrance into the nozzle gallery. The cure adopted, he ventured to guess, was to lighten the reciprocating mass of the injector.

His company, having always designed and manufactured their own fuel-injection equipment for their medium-speed Diesel engines, had always been protagonists of the low inertia injector, i.e. having needle springs acting directly upon the needle without the intermediary of a push rod. Of course, the spring was thus placed into a somewhat uncomfortable position (heat and space), but unorthodox spring wire sections helped with the space and chromium/silicon wire with the temperature

The "hydrodynamic condition" referred to in the paper, he thought, was a spill wave receding too fast, which the normal oscillating system of the injector was incapable of following. A stiffer spring might help in border-line cases. A trick, imparted to him some time ago by Mr. J. F. Alcock (and gratefully acknowledged) was to watch for gas bubbles in the injector leak-off connexion; should gas pass into the nozzle, it would generally pass through the cylindrical lapped part of the needle-nozzle bore causing lacquering of the lap fit and would finally appear in the leak-off pipe, where a plastic tube would facilitate observation. He said that, acting upon this recommendation, his company sometimes used development engineers as "bubble watchers".

He thought the authors should be congratulated on having such confidence in the precision of cathode ray oscillogram interpretation. He had tried something similar and the result had varied between ½ degree and 2 degrees cam angle, depending upon the thickness of the pencil point used and the condition of the interpreter.

He asked if the line pressures in Figs. 13(a) and 13(b) referred to full load conditions. If they did, they were remarkably low. Also, the difference between maximum line pressure and maximum gallery pressure was rather large, being 1,650 lb./sq.in. in Fig. 13(a) and 1,800 lb./sq.in. in Fig. 13(b). He wondered if the edge filter (which, after all, was the major throttling component between line and gallery) could be responsible for this pressure discrepancy. Wall friction could hardly account for it with the customary maximum flow velocities of about 90 m./s.

Dealing with the chapter on heavy fuel operation, he said that he was intrigued by the statement that the lack of cooling ability of preheated fuel should be responsible for high nozzle tip temperatures. The fuel temperature given was 160 deg. F. (71 deg. C.) for a blended fuel of 300 sec. Redwood 1 viscosity; gas oil, not preheated, would reach the nozzle, after passing through the compressive cycle of the fuel pump, not very much cooler. He ventured to think that the hotter nozzle tip of the heavy fuel operated engine was more the result of the slower burning of the heavy fuel, with a resulting larger heat rejection to, and heat absorption from, the nozzle.

He said that he understood that the mechanism of trumpet formation was a function of the lighter fractions of the blended heavy fuels boiling off in the nozzle sac and squeezing the heavier fractions out of the nozzle holes where they carbonized. Carbonization temperatures were much higher than the measured nozzle tip temperature of 392 deg. F. (200 deg. C.). He said that he would be very grateful if the authors could provide information about the precise location of the thermocouple on the nozzle tip. His company had measured nozzle seat temperatures (the thermocouple being located within a millimetre from the nozzle seat) and temperatures obtained on one engine type, depending on engine rating, cylinder head design and fuel, had reached 464 deg. F. (240 deg. C.). This surely indicated nozzle tip temperatures of a far higher order. These nozzles had been uncooled and made from heat-resisting nitriding steel which maintained the seat hardness at elevated temperatures.

He said that he envied the authors the low maximum cylinder pressure of only 1,350 lb./sq.in. at 220 lb./sq.in. b.m.e.p. This should enable them to get away with an injector release pressure of only 2,500 lb./sq.in., resulting in a closing pressure of 1,720 lb./sq.in., thus still having a comfortable margin available above the maximum gas pressure.

In conclusion, he said that the paper would always have a place of honour in his hydraulics department, having already sent several slide rules into a semi-heated condition.

MR. J. F. ALCOCK, O.B.E., B.A., said that the piston crown was described as high-tensile steel, which term covered a lot of compositions. It would be valuable to have either the thermal conductivity or the composition.

Fig. 6 showed a wet-side temperature of over 464 deg. F. (240 deg. C.). He said that it was a rough general rule that one was apt to get coking on the surface if one went over 392 deg. F. (200 deg. C.). He wondered if this had been observed by the authors. He also asked for the velocity of the oil.

Turning to valves in cages, he said that one considerable advantage of the cage was that it very much reduced the flux from the valve seats to the cylinder head. Since the cylinder head was a complex casting, a large concentrated heat flow from the valve via the seat was undesirable. In smaller engines, which did not have caged valves, this was a very common cause of cracking between the seats. He cited the paper of Mr. Fujita* as an example of this.

Turning to valves, he said that he had noticed in the paper that while the temperature of the seat had been reduced by the cooled cage, the temperature difference between the centre of the valve head and the seat was increased. This would increase the thermal stresses, and the risk of seat cracking due to thermal stress. He gathered that the idea of sodium cooling had been discarded, but he thought it might be a useful idea, not from the point of view of cooling the valve seat, but of reducing the thermal stress. Of course, what the sodium-cooled valve did was to pass the flux back to the valve guide, but there it could be coped with quite well.

He then referred to crankcase explosions. These were practically non-existent in small engines, but they did occur from time to time in large engines and were extremely nasty things. It would be valuable to have information on this subject and to know, from the point of view of safety, what the difference was between the trunk-piston and the crosshead engines.

DR. W. P. MANSFIELD was particularly interested in the authors' lubricating oil consumption tests which were briefly described on page 336 of the paper and to which Fig. 21 referred. In some investigations on this subject on smaller engines, the British Internal Combustion Engine Research Institute Ltd. had tried increasing the oil pressure by reducing the area of the bearing surface of the ring, but this had had little effect. However, changes of wall pressure, made by varying the radial thickness of the ring, had a marked effect. He said that it would be interesting to know by what method the authors had varied wall pressure, apart from the change to a spring-loaded ring which was mentioned in the paper.

MR. A. J. S. BAKER (Associate) said that the authors had produced a remarkably full description of what could only be described as an exceptionally well developed engine. Of particular interest to people connected with lubrication research was the systematic work which had gone into piston development. This work had obviously paid off handsomely in the modest temperatures, so clearly denoted in Fig. 6. It would be interesting to see how far the really excellent fuel utilization rates, indicated by the exceptional fuel consumption rates, had contributed to this. For instance, metal temperature comparisons taken at the same time as the variable valve timing tests described in Fig. 10, right-hand side, would perhaps illustrate this point. Fig. 10 itself suggested that an even broader area of minimum specific fuel consumption might be possible with automatically-varied valve timing. He wondered if the authors had considered such a possibility.

Looking at the Fig. 10 data points for a constant b.m.e.p. of 200 lb./sq.in., it appeared that a fairer mean curve would have a pronounced and steepening hog rather than the sag shown by the authors. He asked the authors to justify the mean line they had postulated.

With regard to the important work which had been done to optimize fuel injection characteristics, it was interesting to consider the authors' needle lift/sac pressure relationship conclusions, in the light of the modification which appeared to have been carried out. Apparently the unloading value had been increased between (a) and (b) of Fig. 13. Did the authors attribute this fact to the reduction in incipient secondary lifting indicated in the needle-lift diagram (b)? Presumably the injection rate

* Fujita, H. 1961. "Service Records of Mitsubishi Nagasaki Diesel UE Type Engines and Improvements Made on the Engines". *Trans. I.Mar.E.*, Vol. 73, p. 37.

had also been increased and this had been accommodated by permitting an increase of needle lift. If this were the case, might not needle-lift increases have to be closely controlled in service operation, he asked. Likewise, the fuel-line pressure diagram (b) was presumably taken at the nozzle end of the fuel pipe. He wondered if it had been necessary to tune the fuel pipe length to control the magnitude of the secondary pressure wave so as to eliminate secondary injection.

He thought that the general conclusion to be drawn from the fuel injection work was that engines of comparatively low speed needed the same careful attention as high-speed engines. It would be very interesting to see a comparative investigation in certain large-bore, slow-speed engines, having several nozzles connected to a single fuel-pump element. The results of the careful work done by the authors in this direction were demonstrated in Fig. 14. By extracting data points from the fuel consumption loops, it was interesting to note that the fuelling lines for both the K engine and the K Major at different speeds were virtually identical. The fuelling lines, from around 50 lb./sq.in., b.m.e.p., on the lowest curve, were unusual in their linearity. Perhaps the authors could supply fuel consumptions at very low loads which would give a clearer indication of the likely f.m.e.p. From the data published it was evident that this must be of a very low order and comparable to that obtained with the largest low-speed engines. This point might be worth bringing out since it was fashionable to quote mechanical efficiency for large two-stroke marine engines, and for a given f.m.e.p. this would generally favour the four-stroke engine with its higher b.m.e.p.

He asked the authors to indicate whether the performance curves were obtained with the fuel described at the top of page 333, and if not, he said that he would like them to give details.

He said that the water-cooled, exhaust-valve cage and valve rotators had made a major contribution to operation with low-cost fuels. Presumably some provision had been made to prevent boiling in the small seat-cooling passages in the event of sudden shut-down, as might be expected in main marine engine application.

He thought that the notes on lubricating oil consumption were very relevant, as was the investigation on oil control by the upper scraper ring. He asked the authors to elaborate on this by indicating the degree of control exerted by the other rings in the pack. For instance, could a reduction in radial pressure of the upper scraper be tolerated by increasing the load on the lower one?

The rig described to evaluate piston-ring quality resembled a variety of test rings used for different purposes. Experience with these had indicated considerable scatter of wear results, particularly at high rates of wear. Perhaps the authors could indicate the significance of the weight loss figures they had quoted in Table IV. He wondered whether they had observed a pattern of related ring to bore wear rates for the different material combinations tested. Had any significant differences in piston-ring groove and ring side wear rates been observed when different irons were run in the steel piston crown? The authors had not shown the metallurgy of the piston crown, but other applications of high-tensile steel had suggested that steels containing appreciable nickel contents might produce increased wear rates in the presence of boundary lubrication.

MR. J. A. COWDEROY, B.Sc. (Member) said that, as the K Major engine had been developed specifically for marine propulsion, he had been surprised that Fig. 24 did not show the performance plotted against speed on a propeller law basis. He would particularly like to see the compression pressure included in such a plot, because he had the impression that many builders of marine Diesel engines overlooked the implications of the propeller law, which related power to speed in a ship, particularly when applied to turbocharged engines. If this law were assumed to be a cube law it meant that, if the engine was developing full power at full speed, it was only required to develop as little as 12½ per cent of that power, even at half speed, and as ships not infrequently proceeded at speeds lower than full, this condition did occur now and then.

The engine referred to in the paper had been developed to run on heavy fuel oil. He thought the authors would agree that the turbocharged four-stroke, trunk-piston engine could be troublesome from the point of view of combustion when operating at low loads on heavy fuel, and from some figures he had seen for other engines, which showed a drop in compression pressure, from 665 lb./sq.in., at full load and speed, to 355 lb./sq.in., at 60 per cent speed and 22 per cent of full load, he strongly suspected that the relatively low compression pressure under those conditions was one of the principal reasons for this. Whilst combustion might be quite satisfactory, under those conditions, when the engine was new and in first-class condition, with the accumulation of wear of not only liners, but injection equipment, he thought that the low compression pressure was certainly a contributory factor. He would be glad to have the authors' comments on this.

On the question of the operational control of turbocharged medium-speed engines in ships, particularly in view of the increase in the number of ships with bridge control of the engines, he was convinced that the fuel injection pump rack position should be governed to some degree by the booster pressure. A few years ago, in a certain cross-Channel ferry, which was propelled by two turbocharged Diesel engines under bridge control, it was found, a very short time after the ship had gone into service, that the engine crankcase oil had become very dirty indeed. The reason for this was soon discovered: the bridge control of the engines had been operated on leaving harbour as if it had been an engine room telegraph, with the result that the engines smoked like chimneys until the turbochargers had time to catch up and provide enough air for clean combustion. Under these conditions the oil soon became filled with fuel soot. Instruction to the master as to the correct rate at which to increase engine power soon cured the trouble. He felt that where turbocharged engines were installed in ships, some form of control over the rate of increase of the delivery of fuel to the engine was essential.

MR. C. C. J. FRENCH asked a question concerning thermal stress. The thermal stress rig shown in Fig. 4 was interesting and provided an ingenious method of investigating a problem which was becoming more and more important as engine ratings were increased. This rig was useful in that it was applicable to asymmetric bodies, as well as to those that were bodies of revolution. In this respect the two-piece piston shown in Fig. 6, appeared to be a body of revolution. Computer programmes were now available for calculating the thermal stress of such components. He wondered whether the authors had tried a check calculation to see whether there was any sort of agreement between the rig and a computer. His own rather limited experience so far, with a computer approach, had been more valuable in showing up limitations in the computer programme than in giving realistic piston thermal stresses, the problems being largely the rather complex shape of pistons.

Turning to the inlet-valve wear, he said that he was glad that Dr. Pope, in his presentation, had elaborated on his wear factor, which Mr. French had found somewhat incomprehensible as it stood in the paper. He agreed that lack of lubricant was the main cause of heavy inlet-valve and seat wear in turbocharged engines. It was most interesting that the authors had found thickening the head of the valve so effective in reducing this wear.

Touching on service experience, he said that two years previously a paper* had been presented, giving details of service experience on an engine of very similar size and rating. He thought that everyone looked forward to the time when the authors would be able to give comparable details of exhaust-valve life, cylinder-liner and cylinder-ring wear on the K Major, when operating on residual fuels. In this connexion, if the authors were proved correct in their aim of up to 3,000 hours between servicing of injectors and exhaust valves, this would be a most valuable step forward.

* Henshall, S. H., and Gallois, J. 1964. "Service Performance of S.E.M.T. Pielstick Engines." *Trans. I.Mar.E.*, Vol. 76, p. 445.

Correspondence

COMMANDER E. R. MAY, D.S.C., R.N. (Member) wrote that it was some ten years since the Pielstick PC1 had begun to make its significant contribution to the propulsion of ocean-going ships, and during the whole of this time it had been without any effective medium-speed competitor. The K Major must now be judged by comparison with the Pielstick PC2, with which it would be in direct competition in every field.

Power for power, the British engine was rather larger and heavier than its French competitor. In some applications this would not matter very much. Commander May imagined that the relative first cost of the two engines would be very significant, assuming that they had equal ability to burn heavy fuel. The Pielstick had never been a cheap engine and, in its PC1 form, its exhaust-valve life on heavy fuel did not always prove impressive. Its popularity had stemmed from its introducing high-speed engine standards of accuracy into the marine engine field, with a refreshing freedom from the very heavy maintenance work that marine engineers often experienced on propulsion engines, less well made and indifferently developed.

Over the last few years, the major British medium-speed Diesel firms had caught up the leeway in standards of manufacture, and also had undertaken most impressive programmes of detailed development. It therefore seemed that the K Major would meet international competition successfully, would extend the market gained by the K engine, and join the Pielstick in propelling large merchant ships.

Commander May noticed that the authors had made a rather misleading reference to short-life engines being permissible in naval work. This had never been so (except in motor torpedo boats). Submarine engines were designed and produced by the Admiralty between the two wars in an attempt to produce better—not lighter—engines than those available from industry at the time. After the last war, the Admiralty worked hard to persuade industry to adopt modern standards in development and manufacture of long-life engines up to 9½-in. bore, but success was only achieved gradually and at substantial public expense.

In Germany, before and after the war, and in France at the present day, engines designed partly for naval purposes had met with widespread commercial success. This had come about through recognition that naval and commercial requirements could be designed into the same engine with advantage to all concerned.

Possibly the most remarkable feature of the K Major was that it had achieved so much while retaining cast iron for frame and bedplate. Rigidity was essential to maintain bearing oil film geometry within acceptable limits and cast iron was about twice as flexible as steel. A cast iron frame must have heavier scantlings than a steel frame, the cylinder centres must therefore be further apart, and bending moments increased in consequence. On the other hand, cast iron was cheaper than steel, and development of modern cast irons had done much to make this material more attractive. Fairbanks Morse had used cast iron extensively in their new large opposed-piston, medium-speed engine. Other manufacturers were, the writer believed, using steel for comparable engines and had also chosen the two-stroke, valve-in-head arrangement.

Soon, at least four of these valve-in-head two-stroke engines (one of them British) would be competing with the K Major and the Pielstick in the rapidly expanding world market for large medium-speed engines. It was obvious from this paper that Mirrlees had planned to secure their share of this market.

It would be interesting to know the authors' view on transmission suitable for employing, say, two K Majors to drive a single propeller shaft, and whether their company proposed to offer complete propulsion units—engines and reduction gear.

MR. G. H. HUGHES (Member) commented, in a written contribution, that the increase in power output should in no way alarm prospective users, because even the ultimate aim of 528 b.h.p./cylinder, with 250 lb./sq.in., b.m.e.p., and 1,400 lb./sq. in., peak pressure, represented only 2·98 b.h.p./sq.in. of piston crown—almost identical to the power per square inch on the crown of the Maybach engine with pistons of similar construction.

It would be interesting to know the cooling oil flow rate, (he suggested approximately 1½ gal./b.h.p.-hr.), since crown and ring life depended on adequate cooling and, in this respect, the oil feed through the connecting rod might prove to be the limiting factor. Given adequate cooling, it was known that this form of piston would stand greater power per square inch of crown area, as shown in the two-stroke cycle Ruston and Hornsby A.O. engine, when published figures showed over 5 b.h.p./sq. in.

His company's experience of materials for such piston crowns indicated that thermal fatigue tended to become the limiting factor and this depended principally on coefficient of expansion and thermal conductivity. Had the authors considered one of the high-nickel alloys to minimize the effect of high operating temperatures, or the high-conductivity copper chromium alloys?

A further aid to cooling was increased valve overlap. Had the effect of this been explored with respect to piston crown and piston ring temperatures?

The scraper ring arrangement permitted adequate lubrication of the skirt or crosshead length of the piston, but when oil control became a problem after extended service, there might be a temptation to fit a highly-loaded ring in the skirt groove, with possible risk of seizure. To avoid such possibilities, had the authors considered omitting the skirt-ring altogether and adjusting the upper scraper ring accordingly?

It was noted that three taper-faced rings were fitted below a parallel-faced, chrome-plated ring in the top groove. There might be a tendency to blow-by during the initial running of the engine with this arrangement. Had blow-by readings been taken during test work and had any indications been noted?

An important factor in piston-ring material was compatibility with cylinder liners. Not all materials were suitable in this respect, but might be metallurgically sound and, therefore, of good quality.

It was not surprising, therefore, that a random flake graphite iron had given satisfaction in this size of engine.

With regard to cylinder liner material, was this also random flake graphite? How was the bore machined, and what type of surface was produced?

With regard to the outside diameter, was the liner free from water side attack and what precautions might be taken to deal with this possibility at the higher ratings?

On the question of heat dissipation, was it known what proportion of heat was transferred through the piston crown to the cooling oil and through the piston rings to the cooling water?

MR. J. H. MILTON (Member) wrote that it was stated, on page 327, that to produce a reliable machine one had to pro-

ceed from one successful design to the next, taking care that critical parameters proved in the original design were maintained in the next.

With regard to the critical parameters shown in Table I, it was rather surprising to see that gudgeon pin, or small-end bearings, were not mentioned, as these bearings could be troublesome and also, on occasions, connecting rods had split lengthwise through concentrated eye loading.

Perhaps the authors would care to comment on this subject, and give details of the design of their small-end bearing with particular reference to the bush—whether it was floating or not—and its material.

With regard to the piston design, as shown in Fig. 3, it would be interesting to have the authors' views on the importance of the distance from the crown to the top piston ring, and also further enlightenment on their statements that: a) heat resisting "helicoil" inserts were used to carry the studs and that these acted as a "heat barrier" for these studs; b) that disc springs were fitted under the castle nuts on these studs to increase the resilience of the assembly. Did this mean that they had accepted the fact that movement must take place between the piston crown and the body, and if so, did fretting take place with ensuing leakage of oil across the jointing face?

With further reference to Table I, it was noted that the maximum permissible bearing loads for the main bearings and bottom ends were given as 2,500 and 5,000 lb./sq. in. respectively. Some enlightenment as to how these limitations were arrived at would be of interest

On page 336, under "Space and Weight", the authors made a good case for the cast iron engine, stating that few fabricated structures were able to avoid fillet welds in load-carrying regions, and that the fatigue strength of such welds might be as low as plus or minus 1·2 tons/sq. in., and that even butt welds had only a fatigue strength of plus or minus 3·8 tons/sq. in., compared with 5 tons/sq. in. for a good quality cast iron. If these figures were correct, it was difficult to understand why, apart from the saving in weight, so many other engine builders had adopted fabricated designs, especially as also, in the event of damage resulting from the failure of a bottom-end bolt, a cast iron engine did not suffer distortion and could usually be "patch" repaired, whereas the fabricated structure was usually distorted and had to be renewed.

It was noted that oil was used for piston cooling and lubrication and in this connexion it would be interesting to know if the authors had any relative figures on lubricating oil capacity (e.g. gal./h.p., in circuit) for the engines forming the subject of this paper, as compared with slow-speed, direct-drive Diesels.

Furthermore, in the case of direct-drive, slow-speed engines burning heavy oil, it was found essential, on account of crankcase corrosion, to isolate the cylinder bottoms from the crankcase—what precautions, beyond using an inhibited lubricant, were being taken to prevent such corrosion taking place in the engines produced by the authors' company.

In conclusion, he would be grateful if the authors could briefly state why, in comparison with the builders of large, slow-speed, direct-coupled engines, they had chosen to develop the four-stroke cycle engine instead of the two-stroke cycle engine.

COMMANDER E. B. GOOD, O.B.E., R.N. (Member) wrote that, when a new engine design was introduced, it was natural to compare its rating with those of competitors. A true comparison of ratings should take into account many design features, but an indication of the mechanical and thermal loading problems which the manufacturers had to overcome could be obtained from the output per cubic inch of swept volume and the output per square inch of piston area. These factors had been plotted against cylinder bore, for a number of modern turbocharged engine designs, in Figs. 25 and 26.

The factors for the K Major engine had been plotted at each of the development stages referred to in the paper and it could be seen that these ratings lay neither too adventurously

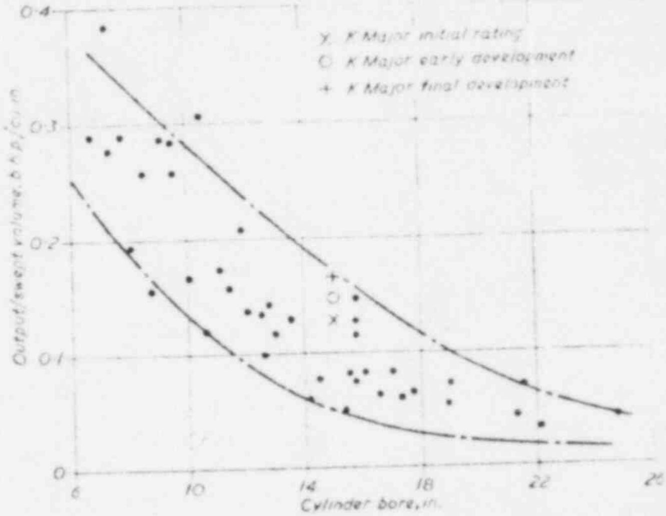

FIG. 25—*Relation between output/swept volume and cylinder bore*

above, nor too much below, the lines marking the upper limit of current design.

Reference had been made to the use of small centrifuges mounted at the engine for the bypass purification of the lubricating oil. A marine installation of more than 3,000 b.h.p. would normally justify the use of a motor-driven centrifuge for the continuous purification of lubricating oil. Such a system also enabled the whole of the lubricating oil charge to be purified in harbour at the end of each trip, a practice adopted by many owners. Commander Good asked the authors whether they considered that the engine-mounted centrifuges would avoid the necessity for a separate motor-driven unit, particularly in a heavy fuel burning installation, and, if so, could they give an indication of the time after which cleaning of the units would be required. It was assumed that provision was made to cut off the flow to individual centrifuges, to permit them to be cleaned while the engine was running.

The attention which had been paid to the design of the fuel injectors and exhaust valves was very welcome. The maintenance of these items probably represented the largest work load for the ship's engineers. It was considered that a period of 5,000 hours between overhauls would not be an unreasonable aim for the exhaust valves.

Shipowners were becoming increasingly concerned about the noise levels in engine rooms and this was reflected in the

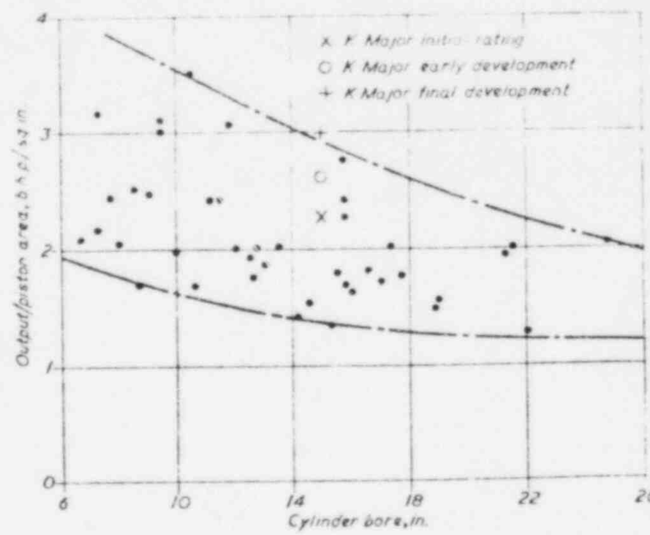

FIG. 26—*Relation between output/piston area and cylinder bore*

number of new ships which had insulated control rooms. With a maximum cylinder pressure 25 per cent greater and a maximum speed 18 per cent greater than its predecessor, the K Major might be expected to be considerably noisier. However, it was possible that the many design changes which had been made had at least partly counteracted the tendency to higher noise levels. It would be useful if the authors could provide any comparative noise measurements for the K and K Major engines.

With the advent of a new medium-speed Diesel engine, it was inevitable that designers of naval machinery installations must ask themselves whether this new engine was suitable for warships. In this respect, section f) on "Space and Weight" was relevant and one could observe that cast iron, whilst it had many admirable properties, was not the best of materials for shock. Perhaps the authors would like to comment on whether they intended to offer a naval version of this engine in due course.

Authors' Replies

Mr. Lowe (replying to the verbal discussion) referred to Mr. Cook's point about 36 cylinders being required for a ship of 18,000 horsepower and said that the paper had tried to show that the power available from that number of cylinders was now twice as great as it was a few years ago, with the same reliability and maintenance requirements.

A comparison of piston-crown temperatures, measured in the engine using templugs, with the temperature distribution produced in the rig and by an electrolytic analogue showed that in the engine, crown temperatures were rather lower than those produced in the rig and predicted by the analogue, while ring-belt temperatures were slightly higher. This difference was attributed to the rather more efficient cooling of the underside of the crown produced by the motion of the piston in the engine. The temperature distribution was proportionately the same in both rig and engine, so that thermal stress measurements should be of the right order.

The authors' company did not use a ratio as high as the 2½ to 3 times increase in fatigue strength of rolled threads over cut threads, mentioned by Mr. Cook. They used a value of 0·25 times the U.T.S. for rolled threads and 0·125 times U.T.S. for machine-cut threads, giving a ratio of 2:1 in favour of rolled threads for fatigue strength.

Both Mr. Cook and Commander Tyrrell had commented on lubricating oils for use with heavy fuel. It was difficult to be precise about the deterioration of oil lubricants because no oil specification defined accurately the requirements of a lubricant for Diesel engines. The type of oil found suitable for this engine was a "good" Supplement 1 level with a quite reasonable alkalinity. The authors used a maximum bulk oil temperature of 170 deg. F. (77 deg. C.), but it must be remembered that this implied that there would be local higher temperatures in the engine of the order of 210 deg. F. (99 deg. C.).

He confirmed that it was not the intention to imply that centri-cast rings were necessarily inferior to sand-cast rings, but the point should be made that the quality of the iron was more difficult to maintain in a centri-cast ring.

Referring to Mr. Henshall's point about specific air flow, he agreed that it was desirable to continue to increase the specific air flow as ratings increased, although the extrapolated figure in Table I showed a small reduction.

There had not been any problem with differential expansion between the steel crown and the cast-iron piston body, probably because the intensive cooling produced a low temperature at the interfaces, as could be seen in Fig. 6. The double line in Figs. 2, 3 and 6 represented a threaded portion which was used to establish the best crown diameter and was a well-known development technique.

The development engine had been run with valve guide diametral clearances as small as 0·001in. to 0·002in. with the force-lubricated guide, and a clearance of 0·003in. to 0·004in. had been arrived at for the final design. Wear rates with the lubricated guide were extremely low and a life of 10,000 hours was expected before replacement of the guide was necessary.

Mr. Gröschel had sounded a word of warning about fuel-injection test rigs and the authors agreed generally with him. Rigs were extremely useful if one was careful in interpreting the results. The suggestion that the problem of back-flow of gas from the cylinder had been prevented by reducing the reciprocating mass in the injector was quite correct and the K Major injector was of the low inertia type, as described by Mr. Gröschel.

The fuel line pressures in Fig. 13 referred to full-load conditions and the authors considered the pressures levels and the difference between line-pressure and gallery pressure to be quite normal in their experience.

He agreed that the higher nozzle temperature for the un-cooled injector with heavy fuel, in Fig. 19, was only partly due to the loss of cooling from the fuel and was also a result of the slower burning of the heavy fuel. The conclusion, however, was unaltered that cooling of the nozzle was necessary with heavy fuel and tests had been carried out, which were too extensive to be fully described in the paper, which showed that the tip temperature was dependent not only on engine load, but also on fuel temperature, water flow quantity and water temperature. The thermocouple for these temperature measurements was located actually at the surface of the nozzle tip.

Mr. Alcock had asked for details of the composition of the high-tensile steel piston crown. This was a 55-ton tensile 1 per cent Cr-Mo steel and he would gladly send exact particulars of the material and of oil velocities to Mr. Alcock. He pointed out that the 464 deg. F. (240 deg. C.) maximum temperature in Fig. 6 was a bulk temperature of the material and that the surface temperature in contact with the oil would be lower. There had been no signs of oil coking on the surface of the cooling chamber.

He agreed that it was very critical to choose the right material, not only for the seat of the valve, but for the valve itself. The relative coefficients of expansion of these two materials and their conductivity were most important.

The subject of crankcase explosions was a general one and not confined particularly to the engine under discussion. Fig. 1 showed that the engine was fitted with explosion doors. His company had experienced one or two cases, in the last seven or eight years, of crankcase explosions in other types of engine where the explosion doors had worked satisfactorily.

Referring to Dr. Mansfield's question about the method of varying the wall pressure of the piston rings, he said that this had been done both by altering the bearing area of a given ring and also by re-designing the ring.

Mr. Baker had suggested that there should be automatic timing on an engine. This was an attractive idea, particularly if combined with automatic timing of fuel injection, but was very difficult to achieve. If some simple and foolproof way of doing this could be found it would be a real achievement.

He said that he could not justify the mean line for the 200 b.m.e.p. data on the right-hand diagram of Fig. 10. The measured points indicated the "hog", suggested by Mr. Baker, but he could not explain why this should be so and hence had merely indicated the downward trend which he thought to be of greater significance.

Referring to the injection diagrams, he said that the un-loading volume had been increased from (a) to (b) in Fig. 1 and had a very strong effect on the tendency for secondary injection. The injection rate had also been increased and, as already stated in the reply to Mr. Gröschel, a low-inertia type of injector had been adopted. The fuel line pressure was measured halfway along the injection pipe in each case and th

injection pipe length was the minimum possible for the engine. Mr. Baker's remarks about the Willan's lines were quite valid and, in fact, the twelve-cylinder engine had shown a mechanical efficiency of 93 per cent. The performance curves were obtained using distillate fuel.

The piston-ring pack, described in the paper, was deliberately designed so that the scraper ring above the pin was more severe than the lower scraper ring. The authors considered it important to maintain an adequate oil film over the body of the piston so that the lower scraper ring was only intended to remove excess oil to prevent the upper ring becoming flooded. The weight loss figures, in Table IV, were only significant in comparison to one another as they were obtained from a rig running completely unlubricated and could not be related to conditions in an engine cylinder. There had not been any controlled tests to measure wear rates in the grooves of the steel crown with piston rings of different irons.

In answer to Mr. Cowderoy's question about compression pressures, he said that, at 525 r.p.m. and full load, the compression pressure was 900 lb./sq.in., coming down linearly with horsepower to 380 lb./sq.in., at 100 r.p.m., i.e., effectively no load. The compression pressure, corresponding to the 60 per cent speed, 22 per cent load condition, quoted by Mr. Cowderoy, was 430 lb./sq.in. He agreed with the contributor that low compression pressure, or rather compression temperature, could contribute to inferior combustion in a worn engine at low load.

Mr. Cowderoy's suggestion that marine propulsion engines should be treated like locomotive engines, from the point of view of preventing the driver from accelerating too fast, was a good point. The necessity for the turbocharger to accelerate was often overlooked when rapid increases in load were called for, and it might well be necessary to apply the locomotive type of fuel rate control to marine engines.

In reply to Mr. French, he said that the computer approach to piston crown thermal stresses had been to construct an electrolytic analogue which, as mentioned earlier, gave quite good agreement with the thermal rig. Work was currently in progress on a digital programme which would calculate stresses directly, whereas the analogue only gave temperature distribution from which stresses could be calculated.

The numerical value of the wear factor for inlet valves obviously applied to one's own engines, but a manufacturer could apply the formula to obtain values from his own engines. In this connexion, he said that the experimental work, from which the wear factor was developed, was described fully in reference (2).

Dr. Pope said that before the meeting closed he would like to make one or two comments about some generalities which had come up during the discussion.

He said that they were getting to the stage in the medium-speed engine industry where the research and development effort of the engine builders was outstripping the component builders, and he could foresee, in the not too distant future, that engine development might well be held up because of lack of blowers and injection equipment. He hoped the supply industry would persevere with enthusiasm for the highly-rated medium-speed Diesel engine as much as the engine builders were.

He thought that the problem of the maintenance of medium-speed Diesel engines in the marine world should be judged objectively. Obviously, in the medium-speed engine one was going to have more parts, but there was a world of difference between handling a 15-in. piston and handling a 30-in. piston. The factors involved were not just the number of parts, but the way in which they could be manipulated and a statistical analysis of what were the major and minor faults. His view was that if this analysis were carried out scientifically one would find that the medium-speed engine could stand on its own, even with regard to maintenance.

With regard to cast iron, he said that each problem must be judged on its merits. The fatigue strength of a good cast iron was near to that of a good steel. There were other advantages in using cast iron. One knew that one could obtain good castings with cast iron for the size of engine he was discussing and it was an easy material to handle. Size for size, his experience had been that cast iron came out cheaper, therefore if one had a material which was as good as another and was cheaper, one had to have a very good reason for not using it. He could only see one reason for precluding its use and that was if weight were a predominating factor. However, when one considered the rest of the engine room equipment, the tankerage and the fuel capacity, one would find that there was a difference of one or two per cent between a welded design and a cast iron design so that, for commercial shipping, this was a marginal consideration.

He pointed out that with an in-line engine with an underslung crankshaft, one could have a very nice stress line pattern which, on the drawing board, looked very attractive and almost impossible to improve upon, but when one came to a "V" engine with side by side connecting rods, so that the opposite liners could not be in line with each other, the stress pattern did not look quite so elegant. He accepted that the cast iron bedplate was more difficult to design because the stress pattern was more complex, but once one had designed it and got a good design one was simply comparing one good design with another. It was also a question of continuing from a well-tried engine to the next generation, without departing from well-proven design principles. His company had now completed its one thousandth K engine and had over 300 of them in marine application. Over a third of the engines were running day in and day out on heavy fuel.

AUTHORS' REPLY TO WRITTEN CONTRIBUTIONS

The authors wrote that they entirely agreed with Commander May's appreciation of the rapidly increasing demand for large medium-speed engines. They had deliberately restricted the paper to the development of the K Major engine itself but their company was certainly proposing to offer complete propulsion units for geared installations with either single or multi-engine inputs.

In reply to Mr. Hughes, the authors wrote that the piston cooling-oil flow was 1·7 gallons/b.h.p.-hr. at the current full-load rating. The cooling design was such that the operating temperatures of the piston crown were well within the thermal fatigue limit of the steel used, as could be seen from the isotherms of Fig. 6. The work that had been carried out on the optimization of valve timing and combustion characteristics, which was described, had been aimed at obtaining the best thermal efficiency from the engine and not at reduction of component temperature by scavenge air cooling, which they considered was a relatively inefficient method of controlling component temperatures. The temperature of the critical parts of the engine, such as exhaust-valve seat, injection nozzle, and top piston-ring groove, was controlled by direct cooling.

As mentioned in the reply to Mr. Baker, the lower scraper ring in the skirt was relatively mild and the upper scraper more severe to ensure adequate lubrication of the piston skirt. The taper-faced compression rings were an advantage in initial running as they bedded in very quickly on a narrow circumferential band. Initial running-in had been the subject of a good deal of investigation on the test bed and the best results had been obtained with a relatively rough liner surface, which was honed to a C.L.A. of about 100μ, the liner being a random flake graphite iron, slightly softer than the piston-ring material.

Calculations of liner frequency and vibration amplitude were included at the design stage to avoid the possibility of water-side attack. The heat dissipation through the piston rings to the cooling water could not be directly measured in the engine because of the heat received directly by the liner from the combustion gases. By reproducing temperatures and total heat flow in the rig, the proportion of heat flowing to the cooling oil was about 75 per cent of the total and the heat to the liner was 25 per cent.

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The Development of a Highly-rated Medium-speed Diesel Engine of 7,000—9,000 Horsepower for Marine Propulsion

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The design considerations and development tests are described which have resulted in the production of the Mirrlees-National K Major engine, which has a current commercial rating of 3,000 to 7,500 b.h.p. in 6 to 18-cylinder units, and a projected future rating of 9,000 b.h.p. in 18 cylinders.

The Mirrlees K engine has been well established for over 12 years, some 980 engines now being in service for power generation and marine propulsion. Of these engines, 250 are operating on heavy fuels with viscosities ranging from 200 to 4,600 seconds Redwood I, representing over 650,000 horsepower. The objective in the design and development of the K Major engine has been to increase the specific power output by 50 per cent and at the same time to maintain or to increase the safety factors possessed by the original K engine. These factors, which determine the ability of the engine to operate on residual fuels with low maintenance and high availability, are discussed and the achievement of the objective is illustrated.

Component parts of the engine are described in turn, with details of the methods of measurement of pressure and temperature levels, air flow and wear rates in test rigs and in a prototype three-cylinder engine which was equipped with special features, such as a camshaft with variable timing, to facilitate development work.

The test results obtained on the first 12-cylinder KV Major engine are shown to confirm the performance expected from the rig and prototype engine tests.

INTRODUCTION

In general, the requirements of a marine propulsion engine are:

a) reliability;
b) low fuel consumption;
c) the ability to burn heavy fuels obtained in any part of the world;
d) low lubricating oil consumption;
e) low maintenance requirements;
f) minimum space and weight in keeping with a) and e).

These requirements are obvious but can only be achieved if certain basic principles in design are followed. The paper is divided into sections, each dealing with one aspect of design which affects these overall qualities.

However, before detailing these definite sections, some observations must be made on the application for which the engine is to be used and its suitability for that application. An engine developing 7,000 to 9,000 b.h.p. in 18 cylinders would be ideal for medium-speed marine propulsion since the power range available would be from 3,000 h.p., in a single six-cylinder engine, to 18,000 h.p. with twin 18-cylinder engines. This power range covers a large section of the marine market, as illustrated in Fig. 1, so that if conditions a) to f) can be achieved a worthwhile market should exist for such an engine.

The initial design study showed that the dimensions of the Mirrlees-National K engine (15-in. bore × 18-in. stroke) would fit this power range very well, if the new design embodied the modern features resulting from research and development which would enable high specific outputs to be obtained whilst retaining economy and reliability. At 500 r.p.m., and 200 lb./sq. in., b.m.e.p., this size of engine would give 402 h.p./cylinder, while at 525 r.p.m., and 220 lb./sq. in., b.m.e.p., 465 h.p./cylinder would be developed. The design of the K Major engine was based on a continuous rating of 250 lb./sq. in., b.m.e.p., at 525 r.p.m., giving 528 h.p./cylinder, and the development programme was planned to achieve this rating, using heavy fuel, in the three stages mentioned.

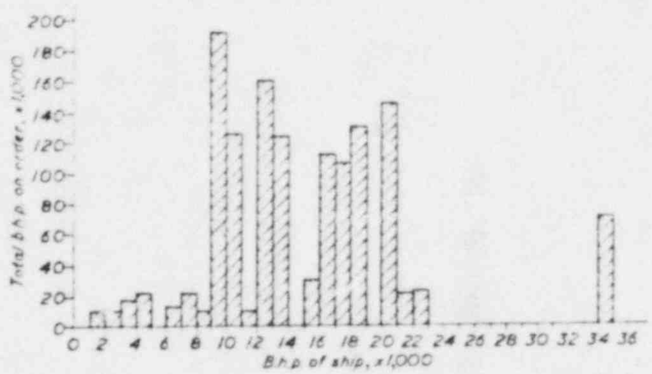


FIG. 1—*Distribution of horsepower for ships over 2,000 d.w.t. on order in Great Britain in March 1965*

At the present time, the K Major is released for the commercial market at a rating of 200 lb./sq. in., b.m.e.p., at 500 r.p.m., and development testing for the second stage of 220 lb./sq. in., b.m.e.p., at 525 r.p.m. is well advanced.

A cross-section of the engine, showing its general construction, is shown in Fig. 2 and the details of the design will be dealt with in the following sections of the paper under the headings a) to f) already given.

* Research and Technical Director, Mirrlees National Ltd.
† Chief Development Engineer, Mirrlees National Ltd.

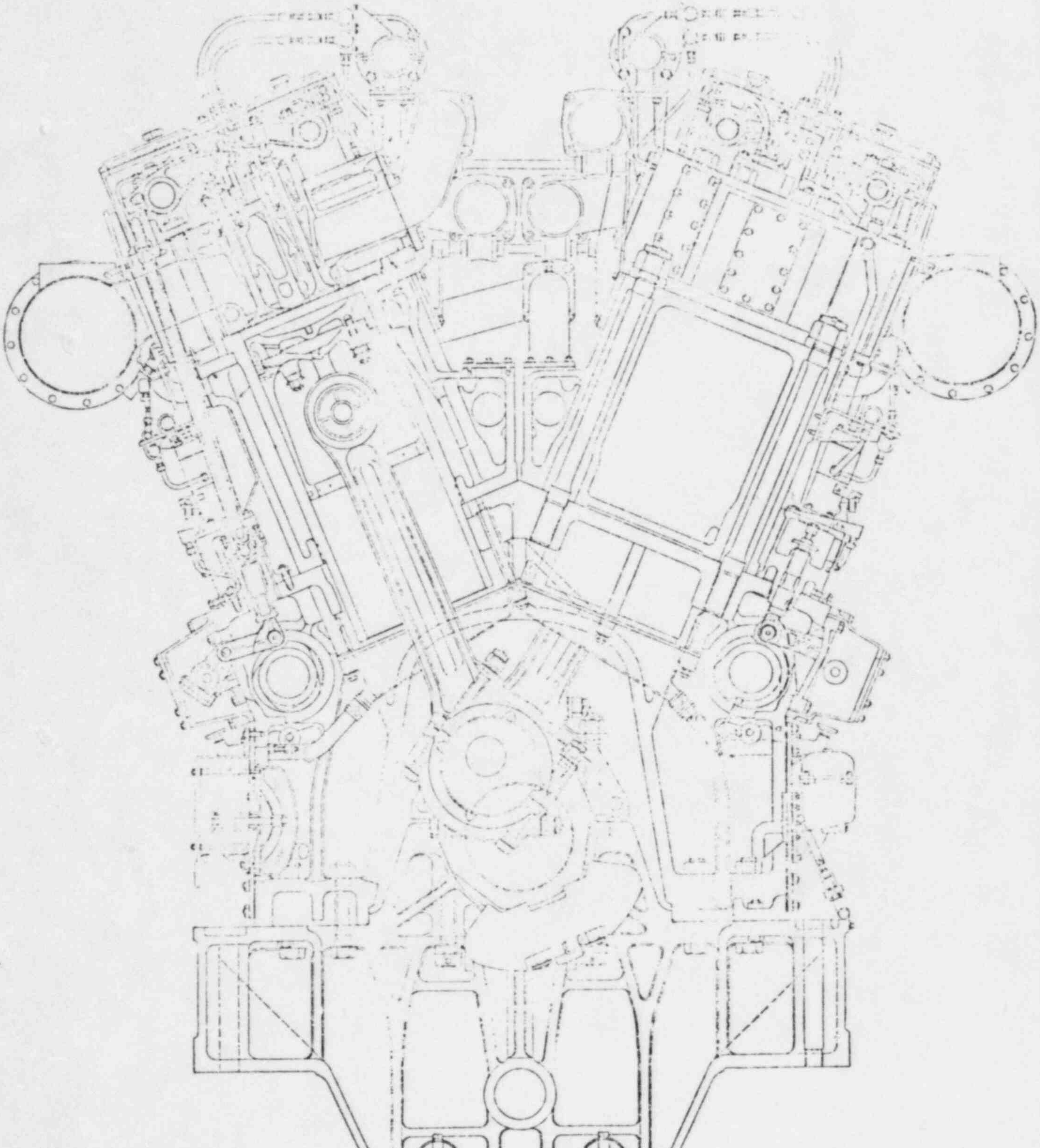


FIG. 2—*KVD Major engine—Cross-section*

a) RELIABILITY

General Considerations

Experience in engineering has shown that one of the surest methods of producing intrinsic reliability in a complex piece of machinery, such as a Diesel engine, is to proceed by a process of logical evolution from one successful design to the next, taking care that the critical parameters, proved in the original design, are maintained in the new design. From the authors' experience of continuous-duty Diesel engines, the critical parameters to be ... fully watched are:

1) exhaust temperature after the valves should not exceed 820 deg. F. (440 deg. C.) with uncooled valve seats and 930 deg. F. (500 deg. C.) with cooled seats;
2) top piston ring groove temperature should not exceed 430 deg. F. (230 deg. C.);
3) injector nozzle tip temperature should not exceed 350 deg. F. (180 deg. C.);
4) exhaust valve seat temperature should not exceed 1,020 deg. F. (550 deg. C.);
5) lubricating oil consumption should not exceed one per cent of fuel consumption at full load and if possible should approach 0·5 per cent of full load fuel consumption;
6) all bearings should be well within their load-carrying capacity;
7) the stressing of all components, both in fatigue and static loading conditions, should be such that an adequate factor of safety exists.

Some of the more important design and performance characteristics of the K Major engine are compared in Table I with those of the earlier and successful K engine, the comparison showing that safe values of the critical parameters have been maintained and, in many cases, improved.

TABLE I

Parameter	Mirrlees KV12 engine	Mirrlees KV12 Major engine		
	154 lb./sq. in. b.m.e.p. 450 r.p.m.	200 lb./sq. in. b.m.e.p. 500 r.p.m.	220 lb./sq. in. b.m.e.p. 525 r.p.m.	250 lb./sq. in. b.m.e.p. 525 r.p.m.
Moment of inertia of bedplate (in.⁴)	18,650	31,430	31,430	31,430
Maximum internal couple, tons. ft.	210	264	290	290
Ratio, maximum couple ÷ moment of inertia	0·0113	0·0084	0·0092	0·0092
Relative stress in crankshaft	1	0·82	0·91	0·96
Maximum cylinder pressure, lb./sq. in.	1,080	1,350	1,350	1,400*
Cylinder head stud stress, tons/sq. in.	8·3	10	10·1	10·4*
Fatigue strength of threads, tons/sq. in.	17	27	27	27
Ratio, stress ÷ fatigue strength	0·49	0·37	0·375	0·385*
He...n stress in fuel ..., lb./sq. in.	262,000	205,000	215,000	225,000*
Main bearing load, lb./sq. in.	814	1,234	1,275	1,330*
Maximum permissible bearing load, lb./sq. in.	1,500	2,500	2,500	2,500
Ratio load/permissible load	0·54	0·49	0·51	0·53*
Large end bearing load, lb./sq. in.	2,400	2,800	2,900	3,050*
Maximum permissible bearing load, lb./sq. in.	2,700	5,000	5,000	5,000
Ratio load/permissible load	0·89	0·56	0·58	0·61*
Maximum stress in piston ÷ U.T.S. of material	0·44	0·33	0·35	0·36*
Top piston ring groove temperature, deg. C.	220	165	185	205*
Exhaust temperature at cylinders, deg. F.	800	810	850	890*
Air flow, lb./b.h.p.-hr.	13·3	13·8	13·9	13·7*
Exhaust valve seat temperature, deg. C.	540	460	490	520*
Specific fuel consumption, lb./b.h.p.-hr.	0·336	0·335	0·336	0·340*
Lubricating oil consumption, lb./b.h.p.-hr. at full load	0·0030	0·0020	0·0020	0·0018*
Weight of engine, lb./b.h.p.	44	30	26	23
Th... area of piston, ...	138	162	162	162
Maximum thrust pressure on piston, lb./sq. in.	35·8	33·5	34·0	34·8*
Depth of cylinder head, in.	11·75	13·5	13·5	13·5
Injector nozzle tip temperature (fuel at 200 deg. F.), deg. C.	177	127	130	136*
Inlet valve seat wear factor	192	156	156	162*

*Extrapolated values

Piston Design

The control of top piston ring groove temperatures by cooling the underside of the crown of the conventional single-piece cast iron piston, used in the K engine, is acceptable up to a rating of about 180 lb./sq. in., b.m.e.p., using a cast iron having a U.T.S. in the ring belt of 17 tons/sq. in., but, above this load, high tensile thermal stresses are produced on the inside wall of the piston behind the ring grooves[1]. For the K Major engine, a two-piece construction has been developed, as illustrated in Fig. 3, which has a high-tensile steel crown and a "Meehanite" skirt. This design incorporates an inner load-carrying boss, so that no pressure load is taken on the outer wall which carries the rings, and the latter may be quite thin, thus reducing the heat-flow path to the piston rings and giving efficient oil cooling of the ring belt, as well as ensuring that the roots of the piston

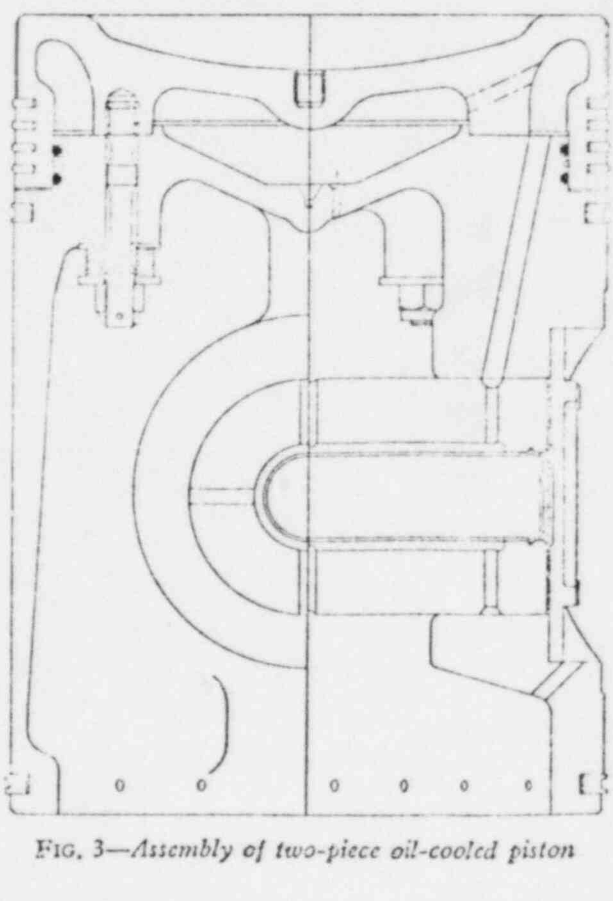


FIG. 3—*Assembly of two-piece oil-cooled piston*

ring grooves are stress-free. The piston crown is retained by four high-tensile studs which have rolled threads to give maximum fatigue strength, and heat-resisting "Helicoil" inserts are used to carry the studs, thus further improving the fatigue strength of the assembly and also acting as a heat barrier for the studs. Disc springs are fitted under the castle nuts to increase the resilience of the assembly and to provide an accurate method of checking the correct pre-load of the studs, this being achieved by measurement of the gap between the two retaining plates for the springs. Lubricating oil is fed, via a drilling in the connecting rod, through the piston pin and to the annulus chamber behind the ring grooves, through which it circulates at high velocity before meeting a transfer drilling to the inner chamber below the piston crown, from where it finally passes down an integral drain drilling in the piston skirt. The returning oil is collected in a cast aluminium tray, supported from the engine column, and is fed through a flexible connexion to a sight-flow and temperature indicator mounted adjacent to the crankcase door.

The thermally-induced and pressure-induced stresses have been thoroughly investigated in test rigs prior to tests in the prototype engine. Fig. 4 is a diagram of the thermal stress rig which is used to simulate the heat flow through the piston which occurs in the engine, heat being supplied by electric immersion heaters using solder as the medium for transferring the heat to the piston crown. Heat transfer through the piston rings is achieved by water-cooling the standard engine liner and oil-cooling of the piston internally is arranged in the same way as in the engine. Thermal stresses are measured by Budd self-temperature compensated strain gauges, having an overall size of ¼in. × ¼in., so that the effect of the gauges on the heat transfer conditions is extremely small. Fig. 5 shows the variation in thermal stress in the piston wall and also the temperature in the region of the top ring groove as a function of brake mean effective pressure for both the original single-piece piston and for the K Major two-piece piston; Fig. 6 illustrates the temperature and stress distri-

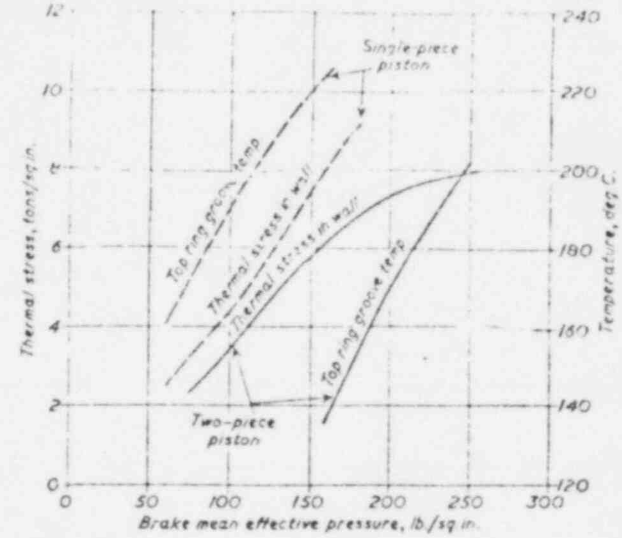


FIG. 5—*Comparison of temperatures and stresses in single-piece and two-piece piston designs*

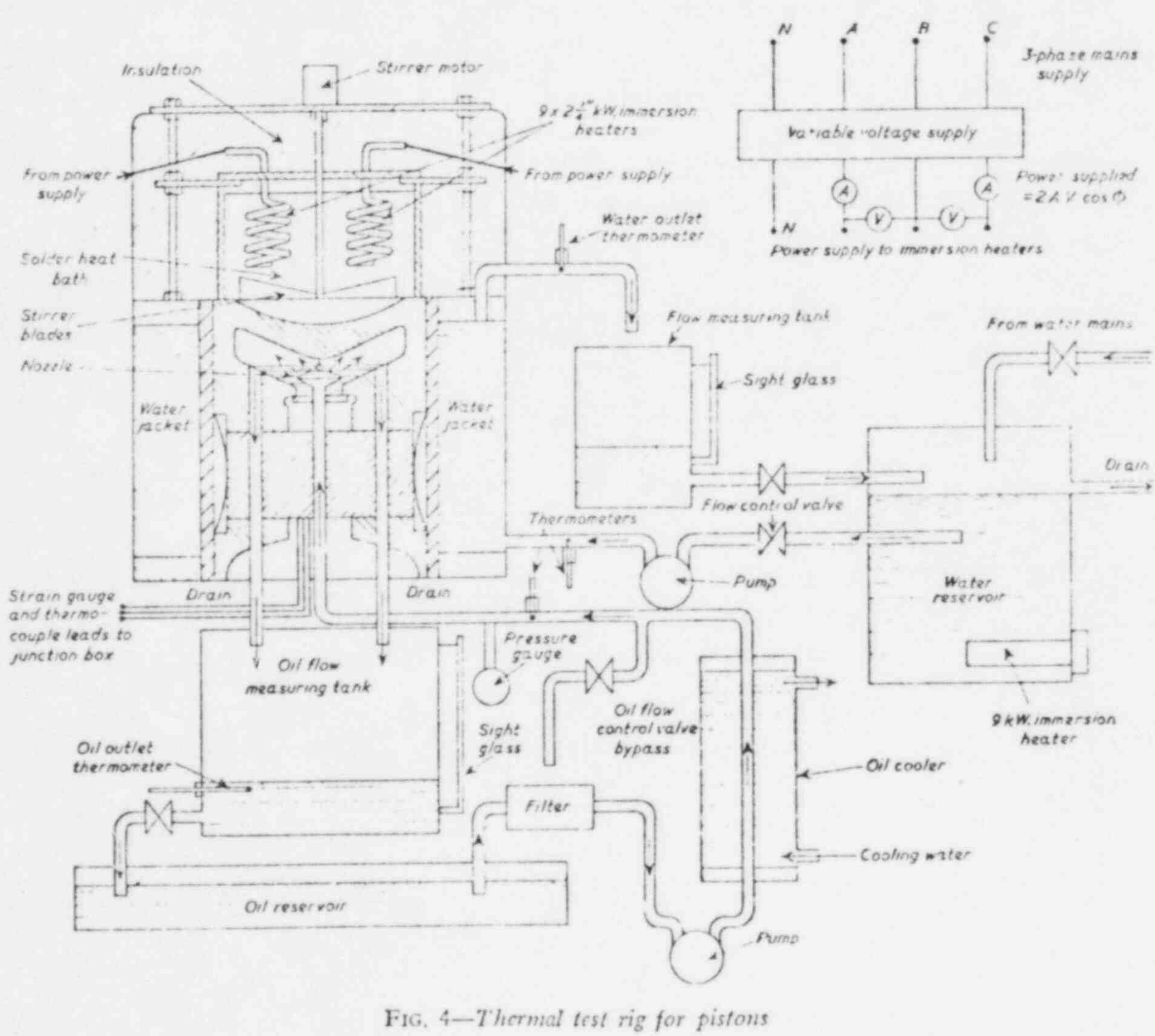


FIG. 4—*Thermal test rig for pistons*

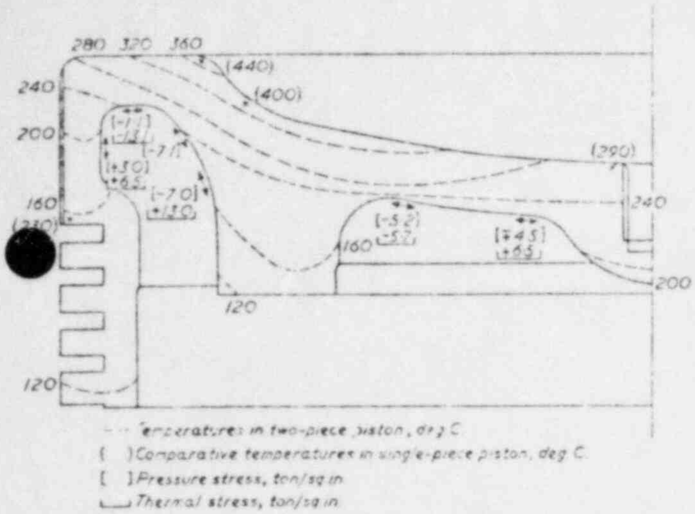
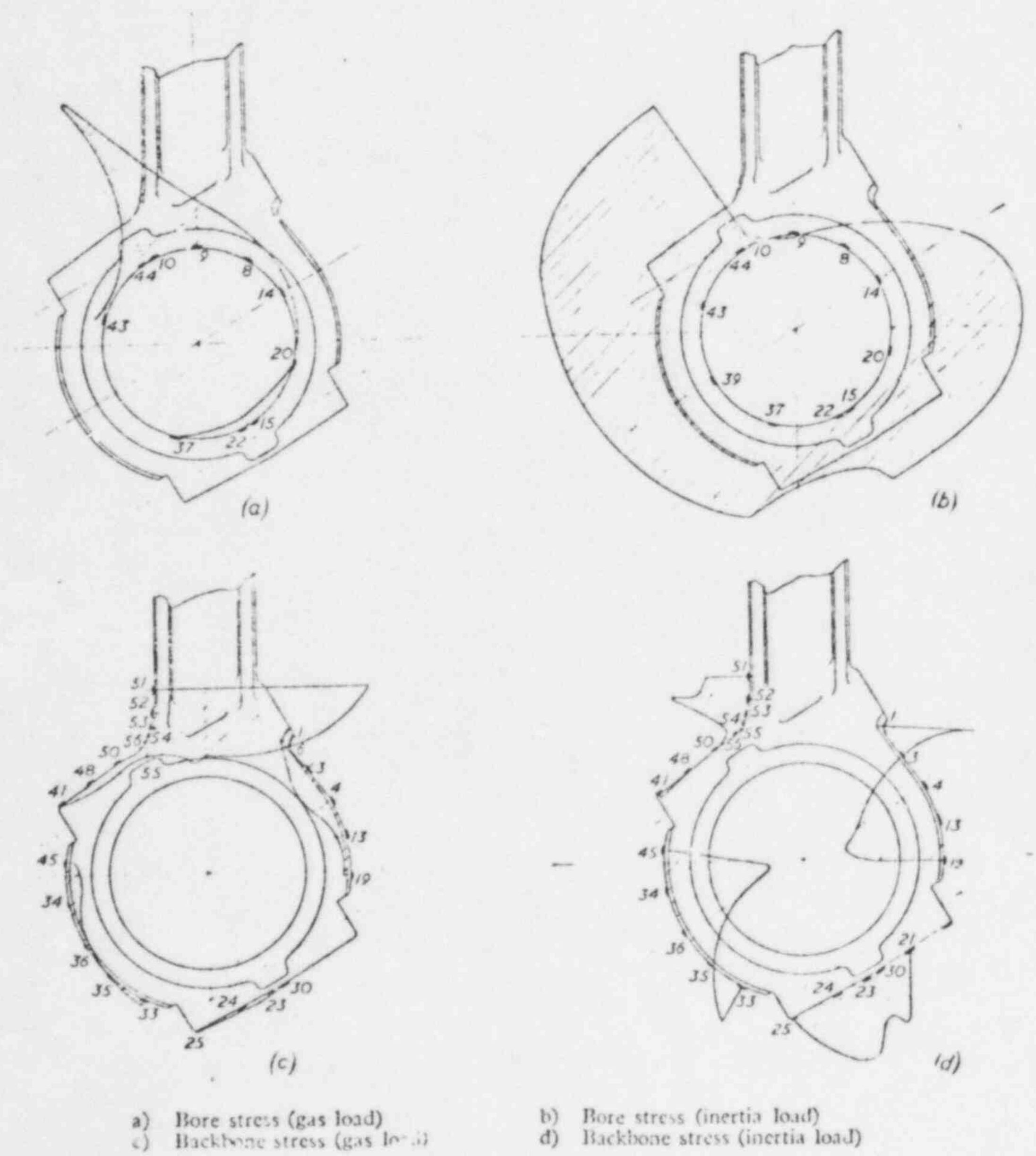


FIG. 6—*Two-piece piston crown temperatures and stresses*

bution in the crown of the two-piece piston. The reduction in top ring groove temperature by some 126 deg. F. (70 deg. C.), achieved by the new design, has made available a wide potential for increase in rating in the future, before any limitation due to lubricating oil break-down is reached. The thermal and pressure stresses are quite modest for the steel crown, which has a U.T.S. of 60 tons/sq. in., at room temperature, so that the factors of safety are much increased over the original single-piece cast iron design.

Connecting Rod

The connecting rods are one-piece stampings with the large-end bearing housing obliquely split at 30 degrees to the rod axis, and carry thin-wall tin-aluminium bearings. This construction permits a crank pin of maximum diameter, consistent with the withdrawal of the connecting rod through the cylinder bore. The optimization of the connecting rod proportions has been assisted by rig tests in a full scale static rig, in which gas loads and inertia loads are simulated by hydraulic pressure and the resulting stresses measured by strain gauges attached to the connecting rod. It was thus possible to reduce the weight of the connecting rod by 15 per cent from that of the original K rod so that, even at



a) Bore stress (gas load)
b) Bore stress (inertia load)
c) Backbone stress (gas load)
d) Backbone stress (inertia load)

FIG. 7—*Connecting rod large-end stress distribution*

the increased speed and load, the connecting rod stresses are lower than in the original design. Fig. 7 shows the stresses in the large-end of the connecting rod under firing pressure and inertia loading, and, in Table II, the safety factors at the most highly stressed points have been listed. In determining these values, allowance has been made for factors which would affect the fatigue strength of the material, such as specimen size effect and surface decarburization where it exists, so that the resulting values indicate the worst conditions and show that the rod design has a large margin of safety.

TABLE II

Gauge No.	Position	Factor of safety
1	Bolt platform radius	3·8
19	Supporting rib	6·8
33	Supporting rib	5·7
51	Base of shank	3·4
55	Shank radius	3·7
59	Bolt platform radius	4·4

Bearings

Main and large-end bearings are thin-wall steel shells lined with tin-aluminium, the increase in bearing loads from the K to the K Major being more than compensated by the improvement in fatigue strength of the bearing material. The actual and permissible bearing loads given in Table I illustrate the increased factor of safety in the new engine, the figures given being the conventional pressures obtained by dividing the maximum bearing load by the projected area of the bearing so that a simple comparison can be made. In the design of the K Major the more accurate methods of calculation, which have been made possible by the use of computers, have been used to assess oil film thickness over the range of speeds and loads so that the true factor of safety is even higher than the simple comparison suggests.

A positive displacement lubricating oil pump is driven from the free end of the engine by a flexible drive and delivers oil through a 15-micron full-flow filter to the main oil gallery cast in the bedplate. An oil-pressure regulating valve is fitted at the engine gallery to ensure that engine oil pressure remains constant, regardless of the degree of contamination of the filter, and a pressure-safety valve at the pump delivery protects the pump in the event of a complete blockage of the system. In addition to the full-flow filter, about five per cent of the flow is bypassed and filtered by small centrifuges mounted at the engine. This dual filtration ensures that carbon and water particles are removed from the lubricating oil and prevents the formation of sludge in the main filters. Tests have shown that a considerable increase in filter life is achieved by this system.

The quantity of lubricating oil circulated through the engine has been determined after thorough development tests to investigate the distribution of oil to main bearings, large-end bearings, piston cooling and other requirements, and the oil quantity has been chosen not only to lubricate but also to cool the main and large-end bearings, thus ensuring that the fatigue strength of the bearing material is maintained at its maximum value.

b) LOW FUEL CONSUMPTION

The importance of adequate air flow in a high-powered Diesel engine cannot be over-emphasized, the air delivered by the turbocharger having to perform the duties of scavenging the cylinder from the products of combustion and of cooling the components in the combustion space region, as well as providing a high mass of trapped air for the combustion process. In recent years the efforts of specialist turbocharger manufacturers to improve turbine and compressor efficiencies have made a substantial contribution to the success of the highly-rated Diesel engine, and the engine manufacturer can play his part by ensuring the maximum utilization of exhaust gas energy and by minimizing flow losses in the porting and ducting.

Air Ports

Air flow tests on the K cylinder head showed that the pressure drop in the inlet passages was made up as follows:

Inlet passage up to valve	11 per cent
Velocity change round valve seat	38 per cent
Loss of velocity head at outlet	35 per cent
Interaction between valves and cylinder wall	11 per cent

The large percentage loss around the valve seat indicated that optimization of the valve head profile and inlet passage shape in this region would be worth while, and the tests also showed that a greater effective flow area could be made available by increasing the valve lift beyond the value of a quarter of valve diameter at which the minimum geometric area becomes constant. Fig. 8 shows the increase in coefficient of discharge beyond the normal L/D ratio of 0·25 and the K Major valve lift was chosen to be 0·3 of the valve diameter, giving an increase in maximum effective area of just over five per cent. This improvement is quite significant when it is remembered that it is effective over a large valve opening period.

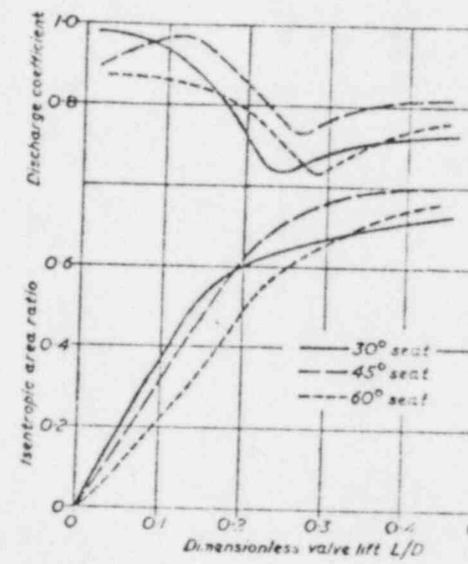


FIG. 8—*Flow characteristics of valves with 30-degree, 45-degree and 60-degree seats*

The effect of varying valve seat angle on flow characteristics was also examined and Fig. 8 shows the characteristics of valves of the same throat area with seats at 30, 45 and 60 degrees to the face of the valve. The 60-degree seat valve is clearly inferior to the other two, and the 30-degree seat is the best at small valve openings, whereas the 45-degree seat is best at large valve openings, while the advantage to be gained by increasing the valve lift beyond $L/D = 0{\cdot}25$ is valid for all values of seat angle.

There are other factors to be considered in choosing seat angle which determine the relative merits of 30 and 45-degree seats, of which the most important is that of useful seat life in service. Here, the conditions for inlet and exhaust valves are quite different, and will be considered separately.

The inlet valve operates in a relatively unlubricated condition at the seat so that seat wear, due to the relative movement of the valve seating face against the face in the cylinder head, as a result of the gas pressure, may be quite appreciable. A "wear factor" was derived theoretically and its validity confirmed by rig and engine tests from which the K Major inlet valve head profile was determined to give the minimum practicable relative movement and hence minimum wear[2]. The "wear factor" is defined as:

$$F_W = \frac{P_m^{\,2} . N . \mu . D^2}{E . B . t . v^2 . b . \cos\theta}$$

where μ = coefficient of friction;
P_m = maximum cylinder pressure;
N = engine speed;
D = valve disc diameter;
θ = seat angle;
E = Young's modulus;
B = wear resistance factor (hardness number);
b = seat width;
t = distance from valve disc face to top of seat;
v = height of valve disc cone.

It can be seen that a decrease in θ, or increase in t and v have the effect of reducing the "wear factor" and the K Major inlet valve head profile was designed from these considerations with a 30-degree seat angle and a stiff valve head. From experience on other engines a wear factor of above 250 gives unsatisfactory life in service and a value of 200 is satisfactory. It will be seen from Table I that the original K engine has a satisfactory value, which is confirmed by service experience, and the K Major has an even bigger safety margin.

The criteria for the seat of the exhaust valve are quite different and will be discussed later in the paper under the heading of "Heavy Fuel Operation".

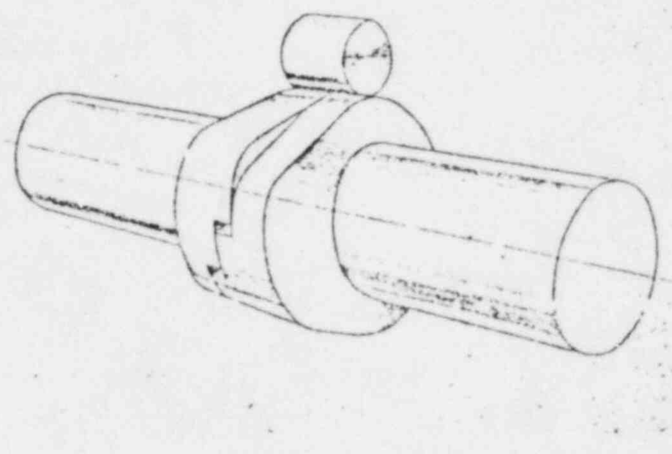


FIG. 9—*Construction of variable timing cam*

Valve Timing

The influence of valve timing on the exhaust, scavenge and charging processes has been examined experimentally on a three-cylinder engine, which was fitted with a special camshaft, in which the timing of both opening and closing of the air and exhaust valves, and of fuel injection were widely variable. Fig. 9 is a pictorial sketch of one of the variable timing cams showing the method by which the valve period is adjusted. Each cam is made in two pieces which are able to rotate relative to each other when hydraulic pressure is applied between the cams and the shaft from a hand pump. Release of the pressure then shrinks the cam on to the shaft to give an interference fit and the two parts of each cam are interlocked to form a bridge over which the cam follower roller can run without any discontinuity of profile. This method of hydraulic mounting allows the whole composite cam to be rotated to any desired position, as well as permitting the opening and closing flanks to be rotated relative to each other. Engine tests have been carried out over a wide range of valve timings, recording overall engine performance and pressure diagrams in the air inlet passages, engine cylinder and exhaust passages, from which optimum cam timings can be determined for any engine speed and load condition.

It will be appreciated that the optimization of valve timing is a complex operation and for a given set of timings it is necessary to match the injection equipment and the turbocharger performance for the engine speed and load range being considered. Some results have been selected from the range of tests carried out on the three-cylinder engine to illustrate the way in which changes in timing can affect the power range over which minimum fuel consumption is achieved.

In Fig. 10, the performance of the three-cylinder engine is shown with all valve timings held constant except the point of exhaust valve opening, the turbocharger match being changed to give the same total air flow. The valve timings were:

	E.V.O. before B.D.C.	E.V.C. after T.D.C.	A.V.O. before T.D.C.	A.V.C. after B.D.C.
Timing A	43 degrees	62 degrees	73 degrees	32 degrees
Timing B	65 degrees	62 degrees	73 degrees	32 degrees
Timing C	75 degrees	62 degrees	73 degrees	32 degrees

The left-hand curves of Fig. 10 show the performance at 450 r.p.m., and since the engine did not have the improved air flow already described in the previous sub-section, the optimum fuel consumption occurs close to the original K rating of 150 lb./sq. in., b.m.e.p. As the exhaust valve opening point is advanced, the position of minimum consumption moves further up the b.m.e.p. scale. This point is more strikingly illustrated in the right-hand curves of Fig. 10 where fuel consumption is plotted against exhaust valve opening point. At the lower rating of 140

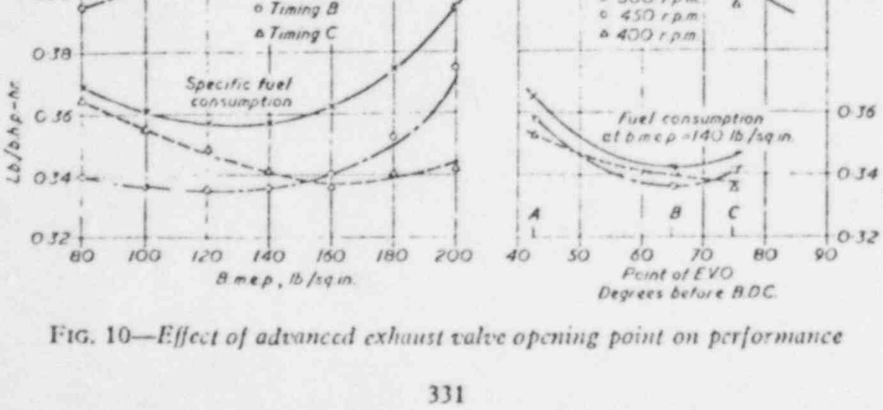


FIG. 10—*Effect of advanced exhaust valve opening point on performance*

sq. in., b.m.e.p., the change in fuel consumption is small but, at 200 lb./sq. in., b.m.e.p., there is a marked reduction in fuel consumption as exhaust valve opening is advanced. It must be emphasized that this illustration is intended to be indicative only of the beneficial effects of early exhaust valve openings. It is important that, at high b.m.e.p. ratings, good thermal efficiency is maintained and in the 12-cylinder K Major engine the minimum fuel consumption occurs at about 200 lb./sq. in., b.m.e.p., as illustrated later in Fig. 14, this result being achieved by improvement in air flow and fuel injection. Fig. 11 shows a low-pressure cylinder and manifold diagram for the 12-cylinder engine at 240 lb./sq. in., b.m.e.p., and 500 r.p.m., and demonstrates the good scavenging and adequate charging of the cylinder which has been obtained. As the development of the engine continues to even higher ratings it will be necessary to move the specific fuel consumption loop still further and the indications from the three-cylinder engine tests are that the advantages of earlier exhaust valve opening will be realized at this stage.

Cam Design

The increase in speed and loading, accompanied by faster opening and closing rates of the air and exhaust valves and the increased lift already described, would be expected to make much

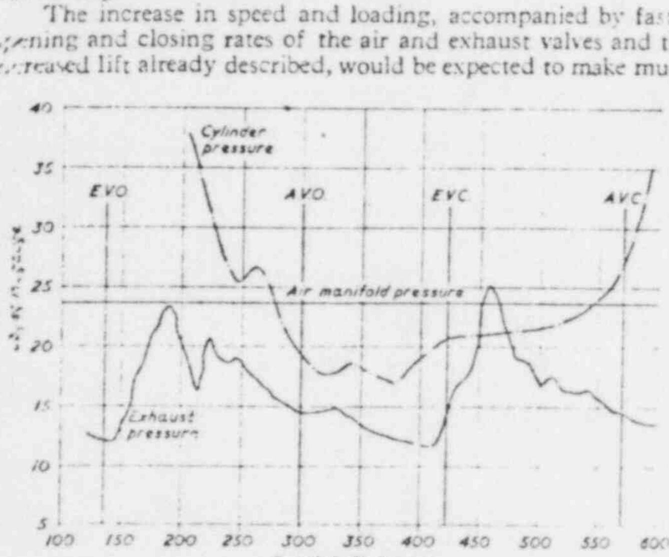


FIG. 11—*Cylinder and manifold low pressure diagrams at 240lb./sq. in., b.m.e.p., and 500 r.p.m.*

greater demands on the air and exhaust cams and follower gear. However, the design of cam profile, to optimize on rates of opening without exceeding established acceleration levels, has been considerably facilitated by the use of computer calculation techniques. The K Major air and exhaust cams are of polynomial profile, the mathematical analysis of the profile by computer calculations making selection of the most desirable curve a relatively simple procedure. The behaviour of the valve gear mechanism under running conditions, to determine the degree and frequency of vibration, has also been programmed and vibration calculations confirmed by a very simple technique of attaching strain gauges to the engine push rods. Fig. 12 is a typical push-rod strain trace which clearly indicates the natural frequency of the valve gear system and confirms that there is no tendency for separation of the valve train to occur.

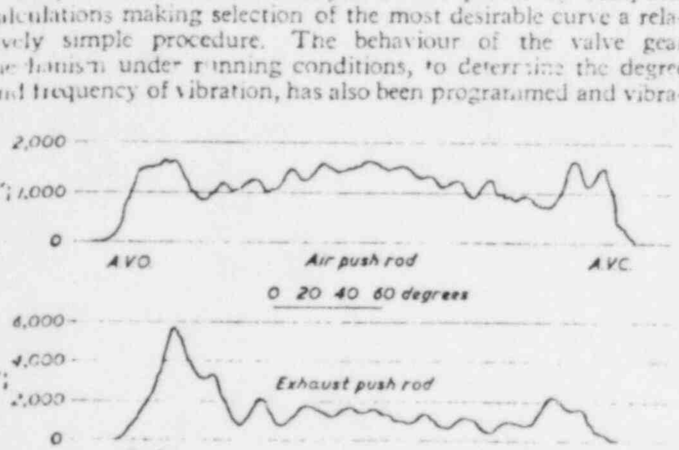


FIG. 12—*Push-rod strain at 200lb/sq. in., b.m.e.p., and 500 r.p.m.*

Fuel Injection

To obtain a maximum rate of injection, the fuel cams are of a profile which gives a constant plunger velocity during the injection period, and the correct matching of the injection equipment was facilitated by the use of test rigs which enabled the injection characteristics to be determined and the design of the injection equipment to be very nearly finalized before engine tests were started, only the confirmation of nozzle spray angle and the number and diameter of nozzle holes of a predetermined area remaining for final decision from the performance of the engine.

These test rigs enable a large number of permutations of fuel cam, pump plunger diameter, delivery valve design, nozzle design, etc., to be tested quickly and cheaply, using conventional methods of electronic indication of needle lift, fuel line pressure and nozzle sac pressure. The latter has proved to be of considerable importance in ensuring long life of injector nozzles by explaining the reason for over-rapid deterioration of nozzles in the K engine under certain service conditions. This phenomenon was a difficult one to explain until, as a result of calculations and rig tests carried out by the fuel injection manufacturer, it was realized that a particular combination of load and speed resulted in a hydrodynamic system in which there was a sudden reduction in fuel pressure in the nozzle sac just before the needle closed, the time interval between the two events being of the order of a quarter of a millisecond. This resulted in a penetration of gas from the cylinder into the nozzle sac during the combustion process, the hot gases impinging on the bottom of the needle and eventually impairing its performance. In Fig. 13 this condition can be seen at (a) on the left, where the pressure in the nozzle sac has fallen down to a low level at a point 16½ degrees after spill closure and there is a period of one degree during which the needle is still off its seat and gas can blow past it into the sac. The rig tests now ensure that the seating of the needle occurs before the sac pressure falls, as illustrated at (b) on the right of Fig. 13. The value of this preliminary rig work was confirmed by the performance produced in the 12-cylinder engine at a very early stage in its development running, many hours of "cut and try" tests to optimize injection equipment being saved. Fig. 14 shows

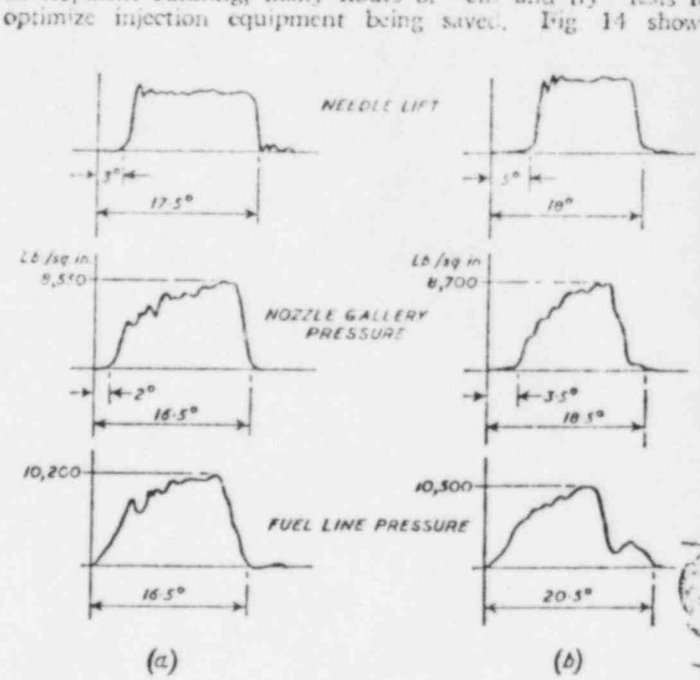


FIG. 13—*Injector needle lift, nozzle gallery pressure and fuel line pressure diagrams*

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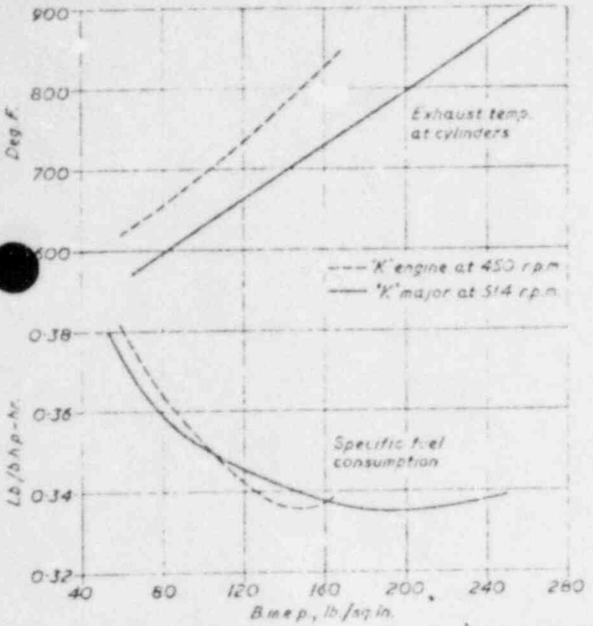


FIG. 14—*K and K Major performance comparison*

the performance of the engine as compared with that of the original K engine, from which it can be seen that the specific fuel consumption of the K Major engine is below 0·34 lb./b.h.p.-hr. over a very wide range of power, i.e., from 140 lb./sq. in. to 250 lb./sq. in., b.m.e.p. The curve also shows that, in spite of the increase in speed, from 450 to 514 r.p.m., and an increase in brake mean effective pressure, from 150 lb./sq. in. to 200 lb./sq. in. (i.e., a power increase of 56 per cent) the same exhaust temperature as in the K engine has been maintained.

c) HEAVY FUEL OPERATION

The operation of a Diesel engine on heavy fuel, the two items which normally deteriorate most rapidly are the injector nozzles and the exhaust valves, and the frequency of servicing of these two items is of predominating importance. In both cases there is a "threshold" of temperature of the critical parts of the components so that, as ratings increase, the design of the component must be improved to maintain safe operating temperature levels.

Exhaust Valves

Exhaust valve life with residual fuels is usually limited by the formation of deposits on the valve seat, resulting from the incombustible constituents of the fuel and largely from the combination of the sodium and vanadium salts present. As the seat deposits build up, they prevent the valve from making full contact on its seat, thus reducing the degree of heat transfer and eventually allowing tracking across the seat between the gaps in the deposits. The left-hand picture of Fig. 15 shows such a condition for an uncooled valve after 600 hours operation at 180 lb./sq. in., b.m.e.p., on a blended fuel of 300 seconds Redwood 1 viscosity with a three per cent sulphur content, 85 p.p.m. sodium and 100 p.p.m. vanadium. The beginning of erosion across the seat face between the deposits can be clearly seen[9].

Although the chemistry of the formation of these deposits is a most complex study, and is beyond the scope of this paper, field experience and engine tests have shown quite clearly that the presence of sodium and vanadium is of great significance and a practical assessment of the temperature range in which deposits are likely to adhere to the seating face of the valve can be made. Table III gives the melting point of possible deposit constituents which are in the temperature range which may appertain in the seat region of an exhaust valve, as is shown in Fig. 16, where the left-hand valve is of the normal uncooled design corresponding to the left-hand illustration of Fig. 15.

TABLE III

Compound	Melting point (deg. C.)
Nickel vanadate $NiO.V_2O_5$	900
Sodium sulphate Na_2SO_4	880
Sodium orthovanadate Na_3VO_4	850
Vanadium pentoxide V_2O_5	675
Sodium pyrovanadate $2Na_2O.V_2O_5$	640
Sodium metavanadate $NaVO_3$	630
Sodium vanadyl vanadate (1.1.5) $Na_2O.V_2O_4.5V_2O_5$	625
Sodium vanadyl vanadate (5.1.11) $5Na_2O.V_2O_4.11V_2O_5$	535

From the Diesel engine designer's point of view, it is sufficient to accept that if the valve seat temperature can be kept below about 1,020 deg. F. (550 deg. C.), adhesion of any of these components will not occur to any appreciable extent so that rapid build-up of the deposits will not be possible. The problem is thus quite different to that of the gas turbine engineer, who has to consider the corrosive effect which occurs at higher temperatures. However, the achievement of low valve seat temperatures at high outputs is not easy and calls for careful attention to details of design and patient development engine testing to achieve the desired result. The three-cylinder prototype engine, shown in Fig. 17, has been used for continuous testing on heavy fuel in the research laboratory for the past three years and more

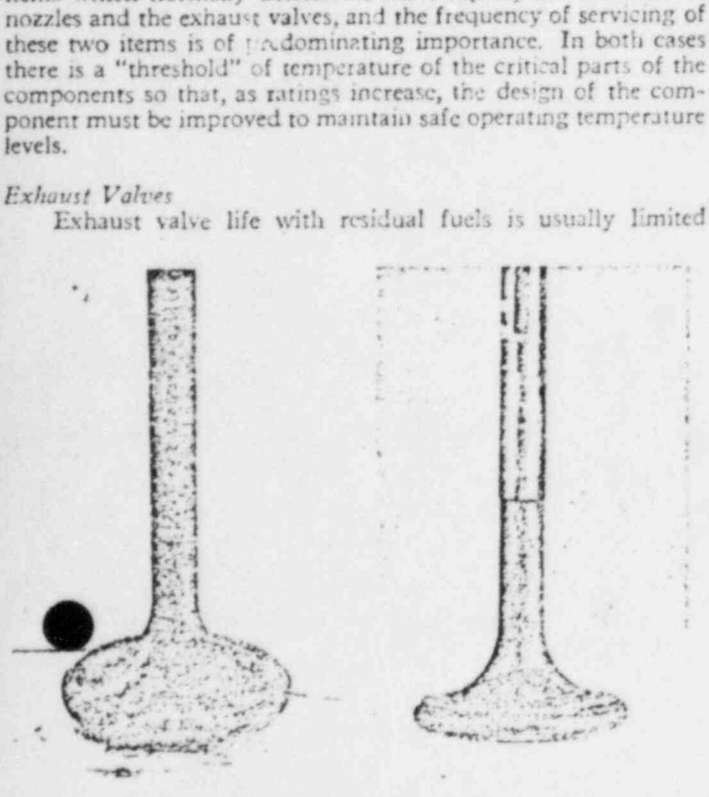
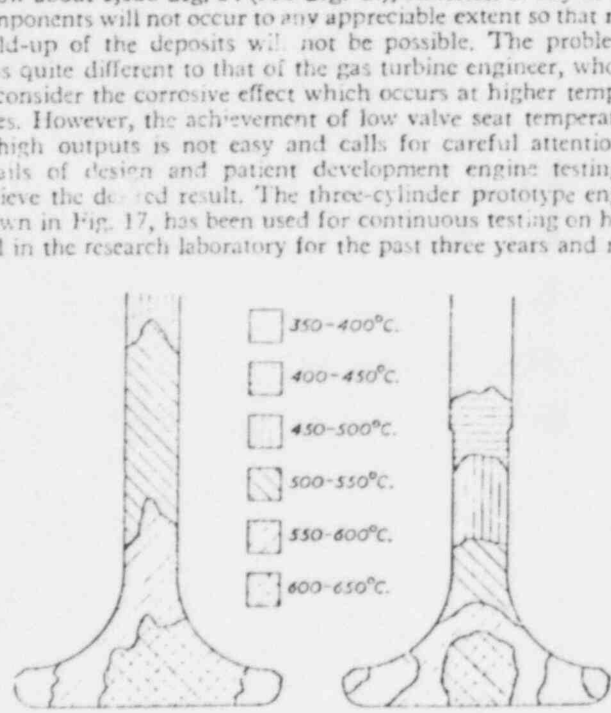


FIG. 15—*Comparison of exhaust valve condition after operation on heavy fuel*



(a) (b)

FIG. 16—*Temperature distribution in exhaust valves with uncooled and cooled cages*

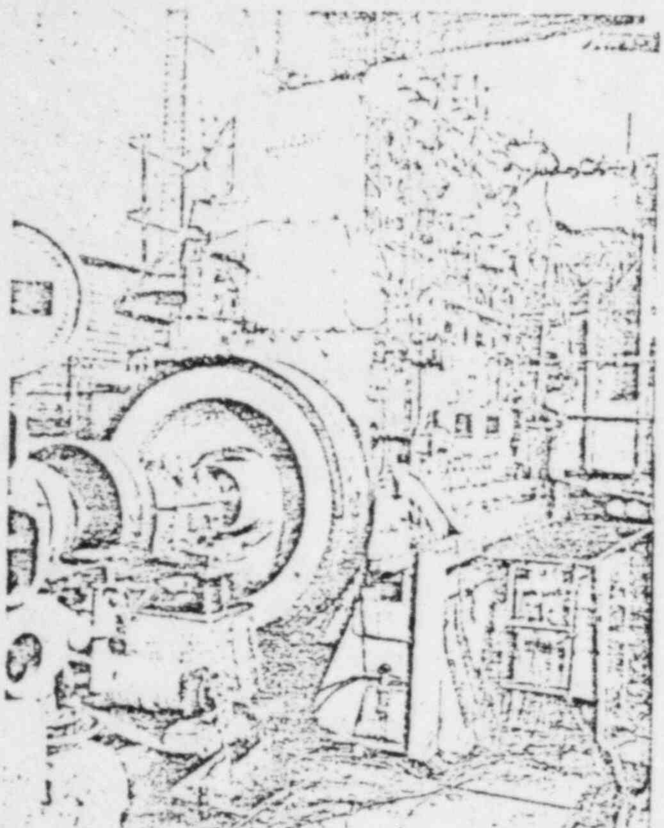


FIG. 17—*Prototype three-cylinder development engine*

than 60 exhaust valve and cage design combinations have been tested, of duration between 300 and 1,300 hours each to determine the effect of different factors in the design. A basic test duration of 500 hours at 200 lb./sq. in., b.m.e.p., loading was chosen and valve seat condition as the main parameter, together with other features such as valve guide wear, was compared with a reference design which was maintained throughout. In many cases the test was prematurely stopped at about the halfway stage where the valve condition was not satisfactory and, in the later stages, valve assemblies were replaced in the engine without re-grinding for second and third runs. Space does not permit a detailed report of the individual tests, but the resulting K Major exhaust valve and cage design will be described in detail to illustrate the factors which were found to be important.

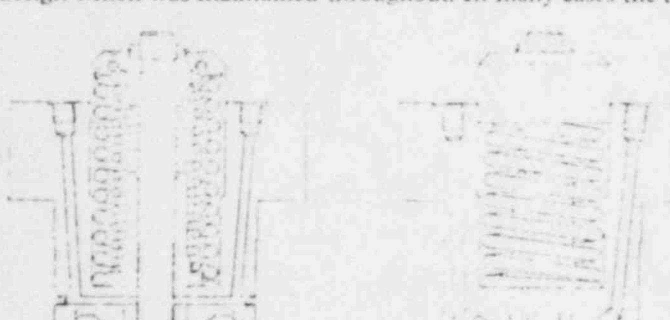


FIG. 18—*Assembly of exhaust valve and water-cooled cage*

Fig. 18 shows the valve and cage and it can be seen that a cooling passage is provided in the cage close to the valve seat. The seat is made of a single piece of Stellite 6, grooved to form the lower part of the cooling passage, and the upper portion is machined in the valve cage which is a three per cent Cr-Mo steel casting, the two being welded together by electron beam welding. The Stellite portion thus provides the facility for simple rebuilding of the seat by oxy-gas deposition of Stellite after a long period of time in service. The valve stem is of increased diameter and the valve of high-conductivity Cr-Ni-Si steel, to assist heat transfer from the head of the valve through the stem, and the valve guide is also surrounded by a water-cooled space, the cooling water passing along a drilled passage from the top of the cage direct to the annulus around the seat, then through a drilling to the space around the guide and via another drilled hole to the outlet at the top of the cage. The heat transfer from the valve to the cooled guide is assisted by the close fit between the valve stem and guide, the previously mentioned development tests having shown that a diametral clearance of 0·012in. resulted in an uncooled guide temperature of 450 deg. F. (232 deg. C.) in its middle position, with rapid deterioration of the lubricant and the formation of hard carbon. Halving the clearance reduced the temperature to 380 deg. F. (193 deg. C.) and, halving it again together with stem lubrication and guide cooling, brought the temperature down to 170 deg. F. (77 deg. C.), i.e., about 10 deg. F. (6 deg. C.) higher than the cooling water temperature, with no deterioration of lubricant and a guide wear rate of less than 0·0005in. in the first thousand hours.

The valve is fitted with a rotator in the top spring carrier which helps mechanically to prevent build-up of seat deposits, but its most important function is to ensure an even temperature distribution around the valve so that there is no local high-temperature region. The flow lubrication of the valve guide takes advantage of this rotation by using the valve itself as a timing device. Two flats are provided on the valve spindle which periodically line up with oil inlet and outlet drillings on the guide as the valve rotates. The linear positioning of the slots only allows the oil to pass while the valve is open and thus the oil space around the valve is only pressurized when the exhaust pulse pressure is present in the valve cage gas passage, the oil acting as a seal against gas penetration up the stem and the exhaust pressure preventing leakage of oil from the guide. It was at first feared that a continuous oil supply to the guide might result in excessive leakage of oil from the bottom of the guide, but this has not proved to be the case and, in fact, the tendency is for the leakage to be upwards as the retardation when the valve meets its seat is greater than the acceleration during opening and the inertia of the oil carries it upwards. This intermittent pressure lubrication of the valve stem makes it possible to use a very small stem/guide bore clearance without any risk of valve sticking, and this helps the heat transfer from the valve to the water-cooled guide. In addition, the danger of stem or guide bore corrosion at low load running conditions is avoided.

Since the stem to guide clearance is important in the heat transfer process, the reduction of guide wear helps to maintain low valve seat temperatures over a long period in service, and many of the development tests were concerned with valve guide material and valve rocker lever geometry to this end. The long guide and the small overhang of the valve head beyond the guide will be noticed in the illustration and were found to be important factors in reducing guide wear, as was the composition of the special "Mechanite" iron which was finally used for the guide material.

Sodium-cooled and water-cooled valves were tested among the many combinations but were found to offer no advantage over the design finally adopted, mainly, it is thought, because of the difficulty, with an internally-cooled valve, of providing cool-

ing passages close enough to the actual seat of the valve. The usual methods of drilling down the centre of the valve stem, although successfully cooling the centre of the head, still leave a fairly high temperature at the seat, and in the case of the internally water-cooled valve, the water connexions to the valve are a difficult problem.

The right-hand valve of Fig. 15 shows the results of this development, the valve having run for 900 hours at 200 lb./sq. in., b.m.e.p., on the same type of fuel as before. The good condition of the seating face shows that no re-grinding is necessary and the valve can operate for a much longer period without attention. The corresponding temperature distribution in the valve head is shown in the right-hand illustration of Fig. 16, and the effect of rating on exhaust valve seat temperature is given in Table I.

The valve development tests also included investigations into the effect of fuel treatment on exhaust valve life and while one fuel additive showed promise, in that the nature of the valve seat deposits was altered, it was not effective enough to justify its adoption. The principle of this additive was that other chemicals were added to the fuel so that the compounds, which were formed during combustion, would have higher melting points than those listed in Table III. It seems likely that, with further development work by the additive manufacturers, there may be some advantage to be gained in the future from this type of additive. Water washing of the fuel, to remove the sodium content, was found to be quite effective and the sodium could be reduced from 90 p.p.m. to about half of this value without difficulty, engine tests showing that the washing had quite an appreciable beneficial effect on the exhaust valve seat condition. As can be quickly calculated from Table III the critical sodium/vanadium ratios in the important temperature zone range from 1:0·74 to 1:13·3, the lower melting point compounds being associated with the latter end of the range, so that a reduction in sodium content may tend to produce the compounds with the lower melting points and, with particular fuel compositions, have an undesirable effect. Thus, with the wide variation in constituents in fuel from different parts of the world, it is difficult to make a clear case for water washing of the fuel.

Injectors

Fuel injection nozzles, when operated at high temperatures, tend to form carbon around the holes in the nozzles, known as "trumpeting", which may interfere with the injection spray pattern and reduce combustion efficiency, thus aggravating the temperature problem. For a time, the carbon formation develops until the "trumpets" become detached from the nozzle and a periodic rise and fall of exhaust temperatures can often be seen as this occurs. The general trend of temperature, however, is upwards and conditions eventually level out at the top end of the exhaust temperature cyclic range. In more extreme cases of high temperature, the needle seat may lose its hardness and the needle rapidly hammers its way into the seat. The temperature at the nozzle tip can be measured by thermocouple and a temperature of about 356 deg. F. (180 deg. C) is considered to be the limit

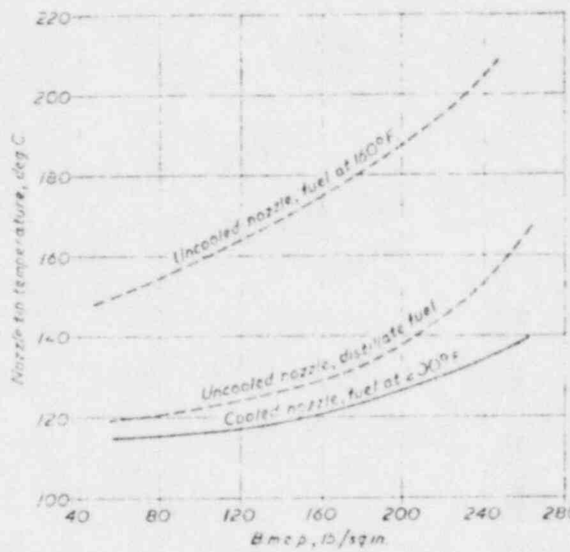


FIG. 19—*Injector nozzle tip temperatures with cooled and uncooled injectors*

for satisfactory operation. In Fig. 19, the middle curve shows the variation of nozzle tip temperature with load for the K Major engine, using an uncooled nozzle and distillate fuel, where the fuel itself has a considerable cooling effect, and there would be no difficulty in operating an uncooled nozzle on this type of fuel up to a load of about 280 lb./sq. in., b.m.e.p. In the upper curve, however, blended fuel of 300 seconds Redwood 1 viscosity was used, with a fuel temperature of 160 deg. F. (71 deg. C.) and it can be seen that the loss in cooling effect from the fuel has limited the acceptable load level to about 180 lb./sq. in., b.m.e.p., and with heavier, and hence hotter, fuels the load limit would be much lower. A water-cooled nozzle is therefore necessary for high ratings on heavy fuel, and the lower curve shows the tip temperature for a cooled nozzle using 1,000 seconds fuel at 200 deg. F. (93 deg. C.) with cooling water at 150 deg. F. (66 deg. C.). It is important that the nozzle should not be over-cooled as cold corrosion can occur at temperatures below 230 deg. F. (110 deg. C.), but this is controlled by the water-circulation system which is separate from that of the engine-cooling water. Fig. 20 shows the cooling system which is a closed circuit serving the injectors and water-cooled seat exhaust valve cages with a thermostatically controlled bypass around the heat exchanger and minimum volume in the system to ensure that correct operating temperatures are reached quickly.

d) LUBRICATING OIL CONSUMPTION

The consumption of lubricating oil in a Diesel engine is an important factor in maintenance costs and it is not always realized that, at a reasonable consumption rate of one per cent of

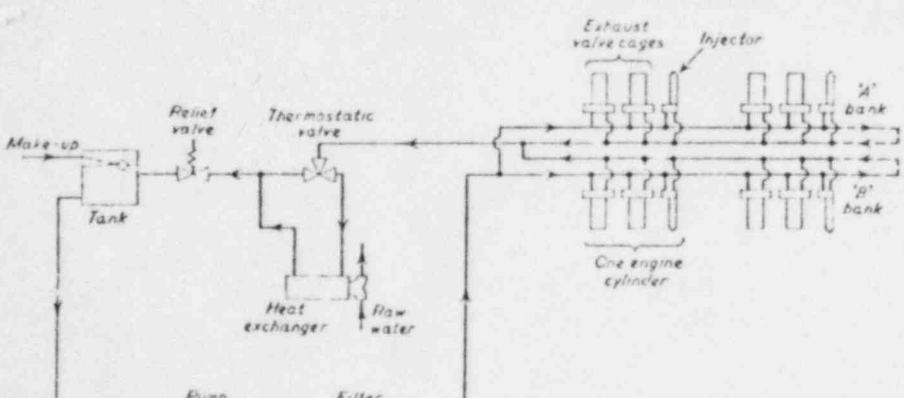


FIG. 20—*Arrangement of injector and valve cage cooling system*

fuel consumption, a 4,000 h.p. engine will burn a quantity of lubricating oil equivalent to its sump capacity in a period of the order of 300 hours. Emphasis is often laid on long periods between oil changes which are extended by an engine with a high oil consumption, whereas the relative importance of oil consumption to oil change period is around 50 to 1. The cost of modern high-duty detergent oils is quite appreciable, so that an oil consumption of one per cent of the fuel consumption represents something like ten per cent of the fuel bill. Not all of this could be saved, of course, but a reduction of 50 per cent in lubricating oil consumption is equivalent to a five per cent saving on the fuel bill, and would be well worth having from the point of view of running costs.

Piston Ring Design

To carry out lubricating oil consumption tests in the relatively short running periods of 500 hours or so in the research laboratory, it was necessary to develop an accurate method of measuring top-up rate and a system was devised, and has proved very successful, whereby consumption can be measured consistently over successive two hour periods and plotted consecutively. The running-in period and the levelling-out to a steady consumption can now be followed and it has been possible to obtain steady state results after a total test period of only 300 hours, which allows much more latitude for testing variations on a ring pack than was previously the case. There is a large number of detail points to be considered such as liner finish, roundness, drainage in the piston, etc., but the basic concept which has been established is to provide a parallel-faced chrome-plated top compression ring, three taper-faced plain compression rings, a relatively mild scraper ring below the gudgeon pin, and a more severe scraper ring above the pin. This ensures that adequate lubrication is available around the body of the piston but that the minimum of oil is allowed to pass up into the combustion space. The consistency of oil consumption measurement has enabled some interesting facts to emerge, and Fig. 21 illustrates one of these—the effect of the wall pressure of the scraper ring above the pin. The left-hand curve is from the three-cylinder, 15-in. bore prototype engine, and the right-hand curve from a

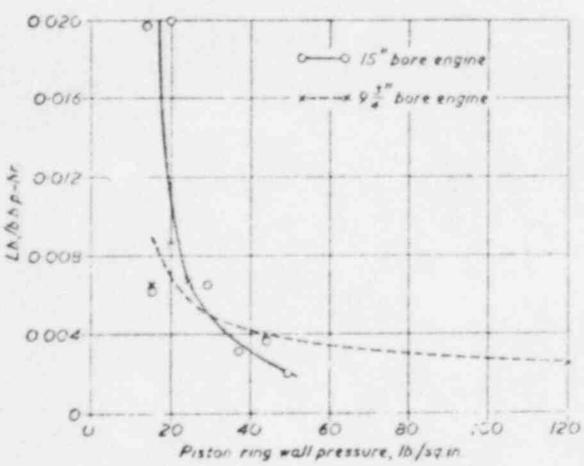


FIG. 21—*Variation of lubricating oil consumption with scraper ring wall pressure*

completely different high-speed engine of 9¾-in. bore, the points marked being the stable lubricating oil consumption achieved after running periods of about 300 hours in each case. Both curves show the same trend of reducing oil consumption with increased ring wall pressure and the tendency for the curves to level out at higher values of wall pressure. The value of wall pressure necessary to achieve a satisfactory consumption can be seen to be much higher for the smaller high-speed engine than for the K Major engine, and in the case of the smaller engine it was necessary to use a spring-loaded conformable scraper ring to achieve the desired consumption. A conventional type of slotted scraper ring was adequate to provide the 50 lb./sq. in. pressure needed for the K Major engine, the resulting consumption, of less than 0·002 lb./b.h.p.-hr., being confirmed in the 12-cylinder engine during development running.

Piston Ring Quality

Consistent oil consumption and low wear rates are largely dependent on the quality of the piston rings, from the point of view of metallurgical structure as well as accuracy of manufacture. Accuracy and good finish in manufacture can be assured by conventional inspection methods, and such methods can easily be extended to give some indication of material quality, such as by measuring the permanent set of the ring at a given load value above that required to close the gap. A simple sample checking method on metallurgical structure was devised in which a small piece of ring is clamped with its working face subjected to a given load and resting on the surface of a ring of liner iron. The ring is then rotated at a standard speed for a fixed time without lubrication and the weight loss of the piece of ring measured. Weight loss is used as a measure of the relative wear resistance of the material and, although "rough and ready", is found to co-relate well with the differences in microstructure of the ring material. Some typical results are given in Table IV, and illustrated in Fig. 22, and show that with the same Brinell hardness, increasing amounts of free ferrite give progressively worse results and these are not improved by increase in phosphorus content within the amounts to comply with mechanical strength requirements[4].

TABLE IV

Sample No.	Structure	Hardness, HB	Weight loss, gm.
A	Greatly undercooled graphite, considerable free ferrite (centricast) 3·15 per cent. T.C., 0·83 per cent P.	210	0·404
B	Some undercooled graphite, a little free ferrite (centricast) 3·20 per cent T.C., 0·40 per cent P.	210	0·185
C	Random uniform medium flake graphite, fully pearlitic (sand-cast) 3·45 per cent T.C., 0·55 per cent P.	210	0·017

e) MAINTENANCE

Engine running times between overhauls are dependent upon the load, duty and running conditions, and the preceding sections have indicated the attention that has been paid to the components which operate under the most arduous conditions. By reducing the critical temperatures of injector nozzles and exhaust valve seats, so that when operating on heavy fuels these temperatures are below the "threshold" values at which deterioration becomes rapid, it has been the aim to achieve periods of 2,000 to 3,000 hours before servicing of injectors or exhaust valves is necessary. Experience on the prototype engine has indicated that this ambition is by no means unreasonable but, of course, true confirmation of success will only come from the accumulation of service experience. Maintenance of other components would not be different from that established over many years, e.g., piston removal annually, complete overhaul every two years, the periods generally being dictated to suit the convenience of the operator rather than by the demands of the engine.

f) SPACE AND WEIGHT

In achieving high engine ratings reliably, the weight per horsepower, and space per horsepower, are naturally reduced and the emphasis on reliability for commercial marine work necessitates a different approach from that which would be appropriate for naval work where light-weight constructions become necessary but short life may be permitted. Sight should not be lost of the importance of low fuel consumption in the considera-

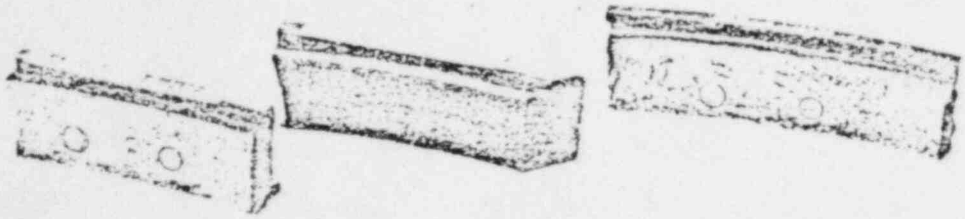


FIG. 22—*Comparison of piston rings after wear rig tests*

tion of weight. A ship refuelling every 3,000 miles, for example, at an average speed of 15 knots, and having engines weighing 34 lb./b.h.p., and a specific fuel consumption of 0·34 lb./b.h.p.-hr., would re-bunker an amount of fuel equivalent to twice the weight of the engines. Thus a five per cent reduction in fuel consumption would be equivalent to a ten per cent reduction in engine weight in addition to the saving in fuel cost.

In the design of the K Major engine, cast iron has been used as the main structural material and, in the authors' experience, has many advantages over fabricated steel designs. Few fabricated structures are able to avoid fillet welds in load-carrying regions and the fatigue strength of such a weld is as low as ≃ 1·2 tons/sq. in. Even butt welds must allow for discontinuity so that their fatigue strength is only ≃ 3·8 tons/sq. in., these values being for good quality welds, the strength of an imperfect weld being, of course, very low indeed. A good quality cast iron has a fatigue strength of over 5 tons/sq. in., and as well as freedom from the notch sensitivity, which so drastically reduces the fatigue strength of a steel structure, cast iron has good internal damping properties and also possesses the useful property of a diminishing E value with increased stress so that stress concentrations are considerably reduced and the material tends to relieve itself of any excessive stresses.

In keeping with the philosophy of designing for maximum

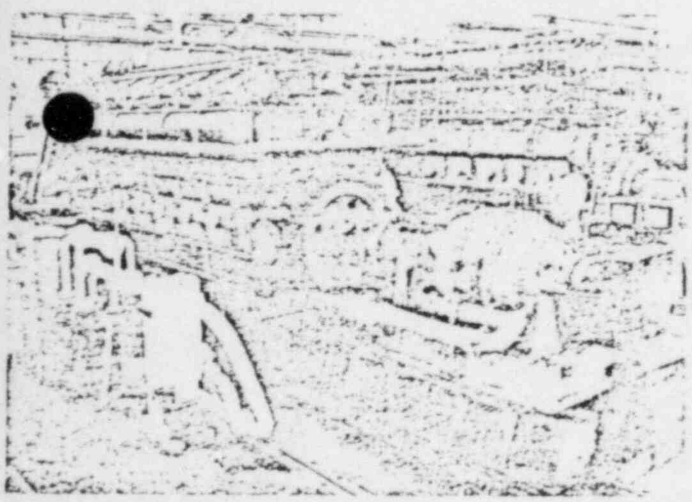


FIG. 23—*Prototype KV Major 12-cylinder engine*

TABLE V

Cylinder bore	15in.
Stroke	18in.
Compression ratio	11.35:1
Maximum r.p.m.	525
Minimum working r.p.m.	125
Continuous rated b.m.e.p.	200 lb./sq. in.
Maximum continuous b.h.p./cylinder	420 b.h.p. (426 cv)
Lubricating oil inlet temperature	150 deg. F. (65 deg. C.)
Lubricating oil outlet temperature	165 deg. F. (74 deg. C.)
Lubricating oil drain tank capacity	650 gal. (2,960 litres)
Fresh and salt water flow rates	5.5 gal./b.h.p.-hr at 50ft. head (25 litres/cv-hr.)
Engine cooling water inlet temperature	155 deg. F. (68 deg. C.)
Engine cooling water outlet temperature	170 deg. F. (77 deg. C.)
Exhaust temperature after turbocharger	800 deg. F. (427 deg. C.)
Starting air pressure	400 lb./sq. in.
Specific fuel consumption	0·335 lb./b.h.p.-hr. (l.c.v. of 18,400 B.t.u/lb.)
Thermal efficiency	42 per cent

TABLE VI—POWER RANGE

	B.m.e.p. lb./sq. in.	No. of cylinders 6	8	9	12	16	18
B.h.p. output at 250 r.p.m.	200	1,200	1,600	1,800	2,400	3,200	3,600
	250	1,500	2,000	2,250	3,000	4,000	4,500
B.h.p. output at 350 r.p.m.	200	1,680	2,240	2,520	3,360	4,480	5,040
	250	2,100	2,800	3,150	4,200	5,600	6,300
B.h.p. output at 450 r.p.m.	200	2,160	2,880	3,240	4,320	5,760	6,480
	250	2,700	3,600	4,050	5,400	7,200	8,100
B.h.p. output at 525 r.p.m.	200	2,520	3,360	3,780	5,040	6,720	7,560
	250	3,150	4,200	4,720	6,300	8,400	9,450
Overall length of engine		20ft. 0in.	24ft. 0in.	26ft. 0in.	24ft. 3in.	29ft. 10in.	32ft. 7in.
Overall width of engine		7ft. 6in.	7ft. 6in.	8ft. 2in.	11ft. 7in.	11ft. 11in.	11ft. 11in.
Overall height of engine		11ft. 6in.	11ft. 6in.	11ft. 6in.	11ft. 11in.	11ft. 11in.	11ft. 11in.
Height above crankshaft C.L.		8ft. 9in.	8ft. 9in.	8ft. 9in.	8ft. 2in.	8ft. 2in.	8ft. 2in.
Engine weight (dry) tons		38	44	48	65	85	95

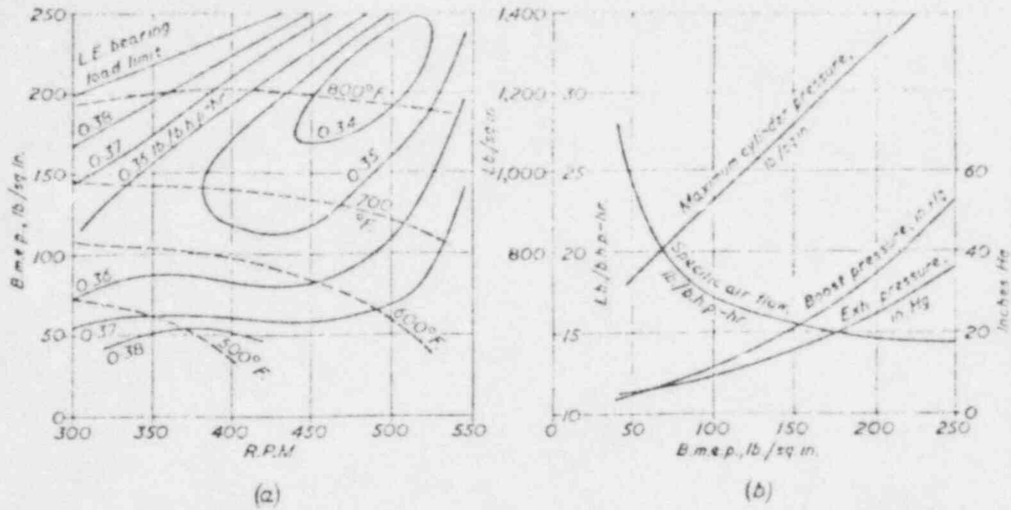


FIG. 24—*KV Major engine performance characteristics*

reliability and easy maintenance, the bedplate type of construction has been retained for the K Major engine and a useful facility has been added by the inclusion of a machined strip on the top surface of the bed so that alignment of the engine can be quickly and accurately checked, and crankshaft deflexion measurements more easily interpreted.

Fig. 23 shows the 12-cylinder KV Major on the test bed, from which the general construction and appearance of the engine can be seen, and Tables V and VI give the specification for the range of engines available at the current commercial rating of 200 lb./sq. in., b.m.e.p., and the future rating of 250 lb./sq. in., b.m.e.p., under normal temperature and pressure conditions with sea water up to 75 deg. F. (24 deg. C.) to the charge air cooler.

CONCLUSIONS

The design and development of a highly-rated medium-speed Diesel engine, to operate economically and reliably on heavy fuels, has been described and it has been shown that, for the K Major engine, the critical parts of the engine, which determine its reliability, have adequate safety margins for its current rating of 200 lb./sq. in., b.m.e.p., and have potential for a substantial increase in rating, to 250 lb./sq. in., b.m.e.p., in the future. The performance of the 12-cylinder prototype engine, beyond the current commercial rating, is illustrated in Fig. 24, curve (a) showing the performance at variable speed and curve (b) the performance at a constant speed of 514 r.p.m., and these, in conjunction with Fig. 14, show clearly the enormous strides which are being made in the Diesel engine industry towards higher specific outputs without exceeding the temperature and pressure levels which past experience has shown to give reliable and trouble-free operation. There is little doubt that in the marine propulsion field there is considerable interest in the use of medium-speed Diesel engines for higher powers than have hitherto been possible, and that within the next few years engines of this type will be available to cover almost the whole range of power demands of British shipping.

ACKNOWLEDGMENTS

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Discussion

MR. R. COOK, M.Sc. (Member of Council) said that, at the present time, the manufacturers of medium-speed engines in Great Britain were making very strenuous efforts to extend their share of the marine market in propulsion machinery. The paper was, therefore, timely and few who had read it would have failed to be impressed by the manner in which the authors and their colleagues were applying the latest knowledge and research techniques to the solution of the problems which arose when such machinery was developed to operate at high ratings on residual fuels. One could hardly doubt that success would attend their efforts, although he suspected that the large direct-drive Diesel would be about for quite a few years to come.

The histogram shown in Fig. 1 was interesting. It would be noted that by far the largest horsepower on order at the present time was between 9,000 and 21,000 s.h.p. per ship. With the machinery described in the paper this implied the use of some 18 to 36 cylinders of 15 in. diameter, each with two exhaust valves, two inlet valves, together with the injection and starting equipment. He said that he could not help wondering whether the modern seagoing engineer, who was perhaps not quite so amenable to long and arduous hard work as his forebears, would take kindly to the never-ending task of top-overhauling such a formidable number of cylinders.

Another point for thought was the effect which such maintenance requirements would have upon the reduction in engine-room staff now being achieved with direct-drive machinery by the application of an increased degree of automatic control. He hoped that some superintendents would comment on these aspects later in the discussion.

He said that some years ago Dr. Pope had made a very thorough theoretical and experimental investigation for the British Shipbuilding Research Association into the causes of failure of pistons, liners and cylinder heads in marine oil engines, with results which had since been published in the Transactions of another Institution. He was not surprised, therefore, to see the attention which the authors had given to thermal and pressure-induced stresses in the design of the two-piece, oil-cooled piston. Presumably, the piston temperatures shown in Fig. 6 were measured on the actual engines and the rig used to check the thermal stresses calculated from these temperature measurements. If so, he wondered whether good correlation was achieved. It would be interesting to know how the temperature distribution in the stationary-rig piston correlated with that on the engine.

He said that he was interested to observe the use of rolled threads on the high-tensile steel studs used for securing the piston crown. Work by B.S.R.A. on the rolling of threads of large mild-steel bolts such as those used in the dynamically-loaded components of direct-drive Diesels, had shown very striking improvements in fatigue strength. Reference to a paper* appearing in the Transactions of the Institute four years ago would show that form-rolling increased the fatigue strength of large forged bolts made from mild steel some 2½ to 3 times, when compared with cut-thread specimens. Rolling of thread roots gave almost as great an increase. The degree of rolling had been found to be not very critical, but his Association was at present investigating more fully the optimum degree for various sizes and pitches. He imagined that with the high-tensile steel material used by the authors, the gain in fatigue strength would not be so great as in the case of mild steel, but it would be interesting if the authors would quote some figures. Form-rolling could be a very cheap method of bolt production, particularly in small sizes.

The means adopted to ensure correct pre-loading of studs was to be commended since there was no doubt that the majority of failures of dynamically-loaded bolts were due to fatigue caused by inadequate tightening. It was not always appreciated that, with a properly-designed bolted connexion, fatigue failure was virtually impossible if the bolt was adequately pre-loaded.

The section in the paper dealing with heavy fuel operation was, of course, of the greatest possible interest, since a solution of the difficult problems involved was essential if the medium-speed engine was to be able to compete with the direct-drive engine, which was so much less fastidious as to its diet. Here again, the authors had given evidence of a careful scientific approach which should go a long way to ensure success. They had commented on the possible use of fuel additives. One could imagine this approach being successful where fuel supplies of constant composition were available, but this was seldom possible in marine practice and the chances of obtaining a cheap additive, which was effective with a wide variety of fuels, seemed somewhat remote. The authors' approach, by tackling the design, was certainly the right one. Their remarks on the drawbacks of water washing were also worth noting.

No reference had been made in this section to sump-oil contamination when using heavy fuels. Presumably this must occur to some degree in this trunk-piston design, and it would be useful if the authors were to give some information on the procedure involved in maintaining the lubricating oil in a suitable condition.

Dr. Pope had, over the years, made many investigations into the properties of cast iron. Few were, therefore, more familiar with its strength and frailties. Sir Harry Ricardo had once referred to cast iron as "the material which served our forefathers so well for lamp posts and kitchen ranges", but he was sure that Sir Harry would be the first to acknowledge the advantages which the authors had enumerated. Its use as the main structural material in the K Major engine had much to commend it, since weight was rarely of paramount importance in merchant ships.

On the subject of cast iron, he said that it might be inferred from the data given in Table IV that centri-cast piston rings were inferior to sand-cast rings. He felt sure that this would not be the authors' intention. Centri-cast rings had been widely employed with success. He took it that the authors' purpose had been simply to show that, with this type of material, under-cooling and consequent presence of free ferrite was most undesirable.

The paper had touched in an interesting manner on so many aspects of Diesel design that to point to ommissions might seem somewhat churlish. He wished, however, that the authors had found it possible to touch on the subject of turbo-

* Cook, R., and McClimont, W. 1961 "The Influence of Screw Forming Methods on the Fatigue Strength of Large Bolts". *Trans. I.Mar.E.*, Vol. 73, p. 417.

charging. Perhaps they might, at some later date, find it possible to give a paper on their experiences in turbocharging up to the 250 lb./sq.in. in b.m.e.p. which was involved in the third stage of the development of the engine.

COMMANDER E. TYRRELL, R.N. (Member), in a contribution read by Mr. T. P. Everett, referred to the successful introduction, by the authors' company, of an engine which he considered met both industrial and marine needs which so far had only been filled by Britain's foreign competitors. As the authors had so rightly said, reliability was the main requirement of a marine propulsion engine, and nobody who had read this paper could fail to be impressed by the systematic way in which Mirrless National had carried out the research and development work necessary to ensure that this engine would operate satisfactorily at the ratings envisaged.

If the medium-speed geared Diesel engine was to compete with the slow-runing direct-coupled engine, it was essential that it should operate satisfactorily while burning heavy fuel, and it was evident that the authors had taken very considerable trouble and had expended a comparatively large sum of money in trying to ensure that this would be the case. There was, however, one point which he felt should have been mentioned in this respect, and that was the effect of various grades and compositions of lubricating oil on the problems associated with the burning of heavy fuel. There was little doubt that trunk-piston engines called for careful selection of the lubricating oil if satisfactory operation with heavy fuel was to be achieved. New types of lubricating oil and testing for quality and make-up by the addition of detergents, anti-oxidants, and alkalis could have a profound effect on the satisfactory operation of trunk-piston engines while burning this type of fuel. The wrong type of lubricant, or one which had been allowed to deteriorate unduly, could give rise to ring-sticking and crankshaft corrosion. The operating temperature of the oil was also important if these defects were to be avoided. Perhaps the authors would like to remark on the type of lubricating oil and its optimum operating temperature for this type of engine burning heavy fuel.

He thought that the title of this paper was slightly misleading. He could not agree that the Mirrlees K Major engine should be regarded as highly rated when operating at the conditions given in the paper. In his opinion there was considerable scope for further advances in b.m.e.p. These were important as they should give worthwhile reductions in the cost per horsepower. Introduction of a reliable engine operating at these higher brake mean effective pressures would do much to increase the competitive power of this type of engine against its competitors abroad and other types of prime mover. In this technically-competitive world, the main object of any Diesel engine manufacturer must be consistently to uprate his engines in order to give better value for money. He must at the same time retain reliability.

Many of those present would be aware that the Ministry of Technology had recently placed a contract with the Yarrow-Admiralty Research Department to investigate the use of medium-speed geared Diesel engines as propulsion units for ocean-going merchant ships. This survey was now almost complete. He thought that it was true to say that the results of this survey would give encouragement to those manufacturers of medium-speed Diesel engines who thought that the medium-speed Diesel had a future and could compete in many ships with the slow-running direct-coupled engine. The report showed that every type of ship and trade must be treated on its merits, but that a shipowner who failed to carry out a detailed economic survey into the possible use of medium-speed Diesel engines, as an alternative to the slow-running direct-coupled engine, did so at his peril.

MR. S. H. HENSHALL, B.Sc. (Member) said that, as an engine builder of medium-speed engines, he found that he was on the side of Dr. Pope in a lot of the things he had said. The paper, however, had been very stimulating and he would like to ask several questions about it.

With regard to Table I, the specific air flow for the 250 lb./sq.in. b.m.e.p. showed a drop compared with lower b.m.e.p., and, although this drop was only a small one, he would have thought it desirable to go on increasing the specific air flow.

Turning to the piston design, he said that the features of it were, in many ways, those with which he agreed, but it was mentioned that it was a steel crown and a cast iron junk. The steel crown probably had a higher coefficient of linear expansion as compared with the cast iron. This meant that there were some problems in its connexion, for instance, it must make life a little difficult for the sealing rings between the two portions. Cold clearance between piston crown and liner must be increased.

Figs. 2, 3 and 6 showed a double line round the junk. He wondered what this signified and whether it was some device to overcome the cold clearance problem.

With regard to exhaust valves and operation on heavy fuels, the importance of losing heat via the stems of the valves was certainly to be considered. In the paper the clearance was mentioned as being of importance. The wear rate was obviously kept down by the ingenious device of continuous lubrication and he said that he would be interested to know what was the maximum allowable clearance of the stem to guide and what sort of life the valve had in this respect. Also, it appeared that the guide could be renewed, although he was not sure whether it was intended to be renewed.

He suggested that there was an argument for not water washing fuel, in that the deposits also occurred on turbocharger blades, and turbochargers could be water washed more easily if the sodium was allowed to remain in the fuel.

On the question of cast iron or steel as the main structural material, he said that steel had its own advantages, and structures could be designed with low stresses where welds occurred. Modern techniques of manufacture and inspection could ensure good quality welds.

He said that surely greater reliability resulted from a design in which the major loads did not have to pass through a joint between the crankcase and bedplate, and easy maintenance was not confined to the bedplate type of construction. The principle of a machined strip used for checking alignment of the engine quickly was also used on engines of fabricated design having undersiung crankshafts and light sumps instead of baseplates.

IR. E. R. GROSCHEL said that he proposed to limit his observations and comments to the fuel injection side of the paper, departing only for a moment in order to fully endorse the authors' statement under "a) Reliability: *General Considerations*", where they said that the surest method of producing intrinsic reliability was to proceed by a process of logical evolution from one successful design to the next, taking care that the critical parameters proved in the original design were maintained in the new design. This statement deserved thunderous applause from both engine manufacturers and engine users, particularly marine engine users.

The use of test rigs for fuel-injection equipment development was, of course, fully appreciated and valuable data regarding performance and life of the equipment might be gained. However, care should be taken, when applying results obtained from injection-equipment test rigs to engine conditions, unless injection into a pressurized medium was strictly simulated. He wondered if this was done in the case where nozzle gallery pressure was found to be lower than the prevailing gas pressure with a needle still open which permitted gas entrance into the nozzle gallery. The cure adopted, he ventured to guess, was to lighten the reciprocating mass of the injector.

His company, having always designed and manufactured their own fuel-injection equipment for their medium-speed Diesel engines, had always been protagonists of the low inertia injector, i.e. having needle springs acting directly upon the needle without the intermediary of a push rod. Of course, the spring was thus placed into a somewhat uncomfortable position (heat and space), but unorthodox spring wire sections helped with the space and chromium/silicon wire with the temperature

The "hydrodynamic condition" referred to in the paper, he thought, was a spill wave receding too fast, which the normal oscillating system of the injector was incapable of following. A stiffer spring might help in border-line cases. A trick, imparted to him some time ago by Mr. J. F. Alcock (and gratefully acknowledged) was to watch for gas bubbles in the injector leak-off connexion; should gas pass into the nozzle, it would generally pass through the cylindrical lapped part of the needle-nozzle bore causing lacquering of the lap fit and would finally appear in the leak-off pipe, where a plastic tube would facilitate observation. He said that, acting upon this recommendation, his company sometimes used development engineers as "bubble watchers".

He thought the authors should be congratulated on having such confidence in the precision of cathode ray oscillogram interpretation. He had tried something similar and the result had varied between ½ degree and 2 degrees cam angle, depending upon the thickness of the pencil point used and the condition of the interpreter.

He asked if the line pressures in Figs. 13(a) and 13(b) referred to full load conditions. If they did, they were remarkably low. Also, the difference between maximum line pressure and maximum gallery pressure was rather large, being 1,650 lb./sq.in. in Fig. 13(a) and 1,800 lb./sq.in. in Fig. 13(b). He wondered if the edge filter (which, after all, was the major throttling component between line and gallery) could be responsible for this pressure discrepancy. Wall friction could hardly account for it with the customary maximum flow velocities of about 90 m./s.

Dealing with the chapter on heavy fuel operation, he said that he was intrigued by the statement that the lack of cooling ability of preheated fuel should be responsible for high nozzle tip temperatures. The fuel temperature given was 160 deg. F. (71 deg. C.) for a blended fuel of 300 sec. Redwood 1 viscosity; gas oil, not preheated, would reach the nozzle, after passing through the compressive cycle of the fuel pump, not very much cooler. He ventured to think that the hotter nozzle tip of the heavy fuel operated engine was more the result of the slower burning of the heavy fuel, with a resulting larger heat rejection to, and heat absorption from, the nozzle.

He said that he understood that the mechanism of trumpet formation was a function of the lighter fractions of the blended heavy fuels boiling off in the nozzle sac and squeezing the heavier fractions out of the nozzle holes where they carbonized. Carbonization temperatures were much higher than the measured nozzle tip temperature of 392 deg. F. (200 deg. C.). He said that he would be very grateful if the authors could provide information about the precise location of the thermocouple on the nozzle tip. His company had measured nozzle seat temperatures (the thermocouple being located within a millimetre from the nozzle seat) and temperatures obtained on one engine type, depending on engine rating, cylinder head design and fuel, had reached 464 deg. F. (240 deg. C.). This surely indicated nozzle tip temperatures of a far higher order. These nozzles had been uncooled and made from heat-resisting nitriding steel which maintained the seat hardness at elevated temperatures.

He said that he envied the authors the low maximum cylinder pressure of only 1,350 lb./sq.in. at 220 lb./sq.in. b.m.e.p. This should enable them to get away with an injector release pressure of only 2,500 lb./sq.in., resulting in a closing pressure of 1,720 lb./sq.in., thus still having a comfortable margin available above the maximum gas pressure.

In conclusion, he said that the paper would always have a place of honour in his hydraulics department, having already sent several slide rules into a semi-heated condition.

Mr. J. F. Alcock, O.B.E., B.A., said that the piston crown was described as high-tensile steel, which term covered a lot of compositions. It would be valuable to have either the thermal conductivity or the composition.

Fig. 6 showed a wet-side temperature of over 464 deg. F. (240 deg. C.). He said that it was a rough general rule that one was apt to get coking on the surface if one went over 392 deg. F. (200 deg. C.). He wondered if this had been observed by the authors. He also asked for the velocity of the oil.

Turning to valves in cages, he said that one considerable advantage of the cage was that it very much reduced the flux from the valve seats to the cylinder head. Since the cylinder head was a complex casting, a large concentrated heat flow from the valve via the seat was undesirable. In smaller engines, which did not have caged valves, this was a very common cause of cracking between the seats. He cited the paper of Mr. Fujita* as an example of this.

Turning to valves, he said that he had noticed in the paper that while the temperature of the seat had been reduced by the cooled cage, the temperature difference between the centre of the valve head and the seat was increased. This would increase the thermal stresses, and the risk of seat cracking due to thermal stress. He gathered that the idea of sodium cooling had been discarded, but he thought it might be a useful idea, not from the point of view of cooling the valve seat, but of reducing the thermal stress. Of course, what the sodium-cooled valve did was to pass the flux back to the valve guide, but there it could be coped with quite well.

He then referred to crankcase explosions. These were practically non-existent in small engines, but they did occur from time to time in large engines and were extremely nasty things. It would be valuable to have information on this subject and to know, from the point of view of safety, what the difference was between the trunk-piston and the crosshead engines.

Dr. W. P. Mansfield was particularly interested in the authors' lubricating oil consumption tests which were briefly described on page 336 of the paper and to which Fig. 21 referred. In some investigations on this subject on smaller engines the British Internal Combustion Engine Research Institute Ltd. had tried increasing the oil pressure by reducing the area of the bearing surface of the ring, but this had had little effect. However, changes of wall pressure, made by varying the radial thickness of the ring, had a marked effect. He said that it would be interesting to know by what method the authors had varied wall pressure, apart from the change to a spring-loaded ring which was mentioned in the paper.

Mr. A. J. S. Baker (Associate) said that the authors had produced a remarkably full description of what could only be described as an exceptionally well developed engine. Of particular interest to people connected with lubrication research was the systematic work which had gone into piston development. This work had obviously paid off handsomely in the modest temperatures, so clearly denoted in Fig. 6. It would be interesting to see how far the really excellent fuel utilization rates, indicated by the exceptional fuel consumption rates, had contributed to this. For instance, metal temperature comparisons taken at the same time as the variable valve timing tests described in Fig. 10, right-hand side, would perhaps illustrate this point. Fig. 10 itself suggested that an even broader area of minimum specific fuel consumption might be possible with automatically-varied valve timing. He wondered if the authors had considered such a possibility.

Looking at the Fig. 10 data points for a constant b.m.e.p. of 200 lb./sq.in., it appeared that a fairer mean curve would have a pronounced and steepening hog rather than the sag shown by the authors. He asked the authors to justify the mean line they had postulated.

With regard to the important work which had been done to optimize fuel injection characteristics, it was interesting to consider the authors' needle lift/sac pressure relationship conclusions, in the light of the modification which appeared to have been carried out. Apparently the unloading value had been increased between (a) and (b) of Fig. 13. Did the authors attribute this fact to the reduction in incipient secondary lifting indicated in the needle-lift diagram (b)? Presumably the injection rate

* Fujita, H. 1961. "Service Records of Mitsubishi Nagasaki Diesel UE Type Engines and Improvements Made on the Engines". *Trans. I.Mar.E.*, Vol. 73, p. 37.

had also been increased and this had been accommodated by permitting an increase of needle lift. If this were the case, might not needle-lift increases have to be closely controlled in service operation, he asked. Likewise, the fuel-line pressure diagram (b) was presumably taken at the nozzle end of the fuel pipe. He wondered if it had been necessary to tune the fuel pipe length to control the magnitude of the secondary pressure wave so as to eliminate secondary injection.

He thought that the general conclusion to be drawn from the fuel injection work was that engines of comparatively low speed needed the same careful attention as high-speed engines. It would be very interesting to see a comparative investigation in certain large-bore, slow-speed engines, having several nozzles connected to a single fuel-pump element. The results of the careful work done by the authors in this direction were demonstrated in Fig. 14. By extracting data points from the fuel consumption loops, it was interesting to note that the fuelling lines for both the K engine and the K Major at different speeds were virtually identical. The fuelling lines, from around 50 lb./sq.in., b.m.e.p., on the lowest curve, were unusual in their linearity. Perhaps the authors could supply fuel consumptions at very low loads which would give a clearer indication of the likely f.m.e.p. From the data published it was evident that this must be of a very low order and comparable to that obtained with the largest low-speed engines. This point might be worth bringing out since it was fashionable to quote mechanical efficiency for large two-stroke marine engines, and for a given f.m.e.p. this would generally favour the four-stroke engine with its higher b.m.e.p.

He asked the authors to indicate whether the performance curves were obtained with the fuel described at the top of page 333, and if not, he said that he would like them to give details.

He said that the water-cooled, exhaust-valve cage and valve rotators had made a major contribution to operation with low-cost fuels. Presumably some provision had been made to prevent boiling in the small seat-cooling passages in the event of sudden shut-down, as might be expected in main marine engine application.

He thought that the notes on lubricating oil consumption were very relevant, as was the investigation on oil control by the upper scraper ring. He asked the authors to elaborate on this by indicating the degree of control exerted by the other rings in the pack. For instance, could a reduction in radial pressure of the upper scraper be tolerated by increasing the load on the lower one?

The rig described to evaluate piston-ring quality resembled a variety of test rings used for different purposes. Experience with these had indicated considerable scatter of wear results, particularly at high rates of wear. Perhaps the authors could indicate the significance of the weight loss figures they had quoted in Table IV. He wondered whether they had observed a pattern of related ring to bore wear rates for the different material combinations tested. Had any significant differences in piston-ring groove and ring side wear rates been observed when different irons were run in the steel piston crown? The authors had not shown the metallurgy of the piston crown, but other applications of high-tensile steel had suggested that steels containing appreciable nickel contents might produce increased wear rates in the presence of boundary lubrication.

MR. J. A. COWDEROY, B.Sc. (Member) said that, as the K Major engine had been developed specifically for marine propulsion, he had been surprised that Fig. 24 did not show the performance plotted against speed on a propeller law basis. He would particularly like to see the compression pressure included in such a plot, because he had the impression that many builders of marine Diesel engines overlooked the implications of the propeller law, which related power to speed in a ship, particularly when applied to turbocharged engines. If this law were assumed to be a cube law it meant that, if the engine was developing full power at full speed, it was only required to develop as little as 12½ per cent of that power, even at half speed, and as ships not infrequently proceeded at speeds lower than full, this condition did occur now and then.

The engine referred to in the paper had been developed to run on heavy fuel oil. He thought the authors would agree that the turbocharged four-stroke, trunk-piston engine could be troublesome from the point of view of combustion when operating at low loads on heavy fuel, and from what figures he had seen for other engines, which showed a drop in compression pressure, from 665 lb./sq.in., at full load and speed, to 355 lb./sq.in., at 60 per cent speed and 22 per cent of full load, he strongly suspected that the relatively low compression pressure under those conditions was one of the principal reasons for this. Whilst combustion might be quite satisfactory, under those conditions, when the engine was new and in first-class condition, with the accumulation of wear of not only liners, but injection equipment, he thought that the low compression pressure was certainly a contributory factor. He would be glad to have the authors' comments on this.

On the question of the operational control of turbocharged medium-speed engines in ships, particularly in view of the increase in the number of ships with bridge control of the engines, he was convinced that the fuel injection pump rack position should be governed to some degree by the booster pressure. A few years ago, in a certain cross-Channel ferry, which was propelled by two turbocharged Diesel engines under bridge control, it was found, a very short time after the ship had gone into service, that the engine crankcase oil had become very dirty indeed. The reason for this was soon discovered: the bridge control of the engines had been operated on leaving harbour as if it had been an engine room telegraph, with the result that the engines smoked like chimneys until the turbochargers had time to catch up and provide enough air for clean combustion. Under these conditions the oil soon became filled with fuel soot. Instruction to the master as to the correct rate at which to increase engine power soon cured the trouble. He felt that where turbocharged engines were installed in ships, some form of control over the rate of increase of the delivery of fuel to the engine was essential.

MR. C. C. J. FRENCH asked a question concerning thermal stress. The thermal stress rig shown in Fig. 4 was interesting and provided an ingenious method of investigating a problem which was becoming more and more important as engine ratings were increased. This rig was useful in that it was applicable to asymmetric bodies, as well as to those that were bodies of revolution. In this respect the two-piece piston shown in Fig. 6, appeared to be a body of revolution. Computer programmes were now available for calculating the thermal stress of such components. He wondered whether the authors had tried a check calculation to see whether there was any sort of agreement between the rig and a computer. His own rather limited experience so far, with a computer approach, had been more valuable in showing up limitations in the computer programme than in giving realistic piston thermal stresses, the problems being largely the rather complex shape of pistons.

Turning to the inlet-valve wear, he said that he was glad that Dr. Pope, in his presentation, had elaborated on his wear factor, which Mr. French had found somewhat incomprehensible as it stood in the paper. He agreed that lack of lubricant was the main cause of heavy inlet-valve and seat wear in turbocharged engines. It was most interesting that the authors had found thickening the head of the valve so effective in reducing this wear.

Touching on service experience, he said that two years previously a paper* had been presented, giving details of service experience on an engine of very similar size and rating. He thought that everyone looked forward to the time when the authors would be able to give comparable details of exhaust-valve life, cylinder-liner and cylinder-ring wear on the K Major, when operating on residual fuels. In this connexion, if the authors were proved correct in their aim of up to 3,000 hours between servicing of injectors and exhaust valves, this would be a most valuable step forward.

* Henshall, S. H., and Gallois, J. 1964. "Service Performance of S.E.M.T. Pielstick Engines." *Trans. I.Mar.E.*, Vol. 76, p. 445.

Correspondence

COMMANDER E. R. MAY, D.S.C., R.N. (Member) wrote that it was some ten years since the Pielstick PC1 had begun to make its significant contribution to the propulsion of ocean-going ships, and during the whole of this time it had been without any effective medium-speed competitor. The K Major must now be judged by comparison with the Pielstick PC2, with which it would be in direct competition in every field.

Power for power, the British engine was rather larger and heavier than its French competitor. In some applications this would not matter very much. Commander May imagined that the relative first cost of the two engines would be very significant, assuming that they had equal ability to burn heavy fuel. The Pielstick had never been a cheap engine and, in its PC1 form, its exhaust-valve life on heavy fuel did not always prove impressive. Its popularity had stemmed from its introducing high-speed engine standards of accuracy into the marine engine field, with a refreshing freedom from the very heavy maintenance work that marine engineers often experienced on propulsion engines, less well made and indifferently developed.

Over the last few years, the major British medium-speed Diesel firms had caught up the leeway in standards of manufacture, and also had undertaken most impressive programmes of detailed development. It therefore seemed that the K Major would meet international competition successfully, would extend the market gained by the K engine, and join the Pielstick in propelling large merchant ships.

Commander May noticed that the authors had made a rather misleading reference to short-life engines being permissible in naval work. This had never been so (except in motor torpedo boats). Submarine engines were designed and produced by the Admiralty between the two wars in an attempt to produce better—not lighter—engines than those available from industry at the time. After the last war, the Admiralty worked hard to persuade industry to adopt modern standards in development and manufacture of long-life engines up to 9¾-in. bore, but success was only achieved gradually and at substantial public expense.

In Germany, before and after the war, and in France at the present day, engines designed partly for naval purposes had met with widespread commercial success. This had come about through recognition that naval and commercial requirements could be designed into the same engine with advantage to all concerned.

Possibly the most remarkable feature of the K Major was that it had achieved so much while retaining cast iron for frame and bedplate. Rigidity was essential to maintain bearing oil film geometry within acceptable limits and cast iron was about twice as flexible as steel. A cast iron frame must have heavier scantlings than a steel frame, the cylinder centres must therefore be further apart, and bending moments increased in consequence. On the other hand, cast iron was cheaper than steel, and development of modern cast irons had done much to make this material more attractive. Fairbanks Morse had used cast iron extensively in their new large opposed-piston, medium-speed engine. Other manufacturers were, the writer believed, using steel for comparable engines and had also chosen the two-stroke, valve-in-head arrangement.

Soon, at least four of these valve-in-head two-stroke engines (one of them British) would be competing with the K Major and the Pielstick in the rapidly expanding world market for large medium-speed engines. It was obvious from this paper that Mirrlees had planned to secure their share of this market.

It would be interesting to know the authors' view on transmission suitable for employing, say, two K Majors to drive a single propeller shaft, and whether their company proposed to offer complete propulsion units—engines and reduction gear.

MR. G. H. HUGHES (Member) commented, in a written contribution, that the increase in power output should in no way alarm prospective users, because even the ultimate aim of 528 b.h.p./cylinder, with 250 lb./sq.in., b.m.e.p., and 1,400 lb./sq. in., peak pressure, represented only 2·98 b.h.p./sq.in. of piston crown—almost identical to the power per square inch on the crown of the Maybach engine with pistons of similar construction.

It would be interesting to know the cooling oil flow rate, (he suggested approximately 1½ gal./b.h.p.-hr.), since crown and ring life depended on adequate cooling and, in this respect, the oil feed through the connecting rod might prove to be the limiting factor. Given adequate cooling, it was known that this form of piston would stand greater power per square inch of crown area, as shown in the two-stroke cycle Ruston and Hornsby A.O. engine, when published figures showed over 5 b.h.p./sq. in.

His company's experience of materials for such piston crowns indicated that thermal fatigue tended to become the limiting factor and this depended principally on coefficient of expansion and thermal conductivity. Had the authors considered one of the high-nickel alloys to minimize the effect of high operating temperatures, or the high-conductivity copper chromium alloys?

A further aid to cooling was increased valve overlap. Had the effect of this been explored with respect to piston crown and piston ring temperatures?

The scraper ring arrangement permitted adequate lubrication of the skirt or crosshead length of the piston, but when oil control became a problem after extended service, there might be a temptation to fit a highly-loaded ring in the skirt groove, with possible risk of seizure. To avoid such possibilities, had the authors considered omitting the skirt-ring altogether and adjusting the upper scraper ring accordingly?

It was noted that three taper-faced rings were fitted below a parallel-faced, chrome-plated ring in the top groove. There might be a tendency to blow-by during the initial running of the engine with this arrangement. Had blow-by readings been taken during test work and had any indications been noted?

An important factor in piston-ring material was compatibility with cylinder liners. Not all materials were suitable in this respect, but might be metallurgically sound and, therefore, of good quality.

It was not surprising, therefore, that a random flake graphite iron had given satisfaction in this size of engine.

With regard to cylinder liner material, was this also random flake graphite? How was the bore machined, and what type of surface was produced?

With regard to the outside diameter, was the liner free from water side attack and what precautions might be taken to deal with this possibility at the higher ratings?

On the question of heat dissipation, was it known what proportion of heat was transferred through the piston crown to the cooling oil and through the piston rings to the cooling water?

MR. J. H. MILTON (Member) wrote that it was stated, on page 327, that to produce a reliable machine one had to pro-

ceed from one successful design to the next, taking care that critical parameters proved in the original design were maintained in the next.

With regard to the critical parameters shown in Table I, it was rather surprising to see that gudgeon pin, or small-end bearings, were not mentioned, as these bearings could be troublesome and also, on occasions, connecting rods had split lengthwise through concentrated eye loading.

Perhaps the authors would care to comment on this subject, and give details of the design of their small-end bearing with particular reference to the bush—whether it was floating or not—and its material.

With regard to the piston design, as shown in Fig. 3, it would be interesting to have the authors' views on the importance of the distance from the crown to the top piston ring, and also further enlightenment on their statements that: a) heat resisting "helicoil" inserts were used to carry the studs and that these acted as a "heat barrier" for these studs; b) that disc springs were fitted under the castle nuts on these studs to increase the resilience of the assembly. Did this mean that they had accepted the fact that movement must take place between the piston crown and the body, and if so, did fretting take place with ensuing leakage of oil across the jointing face?

With further reference to Table I, it was noted that the maximum permissible bearing loads for the main bearings and bottom ends were given as 2,500 and 5,000 lb./sq. in. respectively. Some enlightenment as to how these limitations were arrived at would be of interest.

On page 336, under "Space and Weight", the authors made a good case for the cast iron engine, stating that few fabricated structures were able to avoid fillet welds in load-carrying regions, and that the fatigue strength of such welds might be as low as plus or minus 1·2 tons/sq. in., and that even butt welds had only a fatigue strength of plus or minus 3·8 tons/sq. in., compared with 5 tons/sq. in. for a good quality cast iron. If these figures were correct, it was difficult to understand why, apart from the saving in weight, so many other engine builders had adopted fabricated designs, especially as also, in the event of damage resulting from the failure of a bottom-end bolt, a cast iron engine did not suffer distortion and could usually be "patch" repaired, whereas the fabricated structure was usually distorted and had to be renewed.

It was noted that oil was used for piston cooling and lubrication and in this connexion it would be interesting to know if the authors had any relative figures on lubricating oil capacity (e.g. gal./h.p., in circuit) for the engines forming the subject of this paper, as compared with slow-speed, direct-drive Diesels.

Furthermore, in the case of direct-drive, slow-speed engines burning heavy oil, it was found essential, on account of crankcase corrosion, to isolate the cylinder bottoms from the crankcase—what precautions, beyond using an inhibited lubricant, were being taken to prevent such corrosion taking place in the engines produced by the authors' company.

In conclusion, he would be grateful if the authors could briefly state why, in comparison with the builders of large, slow-speed, direct-coupled engines, they had chosen to develop the four-stroke cycle engine instead of the two-stroke cycle engine.

Commander E. B. Good, O.B.E., R.N. (Member) wrote that, when a new engine design was introduced, it was natural to compare its rating with those of competitors. A true comparison of ratings should take into account many design features, but an indication of the mechanical and thermal loading problems which the manufacturers had to overcome could be obtained from the output per cubic inch of swept volume and the output per square inch of piston area. These factors had been plotted against cylinder bore, for a number of modern turbocharged engine designs, in Figs. 25 and 26.

The factors for the K Major engine had been plotted at each of the development stages referred to in the paper and it could be seen that these ratings lay neither too adventurously above, nor too much below, the lines marking the upper limit of current design.

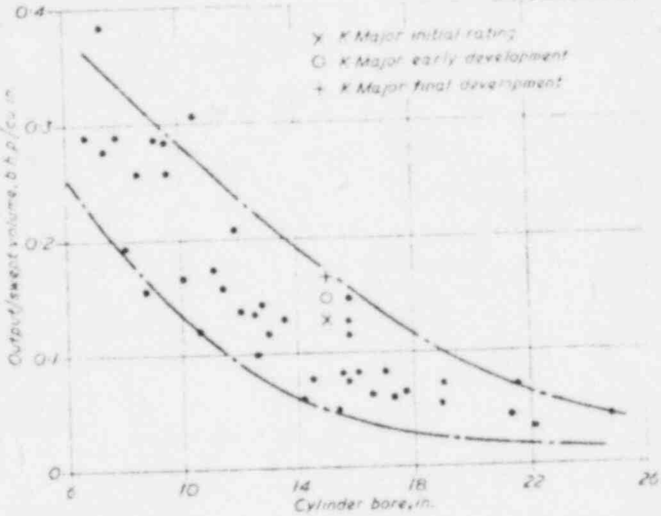

Fig. 25—*Relation between output/swept volume and cylinder bore*

Reference had been made to the use of small centrifuges mounted at the engine for the bypass purification of the lubricating oil. A marine installation of more than 3,000 b.h.p. would normally justify the use of a motor-driven centrifuge for the continuous purification of lubricating oil. Such a system also enabled the whole of the lubricating oil charge to be purified in harbour at the end of each trip, a practice adopted by many owners. Commander Good asked the authors whether they considered that the engine-mounted centrifuges would avoid the necessity for a separate motor-driven unit, particularly in a heavy fuel burning installation, and, if so, could they give an indication of the time after which cleaning of the units would be required. It was assumed that provision was made to cut off the flow to individual centrifuges, to permit them to be cleaned while the engine was running.

The attention which had been paid to the design of the fuel injectors and exhaust valves was very welcome. The maintenance of these items probably represented the largest work load for the ship's engineers. It was considered that a period of 5,000 hours between overhauls would not be an unreasonable aim for the exhaust valves.

Shipowners were becoming increasingly concerned about the noise levels in engine rooms and this was reflected in the

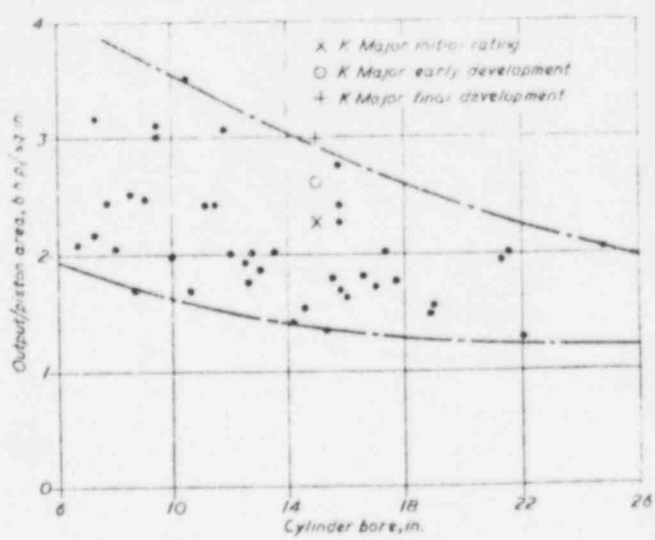

Fig. 26—*Relation between output/piston area and cylinder bore*

number of new ships which had insulated control rooms. With a maximum cylinder pressure 25 per cent greater and a maximum speed 18 per cent greater than its predecessor, the K Major might be expected to be considerably noisier. However, it was possible that the many design changes which had been made had at least partly counteracted the tendency to higher noise levels. It would be useful if the authors could provide any comparative noise measurements for the K and K Major engines.

With the advent of a new medium-speed Diesel engine, it was inevitable that designers of naval machinery installations must ask themselves whether this new engine was suitable for warships. In this respect, section f) on "Space and Weight" was relevant and one could observe that cast iron, whilst it had many admirable properties, was not the best of materials for shock. Perhaps the authors would like to comment on whether they intended to offer a naval version of this engine in due course.

Authors' Replies

Mr. Lowe (replying to the verbal discussion) referred to Mr. Cook's point about 36 cylinders being required for a ship of 18,000 horsepower and said that the paper had tried to show that the power available from that number of cylinders was now twice as great as it was a few years ago, with the same reliability and maintenance requirements.

A comparison of piston-crown temperatures, measured in the engine using templugs, with the temperature distribution produced in the rig and by an electrolytic analogue showed that in the engine, crown temperatures were rather lower than those produced in the rig and predicted by the analogue, while ring-belt temperatures were slightly higher. This difference was attributed to the rather more efficient cooling of the underside of the crown produced by the motion of the piston in the engine. The temperature distribution was proportionately the same in both rig and engine, so that thermal stress measurements should be of the right order.

The authors' company did not use a ratio as high as the 2½ to 3 times increase in fatigue strength of rolled threads over cut threads, mentioned by Mr. Cook. They used a value of 0·25 times the U.T.S. for rolled threads and 0·125 times U.T.S. for machine-cut threads, giving a ratio of 2:1 in favour of rolled threads for fatigue strength.

Both Mr. Cook and Commander Tyrrell had commented on lubricating oils for use with heavy fuel. It was difficult to be precise about the deterioration of oil lubricants because no oil specification defined accurately the requirements of a lubricant for Diesel engines. The type of oil found suitable for this engine was a "good" Supplement 1 level with a quite reasonable alkalinity. The authors used a maximum bulk oil temperature of 170 deg. F. (77 deg. C.) but it must be remembered that this implied that there would be local higher temperatures in the engine of the order of 210 deg. F. (99 deg. C.).

He confirmed that it was not the intention to imply that centri-cast rings were necessarily inferior to sand-cast rings, but the point should be made that the quality of the iron was more difficult to maintain in a centri-cast ring.

Referring to Mr. Henshall's point about specific air flow, he agreed that it was desirable to continue to increase the specific air flow as ratings increased, although the extrapolated figure in Table I showed a small reduction.

There had not been any problem with differential expansion between the steel crown and the cast-iron piston body, probably because the intensive cooling produced a low temperature at the interfaces, as could be seen in Fig. 6. The double line in Figs. 2, 3 and 6 represented a threaded portion which was used to establish the best crown diameter and was a well-known development technique.

The development engine had been run with valve guide diametral clearances as small as 0·001in. to 0·002in. with the force-lubricated guide, and a clearance of 0·003in. to 0·004in. had been arrived at for the final design. Wear rates with the lubricated guide were extremely low and a life of 10,000 hours was expected before replacement of the guide was necessary.

Mr. Gröschel had sounded a word of warning about fuel-injection test rigs and the authors agreed generally with him. Rigs were extremely useful if one was careful in interpreting the results. The suggestion that the problem of back-flow of gas from the cylinder had been prevented by reducing the reciprocating mass in the injector was quite correct and the K Major injector was of the low inertia type, as described by Mr. Gröschel.

The fuel line pressures in Fig. 13 referred to full-load conditions and the authors considered the pressures levels and the difference between line-pressure and gallery pressure to be quite normal in their experience.

He agreed that the higher nozzle temperature for the un-cooled injector with heavy fuel, in Fig. 19, was only partly du[e] to the loss of cooling from the fuel and was also a result of th[e] slower burning of the heavy fuel. The conclusion, however, wa[s] unaltered that cooling of the nozzle was necessary with heavy fuel and tests had been carried out, which were too extensive t[o] be fully described in the paper, which showed that the tip temperature was dependent not only on engine load, but also o[n] fuel temperature, water flow quantity and water temperatur[e]. The thermocouple for these temperature measurements we[re] located actually at the surface of the nozzle tip.

Mr. Alcock had asked for details of the composition o[f] the high-tensile steel piston crown. This was a 55-ton tensil[e] 1 per cent Cr-Mo steel and he would gladly send exact particulars of the material and of oil velocities to Mr. Alcock. He pointed out that the 464 deg. F. (240 deg. C.) maximum temperature in Fig. 6 was a bulk temperature of the materia[l] and that the surface temperature in contact with the oil woul[d] be lower. There had been no signs of oil coking on the surfac[e] of the cooling chamber.

He agreed that it was very critical to choose the righ[t] material, not only for the seat of the valve, but for the valv[e] itself. The relative coefficients of expansion of these tw[o] materials and their conductivity were most important.

The subject of crankcase explosions was a general one an[d] not confined particularly to the engine under discussion. Fi[g]. 1 showed that the engine was fitted with explosion doors. H[is] company had experienced one or two cases, in the last seve[n] or eight years, of crankcase explosions in other types o[f] engine where the explosion doors had worked satisfactorily.

Referring to Dr. Mansfield's question about the metho[d] of varying the wall pressure of the piston rings, he said th[at] this had been done both by altering the bearing area of a giv[en] ring and also by re-designing the ring.

Mr. Baker had suggested that there should be automat[ic] timing on an engine. This was an attractive idea, particular[ly] if combined with automatic timing of fuel injection, but wa[s] very difficult to achieve. If some simple and foolproof way o[f] doing this could be found it would be a real achievement.

He said that he could not justify the mean line for th[e] 200 b.m.e.p. data on the right-hand diagram of Fig. 10. Th[e] measured points indicated the "hog", suggested by Mr. Bake[r], but he could not explain why this should be so and hence ha[d] merely indicated the downward trend which he thought to [be] of greater significance.

Referring to the injection diagrams, he said that the u[n]loading volume had been increased from (a) to (b) in Fig. [ ] and had a very strong effect on the tendency for seconda[ry] injection. The injection rate had also been increased and, [as] already stated in the reply to Mr. Gröschel, a low-inertia ty[pe] of injector had been adopted. The fuel line pressure w[as] measured halfway along the injection pipe in each case and th[e]

injection pipe length was the minimum possible for the engine. Mr. Baker's remarks about the Willan's lines were quite valid and, in fact, the twelve-cylinder engine had shown a mechanical efficiency of 93 per cent. The performance curves were obtained using distillate fuel.

The piston-ring pack, described in the paper, was deliberately designed so that the scraper ring above the pin was more severe than the lower scraper ring. The authors considered it important to maintain an adequate oil film over the body of the piston so that the lower scraper ring was only intended to remove excess oil to prevent the upper ring becoming flooded. The weight loss figures, in Table IV, were only significant in comparison to one another as they were obtained from a rig running completely unlubricated and could not be related to conditions in an engine cylinder. There had not been any controlled tests to measure wear rates in the grooves of the steel crown with piston rings of different irons.

In answer to Mr. Cowderoy's question about compression pressures, he said that, at 525 r.p.m. and full load, the compression pressure was 900 lb./sq.in., coming down linearly with horsepower to 380 lb./sq.in., at 100 r.p.m., i.e., effectively no load. The compression pressure, corresponding to the 60 per cent speed, 22 per cent load condition, quoted by Mr. Cowderoy, was 430 lb./sq.in. He agreed with the contributor that low compression pressure, or rather compression temperature, could contribute to inferior combustion in a worn engine at low load.

Mr. Cowderoy's suggestion that marine propulsion engines should be treated like locomotive engines, from the point of view of preventing the driver from accelerating too fast, was a good point. The necessity for the turbocharger to accelerate was often overlooked when rapid increases in load were called for, and it might well be necessary to apply the locomotive type of fuel rate control to marine engines.

In reply to Mr. French, he said that the computer approach to piston crown thermal stresses had been to construct an electrolytic analogue which, as mentioned earlier, gave quite good agreement with the thermal rig. Work was currently in progress on a digital programme which would calculate stresses directly, whereas the analogue only gave temperature distribution from which stresses could be calculated.

The numerical value of the wear factor for inlet valves obviously applied to one's own engines, but a manufacturer could apply the formula to obtain values from his own engines. In this connexion, he said that the experimental work, from which the wear factor was developed, was described fully in reference (2).

Dr. Pope said that before the meeting closed he would like to make one or two comments about some generalities which had come up during the discussion.

He said that they were getting to the stage in the medium-speed engine industry where the research and development effort of the engine builders was outstripping the component builders, and he could foresee, in the not too distant future, that engine development might well be held up because of lack of blowers and injection equipment. He hoped the supply industry would persevere with enthusiasm for the highly-rated medium-speed Diesel engine as much as the engine builders were.

He thought that the problem of the maintenance of medium-speed Diesel engines in the marine world should be judged objectively. Obviously, in the medium-speed engine one was going to have more parts, but there was a world of difference between handling a 15-in. piston and handling a 30-in. piston. The factors involved were not just the number of parts, but the way in which they could be manipulated and a statistical analysis of what were the major and minor faults. His view was that if this analysis were carried out scientifically one would find that the medium-speed engine could stand on its own, even with regard to maintenance.

With regard to cast iron, he said that each problem must be judged on its merits. The fatigue strength of a good cast iron was near to that of a good steel. There were other advantages in using cast iron. One knew that one could obtain good castings with cast iron for the size of engine he was discussing and it was an easy material to handle. Size for size, his experience had been that cast iron came out cheaper, therefore if one had a material which was as good as another and was cheaper, one had to have a very good reason for not using it. He could only see one reason for precluding its use and that was if weight were a predominating factor. However, when one considered the rest of the engine room equipment, the tankerage and the fuel capacity, one would find that there was a difference of one or two per cent between a welded design and a cast iron design, so that, for commercial shipping, this was a marginal consideration.

He pointed out that with an in-line engine with an underslung crankshaft, one could have a very nice stress line pattern which, on the drawing board, looked very attractive and almost impossible to improve upon, but when one came to a "V" engine with side by side connecting rods, so that the opposite liners could not be in line with each other, the stress pattern did not look quite so elegant. He accepted that the cast iron bedplate was more difficult to design because the stress pattern was more complex, but once one had designed it and got a good design one was simply comparing one good design with another. It was also a question of continuing from a well-tried engine to the next generation, without departing from well-proven design principles. His company had now completed its one thousandth K engine and had over 300 of them in marine application. Over a third of the engines were running day in and day out on heavy fuel.

AUTHORS' REPLY TO WRITTEN CONTRIBUTIONS

The authors wrote that they entirely agreed with Commander May's appreciation of the rapidly increasing demand for large medium-speed engines. They had deliberately restricted the paper to the development of the K Major engine itself but their company was certainly proposing to offer complete propulsion units for geared installations with either single or multi-engine inputs.

In reply to Mr. Hughes, the authors wrote that the piston cooling-oil flow was 1·7 gallons/b.h.p.-hr. at the current full-load rating. The cooling design was such that the operating temperatures of the piston crown were well within the thermal fatigue limit of the steel used, as could be seen from the isotherms of Fig. 6. The work that had been carried out on the optimization of valve timing and combustion characteristics, which was described, had been aimed at obtaining the best thermal efficiency from the engine and not at reduction of component temperature by scavenge air cooling, which they considered was a relatively inefficient method of controlling component temperatures. The temperature of the critical parts of the engine, such as exhaust-valve seat, injection nozzle, and top piston-ring groove, was controlled by direct cooling.

As mentioned in the reply to Mr. Baker, the lower scraper ring in the skirt was relatively mild and the upper scraper more severe to ensure adequate lubrication of the piston skirt. The taper-faced compression rings were an advantage in initial running as they bedded in very quickly on a narrow circumferential band. Initial running-in had been the subject of a good deal of investigation on the test bed and the best results had been obtained with a relatively rough liner surface, which was honed to a C.L.A. of about 100μ, the liner being a random flake graphite iron, slightly softer than the piston-ring material.

Calculations of liner frequency and vibration amplitude were included at the design stage to avoid the possibility of water-side attack. The heat dissipation through the piston rings to the cooling water could not be directly measured in the engine because of the heat received directly by the liner from the combustion gases. By reproducing temperatures and total heat flow in the rig, the proportion of heat flowing to the cooling oil was about 75 per cent of the total and the heat to the liner was 25 per cent.

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The Development of a Highly-rated Medium-speed Diesel Engine of 7,000—9,000 Horsepower for Marine Propulsion

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The design considerations and development tests are described which have resulted in the production of the Mirrlees-National K Major engine, which has a current commercial rating of 3,000 to 7,500 b.h.p. in 6 to 18-cylinder units, and a projected future rating of 9,000 b.h.p. in 18 cylinders.

The Mirrlees K engine has been well established for over 12 years, some 980 engines now being in service for power generation and marine propulsion. Of these engines, 250 are operating on heavy fuels with viscosities ranging from 200 to 4,600 seconds Redwood I, representing over 650,000 horsepower. The objective in the design and development of the K Major engine has been to increase the specific power output by 50 per cent and at the same time to maintain or to increase the safety factors possessed by the original K engine. These factors, which determine the ability of the engine to operate on residual fuels with low maintenance and high availability, are discussed and the achievement of the objective is illustrated.

Component parts of the engine are described in turn, with details of the methods of measurement of pressure and temperature levels, air flow and wear rates in test rigs and in a prototype three-cylinder engine which was equipped with special features, such as a camshaft with variable timing, to facilitate development work.

The test results obtained on the first 12-cylinder KV Major engine are shown to confirm the performance expected from the rig and prototype engine tests.

INTRODUCTION

In general, the requirements of a marine propulsion engine are:

a) reliability;
b) low fuel consumption;
c) the ability to burn heavy fuels obtained in any part of the world;
d) low lubricating oil consumption;
e) low maintenance requirements;
f) minimum space and weight in keeping with a) and e).

These requirements are obvious but can only be achieved if certain basic principles in design are followed. The paper is divided into sections, each dealing with one aspect of design which affects these overall qualities.

However, before detailing these definite sections, some observations must be made on the application for which the engine is to be used and its suitability for that application. An engine developing 7,000 to 9,000 b.h.p. in 18 cylinders would be ideal for medium-speed marine propulsion since the power range available would be from 3,000 h.p., in a single six-cylinder engine, to 18,000 h.p. with twin 18-cylinder engines. This power range covers a large section of the marine market, as illustrated in Fig. 1, so that if conditions a) to f) can be achieved a worthwhile market should exist for such an engine.

The initial design study showed that the dimensions of the Mirrlees-National K engine (15-in. bore × 18-in. stroke) would fit this power range very well, if the new design embodied the modern features resulting from research and development which would enable high specific outputs to be obtained whilst retaining economy and reliability. At 500 r.p.m., and 200 lb./sq. in., b.m.e.p., this size of engine would give 402 h.p./cylinder, while at 525 r.p.m., and 220 lb./sq. in., b.m.e.p., 465 h.p./cylinder would be developed. The design of the K Major engine was based on a continuous rating of 250 lb./sq. in., b.m.e.p., at 525 r.p.m., giving 528 h.p./cylinder, and the development programme was planned to achieve this rating, using heavy fuel, in the three stages mentioned.

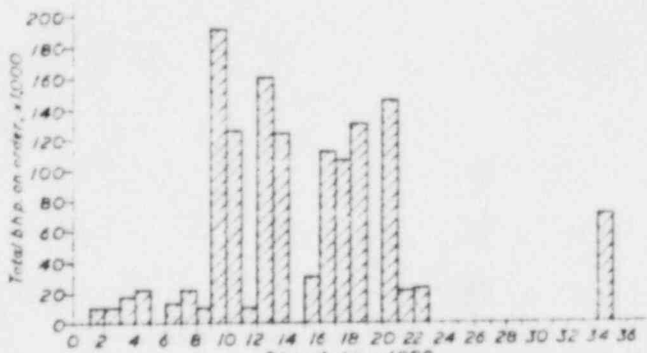

FIG. 1—*Distribution of horsepower for ships over 2,000 d.w.t. on order in Great Britain in March 1965*

At the present time, the K Major is released for the commercial market at a rating of 200 lb./sq. in., b.m.e.p., at 500 r.p.m., and development testing for the second stage of 220 lb./sq. in., b.m.e.p., at 525 r.p.m. is well advanced.

A cross-section of the engine, showing its general construction, is shown in Fig. 2 and the details of the design will be dealt with in the following sections of the paper under the headings a) to f) already given.

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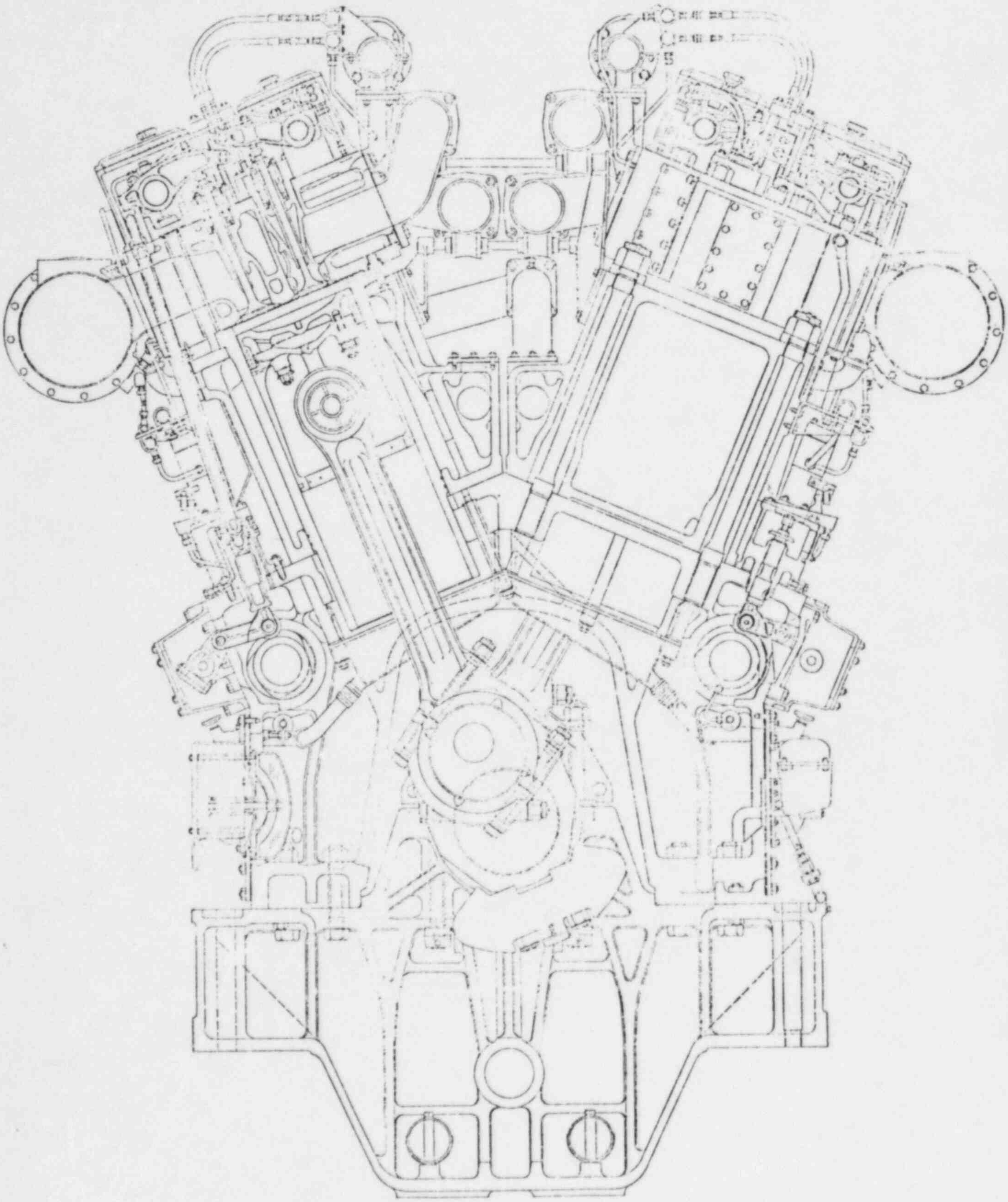


FIG. 2—*KVD Major engine—Cross-section*

a) RELIABILITY

General Considerations

Experience in engineering has shown that one of the surest methods of producing intrinsic reliability in a complex piece of machinery, such as a Diesel engine, is to proceed by a process of logical evolution from one successful design to the next, taking care that the critical parameters, proved in the original design, are maintained in the new design. From the authors' experience of continuous-duty Diesel engines, the critical parameters to be carefully watched are:

1) exhaust temperature after the valves should not exceed 820 deg. F. (440 deg. C.) with uncooled valve seats and 930 deg. F. (500 deg. C.) with cooled seats;
2) top piston ring groove temperature should not exceed 430 deg. F. (230 deg. C.);
3) injector nozzle tip temperature should not exceed 350 deg. F. (180 deg. C.);
4) exhaust valve seat temperature should not exceed 1,020 deg. F. (550 deg. C.);
5) lubricating oil consumption should not exceed one per cent of fuel consumption at full load and if possible should approach 0·5 per cent of full load fuel consumption;
6) all bearings should be well within their load-carrying capacity;
7) the stressing of all components, both in fatigue and static loading conditions, should be such that an adequate factor of safety exists.

Some of the more important design and performance characteristics of the K Major engine are compared in Table I with those of the earlier and successful K engine, the comparison showing that safe values of the critical parameters have been maintained and, in many cases, improved.

TABLE I

Parameter	Mirrlees KV12 engine	Mirrlees KV12 Major engine		
	154 lb./sq. in. b.m.e.p. 450 r.p.m.	200 lb./sq. in. b.m.e.p. 500 r.p.m.	220 lb./sq. in. b.m.e.p. 525 r.p.m.	250 lb./sq. in. b.m.e.p. 525 r.p.m.
Moment of inertia of bedplate (in.^{4})	18,650	31,430	31,430	31,430
Maximum internal couple, tons. ft.	210	264	290	290
Ratio, maximum couple ÷ moment of inertia	0·0113	0·0084	0·0092	0·0092
Relative stress in crankshaft	1	0·82	0·91	0·96
Maximum cylinder pressure, lb./sq. in.	1,080	1,350	1,350	1,400*
Cylinder head stud stress, tons/sq. in.	8·3	10	10·1	10·4*
Fatigue strength of threads, tons/sq. in.	17	27	27	27
Ratio, stress ÷ fatigue strength	0·49	0·37	0·375	0·385*
Hertzian stress in fuel cams, lb./sq. in.	262,000	205,000	215,000	225,000*
Main bearing load, lb./sq. in.	814	1,234	1,275	1,330*
Maximum permissible bearing load, lb./sq. in.	1,500	2,500	2,500	2,500
Ratio load/permissible load	0·54	0·49	0·51	0·53*
Large end bearing load, lb./sq. in.	2,400	2,800	2,900	3,050*
Maximum permissible bearing load, lb./sq. in.	2,700	5,000	5,000	5,000
Ratio load/permissible load	0·89	0·56	0·58	0·61*
Maximum stress in piston ÷ U.T.S. of material	0·44	0·33	0·35	0·36*
Top piston ring groove temperature, deg. C.	220	165	185	205*
Exhaust temperature at cylinders, deg. F.	800	810	850	890*
Air flow, lb./b.h.p.-hr.	13·3	13·8	13·9	13·7*
Exhaust valve seat temperature, deg. C.	540	460	490	520*
Specific fuel consumption, lb./b.h.p.-hr.	0·336	0·335	0·336	0·340*
Lubricating oil consumption, lb./b.h.p.-hr. at full load	0·0030	0·0020	0·0020	0·0018*
Weight of engine, lb./b.h.p.	44	30	26	23
Thrust area of piston, sq. in.	138	162	162	162
Maximum thrust pressure on piston, lb./sq. in.	35·3	33·5	34·0	34·8*
Depth of cylinder head, in.	11·75	13·5	13·5	13·5
Injector nozzle tip temperature (fuel at 200 deg. F.), deg. C.	177	127	130	136*
Inlet valve seat wear factor	192	156	156	162*

*Extrapolated values

Piston Design

The control of top piston ring groove temperatures by cooling the underside of the crown of the conventional single-piece cast iron piston, used in the K engine, is acceptable up to a rating of about 180 lb./sq. in., b.m.e.p., using a cast iron having a U.T.S. in the ring belt of 17 tons/sq. in., but, above this load, high tensile thermal stresses are produced on the inside wall of the piston behind the ring grooves[1]. For the K Major engine, a two-piece construction has been developed, as illustrated in Fig. 3, which has a high-tensile steel crown and a "Meehanite" skirt. This design incorporates an inner load-carrying boss, so that no pressure load is taken on the outer wall which carries the rings, and the latter may be quite thin, thus reducing the heat-flow path to the piston rings and giving efficient oil cooling of the ring belt, as well as ensuring that the roots of the piston

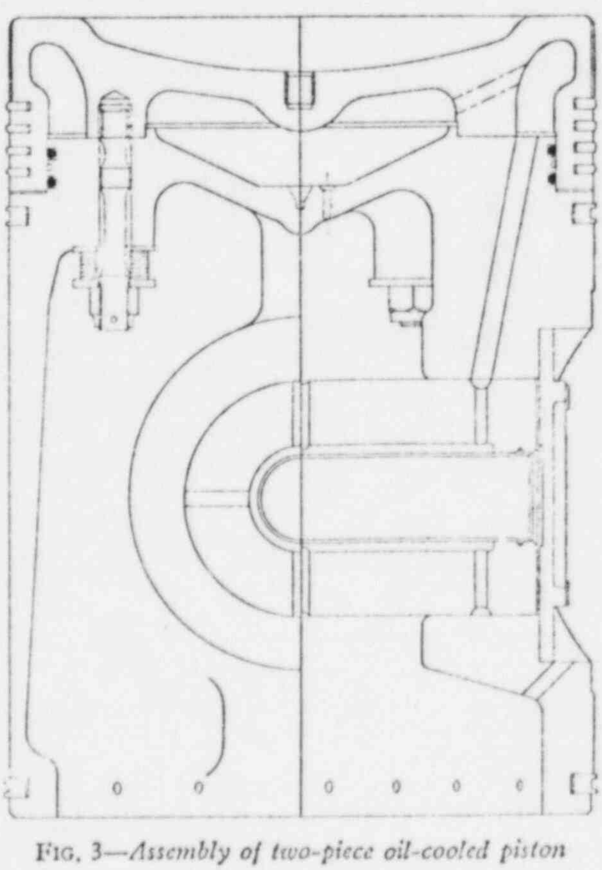


FIG. 3—*Assembly of two-piece oil-cooled piston*

ring grooves are stress-free. The piston crown is retained by four high-tensile studs which have rolled threads to give maximum fatigue strength, and heat-resisting "Helicoil" inserts are used to carry the studs, thus further improving the fatigue strength of the assembly and also acting as a heat barrier for the studs. Disc springs are fitted under the castle nuts to increase the resilience of the assembly and to provide an accurate method of checking the correct pre-load of the studs, this being achieved by measurement of the gap between the two retaining plates for the springs. Lubricating oil is fed, via a drilling in the connecting rod, through the piston pin and to the annulus chamber behind the ring grooves, through which it circulates at high velocity before meeting a transfer drilling to the inner chamber below the piston crown, from where it finally passes down an integral drain drilling in the piston skirt. The returning oil is collected in a cast aluminium tray, supported from the engine column, and is fed through a flexible connexion to a sight-flow and temperature indicator mounted adjacent to the crankcase door.

The thermally-induced and pressure-induced stresses have been thoroughly investigated in test rigs prior to tests in the prototype engine. Fig. 4 is a diagram of the thermal stress rig which is used to simulate the heat flow through the piston which occurs in the engine, heat being supplied by electric immersion heaters using solder as the medium for transferring the heat to the piston crown. Heat transfer through the piston rings is achieved by water-cooling the standard engine liner and oil-cooling of the piston internally is arranged in the same way as in the engine. Thermal stresses are measured by Budd self-temperature compensated strain gauges, having an overall size of ⅛in. × ¼in., so that the effect of the gauges on the heat transfer conditions is extremely small. Fig. 5 shows the variation in thermal stress in the piston wall and also the temperature in the region of the top ring groove as a function of brake mean effective pressure for both the original single-piece piston and for the K Major two-piece piston; Fig. 6 illustrates the temperature and stress distri-

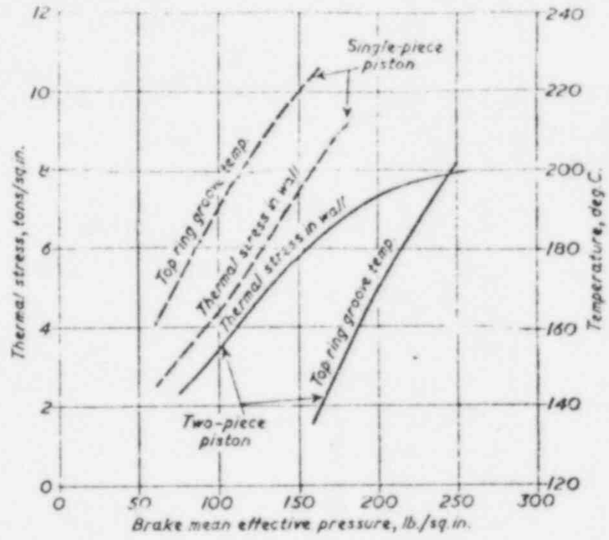


FIG. 5—*Comparison of temperatures and stresses in single-piece and two-piece piston designs*

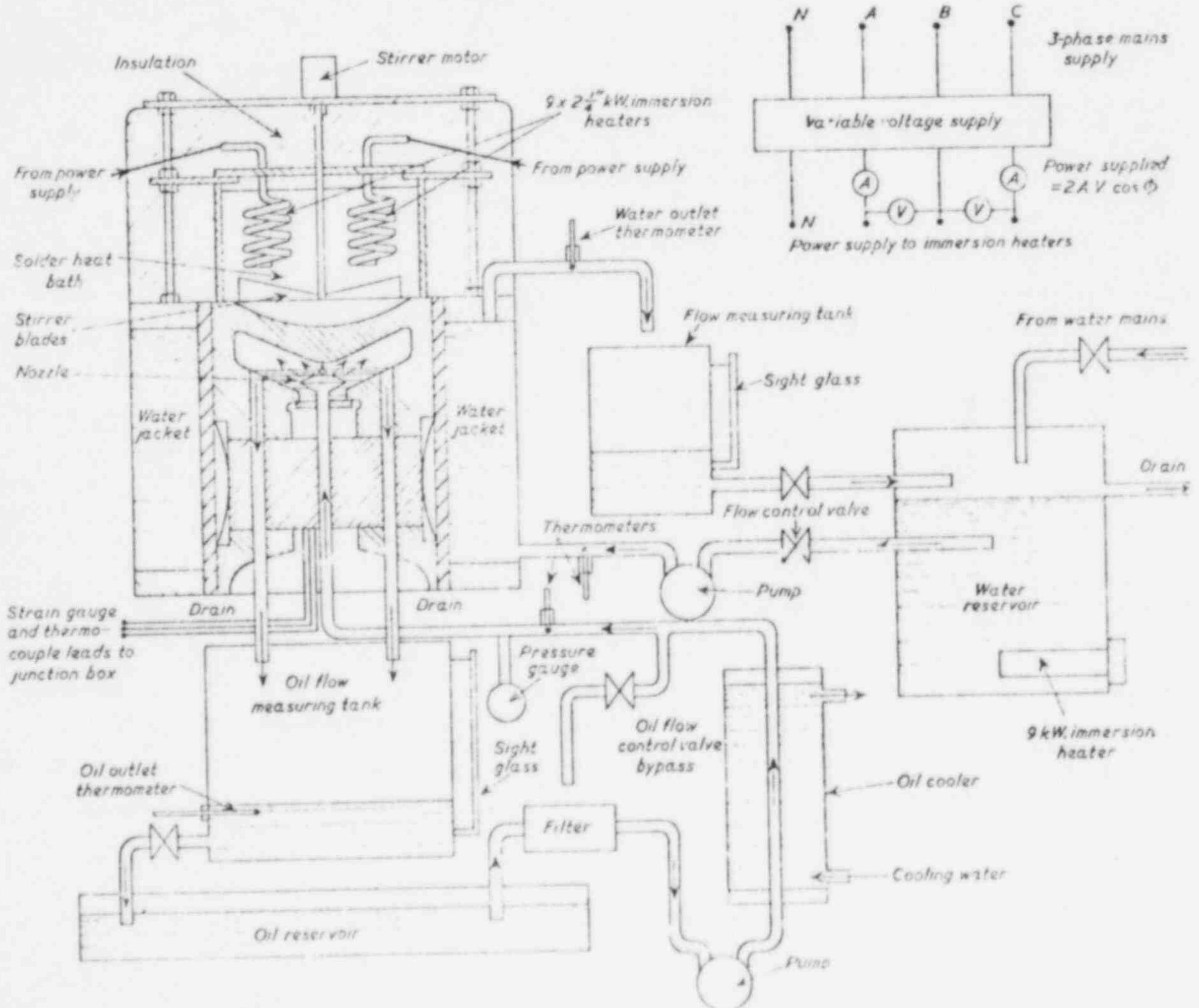


FIG. 4—*Thermal test rig for pistons*

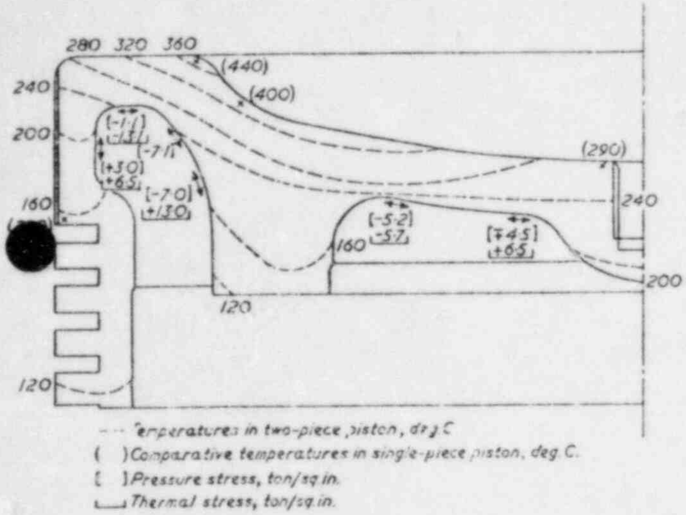
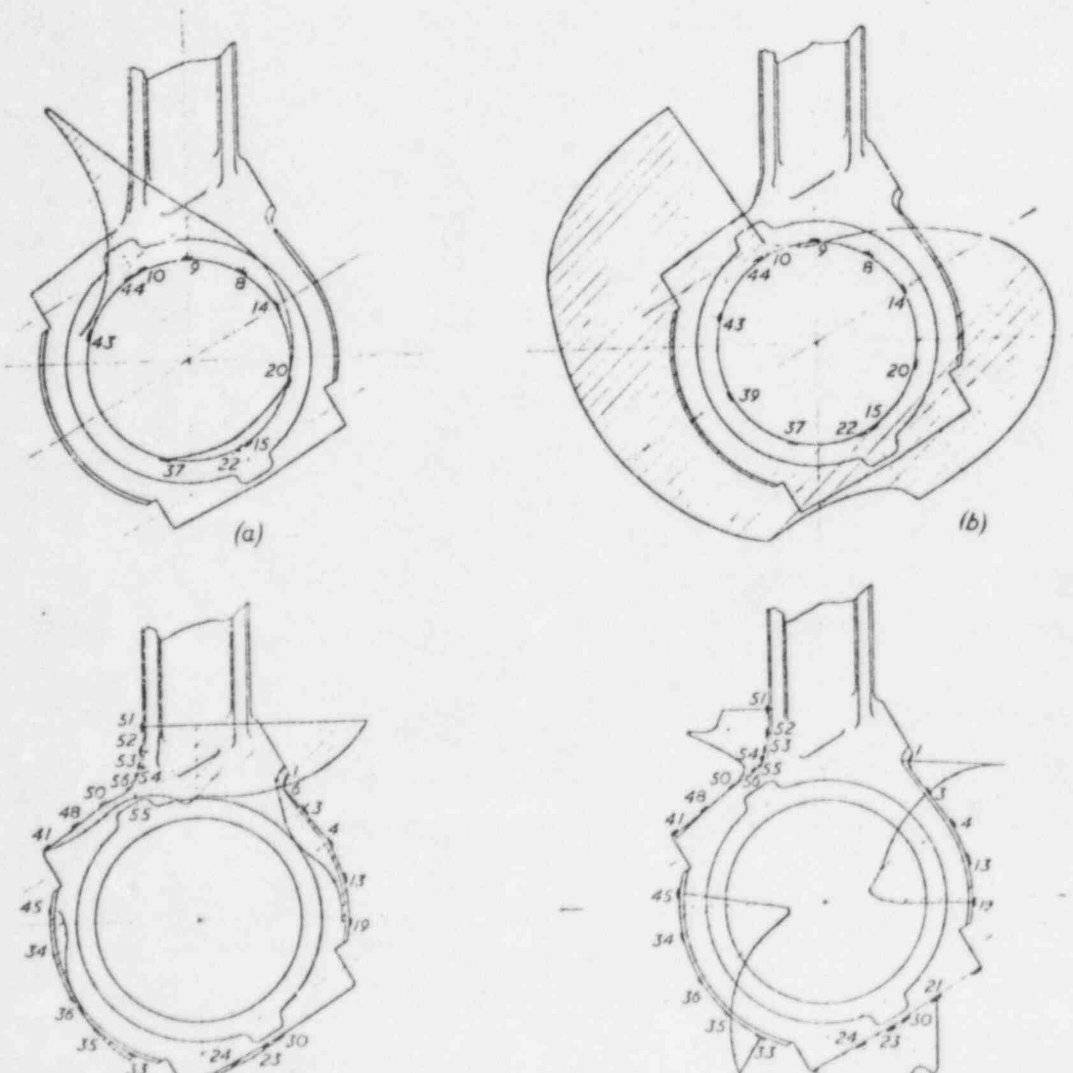


FIG. 6—*Two-piece piston crown temperatures and stresses*

bution in the crown of the two-piece piston. The reduction in top ring groove temperature by some 126 deg. F. (70 deg. C.), achieved by the new design, has made available a wide potential for increase in rating in the future, before any limitation due to lubricating oil break-down is reached. The thermal and pressure stresses are quite modest for the steel crown, which has a U.T.S. of 60 tons/sq. in., at room temperature, so that the factors of safety are much increased over the original single-piece cast iron design.

Connecting Rod

The connecting rods are one-piece stampings with the large-end bearing housing obliquely split at 30 degrees to the rod axis, and carry thin-wall tin-aluminium bearings. This construction permits a crank pin of maximum diameter, consistent with the withdrawal of the connecting rod through the cylinder bore. The optimization of the connecting rod proportions has been assisted by rig tests in a full scale static rig, in which gas loads and inertia loads are simulated by hydraulic pressure and the resulting stresses measured by strain gauges attached to the connecting rod. It was thus possible to reduce the weight of the connecting rod by 15 per cent from that of the original K rod so that, even at



a) Bore stress (gas load)
b) Bore stress (inertia load)
c) Backbone stress (gas load)
d) Backbone stress (inertia load)

FIG. 7—*Connecting rod large-end stress distribution*

the increased speed and load, the connecting rod stresses are lower than in the original design. Fig. 7 shows the stresses in the large-end of the connecting rod under firing pressure and inertia loading, and, in Table II, the safety factors at the most highly stressed points have been listed. In determining these values, allowance has been made for factors which would affect the fatigue strength of the material, such as specimen size effect and surface decarburization where it exists, so that the resulting values indicate the worst conditions and show that the rod design has a large margin of safety.

TABLE II

Gauge No.	Position	Factor of safety
1	Bolt platform radius	3·8
19	Supporting rib	6·8
33	Supporting rib	5·7
51	Base of shank	3·4
55	Shank radius	3·7
59	Bolt platform radius	4·4

Bearings

Main and large-end bearings are thin-wall steel shells lined with tin-aluminium, the increase in bearing loads from the K to the K Major being more than compensated by the improvement in fatigue strength of the bearing material. The actual and permissible bearing loads given in Table I illustrate the increased factor of safety in the new engine, the figures given being the conventional pressures obtained by dividing the maximum bearing load by the projected area of the bearing so that a simple comparison can be made. In the design of the K Major the more accurate methods of calculation, which have been made possible by the use of computers, have been used to assess oil film thickness over the range of speeds and loads so that the true factor of safety is even higher than the simple comparison suggests.

A positive displacement lubricating oil pump is driven from the free end of the engine by a flexible drive and delivers oil through a 15-micron full-flow filter to the main oil gallery cast in the bedplate. An oil-pressure regulating valve is fitted at the engine gallery to ensure that engine oil pressure remains constant, regardless of the degree of contamination of the filter, and a pressure-safety valve at the pump delivery protects the pump in the event of a complete blockage of the system. In addition to the full-flow filter, about five per cent of the flow is bypassed and filtered by small centrifuges mounted at the engine. This dual filtration ensures that carbon and water particles are removed from the lubricating oil and prevents the formation of sludge in the main filters. Tests have shown that a considerable increase in filter life is achieved by this system.

The quantity of lubricating oil circulated through the engine has been determined after thorough development tests to investigate the distribution of oil to main bearings, large-end bearings, piston cooling and other requirements, and the oil quantity has been chosen not only to lubricate but also to cool the main and large-end bearings, thus ensuring that the fatigue strength of the bearing material is maintained at its maximum value.

b) LOW FUEL CONSUMPTION

The importance of adequate air flow in a high-powered Diesel engine cannot be over-emphasized, the air delivered by the turbocharger having to perform the duties of scavenging the cylinder from the products of combustion and of cooling the components in the combustion space region, as well as providing a high mass of trapped air for the combustion process. In recent years the efforts of specialist turbocharger manufacturers to improve turbine and compressor efficiencies have made a substantial contribution to the success of the highly-rated Diesel engine, and the engine manufacturer can play his part by ensuring the maximum utilization of exhaust gas energy and by minimizing flow losses in the porting and ducting.

Air Ports

Air flow tests on the K cylinder head showed that the pressure drop in the inlet passages was made up as follows:

Inlet passage up to valve	11 per cent
Velocity change round valve seat	38 per cent
Loss of velocity head at outlet	35 per cent
Interaction between valves and cylinder wall	11 per cent

The large percentage loss around the valve seat indicated that optimization of the valve head profile and inlet passage shape in this region would be worth while, and the tests also showed that a greater effective flow area could be made available by increasing the valve lift beyond the value of a quarter of valve diameter at which the minimum geometric area becomes constant. Fig. 8 shows the increase in coefficient of discharge beyond the normal L/D ratio of 0·25 and the K Major valve lift was chosen

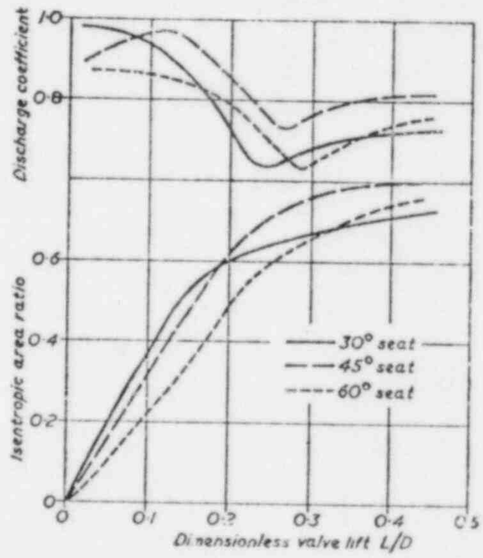


FIG. 8—*Flow characteristics of valves with 30-degree, 45-degree and 60-degree seats*

to be 0·3 of the valve diameter, giving an increase in maximum effective area of just over five per cent. This improvement is quite significant when it is remembered that it is effective over a large valve opening period.

The effect of varying valve seat angle on flow characteristics was also examined and Fig. 8 shows the characteristics of valves of the same throat area with seats at 30, 45 and 60 degrees to the face of the valve. The 60-degree seat valve is clearly inferior to the other two, and the 30-degree seat is the best at small valve openings, whereas the 45-degree seat is best at large valve openings, while the advantage to be gained by increasing the valve lift beyond $L/D = 0{\cdot}25$ is valid for all values of seat angle.

There are other factors to be considered in choosing seat angle which determine the relative merits of 30 and 45-degree seats, of which the most important is that of useful seat life in service. Here, the conditions for inlet and exhaust valves are quite different, and will be considered separately.

The inlet valve operates in a relatively unlubricated condition at the seat so that seat wear, due to the relative movement of the valve seating face against the face in the cylinder head, as a result of the gas pressure, may be quite appreciable. A "wear factor" was derived theoretically and its validity confirmed by rig and engine tests from which the K Major inlet valve head profile was determined to give the minimum practicable relative movement and hence minimum wear[2]. The "wear factor" is defined as:

$$F_W = \frac{P_m^2 . N . \mu . D^3}{E . B . t . v^2 . b . \cos\theta}$$

where μ = coefficient of friction;
P_m = maximum cylinder pressure;
N = engine speed;
D = valve disc diameter;
θ = seat angle;
E = Young's modulus;
B = wear resistance factor (hardness number);
b = seat width;
t = distance from valve disc face to top of seat;
v = height of valve disc cone.

It can be seen that a decrease in θ, or increase in t and v have the effect of reducing the "wear factor" and the K Major inlet valve head profile was designed from these considerations with a 30-degree seat angle and a stiff valve head. From experience on other engines a wear factor of above 250 gives unsatisfactory life in service and a value of 200 is satisfactory. It will be seen from Table I that the original K engine has a satisfactory value, which is confirmed by service experience, and the K Major has an even bigger safety margin.

The criteria for the seat of the exhaust valve are quite different and will be discussed later in the paper under the heading of "Heavy Fuel Operation".

Valve Timing

The influence of valve timing on the exhaust, scavenge and charging processes has been examined experimentally on a three-cylinder engine, which was fitted with a special camshaft, in which the timing of both opening and closing of the air and exhaust valves, and of fuel injection were widely variable. Fig. 9 is a pictorial sketch of one of the variable timing cams showing the method by which the valve period is adjusted. Each cam is made in two pieces which are able to rotate relative to each other when hydraulic pressure is applied between the cams and the shaft from a hand pump. Release of the pressure then shrinks the cam on to the shaft to give an interference fit and the two parts of each cam are interlocked to form a bridge over which the cam follower roller can run without any discontinuity of profile. This method of hydraulic mounting allows the whole composite cam to be rotated to any desired position, as well as permitting the opening and closing flanks to be rotated relative to each other. Engine tests have been carried out over a wide range of valve timings, recording overall engine performance and pressure diagrams in the air inlet passages, engine cylinder and exhaust passages, from which optimum cam timings can be determined for any engine speed and load condition.

It will be appreciated that the optimization of valve timing is a complex operation and for a given set of timings it is necessary to match the injection equipment and the turbocharger performance for the engine speed and load range being considered. Some results have been selected from the range of tests carried out on the three-cylinder engine to illustrate the way in which changes in timing can affect the power range over which minimum fuel consumption is achieved.

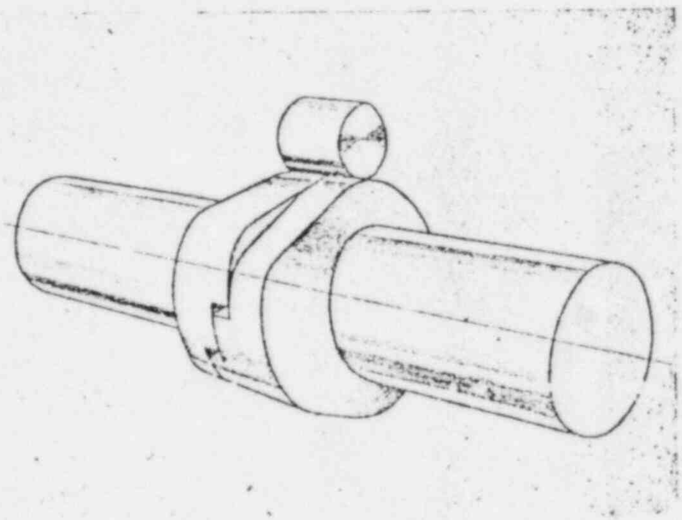


FIG. 9—*Construction of variable timing cam*

In Fig. 10, the performance of the three-cylinder engine is shown with all valve timings held constant except the point of exhaust valve opening, the turbocharger match being changed to give the same total air flow. The valve timings were:

	E.V.O. before B.D.C.	E.V.C. after T.D.C.	A.V.O. before T.D.C.	A.V.C. after B.D.C.
Timing A	43 degrees	62 degrees	73 degrees	32 degrees
Timing B	65 degrees	62 degrees	73 degrees	32 degrees
Timing C	75 degrees	62 degrees	73 degrees	32 degrees

The left-hand curves of Fig. 10 show the performance at 450 r.p.m., and since the engine did not have the improved air flow already described in the previous sub-section, the optimum fuel consumption occurs close to the original K rating of 150 lb./sq. in., b.m.e.p. As the exhaust valve opening point is advanced, the position of minimum consumption moves further up the b.m.e.p. scale. This point is more strikingly illustrated in the right-hand curves of Fig. 10 where fuel consumption is plotted against exhaust valve opening point. At the lower rating of 140

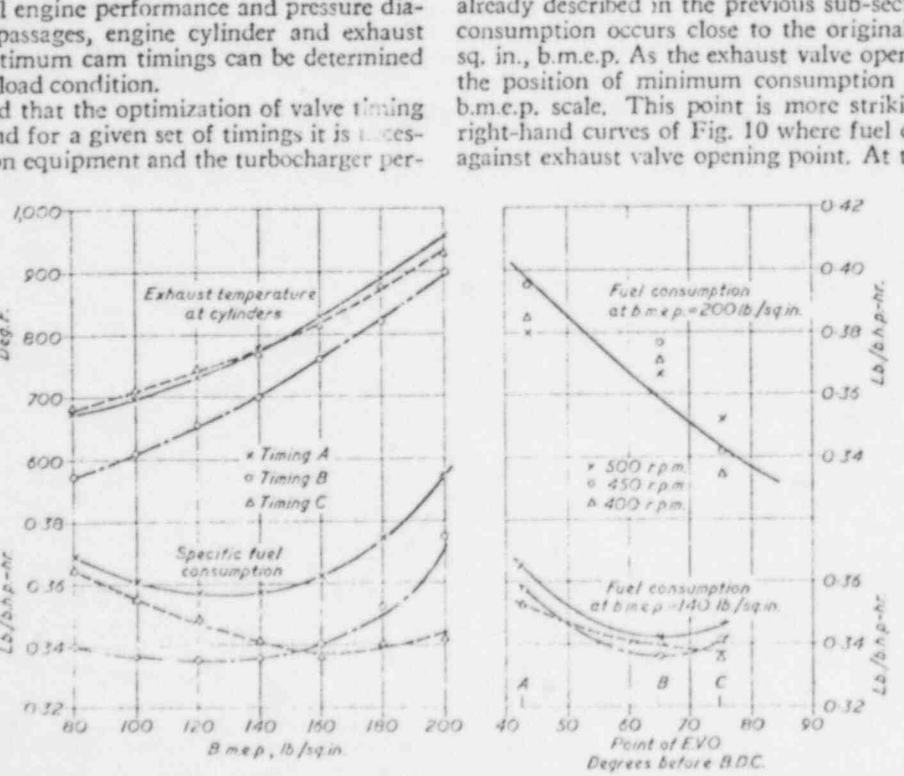


FIG. 10—*Effect of advanced exhaust valve opening point on performance*

... /sq. in., b.m.e.p., the change in fuel consumption is small but, at 200 lb./sq. in., b.m.e.p., there is a marked reduction in fuel consumption as exhaust valve opening is advanced. It must be emphasized that this illustration is intended to be indicative only of the beneficial effects of early exhaust valve openings. It is important that, at high b.m.e.p. ratings, good thermal efficiency is maintained and in the 12-cylinder K Major engine the minimum fuel consumption occurs at about 200 lb./sq. in., b.m.e.p., as illustrated later in Fig. 14, this result being achieved by improvement in air flow and fuel injection. Fig. 11 shows a low-pressure cylinder and manifold diagram for the 12-cylinder engine at 240 lb./sq. in., b.m.e.p., and 500 r.p.m., and demonstrates the good scavenging and adequate charging of the cylinder which has been obtained. As the development of the engine continues to even higher ratings it will be necessary to move the specific fuel consumption loop still further and the indications from the three-cylinder engine tests are that the advantages of earlier exhaust valve opening will be realized at this stage.

Cam Design

The increase in speed and loading, accompanied by faster opening and closing rates of the air and exhaust valves and the increased lift already described, would be expected to make much greater demands on the air and exhaust cams and follower gear. However, the design of cam profile, to optimize on rates of opening without exceeding established acceleration levels, has been considerably facilitated by the use of computer calculation techniques. The K Major air and exhaust cams are of polynomial profile, the mathematical analysis of the profile by computer calculations making selection of the most desirable curve a relatively simple procedure. The behaviour of the valve gear mechanism under running conditions, to determine the degree and frequency of vibration, has also been programmed and vibration calculations confirmed by a very simple technique of attaching strain gauges to the engine push rods. Fig. 12 is a typical push-rod strain trace which clearly indicates the natural frequency of the valve gear system and confirms that there is no tendency for separation of the valve train to occur.

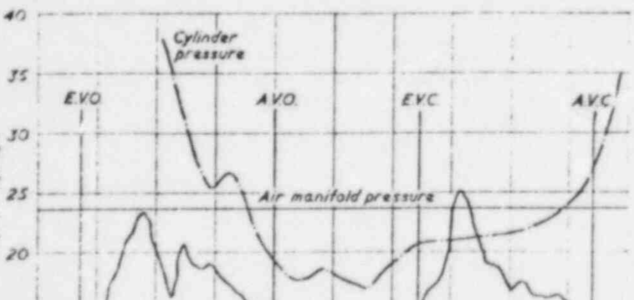


FIG. 11—*Cylinder and manifold low pressure diagrams at 240lb./sq. in., b.m.e.p., and 500 r.p.m.*

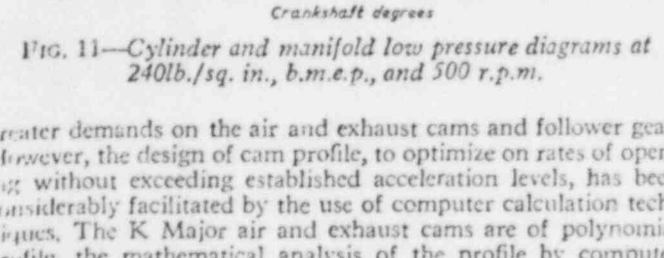


FIG. 12—*Push-rod strain at 200lb/sq. in., b.m.e.p., and 500 r.p.m.*

Fuel Injection

To obtain a maximum rate of injection, the fuel cams are of a profile which gives a constant plunger velocity during the injection period, and the correct matching of the injection equipment was facilitated by the use of test rigs which enabled the injection characteristics to be determined and the design of the injection equipment to be very nearly finalized before engine tests were started, only the confirmation of nozzle spray angle and the number and diameter of nozzle holes of a predetermined area remaining for final decision from the performance of the engine.

These test rigs enable a large number of permutations of fuel cam, pump plunger diameter, delivery valve design, nozzle design, etc., to be tested quickly and cheaply, using conventional methods of electronic indication of needle lift, fuel line pressure and nozzle sac pressure. The latter has proved to be of considerable importance in ensuring long life of injector nozzles by explaining the reason for over-rapid deterioration of nozzles in the K engine under certain service conditions. This phenomenon was a difficult one to explain until, as a result of calculations and rig tests carried out by the fuel injection manufacturer, it was realized that a particular combination of load and speed resulted in a hydrodynamic system in which there was a sudden reduction in fuel pressure in the nozzle sac just before the needle closed, the time interval between the two events being of the order of a quarter of a millisecond. This resulted in a penetration of gas from the cylinder into the nozzle sac during the combustion process, the hot gases impinging on the bottom of the needle and eventually impairing its performance. In Fig. 13 this condition can be seen at (a) on the left, where the pressure in the nozzle sac has fallen down to a low level at a point 16½ degrees after spill closure and there is a period of one degree during which the needle is still off its seat and gas can blow past it into the sac. The rig tests now ensure that the seating of the needle occurs before the sac pressure falls, as illustrated at (b) on the right of Fig. 13. The value of this preliminary rig work was confirmed by the performance produced in the 12-cylinder engine at a very early stage in its development running, many hours of "cut and try" tests to optimize injection equipment being saved. Fig. 14 shows

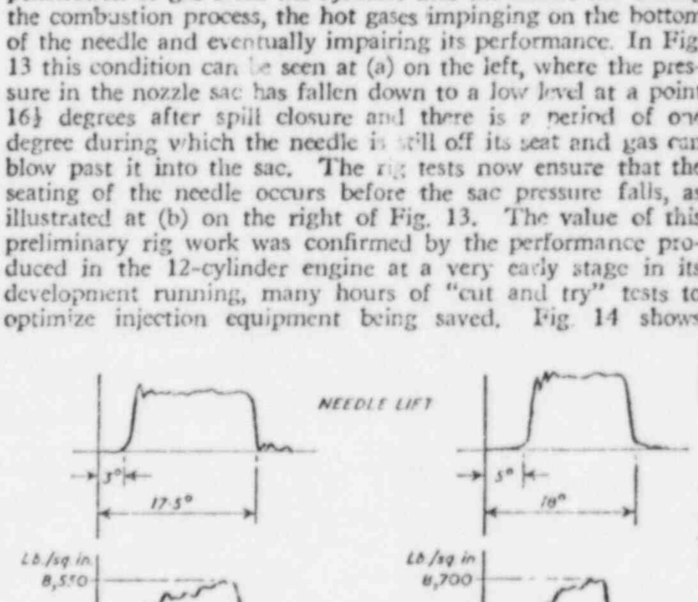


FIG. 13—*Injector needle lift, nozzle gallery pressure and fuel line pressure diagrams*

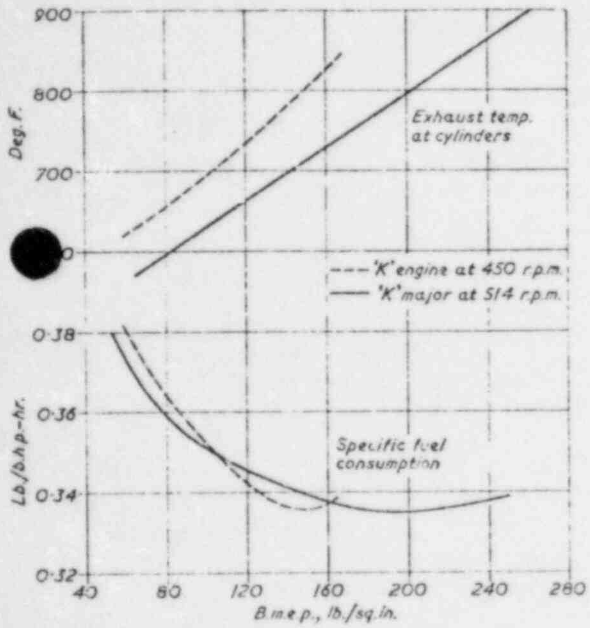


FIG. 14—*K and K Major performance comparison*

the performance of the engine as compared with that of the original K engine, from which it can be seen that the specific fuel consumption of the K Major engine is below 0·34 lb./b.h.p.-hr. over a very wide range of power, i.e., from 140 lb./sq. in. to 250 lb./sq. in., b.m.e.p. The curve also shows that, in spite of the increase in speed, from 450 to 514 r.p.m., and an increase in brake mean effective pressure, from 150 lb./sq. in. to 200 lb./sq. in. (i.e., a power increase of 50 per cent) the same exhaust temperature as in the K engine has been maintained.

c) HEAVY FUEL OPERATION

In the operation of a Diesel engine on heavy fuel, the two items which normally deteriorate most rapidly are the injector nozzles and the exhaust valves, and the frequency of servicing of these two items is of predominating importance. In both cases there is a "threshold" of temperature of the critical parts of the components so that, as ratings increase, the design of the component must be improved to maintain safe operating temperature levels.

Exhaust Valves

Exhaust valve life with residual fuels is usually limited by the formation of deposits on the valve seat, resulting from the incombustible constituents of the fuel and largely from the combination of the sodium and vanadium salts present. As the seat deposits build up, they prevent the valve from making full contact on its seat, thus reducing the degree of heat transfer and eventually allowing tracking across the seat between the gaps in the deposits. The left-hand picture of Fig. 15 shows such a condition for an uncooled valve after 600 hours operation at 180 lb./sq. in., b.m.e.p., on a blended fuel of 300 seconds Redwood 1 viscosity with a three per cent sulphur content, 85 p.p.m. sodium and 100 p.p.m. vanadium. The beginning of erosion across the seat face between the deposits can be clearly seen(3).

Although the chemistry of the formation of these deposits is a most complex study, and is beyond the scope of this paper, field experience and engine tests have shown quite clearly that the presence of sodium and vanadium is of great significance and a practical assessment of the temperature range in which deposits are likely to adhere to the seating face of the valve can be made. Table III gives the melting point of possible deposit constituents which are in the temperature range which may appertain in the seat region of an exhaust valve, as is shown in Fig. 16, where the left-hand valve is of the normal uncooled design corresponding to the left-hand illustration of Fig. 15.

TABLE III

Compound	Melting point (deg. C.)
Nickel vanadate $N_iO.V_2O_5$	900
Sodium sulphate Na_2SO_4	880
Sodium orthovanadate Na_3VO_4	850
Vanadium pentoxide V_2O_5	675
Sodium pyrovanadate $2Na_2O.V_2O_5$	640
Sodium metavanadate $NaVO_3$	630
Sodium vanadyl vanadate (1.1.5) $Na_2O.V_2O_4.5V_2O_5$	625
Sodium vanadyl vanadate (5.1.11) $5Na_2O.V_2O_4.11V_2O_5$	535

From the Diesel engine designer's point of view, it is sufficient to accept that if the valve seat temperature can be kept below about 1,020 deg. F. (550 deg. C.), adhesion of any of these components will not occur to any appreciable extent so that rapid build-up of the deposits will not be possible. The problem is thus quite different to that of the gas turbine engineer, who has to consider the corrosive effect which occurs at higher temperatures. However, the achievement of low valve seat temperatures at high outputs is not easy and calls for careful attention to details of design and patient development engine testing to achieve the desired result. The three-cylinder prototype engine, shown in Fig. 17, has been used for continuous testing on heavy fuel in the research laboratory for the past three years and more

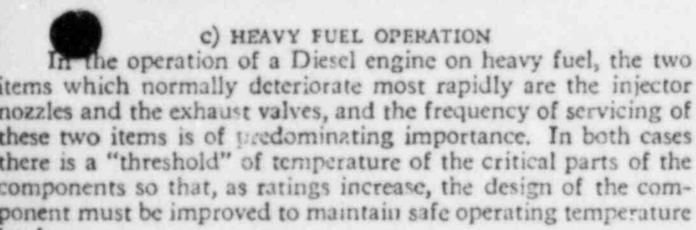


FIG. 15—*Comparison of exhaust valve condition after operation on heavy fuel*

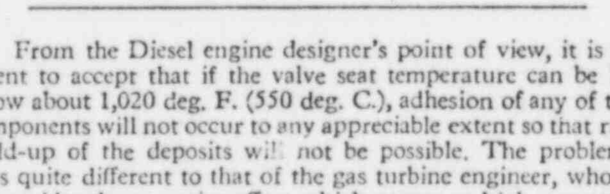


FIG. 16—*Temperature distribution in exhaust valves with uncooled and cooled cages*

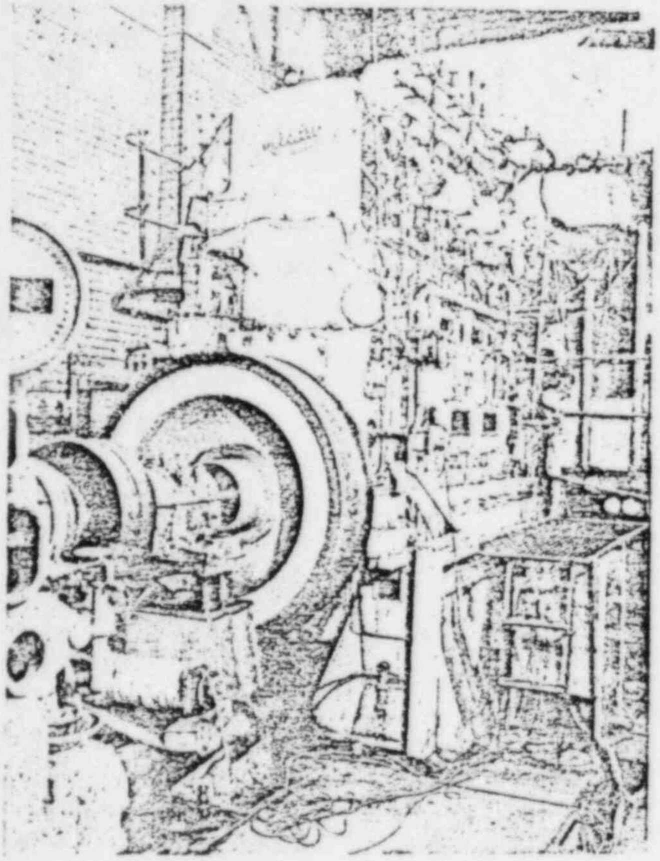


FIG. 17—*Prototype three-cylinder development engine*

than 60 exhaust valve and cage design combinations have been tested, of duration between 300 and 1,300 hours each to determine the effect of different factors in the design. A basic test duration of 500 hours at 200 lb./sq. in., b.m.e.p., loading was chosen and valve seat condition as the main parameter, together with other features such as valve guide wear, was compared with a reference design which was maintained throughout. In many cases the test was prematurely stopped at about the halfway stage where the valve condition was not satisfactory and, in the later stages, valve assemblies were replaced in the engine without re-grinding for second and third runs. Space does not permit a detailed report of the individual tests, but the resulting K Major exhaust valve and cage design will be described in detail to illustrate the factors which were found to be important.

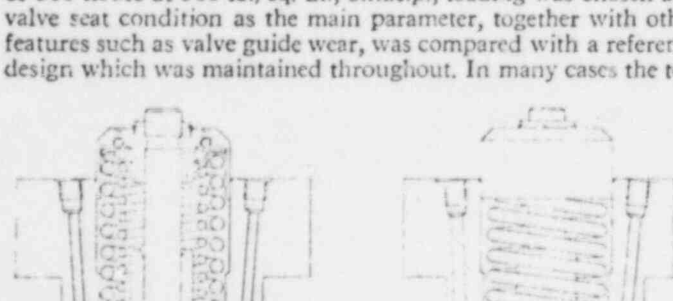


FIG. 18—*Assembly of exhaust valve and water-cooled cage*

Fig. 18 shows the valve and cage and it can be seen that a cooling passage is provided in the cage close to the valve seat. The seat is made of a single piece of Stellite 6, grooved to form the lower part of the cooling passage, and the upper portion is machined in the valve cage which is a three per cent Cr-Mo steel casting, the two being welded together by electron beam welding. The Stellite portion thus provides the facility for simple rebuilding of the seat by oxy-gas deposition of Stellite after a long period of time in service. The valve stem is of increased diameter and the valve of high-conductivity Cr-Ni-Si steel, to assist heat transfer from the head of the valve through the stem, and the valve guide is also surrounded by a water-cooled space, the cooling water passing along a drilled passage from the top of the cage direct to the annulus around the seat, then through a drilling to the space around the guide and via another drilled hole to the outlet at the top of the cage. The heat transfer from the valve to the cooled guide is assisted by the close fit between the valve stem and guide, the previously mentioned development tests having shown that a diametral clearance of 0·012in. resulted in an uncooled guide temperature of 450 deg. F. (232 deg. C.) in its middle position, with rapid deterioration of the lubricant and the formation of hard carbon. Halving the clearance reduced the temperature to 380 deg. F. (193 deg. C.) and, halving it again together with stem lubrication and guide cooling, brought the temperature down to 170 deg. F. (77 deg. C.), i.e., about 10 deg. F. (6 deg. C.) higher than the cooling water temperature, with no deterioration of lubricant and a guide wear rate of less than 0·0005in. in the first thousand hours.

The valve is fitted with a rotator in the top spring carrier which helps mechanically to prevent build-up of seat deposits, but its most important function is to ensure an even temperature distribution around the valve so that there is no local high-temperature region. The flow lubrication of the valve guide takes advantage of this rotation by using the valve itself as a timing device. Two flats are provided on the valve spindle which periodically line up with oil inlet and outlet drillings on the guide as the valve rotates. The linear positioning of the slots only allows the oil to pass while the valve is open and thus the oil space around the valve is only pressurized when the exhaust pulse pressure is present in the valve cage gas passage, the oil acting as a seal against gas penetration up the stem and the exhaust pressure preventing leakage of oil from the guide. It was at first feared that a continuous oil supply to the guide might result in excessive leakage of oil from the bottom of the guide, but this has not proved to be the case and, in fact, the tendency is for the leakage to be upwards as the retardation when the valve meets its seat is greater than the acceleration during opening and the inertia of the oil carries it upwards. This intermittent pressure lubrication of the valve stem makes it possible to use a very small stem/guide bore clearance without any risk of valve sticking, and this helps the heat transfer from the valve to the water-cooled guide. In addition, the danger of stem or guide bore corrosion at low load running conditions is avoided.

Since the stem to guide clearance is important in the heat transfer process, the reduction of guide wear helps to maintain low valve seat temperatures over a long period in service, and many of the development tests were concerned with valve guide material and valve rocker lever geometry to this end. The long guide and the small overhang of the valve head beyond the guide will be noticed in the illustration and were found to be important factors in reducing guide wear, as was the composition of the special "Mechanite" iron which was finally used for the guide material.

Sodium-cooled and water-cooled valves were tested among the many combinations but were found to offer no advantage over the design finally adopted, mainly, it is thought, because of the difficulty, with an internally-cooled valve, of providing cool-

ing passages close enough to the actual seat of the valve. The usual methods of drilling down the centre of the valve stem, although successfully cooling the centre of the head, still leave a fairly high temperature at the seat, and in the case of the internally water-cooled valve, the water connexions to the valve are a difficult problem.

The right-hand valve of Fig. 15 shows the results of this development, the valve having run for 900 hours at 200 lb./sq. in., b.m.e.p., on the same type of fuel as before. The good condition of the seating face shows that no re-grinding is necessary and the valve can operate for a much longer period without attention. The corresponding temperature distribution in the valve head is shown in the right-hand illustration of Fig. 16, and the effect of rating on exhaust valve seat temperature is given in Table I.

The valve development tests also included investigations into the effect of fuel treatment on exhaust valve life and while one fuel additive showed promise, in that the nature of the valve seat deposits was altered, it was not effective enough to justify its adoption. The principle of this additive was that other chemicals were added to the fuel so that the compounds, which were formed during combustion, would have higher melting points than those listed in Table III. It seems likely that, with further development work by the additive manufacturers, there may be some advantage to be gained in the future from this type of additive. Water washing of the fuel, to remove the sodium content, was found to be quite effective and the sodium could be reduced from 90 p.p.m. to about half of this value without difficulty, engine tests showing that the washing had quite an appreciable beneficial effect on the exhaust valve seat condition. As can be quickly calculated from Table III the critical sodium/vanadium ratios in the important temperature zone range from 1:0·74 to 1:13·3, the lower melting point compounds being associated with the latter end of the range, so that a reduction in sodium content may tend to produce the compounds with the lower melting points and, with particular fuel compositions, have an undesirable effect. Thus, with the wide variation in constituents in fuel from different parts of the world, it is difficult to make a clear case for water washing of the fuel.

Injectors

Fuel injection nozzles, when operated at high temperatures, tend to form carbon around the holes in the nozzles, known as "trumpeting", which may interfere with the injection spray pattern and reduce combustion efficiency, thus aggravating the temperature problem. For a time, the carbon formation develops until the "trumpets" become detached from the nozzle and a periodic rise and fall of exhaust temperatures can often be seen as this occurs. The general trend of temperature, however, is upwards and conditions eventually level out at the top end of the exhaust temperature cyclic range. In more extreme cases of high temperature, the needle seat may lose its hardness and the needle rapidly hammers its way into the seat. The temperature at the nozzle tip can be measured by thermocouple and a temperature of about 356 deg. F. (180 deg. C) is considered to be the limit

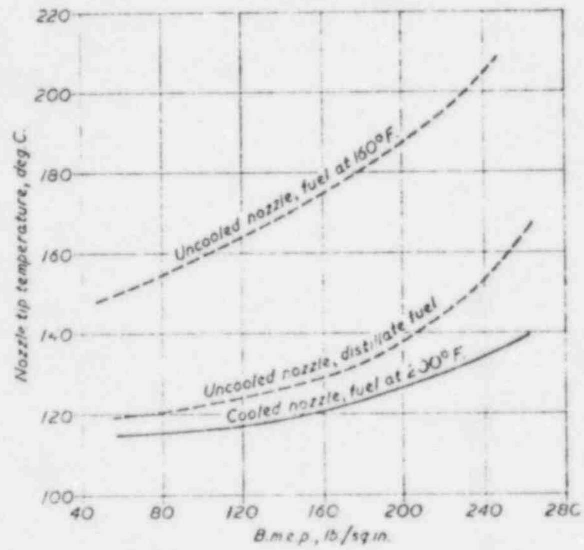


FIG. 19—*Injector nozzle tip temperatures with cooled and uncooled injectors*

for satisfactory operation. In Fig. 19, the middle curve shows the variation of nozzle tip temperature with load for the K Major engine, using an uncooled nozzle and distillate fuel, where the fuel itself has a considerable cooling effect, and there would be no difficulty in operating an uncooled nozzle on this type of fuel up to a load of about 280 lb./sq. in., b.m.e.p. In the upper curve, however, blended fuel of 300 seconds Redwood 1 viscosity was used, with a fuel temperature of 160 deg. F. (71 deg. C.) and it can be seen that the loss in cooling effect from the fuel has limited the acceptable load level to about 180 lb./sq. in., b.m.e.p., and with heavier, and hence hotter, fuels the load limit would be much lower. A water-cooled nozzle is therefore necessary for high ratings on heavy fuel, and the lower curve shows the tip temperature for a cooled nozzle using 1,000 seconds fuel at 200 deg. F. (93 deg. C.) with cooling water at 150 deg. F. (66 deg. C.). It is important that the nozzle should not be over-cooled as cold corrosion can occur at temperatures below 230 deg. F. (110 deg. C.), but this is controlled by the water-circulation system which is separate from that of the engine-cooling water. Fig. 20 shows the cooling system which is a closed circuit serving the injectors and water-cooled seat exhaust valve cages with a thermostatically controlled bypass around the heat exchanger and minimum volume in the system to ensure that correct operating temperatures are reached quickly.

d) LUBRICATING OIL CONSUMPTION

The consumption of lubricating oil in a Diesel engine is an important factor in maintenance costs and it is not always realized that, at a reasonable consumption rate of one per cent of

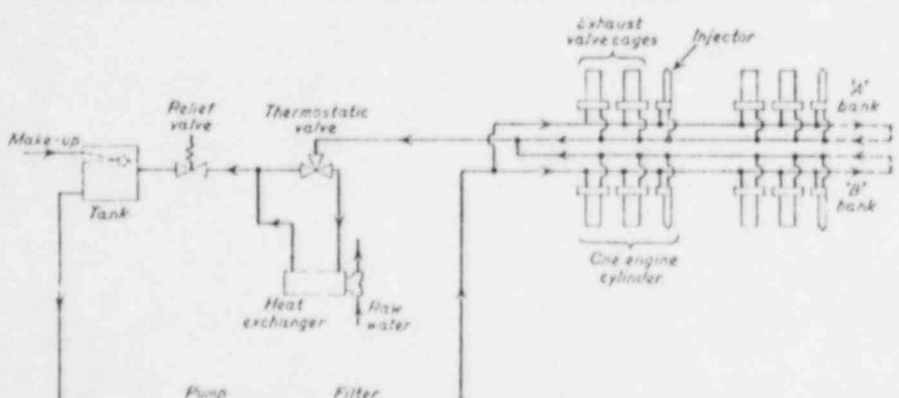


FIG. 20—*Arrangement of injector and valve cage cooling system*

fuel consumption, a 4,000 h.p. engine will burn a quantity of lubricating oil equivalent to its sump capacity in a period of the order of 300 hours. Emphasis is often laid on long periods between oil changes which are extended by an engine with a high oil consumption, whereas the relative importance of oil consumption to oil change period is around 50 to 1. The cost of modern high-duty detergent oils is quite appreciable, so that an oil consumption of one per cent of the fuel consumption represents something like ten per cent of the fuel bill. Not all of this could be saved, of course, but a reduction of 50 per cent in lubricating oil consumption is equivalent to a five per cent saving on the fuel bill, and would be well worth having from the point of view of running costs.

Piston Ring Design

To carry out lubricating oil consumption tests in the relatively short running periods of 500 hours or so in the research laboratory, it was necessary to develop an accurate method of measuring top-up rate and a system was devised, and has proved very successful, whereby consumption can be measured consistently over successive two hour periods and plotted consecutively. The running-in period and the levelling-out to a steady consumption can now be followed and it has been possible to obtain steady state results after a total test period of only 300 hours, which allows much more latitude for testing variations on a ring pack than was previously the case. There is a large number of detail points to be considered such as liner finish, roundness, drainage in the piston, etc., but the basic concept which has been established is to provide a parallel-faced chrome-plated top compression ring, three taper-faced plain compression rings, a relatively mild scraper ring below the gudgeon pin, and a more severe scraper ring above the pin. This ensures that adequate lubrication is available around the body of the piston but that the minimum of oil is allowed to pass up into the combustion space. The consistency of oil consumption measurement has enabled some interesting facts to emerge, and Fig. 21 illustrates one of these—the effect of the wall pressure of the scraper ring above the pin. The left-hand curve is from the three-cylinder, 15-in. bore prototype engine, and the right-hand curve from a

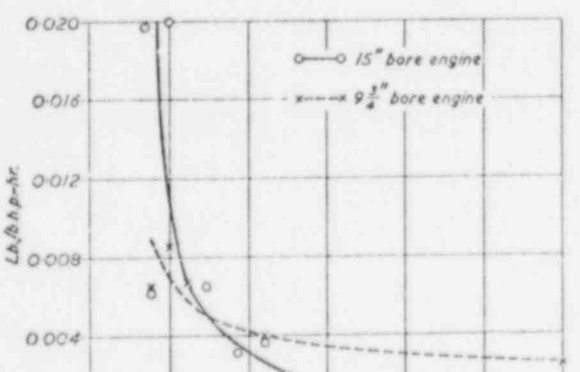


FIG. 21—*Variation of lubricating oil consumption with scraper ring wall pressure*

completely different high-speed engine of $9\frac{3}{4}$-in. bore, the points marked being the stable lubricating oil consumption achieved after running periods of about 300 hours in each case. Both curves show the same trend of reducing oil consumption with increased ring wall pressure and the tendency for the curves to level out at higher values of wall pressure. The value of wall pressure necessary to achieve a satisfactory consumption can be seen to be much higher for the smaller high-speed engine than for the K Major engine, and in the case of the smaller engine it was necessary to use a spring-loaded conformable scraper ring to achieve the desired consumption. A conventional type of slotted scraper ring was adequate to provide the 50 lb./sq. in. pressure needed for the K Major engine, the resulting consumption, of less than 0·002 lb./b.h.p.-hr., being confirmed in the 12-cylinder engine during development running.

Piston Ring Quality

Consistent oil consumption and low wear rates are largely dependent on the quality of the piston rings, from the point of view of metallurgical structure as well as accuracy of manufacture. Accuracy and good finish in manufacture can be assured by conventional inspection methods, and such methods can easily be extended to give some indication of material quality, such as by measuring the permanent set of the ring at a given load value above that required to close the gap. A simple sample checking method on metallurgical structure was devised in which a small piece of ring is clamped with its working face subjected to a given load and resting on the surface of a ring of liner iron. The ring is then rotated at a standard speed for a fixed time without lubrication and the weight loss of the piece of ring measured. Weight loss is used as a measure of the relative wear resistance of the material and, although "rough and ready", is found to co-relate well with the differences in microstructure of the ring material. Some typical results are given in Table IV, and illustrated in Fig. 22, and show that with the same Brinell hardness, increasing amounts of free ferrite give progressively worse results and these are not improved by increase in phosphorus content within the amounts to comply with mechanical strength requirements[4].

TABLE IV

Sample No.	Structure	Hardness, HB	Weight loss, gm.
A	Greatly undercooled graphite, considerable free ferrite (centricast) 3·15 per cent. T.C., 0·83 per cent P.	210	0·404
B	Some undercooled graphite, a little free ferrite (centricast) 3·20 per cent T.C., 0·40 per cent P.	210	0·185
C	Random uniform medium flake graphite, fully pearlitic (sandcast) 3·45 per cent T.C., 0·55 per cent P.	210	0·017

e) MAINTENANCE

Engine running times between overhauls are dependent upon the load, duty and running conditions, and the preceding sections have indicated the attention that has been paid to the components which operate under the most arduous conditions. By reducing the critical temperatures of injector nozzles and exhaust valve seats, so that when operating on heavy fuels these temperatures are below the "threshold" values at which deterioration becomes rapid, it has been the aim to achieve periods of 2,000 to 3,000 hours before servicing of injectors or exhaust valves is necessary. Experience on the prototype engine has indicated that this ambition is by no means unreasonable but, of course, true confirmation of success will only come from the accumulation of service experience. Maintenance of other components would not be different from that established over many years, e.g., piston removal annually, complete overhaul every two years, the periods generally being dictated to suit the convenience of the operator rather than by the demands of the engine.

f) SPACE AND WEIGHT

In achieving high engine ratings reliably, the weight per horsepower, and space per horsepower, are naturally reduced and the emphasis on reliability for commercial marine work necessitates a different approach from that which would be appropriate for naval work where light-weight constructions become necessary but short life may be permitted. Sight should not be lost of the importance of low fuel consumption in the considera-

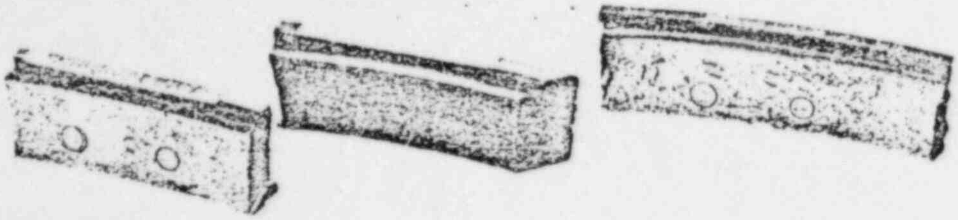


FIG. 22—*Comparison of piston rings after wear rig tests*

tion of weight. A ship refuelling every 3,000 miles, for example, at an average speed of 15 knots, and having engines weighing 34 lb./b.h.p., and a specific fuel consumption of 0·34 lb./b.h.p.-hr., would re-bunker an amount of fuel equivalent to twice the weight of the engines. Thus a five per cent reduction in fuel consumption would be equivalent to a ten per cent reduction in engine weight in addition to the saving in fuel cost.

In the design of the K Major engine, cast iron has been used as the main structural material and, in the authors' experience, has many advantages over fabricated steel designs. Few fabricated structures are able to avoid fillet welds in load-carrying regions and the fatigue strength of such a weld is as low as $\doteq$ 1·2 tons/sq. in. Even butt welds must allow for discontinuity so that their fatigue strength is only $\doteq$ 3·8 tons/sq. in., these values being for good quality welds, the strength of an imperfect weld being, of course, very low indeed. A good quality cast iron has a fatigue strength of over 5 tons/sq. in., and as well as freedom from the notch sensitivity, which so drastically reduces the fatigue strength of a steel structure, cast iron has good internal damping properties and also possesses the useful property of a diminishing E value with increased stress so that stress concentrations are considerably reduced and the material tends to relieve itself of any excessive stresses.

In keeping with the philosophy of designing for maximum

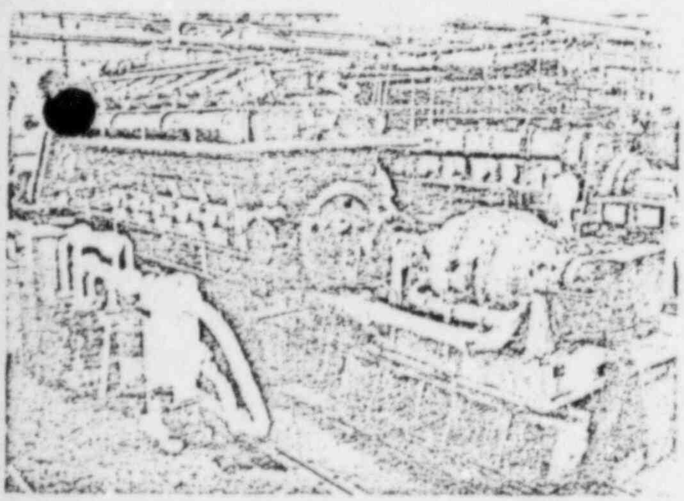


FIG. 23—*Prototype KV Major 12-cylinder engine*

TABLE V

Cylinder bore	15in.
Stroke	18in.
Compression ratio	11.35:1
Maximum r.p.m.	525
Minimum working r.p.m.	125
Continuous rated b.m.e.p.	200 lb./sq. in.
Maximum continuous b.h.p./cylinder	420 b.h.p. (426 cv)
Lubricating oil inlet temperature	150 deg. F. (65 deg. C.)
Lubricating oil outlet temperature	165 deg. F. (74 deg. C.)
Lubricating oil drain tank capacity	650 gal. (2,960 litres)
Fresh and salt water flow rates	5.5 gal./b.h.p.-hr at 50ft. head (25 litres/cv-hr.)
Engine cooling water inlet temperature	155 deg. F. (68 deg. C.)
Engine cooling water outlet temperature	170 deg. F. (77 deg. C.)
Exhaust temperature after turbocharger	800 deg. F. (427 deg. C.)
Starting air pressure	400 lb./sq. in.
Specific fuel consumption	0·335 lb./b.h.p.-hr. (l.c.v. of 18,400 B.t.u/lb.)
Thermal efficiency	42 per cent

TABLE VI—POWER RANGE

	B.m.e.p. lb./sq. in.	No. of cylinders 6	8	9	12	16	18
B.h.p. output at 250 r.p.m.	200	1,200	1,600	1,800	2,400	3,200	3,600
	250	1,500	2,000	2,250	3,000	4,000	4,500
B.h.p. output at 350 r.p.m.	200	1,680	2,240	2,520	3,360	4,480	5,040
	250	2,100	2,800	3,150	4,200	5,600	6,300
B.h.p. output at 450 r.p.m.	200	2,160	2,880	3,240	4,320	5,760	6,480
	250	2,700	3,600	4,050	5,400	7,200	8,100
B.h.p. output at 525 r.p.m.	200	2,520	3,360	3,780	5,040	6,720	7,560
	250	3,150	4,200	4,720	6,300	8,400	9,450
Overall length of engine		20ft. 0in.	24ft. 0in.	26ft. 0in.	24ft. 3in.	29ft. 10in.	32ft. 7in.
Overall width of engine		7ft. 6in.	7ft. 6in.	8ft. 2in.	11ft. 7in.	11ft. 11in.	11ft. 11in.
Overall height of engine		11ft. 6in.	11ft. 6in.	11ft. 6in.	11ft. 11in.	11ft. 11in.	11ft. 11in.
Height above crankshaft C.L.		8ft. 9in.	8ft. 9in.	8ft. 9in.	8ft. 2in.	8ft 2in.	8ft. 2in.
Engine weight (dry) tons		38	44	48	65	85	95

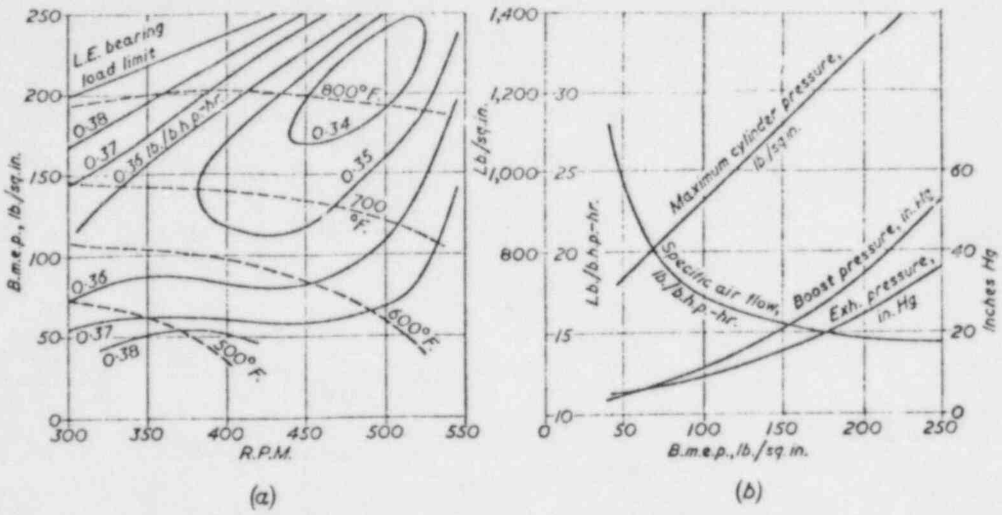


FIG. 24—*KV Major engine performance characteristics*

reliability and easy maintenance, the bedplate type of construction has been retained for the K Major engine and a useful facility has been added by the inclusion of a machined strip on the top surface of the bed so that alignment of the engine can be quickly and accurately checked, and crankshaft deflexion measurements more easily interpreted.

Fig. 23 shows the 12-cylinder KV Major on the test bed, from which the general construction and appearance of the engine can be seen, and Tables V and VI give the specification for the range of engines available at the current commercial rating of 200 lb./sq. in., b.m.e.p., and the future rating of 250 lb./sq. in., b.m.e.p., under normal temperature and pressure conditions with sea water up to 75 deg. F. (24 deg. C.) to the charge air cooler.

CONCLUSIONS

The design and development of a highly-rated medium-speed Diesel engine, to operate economically and reliably on heavy fuels, has been described and it has been shown that, for the K Major engine, the critical parts of the engine, which determine its reliability, have adequate safety margins for its current rating of 200 lb./sq. in., b.m.e.p., and have potential for a substantial increase in rating, to 250 lb./sq. in., b.m.e.p., in the future. The performance of the 12-cylinder prototype engine, beyond the current commercial rating, is illustrated in Fig. 24, curve (a) showing the performance at variable speed and curve (b) the performance at a constant speed of 514 r.p.m., and these, in conjunction with Fig. 14, show clearly the enormous strides which are being made in the Diesel engine industry towards higher specific outputs without exceeding the temperature and pressure levels which past experience has shown to give reliable and trouble-free operation. There is little doubt that in the marine propulsion field there is considerable interest in the use of medium-speed Diesel engines for higher powers than have hitherto been possible, and that within the next few years engines of this type will be available to cover almost the whole range of power demands of British shipping.

ACKNOWLEDGMENTS

The authors wish to express their thanks to the Board of Mirrlees National Limited for permission to publish the information contained in this paper and acknowledge the assistance given by Bryce Berger Ltd. and by their colleagues in the compilation of data.

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Discussion

MR. R. COOK, M.Sc. (Member of Council) said that, at the present time, the manufacturers of medium-speed engines in Great Britain were making very strenuous efforts to extend their share of the marine market in propulsion machinery. The paper was, therefore, timely and few who had read it would have failed to be impressed by the manner in which the authors and their colleagues were applying the latest knowledge and research techniques to the solution of the problems which arose when such machinery was developed to operate at high ratings on residual fuels. One could hardly doubt that success would attend their efforts, although he suspected that the large direct-drive Diesel would be about for quite a few years to come.

The histogram shown in Fig. 1 was interesting. It would be noted that by far the largest horsepower on order at the present time was between 9,000 and 21,000 s.h.p. per ship. With the machinery described in the paper this implied the use of some 18 to 36 cylinders of 15 in. diameter, each with two exhaust valves, two inlet valves, together with the injection and starting equipment. He said that he could not help wondering whether the modern seagoing engineer, who was perhaps not quite so amenable to long and arduous hard work as his forebears, would take kindly to the never-ending task of top-overhauling such a formidable number of cylinders.

Another point for thought was the effect which such maintenance requirements would have upon the reduction in engine-room staff now being achieved with direct-drive machinery by the application of an increased degree of automatic control. He hoped that some superintendents would comment on these aspects later in the discussion.

He said that some years ago Dr. Pope had made a very thorough theoretical and experimental investigation for the British Shipbuilding Research Association into the causes of failure of pistons, liners and cylinder heads in marine oil engines, with results which had since been published in the Transactions of another Institution. He was not surprised, therefore, to see the attention which the authors had given to thermal and pressure-induced stresses in the design of the two-piece, oil-cooled piston. Presumably, the piston temperatures shown in Fig. 6 were measured on the actual engines and the rig used to check the thermal stresses calculated from these temperature measurements. If so, he wondered whether good correlation was achieved. It would be interesting to know how the temperature distribution in the stationary-rig piston correlated with that on the engine.

He said that he was interested to observe the use of rolled threads on the high-tensile steel studs used for securing the piston crown. Work by B.S.R.A. on the rolling of threads of large mild-steel bolts such as those used in the dynamically-loaded components of direct-drive Diesels, had shown very striking improvements in fatigue strength. Reference to a paper* appearing in the Transactions of the Institute four years ago would show that form-rolling increased the fatigue strength of large forged bolts made from mild steel some 2½ to 3 times when compared with cut-thread specimens. Rolling of thread roots gave almost as great an increase. The degree of rolling had been found to be not very critical, but his Association was at present investigating more fully the optimum degree for various sizes and pitches. He imagined that with the high-tensile steel material used by the authors, the gain in fatigue strength would not be so great as in the case of mild steel, but it would be interesting if the authors would quote some figures. Form-rolling could be a very cheap method of bolt production, particularly in small sizes.

The means adopted to ensure correct pre-loading of studs was to be commended since there was no doubt that the majority of failures of dynamically-loaded bolts were due to fatigue caused by inadequate tightening. It was not always appreciated that, with a properly-designed bolted connexion, fatigue failure was virtually impossible if the bolt was adequately pre-loaded.

The section in the paper dealing with heavy fuel operation was, of course, of the greatest possible interest, since a solution of the difficult problems involved was essential if the medium-speed engine was to be able to compete with the direct-drive engine, which was so much less fastidious as to its diet. Here again, the authors had given evidence of a careful scientific approach which should go a long way to ensure success. They had commented on the possible use of fuel additives. One could imagine this approach being successful where fuel supplies of constant composition were available, but this was seldom possible in marine practice and the chances of obtaining a cheap additive, which was effective with a wide variety of fuels, seemed somewhat remote. The authors' approach, by tackling the design, was certainly the right one. Their remarks on the drawbacks of water washing were also worth noting.

No reference had been made in this section to sump-oil contamination when using heavy fuels. Presumably this must occur to some degree in this trunk-piston design, and it would be useful if the authors were to give some information on the procedure involved in maintaining the lubricating oil in a suitable condition.

Dr. Pope had, over the years, made many investigations into the properties of cast iron. Few were, therefore, more familiar with its strength and frailties. Sir Harry Ricardo had once referred to cast iron as "the material which served our forefathers so well for lamp posts and kitchen ranges", but he was sure that Sir Harry would be the first to acknowledge the advantages which the authors had enumerated. Its use as the main structural material in the K Major engine had much to commend it, since weight was rarely of paramount importance in merchant ships.

On the subject of cast iron, he said that it might be inferred from the data given in Table IV that centri-cast piston rings were inferior to sand-cast rings. He felt sure that this would not be the authors' intention. Centri-cast rings had been widely employed with success. He took it that the authors' purpose had been simply to show that, with this type of material, undercooling and consequent presence of free ferrite was most undesirable.

The paper had touched in an interesting manner on so many aspects of Diesel design that to point to ommissions might seem somewhat churlish. He wished, however, that the authors had found it possible to touch on the subject of turbo-

* Cook, R., and McClimont, W. 1961. "The Influence of Screw Forming Methods on the Fatigue Strength of Large Bolts". Trans. I.Mar.E., Vol. 73, p. 417.

charging. Perhaps they might, at some later date, find it possible to give a paper on their experiences in turbocharging up to the 250 lb./sq.in. in b.m.e.p. which was involved in the third stage of the development of the engine.

COMMANDER E. TYRRELL, R.N. (Member), in a contribution read by Mr. T. P. Everett, referred to the successful introduction, by the authors' company, of an engine which he considered met both industrial and marine needs which so far had only been filled by Britain's foreign competitors. As the authors had so rightly said, reliability was the main requirement of a marine propulsion engine, and nobody who had read this paper could fail to be impressed by the systematic way in which Mirrless National had carried out the research and development work necessary to ensure that this engine would operate satisfactorily at the ratings envisaged.

If the medium-speed geared Diesel engine was to compete with the slow-runing direct-coupled engine, it was essential that it should operate satisfactorily while burning heavy fuel, and it was evident that the authors had taken very considerable trouble and had expended a comparatively large sum of money in trying to ensure that this would be the case. There was, however, one point which he felt should have been mentioned in this respect, and that was the effect of various grades and compositions of lubricating oil on the problems associated with the burning of heavy fuel. There was little doubt that trunk-piston engines called for careful selection of the lubricating oil if satisfactory operation with heavy fuel was to be achieved. New types of lubricating oil and testing for quality and make-up by the addition of detergents, anti-oxidants, and alkalis could have a profound effect on the satisfactory operation of trunk-piston engines while burning this type of fuel. The wrong type of lubricant, or one which had been allowed to deteriorate unduly, could give rise to ring-sticking and crankshaft corrosion. The operating temperature of the oil was also important if these defects were to be avoided. Perhaps the authors would like to remark on the type of lubricating oil and its optimum operating temperature for this type of engine burning heavy fuel.

He thought that the title of this paper was slightly misleading. He could not agree that the Mirrlees K Major engine should be regarded as highly rated when operating at the conditions given in the paper. In his opinion there was considerable scope for further advances in b.m.e.p. These were important as they should give worthwhile reductions in the cost per horsepower. Introduction of a reliable engine operating at these higher brake mean effective pressures would do much to increase the competitive power of this type of engine against its competitors abroad and other types of prime mover. In this technically-competitive world, the main object of any Diesel engine manufacturer must be consistently to uprate his engines in order to give better value for money. He must at the same time retain reliability.

Many of those present would be aware that the Ministry of Technology had recently placed a contract with the Yarrow-Admiralty Research Department to investigate the use of medium-speed geared Diesel engines as propulsion units for ocean-going merchant ships. This survey was now almost complete. He thought that it was true to say that the results of this survey would give encouragement to those manufacturers of medium-speed Diesel engines who thought that the medium-speed Diesel had a future and could compete in many ships with the slow-running direct-coupled engine. The report showed that every type of ship and trade must be treated on its merits, but that a shipowner who failed to carry out a detailed economic survey into the possible use of medium-speed Diesel engines, as an alternative to the slow-running direct-coupled engine, did so at his peril.

MR. S. H. HENSHALL, B.Sc. (Member) said that, as an engine builder of medium-speed engines, he found that he was on the side of Dr. Pope in a lot of the things he had said. The paper, however, had been very stimulating and he would like to ask several questions about it.

With regard to Table I, the specific air flow for the 250 lb./sq.in. b.m.e.p. showed a drop compared with lower b.m.e.p., and, although this drop was only a small one, he would have thought it desirable to go on increasing the specific air flow.

Turning to the piston design, he said that the features of it were, in many ways, those with which he agreed, but it was mentioned that it was a steel crown and a cast iron junk. The steel crown probably had a higher coefficient of linear expansion as compared with the cast iron. This meant that there were some problems in its connexion, for instance, it must make life a little difficult for the sealing rings between the two portions. Cold clearance between piston crown and liner must be increased.

Figs. 2, 3 and 6 showed a double line round the junk. He wondered what this signified and whether it was some device to overcome the cold clearance problem.

With regard to exhaust valves and operation on heavy fuels, the importance of losing heat via the stems of the valves was certainly to be considered. In the paper the clearance was mentioned as being of importance. The wear rate was obviously kept down by the ingenious device of continuous lubrication, and he said that he would be interested to know what was the maximum allowable clearance of the stem to guide and what sort of life the valve had in this respect. Also, it appeared that the guide could be renewed, although he was not sure whether it was intended to be renewed.

He suggested that there was an argument for not water washing fuel, in that the deposits also occurred on turbocharger blades, and turbochargers could be water washed more easily if the sodium was allowed to remain in the fuel.

On the question of cast iron or steel as the main structural material, he said that steel had its own advantages, and structures could be designed with low stresses where welds occurred. Modern techniques of manufacture and inspection could ensure good quality welds.

He said that surely greater reliability resulted from a design in which the major loads did not have to pass through a joint between the crankcase and bedplate, and easy maintenance was not confined to the bedplate type of construction. The principle of a machined strip used for checking alignment of the engine quickly was also used on engines of fabricated design having underslung crankshafts and light sumps instead of baseplates.

IR. E. R. GRÖSCHEL said that he proposed to limit his observations and comments to the fuel injection side of the paper, departing only for a moment in order to fully endorse the authors' statement under "a) Reliability: *General Considerations*", where they said that the surest method of producing intrinsic reliability was to proceed by a process of logical evolution from one successful design to the next, taking care that the critical parameters proved in the original design were maintained in the new design. This statement deserved thunderous applause from both engine manufacturers and engine users, particularly marine engine users.

The use of test rigs for fuel-injection equipment development was, of course, fully appreciated and valuable data regarding performance and life of the equipment might be gained. However, care should be taken, when applying results obtained from injection-equipment test rigs to engine conditions, unless injection into a pressurized medium was strictly simulated. He wondered if this was done in the case where nozzle gallery pressure was found to be lower than the prevailing gas pressure with a needle still open which permitted gas entrance into the nozzle gallery. The cure adopted, he ventured to guess, was to lighten the reciprocating mass of the injector.

His company, having always designed and manufactured their own fuel-injection equipment for their medium-speed Diesel engines, had always been protagonists of the low inertia injector, i.e. having needle springs acting directly upon the needle without the intermediary of a push rod. Of course, the spring was thus placed into a somewhat uncomfortable position (heat and space), but unorthodox spring wire sections helped with the space and chromium/silicon wire with the temperature.

The "hydrodynamic condition" referred to in the paper, he thought, was a spill wave receding too fast, which the normal oscillating system of the injector was incapable of following. A stiffer spring might help in border-line cases. A trick, imparted to him some time ago by Mr. J. F. Alcock (and gratefully acknowledged) was to watch for gas bubbles in the injector leak-off connexion; should gas pass into the nozzle, it would generally pass through the cylindrical lapped part of the needle-nozzle bore causing lacquering of the lap fit and would finally appear in the leak-off pipe, where a plastic tube would facilitate observation. He said that, acting upon this recommendation, his company sometimes used development engineers as "bubble watchers".

He thought the authors should be congratulated on having such confidence in the precision of cathode ray oscillogram interpretation. He had tried something similar and the result had varied between ½ degree and 2 degrees cam angle, depending upon the thickness of the pencil point used and the condition of the interpreter.

He asked if the line pressures in Figs. 13(a) and 13(b) referred to full load conditions. If they did, they were remarkably low. Also, the difference between maximum line pressure and maximum gallery pressure was rather large, being 1,650 lb./sq.in. in Fig. 13(a) and 1,800 lb./sq.in. in Fig. 13(b). He wondered if the edge filter (which, after all, was the major throttling component between line and gallery) could be responsible for this pressure discrepancy. Wall friction could hardly account for it with the customary maximum flow velocities of about 90 m./s.

Dealing with the chapter on heavy fuel operation, he said that he was intrigued by the statement that the lack of cooling ability of preheated fuel should be responsible for high nozzle tip temperatures. The fuel temperature given was 160 deg. F. (71 deg. C.) for a blended fuel of 300 sec. Redwood 1 viscosity; gas oil, not preheated, would reach the nozzle, after passing through the compressive cycle of the fuel pump, not very much cooler. He ventured to think that the hotter nozzle tip of the heavy fuel operated engine was more the result of the slower burning of the heavy fuel, with a resulting larger heat rejection to, and heat absorption from, the nozzle.

He said that he understood that the mechanism of trumpet formation was a function of the lighter fractions of the blended heavy fuels boiling off in the nozzle sac and squeezing the heavier fractions out of the nozzle holes where they carbonized. Carbonization temperatures were much higher than the measured nozzle tip temperature of 392 deg. F. (200 deg. C.). He said that he would be very grateful if the authors could provide information about the precise location of the thermocouple on the nozzle tip. His company had measured nozzle seat temperatures (the thermocouple being located within a millimetre from the nozzle seat) and temperatures obtained on one engine type, depending on engine rating, cylinder head design and fuel, had reached 464 deg. F. (240 deg. C.). This surely indicated nozzle tip temperatures of a far higher order. These nozzles had been uncooled and made from heat-resisting nitriding steel which maintained the seat hardness at elevated temperatures.

He said that he envied the authors the low maximum cylinder pressure of only 1,350 lb./sq.in. at 220 lb./sq.in. b.m.e.p. This should enable them to get away with an injector release pressure of only 2,500 lb./sq.in., resulting in a closing pressure of 1,720 lb./sq.in., thus still having a comfortable margin available above the maximum gas pressure.

In conclusion, he said that the paper would always have a place of honour in his hydraulics department, having already sent several slide rules into a semi-heated condition.

Mr. J. F. Alcock, O.B.E., B.A., said that the piston crown was described as high-tensile steel, which term covered a lot of compositions. It would be valuable to have either the thermal conductivity or the composition.

Fig. 6 showed a wet-side temperature of over 464 deg. F. (240 deg. C.). He said that it was a rough general rule that one was apt to get coking on the surface if one went over 392 deg. F. (200 deg. C.). He wondered if this had been observed by the authors. He also asked for the velocity of the oil.

Turning to valves in cages, he said that one considerable advantage of the cage was that it very much reduced the flux from the valve seats to the cylinder head. Since the cylinder head was a complex casting, a large concentrated heat flow from the valve via the seat was undesirable. In smaller engines, which did not have caged valves, this was a very common cause of cracking between the seats. He cited the paper of Mr. Fujita* as an example of this.

Turning to valves, he said that he had noticed in the paper that while the temperature of the seat had been reduced by the cooled cage, the temperature difference between the centre of the valve head and the seat was increased. This would increase the thermal stresses, and the risk of seat cracking due to thermal stress. He gathered that the idea of sodium cooling had been discarded, but he thought it might be a useful idea, not from the point of view of cooling the valve seat, but of reducing the thermal stress. Of course, what the sodium-cooled valve did was to pass the flux back to the valve guide, but there it could be coped with quite well.

He then referred to crankcase explosions. These were practically non-existent in small engines, but they did occur from time to time in large engines and were extremely nasty things. It would be valuable to have information on this subject and to know, from the point of view of safety, what the difference was between the trunk-piston and the crosshead engines.

Dr. W. P. Mansfield was particularly interested in the authors' lubricating oil consumption tests which were briefly described on page 336 of the paper and to which Fig. 21 referred. In some investigations on this subject on smaller engines the British Internal Combustion Engine Research Institute Ltd. had tried increasing the oil pressure by reducing the area of the bearing surface of the ring, but this had had little effect. However, changes of wall pressure, made by varying the radial thickness of the ring, had a marked effect. He said that it would be interesting to know by what method the authors had varied wall pressure, apart from the change to a spring-loaded ring which was mentioned in the paper.

Mr. A. J. S. Baker (Associate) said that the authors had produced a remarkably full description of what could only be described as an exceptionally well developed engine. Of particular interest to people connected with lubrication research was the systematic work which had gone into piston development. This work had obviously paid off handsomely in the modest temperatures, so clearly denoted in Fig. 6. It would be interesting to see how far the really excellent fuel utilization rates, indicated by the exceptional fuel consumption rates, had contributed to this. For instance, metal temperature comparisons taken at the same time as the variable valve timing tests described in Fig. 10, right-hand side, would perhaps illustrate this point. Fig. 10 itself suggested that an even broader area of minimum specific fuel consumption might be possible with automatically-varied valve timing. He wondered if the authors had considered such a possibility.

Looking at the Fig. 10 data points for a constant b.m.e.p. of 200 lb./sq.in., it appeared that a fairer mean curve would have a pronounced and steepening hog rather than the sag shown by the authors. He asked the authors to justify the mean line they had postulated.

With regard to the important work which had been done to optimize fuel injection characteristics, it was interesting to consider the authors' needle lift/sac pressure relationship conclusions, in the light of the modification which appeared to have been carried out. Apparently the unloading value had been increased between (a) and (b) of Fig. 13. Did the authors attribute this fact to the reduction in incipient secondary lifting indicated in the needle-lift diagram (b)? Presumably the injection rate

* Fujita, H. 1961. "Service Records of Mitsubishi Nagasaki Diesel UE Type Engines and Improvements Made on the Engines". *Trans. I.Mar.E.*, Vol. 73, p. 37.

had also been increased and this had been accommodated by permitting an increase of needle lift. If this were the case, might not needle-lift increases have to be closely controlled in service operation, he asked. Likewise, the fuel-line pressure diagram (b) was presumably taken at the nozzle end of the fuel pipe. He wondered if it had been necessary to tune the fuel pipe length to control the magnitude of the secondary pressure wave so as to eliminate secondary injection.

He thought that the general conclusion to be drawn from the fuel injection work was that engines of comparatively low speed needed the same careful attention as high-speed engines. It would be very interesting to see a comparative investigation in certain large-bore, slow-speed engines, having several nozzles connected to a single fuel-pump element. The results of the careful work done by the authors in this direction were demonstrated in Fig. 14. By extracting data points from the fuel consumption loops, it was interesting to note that the fuelling lines for both the K engine and the K Major at different speeds were virtually identical. The fuelling lines, from around 50 lb./sq.in., b.m.e.p., on the lowest curve, were unusual in their linearity. Perhaps the authors could supply fuel consumptions at very low loads which would give a clearer indication of the likely f.m.e.p. From the data published it was evident that this must be of a very low order and comparable to that obtained with the largest low-speed engines. This point might be worth bringing out since it was fashionable to quote mechanical efficiency for large two-stroke marine engines, and for a given f.m.e.p. this would generally favour the four-stroke engine with its higher b.m.e.p.

He asked the authors to indicate whether the performance curves were obtained with the fuel described at the top of page 333, and if not, he said that he would like them to give details.

He said that the water-cooled, exhaust-valve cage and valve rotators had made a major contribution to operation with low-cost fuels. Presumably some provision had been made to prevent boiling in the small seat-cooling passages in the event of sudden shut-down, as might be expected in main marine engine application.

He thought that the notes on lubricating oil consumption were very relevant, as was the investigation on oil control by the upper scraper ring. He asked the authors to elaborate on this by indicating the degree of control exerted by the other rings in the pack. For instance, could a reduction in radial pressure of the upper scraper be tolerated by increasing the load on the lower one?

The rig described to evaluate piston-ring quality resembled a variety of test rings used for different purposes. Experience with these had indicated considerable scatter of wear results, particularly at high rates of wear. Perhaps the authors could indicate the significance of the weight loss figures they had quoted in Table IV. He wondered whether they had observed a pattern of related ring to bore wear rates for the different material combinations tested. Had any significant differences in piston-ring groove and ring side wear rates been observed when different irons were run in the steel piston crown? The authors had not shown the metallurgy of the piston crown, but other applications of high-tensile steel had suggested that steels containing appreciable nickel contents might produce increased wear rates in the presence of boundary lubrication.

Mr. J. A. Cowderoy, B.Sc. (Member) said that, as the K Major engine had been developed specifically for marine propulsion, he had been surprised that Fig. 24 did not show the performance plotted against speed on a propeller law basis. He would particularly like to see the compression pressure included in such a plot, because he had the impression that many builders of marine Diesel engines overlooked the implications of the propeller law, which related power to speed in a ship, particularly when applied to turbocharged engines. If this law were assumed to be a cube law it meant that, if the engine was developing full power at full speed, it was only required to develop as little as 12½ per cent of that power, even at half speed, and as ships not infrequently proceeded at speeds lower than full, this condition did occur now and then.

The engine referred to in the paper had been developed to run on heavy fuel oil. He thought the authors would agree that the turbocharged four-stroke, trunk-piston engine could be troublesome from the point of view of combustion when operating at low loads on heavy fuel, and from some figures he had seen for other engines, which showed a drop in compression pressure, from 665 lb./sq.in., at full load and speed, to 355 lb./sq.in., at 60 per cent speed and 22 per cent of full load, he strongly suspected that the relatively low compression pressure under those conditions was one of the principal reasons for this. Whilst combustion might be quite satisfactory, under those conditions, when the engine was new and in first-class condition, with the accumulation of wear of not only liners, but injection equipment, he thought that the low compression pressure was certainly a contributory factor. He would be glad to have the authors' comments on this.

On the question of the operational control of turbocharged medium-speed engines in ships, particularly in view of the increase in the number of ships with bridge control of the engines, he was convinced that the fuel injection pump rack position should be governed to some degree by the booster pressure. A few years ago, in a certain cross-Channel ferry, which was propelled by two turbocharged Diesel engines under bridge control, it was found, a very short time after the ship had gone into service, that the engine crankcase oil had become very dirty indeed. The reason for this was soon discovered: the bridge control of the engines had been operated on leaving harbour as if it had been an engine room telegraph, with the result that the engines smoked like chimneys until the turbochargers had time to catch up and provide enough air for clean combustion. Under these conditions the oil soon became filled with fuel soot. Instruction to the master as to the correct rate at which to increase engine power soon cured the trouble. He felt that where turbocharged engines were installed in ships, some form of control over the rate of increase of the delivery of fuel to the engine was essential.

Mr. C. C. J. French asked a question concerning thermal stress. The thermal stress rig shown in Fig. 4 was interesting and provided an ingenious method of investigating a problem which was becoming more and more important as engine ratings were increased. This rig was useful in that it was applicable to asymmetric bodies, as well as to those that were bodies of revolution. In this respect the two-piece piston shown in Fig. 6, appeared to be a body of revolution. Computer programmes were now available for calculating the thermal stress of such components. He wondered whether the authors had tried a check calculation to see whether there was any sort of agreement between the rig and a computer. His own rather limited experience so far, with a computer approach, had been more valuable in showing up limitations in the computer programme than in giving realistic piston thermal stresses, the problems being largely the rather complex shape of pistons.

Turning to the inlet-valve wear, he said that he was glad that Dr. Pope, in his presentation, had elaborated on his wear factor, which Mr. French had found somewhat incomprehensible as it stood in the paper. He agreed that lack of lubricant was the main cause of heavy inlet-valve and seat wear in turbocharged engines. It was most interesting that the authors had found thickening the head of the valve so effective in reducing this wear.

Touching on service experience, he said that two years previously a paper* had been presented, giving details of service experience on an engine of very similar size and rating. He thought that everyone looked forward to the time when the authors would be able to give comparable details of exhaust-valve life, cylinder-liner and cylinder-ring wear on the K Major, when operating on residual fuels. In this connexion, if the authors were proved correct in their aim of up to 3,000 hours between servicing of injectors and exhaust valves, this would be a most valuable step forward.

* Henshall, S. H., and Gallois, J. 1964. "Service Performance of S.E.M.T. Pielstick Engines." *Trans. I.Mar.E.*, Vol. 76, p. 445.

Correspondence

[illegible]MANDER E. R. MAY, D.S.C., R.N. (Member) wrote that it was [illegible]e ten years since the Pielstick PC1 had begun to make its significant contribution to the propulsion of ocean-going ships, and during the whole of this time it had been without any effective medium-speed competitor. The K Major must now be judged by comparison with the Pielstick PC2, with which it would be in direct competition in every field.

Power for power, the British engine was rather larger and heavier than its French competitor. In some applications this would not matter very much. Commander May imagined that the relative first cost of the two engines would be very significant, assuming that they had equal ability to burn heavy fuel. The Pielstick had never been a cheap engine and, in its PC1 form, its exhaust-valve life on heavy fuel did not always prove impressive. Its popularity had stemmed from its introducing high-speed engine standards of accuracy into the marine engine field, with a refreshing freedom from the very heavy maintenance work that marine engineers often experienced on propulsion engines, less well made and indifferently developed.

Over the last few years, the major British medium-speed Diesel firms had caught up the leeway in standards of manufacture, and also had undertaken most impressive programmes of detailed development. It therefore seemed that the K Major would meet international competition successfully, would extend the market gained by the K engine, and join the Pielstick in propelling large merchant ships.

[illegible]mmander May noticed that the authors had made a ra[illegible] misleading reference to short-life engines being permissible in naval work. This had never been so (except in motor torpedo boats). Submarine engines were designed and produced by the Admiralty between the two wars in an attempt to produce better—not lighter—engines than those available from industry at the time. After the last war, the Admiralty worked hard to persuade industry to adopt modern standards in development and manufacture of long-life engines up to 9¾-in. bore, but success was only achieved gradually and at substantial public expense.

In Germany, before and after the war, and in France at the present day, engines designed partly for naval purposes had met with widespread commercial success. This had come about through recognition that naval and commercial requirements could be designed into the same engine with advantage to all concerned.

Possibly the most remarkable feature of the K Major was that it had achieved so much while retaining cast iron for frame and bedplate. Rigidity was essential to maintain bearing oil film geometry within acceptable limits and cast iron was about twice as flexible as steel. A cast iron frame must have heavier scantlings than a steel frame, the cylinder centres must therefore be further apart, and bending moments increased in consequence. On the other hand, cast iron was cheaper than steel, and development of modern cast irons had done much to make this [illegible]erial more attractive. Fairbanks Morse had used cast iron exte[illegible]ly in their new large opposed-piston, medium-speed engine. Other manufacturers were, the writer believed, using steel for comparable engines and had also chosen the two-stroke, valve-in-head arrangement.

Soon, at least four of these valve-in-head two-stroke engines (one of them British) would be competing with the K Major and the Pielstick in the rapidly expanding world market for large medium-speed engines. It was obvious from this paper that Mirrlees had planned to secure their share of this market.

It would be interesting to know the authors' view on transmission suitable for employing, say, two K Majors to drive a single propeller shaft, and whether their company proposed to offer complete propulsion units—engines and reduction gear.

MR. G. H. HUGHES (Member) commented, in a written contribution, that the increase in power output should in no way alarm prospective users, because even the ultimate aim of 528 b.h.p./cylinder, with 250 lb./sq.in., b.m.e.p., and 1,400 lb./sq. in., peak pressure, represented only 2.98 b.h.p./sq.in. of piston crown—almost identical to the power per square inch on the crown of the Maybach engine with pistons of similar construction.

It would be interesting to know the cooling oil flow rate, (he suggested approximately 1½ [illegible] gal./b.h.p.-hr.), since crown and ring life depended on adequate cooling and, in this respect, the oil feed through the connecting rod might prove to be the limiting factor. Given adequate cooling, it was known that this form of piston would stand greater power per square inch of crown area, as shown in the two-stroke cycle Ruston and Hornsby A.O. engine, when published figures showed over 5 b.h.p./sq. in.

His company's experience of materials for such piston crowns indicated that thermal fatigue tended to become the limiting factor and this depended principally on coefficient of expansion and thermal conductivity. Had the authors considered one of the high-nickel alloys to minimize the effect of high operating temperatures, or the high-conductivity copper chromium alloys?

A further aid to cooling was increased valve overlap. Had the effect of this been explored with respect to piston crown and piston ring temperatures?

The scraper ring arrangement permitted adequate lubrication of the skirt or crosshead length of the piston, but when oil control became a problem after extended service, there might be a temptation to fit a highly-loaded ring in the skirt groove, with possible risk of seizure. To avoid such possibilities, had the authors considered omitting the skirt-ring altogether and adjusting the upper scraper ring accordingly?

It was noted that three taper-faced rings were fitted below a parallel-faced, chrome-plated ring in the top groove. There might be a tendency to blow-by during the initial running of the engine with this arrangement. Had blow-by readings been taken during test work and had any indications been noted?

An important factor in piston-ring material was compatibility with cylinder liners. Not all materials were suitable in this respect, but might be metallurgically sound and, therefore, of good quality.

It was not surprising, therefore, that a random flake graphite iron had given satisfaction in this size of engine.

With regard to cylinder liner material, was this also random flake graphite? How was the bore machined, and what type of surface was produced?

With regard to the outside diameter, was the liner free from water side attack and what precautions might be taken to deal with this possibility at the higher ratings?

On the question of heat dissipation, was it known what proportion of heat was transferred through the piston crown to the cooling oil and through the piston rings to the cooling water?

MR. J. H. MILTON (Member) wrote that it was stated, on page 32[illegible], that to produce a reliable machine one had to pro-

ceed from one successful design to the next, taking care that critical parameters proved in the original design were maintained in the next.

With regard to the critical parameters shown in Table I, it was rather surprising to see that gudgeon pin, or small-end bearings, were not mentioned, as these bearings could be troublesome and also, on occasions, connecting rods had split lengthwise through concentrated eye loading.

Perhaps the authors would care to comment on this subject, and give details of the design of their small-end bearing with particular reference to the bush—whether it was floating or not —and its material.

With regard to the piston design, as shown in Fig. 3, it would be interesting to have the authors' views on the importance of the distance from the crown to the top piston ring, and also further enlightenment on their statements that: a) heat resisting "helicoil" inserts were used to carry the studs and that these acted as a "heat barrier" for these studs; b) that disc springs were fitted under the castle nuts on these studs to increase the resilience of the assembly. Did this mean that they had accepted the fact that movement must take place between the piston crown and the body, and if so, did fretting take place with ensuing leakage of oil across the jointing face?

With further reference to Table I, it was noted that the maximum permissible bearing loads for the main bearings and bottom ends were given as 2,500 and 5,000 lb./sq. in. respectively. Some enlightenment as to how these limitations were arrived at would be of interest

On page 336, under "Space and Weight", the authors made a good case for the cast iron engine, stating that few fabricated structures were able to avoid fillet welds in load-carrying regions, and that the fatigue strength of such welds might be as low as plus or minus 1·2 tons/sq. in., and that even butt welds had only a fatigue strength of plus or minus 3·8 tons/sq. in., compared with 5 tons/sq. in. for a good quality cast iron. If these figures were correct, it was difficult to understand why, apart from the saving in weight, so many other engine builders had adopted fabricated designs, especially as also, in the event of damage resulting from the failure of a bottom-end bolt, a cast iron engine did not suffer distortion and could usually be "patch" repaired, whereas the fabricated structure was usually distorted and had to be renewed.

It was noted that oil was used for piston cooling and lubrication and in this connexion it would be interesting to know if the authors had any relative figures on lubricating oil capacity (e.g. gal./h.p., in circuit) for the engines forming the subject of this paper, as compared with slow-speed, direct-drive Diesels.

Furthermore, in the case of direct-drive, slow-speed engines burning heavy oil, it was found essential, on account of crankcase corrosion, to isolate the cylinder bottoms from the crankcase—what precautions, beyond using an inhibited lubricant, were being taken to prevent such corrosion taking place in the engines produced by the authors' company.

In conclusion, he would be grateful if the authors could briefly state why, in comparison with the builders of large, slow-speed, direct-coupled engines, they had chosen to develop the four-stroke cycle engine instead of the two-stroke cycle engine.

COMMANDER E. B. GOOD, O.B.E., R.N. (Member) wrote that, when a new engine design was introduced, it was natural to compare its rating with those of competitors. A true comparison of ratings should take into account many design features, but an indication of the mechanical and thermal loading problems which the manufacturers had to overcome could be obtained from the output per cubic inch of swept volume and the output per square inch of piston area. These factors had been plotted against cylinder bore, for a number of modern turbocharged engine designs, in Figs. 25 and 26.

The factors for the K Major engine had been plotted at each of the development stages referred to in the paper and it could be seen that these ratings lay neither too adventurously

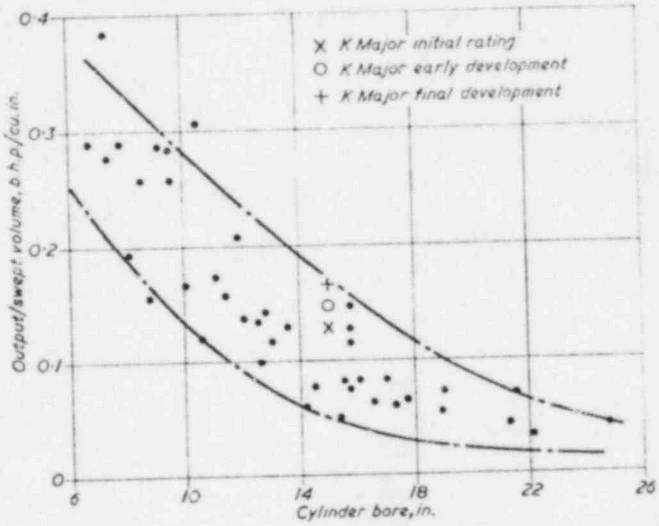

FIG. 25—*Relation between output/swept volume and cylinder bore*

above, nor too much below, the lines marking the upper limit of current design.

Reference had been made to the use of small centrifuges mounted at the engine for the bypass purification of the lubricating oil. A marine installation of more than 3,000 b.h.p. would normally justify the use of a motor-driven centrifuge for the continuous purification of lubricating oil. Such a system also enabled the whole of the lubricating oil charge to be purified in harbour at the end of each trip, a practice adopted by many owners. Commander Good asked the authors whether they considered that the engine-mounted centrifuges would avoid the necessity for a separate motor-driven unit, particularly in a heavy fuel burning installation, and, if so, could they give an indication of the time after which cleaning of the units would be required. It was assumed that provision was made to cut off the flow to individual centrifuges, to permit them to be cleaned while the engine was running.

The attention which had been paid to the design of the fuel injectors and exhaust valves was very welcome. The maintenance of these items probably represented the largest work load for the ship's engineers. It was considered that a period of 5,000 hours between overhauls would not be an unreasonable aim for the exhaust valves.

Shipowners were becoming increasingly concerned about the noise levels in engine rooms and this was reflected in the

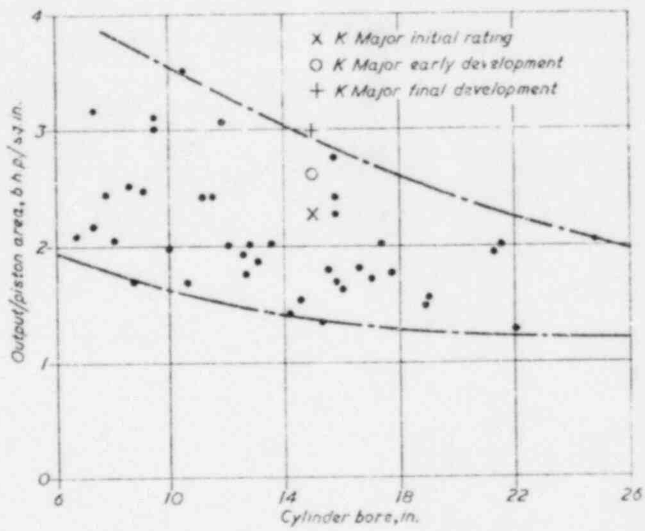

FIG. 26—*Relation between output/piston area and cylinder bore*

number of new ships which had insulated control rooms. With a maximum cylinder pressure 25 per cent greater and a maximum speed 18 per cent greater than its predecessor, the K Major might be expected to be considerably noisier. However, it was possible that the many design changes which had been made had at least partly counteracted the tendency to higher noise levels. It would be useful if the authors could provide any comparative noise measurements for the K and K Major engines.

With the advent of a new medium-speed Diesel engine, it was inevitable that designers of naval machinery installations must ask themselves whether this new engine was suitable for warships. In this respect, section f) on "Space and Weight" was relevant and one could observe that cast iron, whilst it had many admirable properties, was not the best of materials for shock. Perhaps the authors would like to comment on whether they intended to offer a naval version of this engine in due course.

Authors' Replies

Mr. Lowe (replying to the verbal discussion) referred to Mr. Cook's point about 36 cylinders being required for a ship of 18,000 horsepower and said that the paper had tried to show that the power available from that number of cylinders was now twice as great as it was a few years ago, with the same reliability and maintenance requirements.

A comparison of piston-crown temperatures, measured in the engine using templugs, with the temperature distribution produced in the rig and by an electrolytic analogue showed that in the engine, crown temperatures were rather lower than those produced in the rig and predicted by the analogue, while ring-belt temperatures were slightly higher. This difference was attributed to the rather more efficient cooling of the underside of the crown produced by the motion of the piston in the engine. The temperature distribution was proportionately the same in both rig and engine, so that thermal stress measurements should be of the right order.

The authors' company did not use a ratio as high as the $2\frac{1}{2}$ to 3 times increase in fatigue strength of rolled threads over cut threads, mentioned by Mr. Cook. They used a value of 0·25 times the U.T.S. for rolled threads and 0·125 times U.T.S. for machine-cut threads, giving a ratio of 2:1 in favour of rolled threads for fatigue strength.

Both Mr. Cook and Commander Tyrrell had commented on lubricating oils for use with heavy fuel. It was difficult to be precise about the deterioration of oil lubricants because no oil specification defined accurately the requirements of a lubricant for Diesel engines. The type of oil found suitable for this engine was a "good" Supplement 1 level with a quite reasonable alkalinity. The authors used a maximum bulk oil temperature of 170 deg. F. (77 deg. C.) but it must be remembered that this implied that there would be local higher temperatures in the engine of the order of 210 deg. F. (99 deg. C.).

He confirmed that it was not the intention to imply that centri-cast rings were necessarily inferior to sand-cast rings, but the point should be made that the quality of the iron was more difficult to maintain in a centri-cast ring.

Referring to Mr. Henshall's point about specific air flow, he agreed that it was desirable to continue to increase the specific air flow as ratings increased, although the extrapolated figure in Table I showed a small reduction.

There had not been any problem with differential expansion between the steel crown and the cast-iron piston body, probably because the intensive cooling produced a low temperature at the interfaces, as could be seen in Fig. 6. The double line in Figs. 2, 3 and 6 represented a threaded portion which was used to establish the best crown diameter and was a well-known development technique.

The development engine had been run with valve guide diametral clearances as small as 0·001in. to 0·002in. with the force-lubricated guide, and a clearance of 0·003in. to 0·004in. had been arrived at for the final design. Wear rates with the lubricated guide were extremely low and a life of 10,000 hours was expected before replacement of the guide was necessary.

Mr. Gröschel had sounded a word of warning about fuel-injection test rigs and the authors agreed generally with him. Rigs were extremely useful if one was careful in interpreting the results. The suggestion that the problem of back-flow of gas from the cylinder had been prevented by reducing the reciprocating mass in the injector was quite correct and the K Major injector was of the low inertia type, as described by Mr. Gröschel.

The fuel line pressures in Fig. 13 referred to full-load conditions and the authors considered the pressures levels and the difference between line-pressure and gallery pressure to be quite normal in their experience.

He agreed that the higher nozzle temperature for the uncooled injector with heavy fuel, in Fig. 19, was only partly due to the loss of cooling from the fuel and was also a result of the slower burning of the heavy fuel. The conclusion, however, was unaltered that cooling of the nozzle was necessary with heavy fuel and tests had been carried out, which were too extensive to be fully described in the paper, which showed that the tip temperature was dependent not only on engine load, but also on fuel temperature, water flow quantity and water temperature. The thermocouple for these temperature measurements was located actually at the surface of the nozzle tip.

Mr. Alcock had asked for details of the composition of the high-tensile steel piston crown. This was a 55-ton tensile 1 per cent Cr-Mo steel and he would gladly send exact particulars of the material and of oil velocities to Mr. Alcock. He pointed out that the 464 deg. F. (240 deg. C.) maximum temperature in Fig. 6 was a bulk temperature of the material and that the surface temperature in contact with the oil would be lower. There had been no signs of oil coking on the surface of the cooling chamber.

He agreed that it was very critical to choose the right material, not only for the seat of the valve, but for the valve itself. The relative coefficients of expansion of these two materials and their conductivity were most important.

The subject of crankcase explosions was a general one and not confined particularly to the engine under discussion. Fig. 1 showed that the engine was fitted with explosion doors. His company had experienced one or two cases, in the last seven or eight years, of crankcase explosions in other types of engine where the explosion doors had worked satisfactorily.

Referring to Dr. Mansfield's question about the method of varying the wall pressure of the piston rings, he said that this had been done both by altering the bearing area of a given ring and also by re-designing the ring.

Mr. Baker had suggested that there should be automatic timing on an engine. This was an attractive idea, particularly if combined with automatic timing of fuel injection, but was very difficult to achieve. If some simple and foolproof way of doing this could be found it would be a real achievement.

He said that he could not justify the mean line for the 200 b.m.e.p. data on the right-hand diagram of Fig. 10. The measured points indicated the "hog", suggested by Mr. Baker, but he could not explain why this should be so and hence had merely indicated the downward trend which he thought to be of greater significance.

Referring to the injection diagrams, he said that the unloading volume had been increased from (a) to (b) in Fig. 13 and had a very strong effect on the tendency for secondary injection. The injection rate had also been increased and, as already stated in the reply to Mr. Gröschel, a low-inertia type of injector had been adopted. The fuel line pressure was measured halfway along the injection pipe in each case and the

injection pipe length was the minimum possible for the engine. Mr. Baker's remarks about the Willan's lines were quite valid and, in fact, the twelve-cylinder engine had shown a mechanical efficiency of 93 per cent. The performance curves were obtained using distillate fuel.

The piston-ring pack, described in the paper, was deliberately designed so that the scraper ring above the pin was more severe than the lower scraper ring. The authors considered it important to maintain an adequate oil film over the body of the piston so that the lower scraper ring was only intended to remove excess oil to prevent the upper ring becoming flooded. The weight loss figures, in Table IV, were only significant in comparison to one another as they were obtained from a rig running completely unlubricated and could not be related to conditions in an engine cylinder. There had not been any controlled tests to measure wear rates in the grooves of the steel crown with piston rings of different irons.

In answer to Mr. Cowderoy's question about compression pressures, he said that, at 525 r.p.m. and full load, the compression pressure was 900 lb./sq.in., coming down linearly with horsepower to 380 lb./sq.in., at 100 r.p.m., i.e., effectively no load. The compression pressure, corresponding to the 60 per cent speed, 22 per cent load condition, quoted by Mr. Cowderoy, was 430 lb./sq.in. He agreed with the contributor that low compression pressure, or rather compression temperature, could contribute to inferior combustion in a worn engine at low load.

Mr. Cowderoy's suggestion that marine propulsion engines should be treated like locomotive engines, from the point of view of preventing the driver from accelerating too fast, was a good point. The necessity for the turbocharger to accelerate was often overlooked when rapid increases in load were called for, and it might well be necessary to apply the locomotive type of fuel rate control to marine engines.

In reply to Mr. French, he said that the computer approach to piston crown thermal stresses had been to construct an electrolytic analogue which, as mentioned earlier, gave quite good agreement with the thermal rig. Work was currently in progress on a digital programme which would calculate stresses directly, whereas the analogue only gave temperature distribution from which stresses could be calculated.

The numerical value of the wear factor for inlet valves obviously applied to one's own engines, but a manufacturer could apply the formula to obtain values from his own engines. In this connexion, he said that the experimental work, from which the wear factor was developed, was described fully in reference (2).

Dr. Pope said that before the meeting closed he would like to make one or two comments about some generalities which had come up during the discussion.

He said that they were getting to the stage in the medium-speed engine industry where the research and development effort of the engine builders was outstripping the component builders, and he could foresee, in the not too distant future, that engine development might well be held up because of lack of blowers and injection equipment. He hoped the supply industry would persevere with enthusiasm for the highly-rated medium-speed Diesel engine as much as the engine builders were.

He thought that the problem of the maintenance of medium-speed Diesel engines in the marine world should be judged objectively. Obviously, in the medium-speed engine one was going to have more parts, but there was a world of difference between handling a 15-in. piston and handling a 30-in. piston. The factors involved were not just the number of parts, but the way in which they could be manipulated and a statistical analysis of what were the major and minor faults. His view was that if this analysis were carried out scientifically one would find that the medium-speed engine could stand on its own, even with regard to maintenance.

With regard to cast iron, he said that each problem must be judged on its merits. The fatigue strength of a good cast iron was near to that of a good steel. There were other advantages in using cast iron. One knew that one could obtain good castings with cast iron for the size of engine he was discussing and it was an easy material to handle. Size for size, his experience had been that cast iron came out cheaper, therefore if one had a material which was as good as another and was cheaper, one had to have a very good reason for not using it. He could only see one reason for precluding its use and that was if weight were a predominating factor. However, when one considered the rest of the engine room equipment, the tankerage and the fuel capacity, one would find that there was a difference of one or two per cent between a welded design and a cast iron design, so that, for commercial shipping, this was a marginal consideration.

He pointed out that with an in-line engine with an underslung crankshaft, one could have a very nice stress line pattern which, on the drawing board, looked very attractive and almost impossible to improve upon, but when one came to a "V" engine with side by side connecting rods, so that the opposite liners could not be in line with each other, the stress pattern did not look quite so elegant. He accepted that the cast iron bedplate was more difficult to design because the stress pattern was more complex, but once one had designed it and got a good design one was simply comparing one good design with another. It was also a question of continuing from a well-tried engine to the next generation, without departing from well-proven design principles. His company had now completed its one thousandth K engine and had over 300 of them in marine application. Over a third of the engines were running day in and day out on heavy fuel.

AUTHORS' REPLY TO WRITTEN CONTRIBUTIONS

The authors wrote that they entirely agreed with Commander May's appreciation of the rapidly increasing demand for large medium-speed engines. They had deliberately restricted the paper to the development of the K Major engine itself but their company was certainly proposing to offer complete propulsion units for geared installations with either single or multi-engine inputs.

In reply to Mr. Hughes, the authors wrote that the piston cooling-oil flow was 1·7 gallons/b.h.p.-hr. at the current full-load rating. The cooling design was such that the operating temperatures of the piston crown were well within the thermal fatigue limit of the steel used, as could be seen from the isotherms of Fig. 6. The work that had been carried out on the optimization of valve timing and combustion characteristics, which was described, had been aimed at obtaining the best thermal efficiency from the engine and not at reduction of component temperature by scavenge air cooling, which they considered was a relatively inefficient method of controlling component temperatures. The temperature of the critical parts of the engine, such as exhaust-valve seat, injection nozzle, and top piston-ring groove, was controlled by direct cooling.

As mentioned in the reply to Mr. Baker, the lower scraper ring in the skirt was relatively mild and the upper scraper more severe to ensure adequate lubrication of the piston skirt. The taper-faced compression rings were an advantage in initial running as they bedded in very quickly on a narrow circumferential band. Initial running-in had been the subject of a good deal of investigation on the test bed and the best results had been obtained with a relatively rough liner surface, which was honed to a C.L.A. of about 100μ, the liner being a random flake graphite iron, slightly softer than the piston-ring material.

Calculations of liner frequency and vibration amplitude were included at the design stage to avoid the possibility of water-side attack. The heat dissipation through the piston rings to the cooling water could not be directly measured in the engine because of the heat received directly by the liner from the combustion gases. By reproducing temperatures and total heat flow in the rig, the proportion of heat flowing to the cooling oil was about 75 per cent of the total and the heat to the liner was 25 per cent.

NUCLEAR REGULATORY COMMISSION

Suffolk County

Docket No. ______ Official Exh. No. Diesel-71

In the matter of ______

Staff ______ IDENTIFIED ______

Applicant ______ RECEIVED ______

Intervenor ______ REJECTED ______

Cont'g Off'r ______

Contractor ______ DATE ______

Other ______ Witness ______

Reporter Dohogne ______

A photo of a piston removed from EDG 103
Taken by Aneesh Bakshi in June 1984
at SNPS
(SCUFFING)