EVALUATION OF THE OPERATIONAL AND MAINTENANCE HISTORY OF, AND RECENT MODIFICATIONS TO, THE MAIN ENGINES IN THE M.V. COLUMBIA

SES Report No. 123-01

Prepared For:

Department of Transportation and Public Facilities
State of Alaska
Division of Marine Highway Systems
P.O. Box 1467
Juneau, Alaska 99802

Prepared By:

Seaworthy Engine Systems, Inc.
36 Main Street
Essex, Connecticut 06426

8401260415 XA

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- ERRATA SHEET -

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SES REPORT NO. 123-01, APRIL 1983

1. Add: Pg 2-2 (after last item in 3.):
   - The new C-17 turbochargers must be properly matched to the engine revised performance ratings.
   - The turbocharger installation shall include such modifications as necessary to maintain the waste heat recovery systems at present output level.

2. Add: Pg 2-2:

   Word "Carbon"
   - combustion chamber carbon deposits

3. Correct: Pg 2-7 (Table 2.2):

   Column Predicted By Fuel Rack(2)
   Entry should read: 28.3/6200 BHP

4. Correct: Pg 2-12 (2nd paragraph, line 11):

   Omit "the fact that" where repeated

5. Correct: Pg 2-14 (Table 2.4):

   Column 385/4680/135.2-Halter Marine Test
   Entry should read: 390/5521/147

6. Correct: Pg 2-16 (1st paragraph, line 13):

   Sentence should read: Rating of 5284 BHP/385 ERPM

7. Correct: Pg 2-18 (lowest curve):

   Entry should read: 3/25/83 Test Data, BHP By Rack, Figure 2.1, Fuel By Rack (Ref. No. 3)

8. Correct: Pg 2-21 (1st paragraph, line 4):

   Sentence should read: ....Appendices F and G:

9. Add: Pg 2-24 (Table 2.7)
   - Omitted - η
   - Vertical columns should read: Left Bank η TC
     Right Bank η TC
     Average η TC

10. Correct: Pg 3-32 (1st paragraph, line 3):

    Omit "generating time"

    Sentence should read: Original planning records show that for the five (5)................

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1.0 INTRODUCTION AND EXECUTIVE SUMMARY

1.1 Background

In support of a major engineering change for the power plant in the passenger and vehicle ferry, M.V. COLUMBIA, consisting of the installation of new turbochargers and the de-rating of the main propulsion engines, Seaworthy Engine Systems, Inc., was retained by the State of Alaska, Department of Transportation and Public Facilities, Division of Marine Highway Systems, and tasked with a review of the adequacy of the de-rating and evaluation of the post de-rating trial performance of the vessel's main propulsion engines. As an additional (and related) task, Seaworthy was also requested to review historical main engine component failures, and where available, associated costs to provide further insight as to the ultimate adequacy of the engine de-rating in terms of anticipated improvements in reliability, performance and associated operating economics.

The M.V. COLUMBIA was delivered as a combination vehicle and passenger ferry by Lockheed Shipbuilding Company in 1974 for the Southeastern Alaska/Seattle, Washington service. She is 418 feet long, overall, having an 85.13 foot beam and a depth of 24 feet. At a full load displacement of 7745 Long Tons, the vessel has a draft of 17.6 feet. The ship is propelled by a twin shaft medium speed diesel engine propulsion plant supported by three (3) 900 KW auxiliary diesel generators, a combination waste heat recovered/oil-fired steam generating system and two (2) saltwater distillers. Each main propulsion shaft is fitted with an Allis-Chalmers/Escher-Wyss controllable and reversible pitch propeller capable of delivering a maximum of 9900 HP at 250 SRPM, driven by a single engine through a 1.8:1 ratio single stage reduction gear.

The two (2) V-type turbocharged main engines are DeLaval-Enterprise model DMRV-16-4 units (serial nos. 72034 Port, 72033 Stbd.), each capable of developing a maximum of 9200 BHP at 450 RPM (prior to de-rating).

1.2 Executive Summary

Scope/Objective: To evaluate the historical operation and maintenance and repair of, and the recent de-rating of, the main engine in the State of Alaska Vessel, M.V. COLUMBIA, by the completion of the following tasks:

1. Observation and evaluation of the vessel's sea trial after de-rating, held on March 24-25, 1983.

2. Review and summarize historical main engine component failures to date and related maintenance and repair records, including cost data, where available.

3. Analyze and review the existing engine de-rating to identify and quantify, where possible:
   * Adequacy of the de-rating modifications
   * Additional modifications required to ensure engine reliability
   * Cost in time and dollars to make additional modifications
   * Engine life expectancy once de-rating and additional

required modifications are completed

* Cost effectiveness of re-engining the M.V. COLUMBIA versus continued operation of the de-rated engines.

Supporting Documentation/Results: The method of approach, supporting documentation and data and results of the completion of the required scope of work are presented in detail in Sections 2.0, 3.0, 4.0 and 5.0 and the Appendices of this report.

Conclusions and Recommendations: Supported and substantiated by data and documentation contained in preceeding sections of the report, the following pertinent conclusions and recommendations have been extracted from Section 6.0.

* Sea Trial Performance:

    1. The engines as de-rated by TDI failed to develop the required power outputs as specified in the work scope of the contract authorizing this work.

    2. The turbochargers, as indicated by surge problems observed during the trials and on subsequent voyages, are not properly matched to the new de-rated engine operating profile. Emperical data presented in Section 2.0 further supports this conclusion.

    3. Numerous other problems of a smaller magnitude also identified in Section 2.0, have developed as a result of the de-rating work and for the most part are unresolved.

    4. Adequate air flow appears to have been provided to the engines by the new turbochargers. Brake Mean Effective

Pressures at the new operating outputs are equal to, or less than, those specified in the de-rating contract.

5. It is possible that some minor portion of the turbocharger surge problem is related to the difficulties being encountered with the pitch scheduling portion of the main engine control system. TDI should be required to assist and work closely with Mathers Controls to establish responsibility for and correct this situation.

6. Based on the above described performance, TDI should be put on notice that the de-rating work to date is unacceptable and payment withheld.

* <u>Adequacy of the Engine De-Rating</u>:

1. Based on a review of main engine historical maintenance and repair data and a comparison of engine component failure frequency and mode with the modification accomplished as a result of the de-rating effort, it is anticipated that only minimal overall improvement in failure rates and time between failures or overhauls will occur. The most significant portion of this improvement will occur for those components directly impacted by the improved combustion process which results from the increased availability of air blown for combustion.

2. It is believed that for the remainder of the engine component failures identified in Sections 3.0 and 4.0, those not directly influenced by increased air flow, little or no change in failure rate, and probably no

more than would be obtained by simply running the original engines at a reduced output without officially de-rating, will occur. These component failures include:

- Cylinder heads - design and manufacturing defects
- Cylinder liner distortion and wear - due to block distortion
- Piston ring distortion and wear - due to block distortion
- Cylinder blocks - distortion and cracking
- Connecting rod bearings - design of articulated connecting rod assembly
- Main bearings - premature wear, high loading
- Camshafts - premature wear

3. It is estimated that when equated to dollars, the reduction in main engine maintenance and repair historical average annual cost resulting from de-rating may approach twenty-five percent (25%).

4. The existing de-rated engines,* after incorporation of the additional modifications identified in this report, can be kept running almost indefinitely if AMSH is willing to continue to maintain them at the same expensive rate, in terms of time and money.

* Additional Modifications:

1. Numerous additional modifications have been identified in Section 5.0 and should be incorporated to enhance the future reliable and efficient operation of the

de-rated engines. Some of the more important of these modifications are a result of, and not in addition to, the de-rating effort. The most significant of these is the turbocharger mismatch which should be rectified by TDI by installing new matched turbochargers at no additional cost to the de-rating contract..

* <u>Economic Evaluation of Re-engining of the M/V COLUMBIA</u>:

1. Re-engining of the COLUMBIA for operation on Marine Diesel Oil, MDO, depending on the acquisition cost estimate/remaining vessel life combination considered, can offer a significant economic advantage over continued operation of the existing de-rated engines on MDO.

2. Re-engining of the vessel to operate on Heavy Fuel Oil, HFO, is a clearly superior economic alternative compared to both re-engining for MDO operation or continued operation of the de-rated engines on MDO, regardless of the acquisition cost/investment period combination considered in the economic analysis presented in Section 5.0.

Based on the technical analysis and evaluation conducted and documented in this report and the results derived for the range of estimated re-engining acquisition cost/remaining vessel life combinations considered as part of the economic analysis, it is recommended that the M/V COLUMBIA be re-engined for HFO operation at the earliest opportunity.

2.0 ANALYSIS OF DE-RATED ENGINES AND NEW TURBOCHARGERS

The assessment of engine and turbocharger performance after de-rating and installation of the new turbos is based on design and shop test data provided by Transamerica DeLaval, Inc. (TDI) and sea trial observations and data obtained jointly by TDI, Seaworthy and AMHS personnel during underway tests on the COLUMBIA on March 24 and 25, 1983. Subsequent operating problems reported by the crew (up to the time of report preparation) during the vessel's initial voyage of the season, commencing April 1, 1983, are also commented on in this section. Briefly, the scope of work to be carried out by TDI as a part of the engine de-rating process or in conjunction with this work and which impacts engine/turbo performance, as defined in State of Alaska Delivery Order 707573 (Reference No. 1) included:

1. De-rating of the main propulsion engines from 9200 BHP/ 450 ERPM each to the following operating conditions and limits:
   - Idle Speed: 300 ERPM
   - Design Service Rating: 5284 BHP @ 384 ERPM
   - Maximum Continuous Rating: 6164 BHP @ 403 ERPM
   - 10% Overload Rating: 6791 BHP @ 403 ERPM
2. Reduction in brake mean effective pressure from 213 PSI to approximately 158 PSI.
3. Procurement and installation of new DeLaval-Enterprise C-17-123 turbochargers, two (2) per engine, four (4) total,

having the following performance characteristics:

- Response time from 40 to 100% load; 6 to 7 seconds
- Response during rapid propeller de-pitching (engine unloading) to be at least as good as the original Elliot G-90 series units being replaced.

4. Installation of a Trabon lubricating oil system to seal against exhaust valve stem and guide soot and exhaust gas blow by.

The anticipated improvement in engine performance and reliability resulting from the incorporation of the above described modifications would reasonably be expected to be manifested by reductions in:

- smoke level
- combustion chamber deposits
- lube oil contamination
- cylinder liner wear
- exhaust valve/guide blow by

The discussion of various aspects of the de-rated engine and new turbocharger performance in the following paragraphs deals largely with the results of various computations and comparison of data obtained from the previously mentioned sea trials and design or ship test data provided by TDI for the installed and/or comparable engines and turbos. While these results are felt to be directionally indicative of current engine and turbo performance, the absolute values shown in certain instances should be viewed with some reservation due to the nature of the trial data obtained and the available engine design and operating baseline comparative performance information. These qualifications are summarized briefly below.

* March 24-25, 1983 Sea Trials:

  - Due to the lateness in completing the work associated with engine de-rating and the limited time available to plan and establish rigorous trials, test procedures and install test equipment, the testing performed was quite cursory and unusually brief. (See Appendix A for Trials Agenda).

  - Only the starboard engine and its shaft line were instrumented. As a result, data and calculations have reasonably been assumed as typical for both engines.

  - Actual sea trials were compressed from a time standpoint. Thus, various tests were conducted simultaneously with or at the expense of others. For example, pitch/load control systems test and adjustments were conducted simultaneously with steady state power runs for engine performance evaluation. Difficulty with the control system actually caused certain runs to be aborted or shortened. In general, the time alloted for data gathering at each engine load point was felt to be less than desirable (see Table 2.1).

  - The fuel oil meters fitted on the starboard engine for the test were of questionable accuracy, despite an attempt by Todd Shipyards to calibrate them prior to trials on March 25, 1983. A better selection could have been made if adequate time had been available.

TABLE 2.1

SCHEDULED vs ACTUAL ENGINE PERFORMANCE
LOAD POINT TEST DURATION

LOAD POINT SRPM/ERPM/BHP	SCHEDULED TIME	ACTUAL TIME (COMMENTS)
167/300/2500	1 Hour	3/24/83: 1 hour, maneuvering, engine break-in
184/330/3300	1 Hour	3/25/83: 1 hour, 5 minutes. F.O. meters out of calibration vessel turning frequently. Testing halted due to control problems and port engine intercooler transition ducting leak.
202/363/4300	1 Hour	3/25/83: 1 hour, Seawater cooling system on hand control (off/on) due to problems with thermostatic control valves in various cooling loops which continued throughout the trial, F.O. meters recalibrated on evening of 3/24/83 by Todd.
214/385/5300	1 Hour	3/25/83: Test aborted and restarted twice due to maneuvering requirements and engine control system problems over a two hour period. Only final 25 minutes are felt to be representative of steady state operation.
224/403/6200 (Maximum Continuous Rating)	4 Hours	3/25/83: 1 hour, 40 minutes. Cut short upon return to dock.
224/403/6800 (10% Overload)	1 Hour	Not Run. Engine could not reach overlaod at 403 ERPM with propeller on maximum pitch.

* Design Predicted Performance Data:

- Apparently there is no TDI published standardized fuel consumption map of brake specific fuel consumption rate versus BMEP or BHP and speed. (Requested by Seaworthy).

- C-17-123 turbocharger and engine performance for the COLUMBIA's de-rated engines is predicted with no shop test comparative basis available.
- Rack setting versus engine speed and power was predicted and, by necessity, not confirmed on a test stand prior to trials.
- There appeared to be little coordination between TDI and Mathers Controls prior to trials relative to integration of pitch control schedule with the performance characteristics of the new turbos and de-rated engines.

2.1 Engine Performance

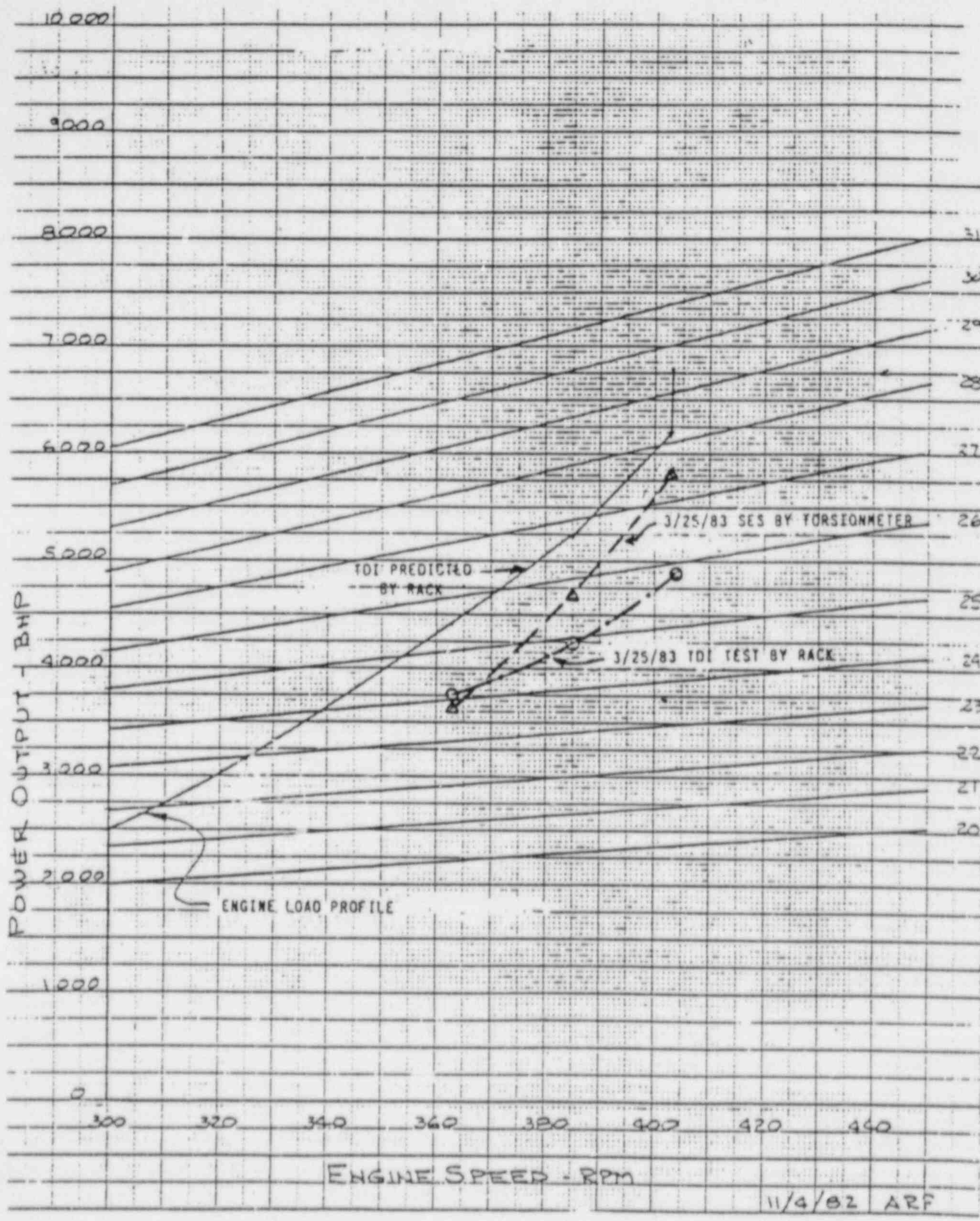
Engine performance evaluation as discussed here consisted of a review and comparison of data obtained during the sea trial with predicted values or test stand information from TDI and past performance information for the original engine configuration at similar outputs. Specifically, engine power output correlated satisfactorily with predicted and observed vessel speed/power data, brake mean effective pressure, fuel consumption and apparent combustion quality are addressed in the following paragraphs. Sea trial data gathered on March 24-25, 1983 by both TDI and Seaworthy is contained in Appendix B.

Power Output: Figure 2.1 presents plots of brake horse power (BHP) produced by the de-rated main engines, including power as predicted by TDI versus fuel rack setting and ERPM, power determined as a result of average rack setting observed during trials versus ERPM and a final curve of BHP developed, corrected

Figure 2.1

Fuel Rack vs Engine Speed DMRV-16-4 with C-17 Turbo

M.V. COLUMBIA



from trial shaft horsepower (SHP) measurements made by Seaworthy assuming a 98% gearing and shafting mechanical efficiency. (The theory of operation of the torsionmeter utilized and sample traces of recorded data are contained in Appendix C). Table 2.2 presents a comparison of these values at test points from 360 ERPM and above. The original predicted load curve of fuel rack versus BHP and ERPM assumes the propeller law's approximate full pitch cubic relationship. During the trials, very close to full pitch was applied to the propellers from 360 ERPM on up, equating to somewhere between 1.0 and 1.1 pitch to diameter (P/D) ratio as reported by Mathers Controls personnel who were onboard testing and adjusting the engine controls during this period.

TABLE 2.2

COMPARISON OF STARBOARD ENGINE POWER OUT; PREDICTED BY FUEL RACK AND ERPM VERSUS OBSERVED FUEL RACK AND ERPM AND AS MEASURED AT THE SHAFT BY TORSIONMETER, MARCH 25, 1983

ERPM/SRPM(1)	PREDICTED BY FUEL RACK(2)	BY OBSERVED AVERAGE FUEL RACK	BY TORSION-METER(3)
363/202	25.55/4400 BHP	24.1/3750 BHP	3640 BHP
385/215	26.7/5250 BHP	24.9/4250 BHP	4680 BHP
403/224	28.3/16200 BHP	25.85/4900 BHP	5930 BHP

(1) Propeller pitch at maximum, $1.0 \leq P/D \leq 1.1$

(2) From Figure 2.1

(3) Corrected from measured SHP values assuming a 98% gear/shafting mechanical efficiency

From an inspection of the data presented in Figure 2.1 and Table 2.2, neither power determined from actual fuel rack setting and ERPM or as measured at the shaft very closely matched the predicted engine load profile, shown in Figure 2.1. Further, at all test ERPM's but 363, TDI observed rack BHP falls below that determined from measured SHP. Relative to contractual performance, TDI test data and resultant plotted Brake Horsepower data fails to meet anticipated outputs for Design, Maximum Continuous and 10% overload service ratings of 385 ERPM/5284 BHP, 403 ERPM/6164 BHP and 403 ERPM/6791 BHP, respectively.

Additionally, per the torsionmeter, the Maximum Continuous Rating of 6164 BHP at 403 ERPM appears not to have been met based on Seaworthy's measured SHP data at this load. Also, the 10% overload capability at 403 ERPM could not be demonstrated as the propellers were on full pitch from at least 360 ERPM on up. Thus, the only way that load could have been increased was to increase engine speed above the new limit of 403 ERPM established for the de-rated engines. Further, with the propellers on full pitch, engine/shaft RPM and power output would be expected to more closely follow the propeller law cubic speed/power relationship as is the case with the speed/power curve plotted from measured SHP and ERPM, as compared with the speed/power curve obtained utilizing average fuel pump rack setting and observed ERPM. Also, the fuel rack/BHP/ERPM load curve shown by TDI in Figure 2.1 is a predicted one, never previously verified by actual tests for the COLUMBIA's engines. In addition, it is noted here that the adjustment and calibration of the starboard engine's number one right bank fuel pump and rack assembly had been altered and never reset prior to the trials. This fuel rack position is used as the master command signal indicator for that engine's speed and load control program. In consideration of the previously discussed factors, the data obtained from the torsionmeter reading is felt to be more closely representative of actual power produced.

Vessel Speed: To further evaluate and verify the speed and power relationships derived from the test data from the shaft torsionmeter and from fuel rack settings, a comparison of vessel speed over the ground taken for each test run from the bridge was plotted versus rack and torsionmeter power outputs and compared with predicted vessel speed/power curves for propeller P/D ratios of 1.0 and 1.1, as presented on pages 19 and 20 in Morris Guralnick Associates, Inc. report, "Performance Predictions and Engine Selection Criteria for the M.V. COLUMBIA", dated June 1982 (Reference No. 2). The results of these comparisons are shown graphically in Figure 2.2 and 2.3. By inspection of Figure 2.2, the shaft horsepowers and SRPM's plotted for the torsionmeter data are much more consistent with the shape of, and fall very closely to, the predicted P/D = 1.0 curve while the plot of the rack determined SHP versus SRPM falls well below the P/D = 1.0 line, which would indicate a P/D ratio of less than 1.0 in contradiction to the pitch carried during the trials as reported by Mathers. Trial and predicted data

Figure 2.2

Plot of Trial vs Predicted SHP and SRPM, M.V. COLUMBIA

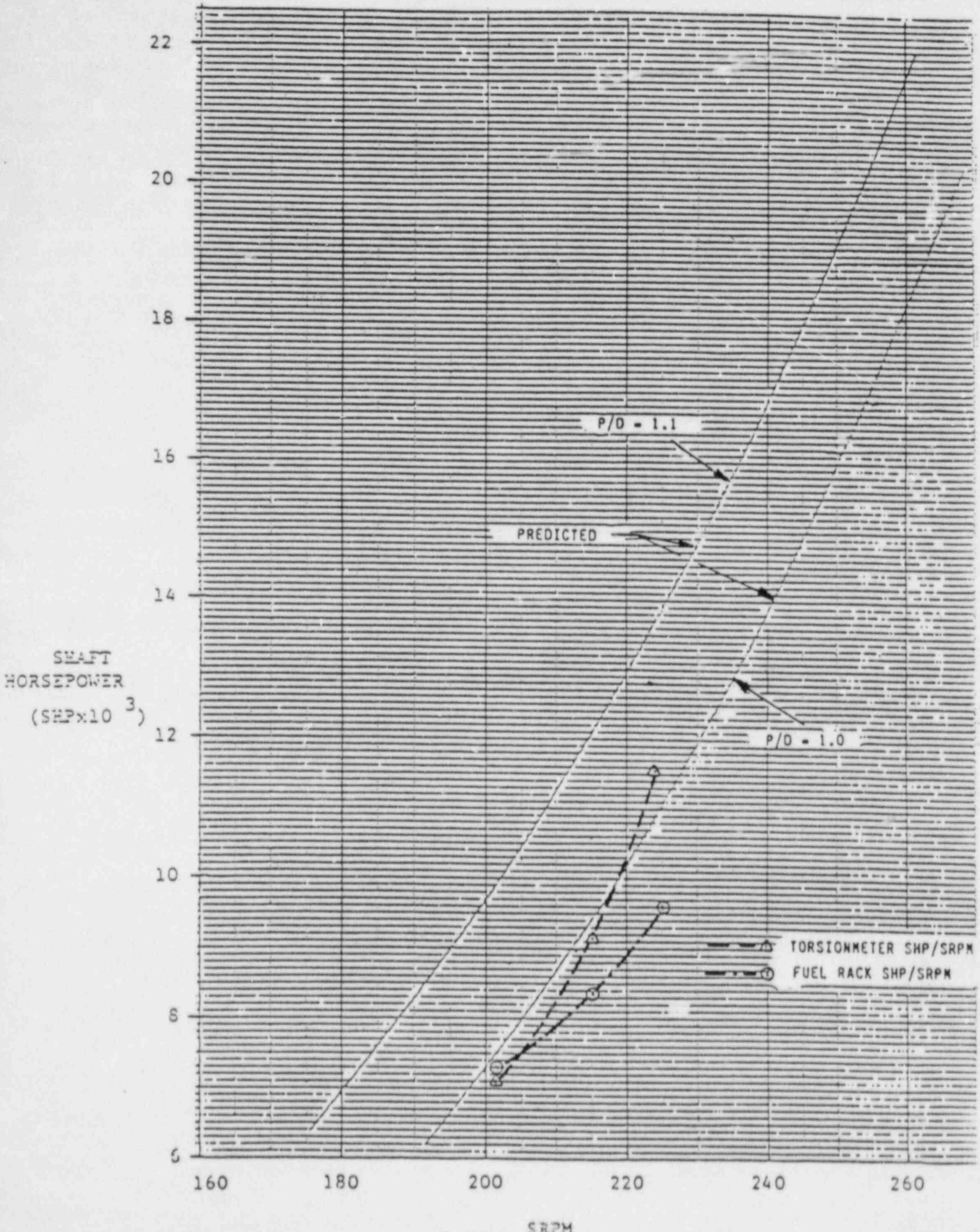
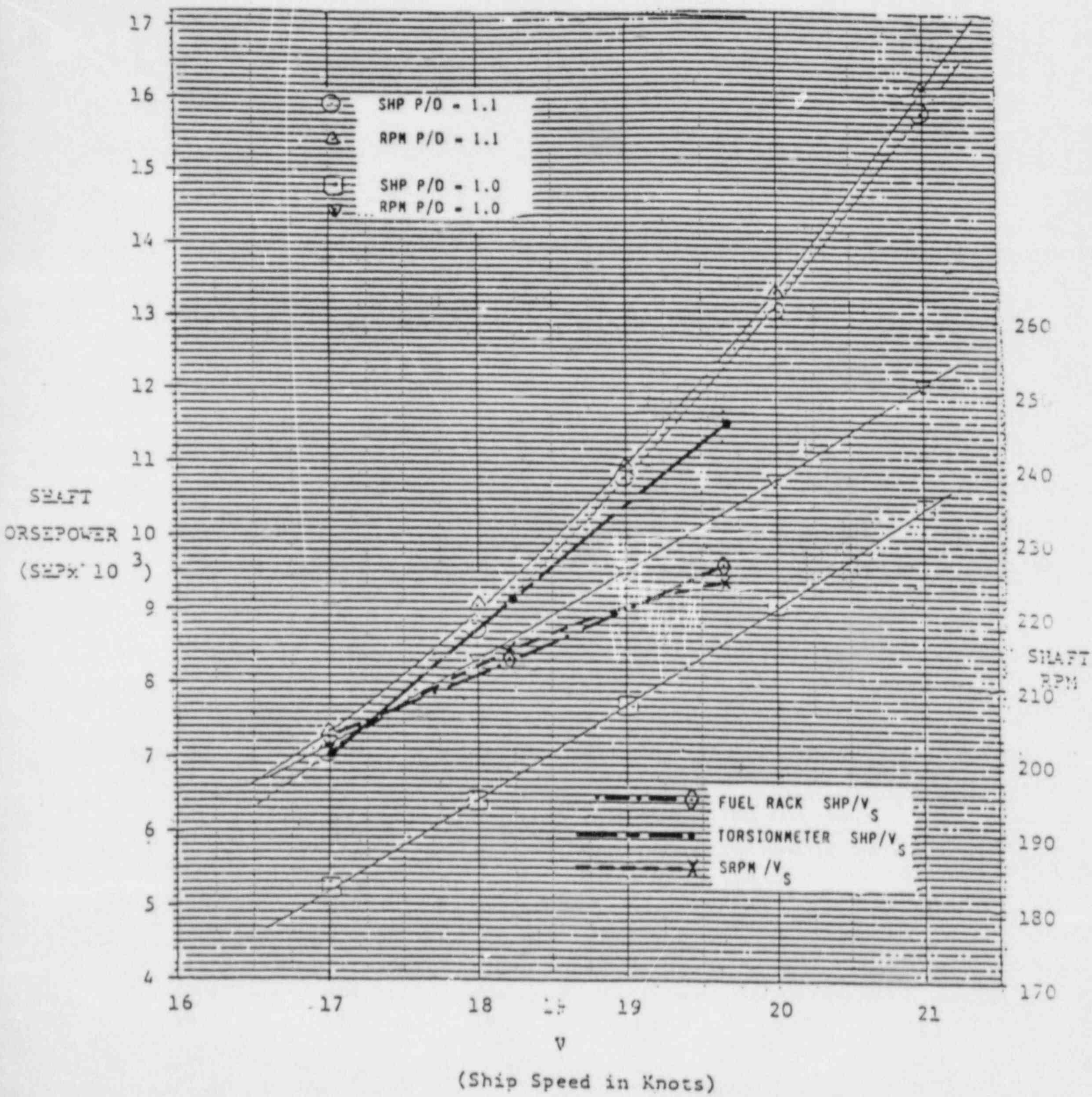


Figure 2.3

Comparison of Predicted and Trial Speed Power Data, M.V. COLUMBIA



plotted for vessel speed and SHP for P/D ratios is shown in Figure 2.3. Here also, the observed uncorrected speed/SHP points recorded during the test from the torsionmeter show a much closer agreement with vessel speeds predicted for P/D ratios of 1.0 and 1.1 than speed/SHP data based on and plotted for fuel rack settings.

The predicted speed, SRPM and SHP data plotted in Figures 2.2 and 2.3 was extrapolated (in Reference No. 2) from model test data and initial delivery sea trial data. It also contains adjustments for estimates of increased hull roughness as a function of time out of drydock and additional wetted surface areas which would result from a planned lengthening of the existing skeg to improve the vessel's manuevering characteristics. It is estimated that these adjustments increased required SHP by three (3) to four (4) percent over what would be the case for the hull at the time of testing on March 25, 1983. This results from the fact that the fact that the hull was freshly painted and that the skeg had not been lengthened. The ship speed and shaft horsepower data recorded during the test was taken with the vessel operating at a draft of 13'5" FWD and 16'0" Aft, resulting in a mean draft of 14'8.5". The draft on which predicted speed and power curves were based was 16.0 feet even keel. Again, based on a comparison of the observed trial vessel speed/power data with predicted vessel speed/power data, the BHP's determined from the torsionmeter readings appear to be more closely representative of actual power produced than do the equivalent rack setting values of BHP.

Brake Mean Effective Pressures: Brake Mean Effective Pressures were calculated for BHP's as determined from fuel rack and ERPM data and from SHP and ERPM data recorded at 363, 385 and 403 ERPM on March 25, 1983. Formulae and sample calculations are presented in Appendix D. The results of these calculations are contained in Table 2.3.

TABLE 2.3

COMPUTED BMEP'S FROM TRIAL RESULTS FOR
DE-RATED STARBOARD ENGINE, MARCH 25, 1983

ERPM	PREDICTED RACK (1) BHP/BMEP, PSI	OBSERVED RACK (1) BHP/BMEP, PSI	TORSIONMETER (2) BHP/BMEP, PSI
363	4400/125.8	3750/107.2	3640/104.1
385	5250/141.5	4250/114.6	4680/126.2
403	6200/159.6	4900/126.2	5930/152.8

(1) From Figure 2.1

(2) 98% gearing and shafting mechanical efficiency assumed

As can be seen from the results, computed BMEP's, regardless of power measurement results utilized, did not exceed the maximum limit set by the de-rating contract workscope of 158 PSI.

Further, a review of COLUMBIA's starboard engine test stand data and June 1981 sea trial data contained in TDI's report, "Shipboard Test, M.V. COLUMBIA, Starboard Engine, S/N 72033", August 31, 1981 (Reference No. 3) and test stand data for a similar DMRV-16-4 engine (Reference No. 4) was conducted to determine BMEP's at ERPM's similar to those run during the March 25, 1983 test. A review of past engine room log data for the COLUMBIA's starboard engine was also performed in an attempt to establish a typical load profile from which BMEP's could also be computed. The results of these investigations are presented in Table 2.4.

TABLE 2.4

COMPARISON OF VARIOUS DMRV-16-4 ENGINE BMEP'S AT SIMILAR LOADS

ENGINE	MARCH 25, 1983 TRIAL ERPM/BHP/BMEP		
	363/3640/111.5	385/4680/135.2	403/5930/152.8
1. Halter Marine Test Stand, 12/78(1)	390/2366/63	390/5521/~~2~~147	390/6704/178.5
2. COLUMBIA: Stbd Engine Test Stand 7/72(2)	320/4089/137.2	360/5814/167.7	400/6460/167.7
3. COLUMBIA: Stbd Engine Sea Trial, 7/81(2)	347/2700/80.8	368/3950/127.3	401/7270/194
4. COLUMBIA: Stbd Engine(2)			399/7500/204.6
5-6/81	-----	-----	399/7500/194.
7/80	-----	-----	396/7500/196.6
7/77	-----	-----	400/7300/189.4
6/74	-----	-----	430/7400/178.7

(1) With DE C-17-123 turbos
(2) With Elliot 6-90 series turbos

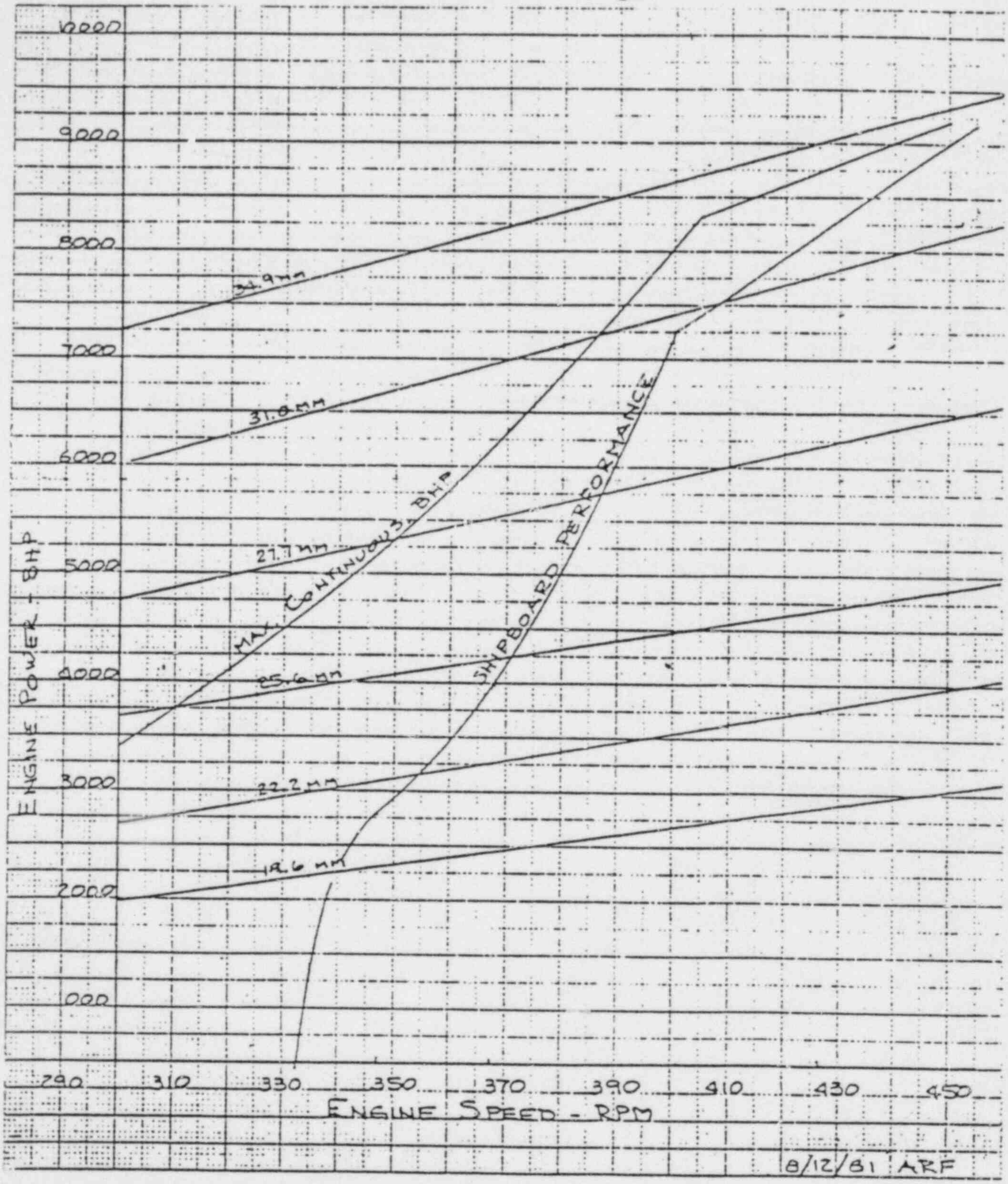
BMEP's for various COLUMBIA voyages were computed from log book ERPM and rack settings per the July 1981 sea trial load curve for the original engine rating shown in Figure 2.4 and taken from Reference No. 3. A comparison of BMEP's shown indicates the following. First, it appears that the engine has been operated full away at combinations of ERPM, propeller pitch and BHP which result in BMEP's ranging from approximately 175 to 200

Figure 2.4

Locked Rack Test DMRV-16-4-72033

9,200 HP@ 450 RPM

(Prior To De-rating)



PSI as can be best determined utilizing Figure 2.4. (However, it should be noted that frequent changes in the pitch program portion of the throttle control system, as reported, make it difficult to conclude that the load curve shown in Figure 2.4 is 100% representative of engine load profile from delivery in 1974 up until de-rating in early 1983.) Secondly, from Tables 2.3 and 2.4, operation at ERPM's and BHP's, as shown from the COLUMBIA and Halter Marine engine test stand data, which are somewhat similar to the predicted de-rated engine load profile, should produce BMEP's which are ten (10) to twenty (20) percent lower at the projected maximum continuous rating of 6164 BHP for the de-rated engines. Operation at the new Design Service Rating of 5284 BHP/355 ERPM should result in a reduction of from twenty (20) to thirty (30) percent in BMEP's compared to past operating loads.

Combustion Quality: As a qualitative assessment based on smoke and particulate emission determined from a Bosch smoke test apparatus and visual observation of the stack at various steady state loads, it appears that the combination of new turbochargers and the engine de-rating have significantly improved the combustion process. Stack emissions were virtually clear up to the maximum load point at 403 ERPM where a very slight haze was observed. Further, data shown in Table 2.5, which compares the smoke results from the July 1981 sea trial with those taken on March 25, 1983, also indicates a substantial reduction in visable smoke.

TABLE 2.5

COMPARISON OF SMOKE TEST RESULTS
JULY 1981 vs MARCH 25, 1983

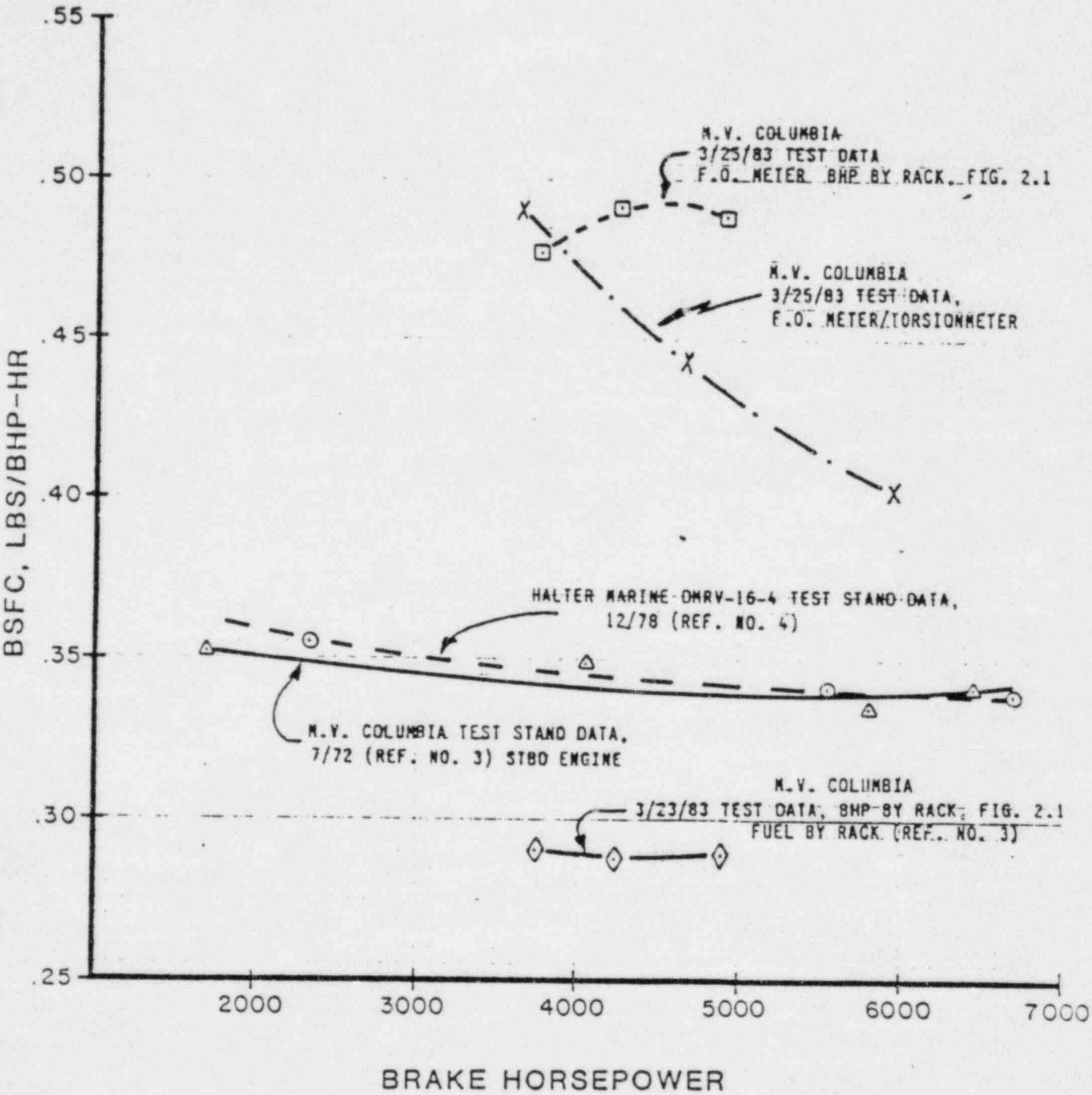
JULY 1981	MARCH 25, 1983
332 ERPM/0.5 BSN	363 ERPM/0.3 BSN
367 ERPM/0.4 BSN	385 ERPM/0.225 BSN
401 ERPM/0.8 BSN	403 ERPM/0.33 BSN

The reduction in exhaust gas smoke level, while indicative of an improvement in combustion quality, cannot be utilized as an absolute indicator of combustion efficiency or the completeness with which the potential chemical energy in the fuel is converted to heat via combustion in the engine's cylinders. It is possible to have a significant amount of fuel in various stages of oxidation exit in the cylinders with the exhaust gases in a clear state, if sufficient air is being supplied by the turbos.

Fuel Consumption: Figure 2.5 presents plots of brake specific fuel consumption rate (BSFC) in LBS/BHP-HR for various conditions. Briefly, BSFC can be viewed as an indicator of how efficiently an engine converts the energy in a pound of fuel to a unit of power, the lower the BSFC the more efficient the engine. First, as shown, test stand BSFC curves for the COLUMBIA's starboard engine and a similar newer DMRV-16-4 engine delivered to Halter Marine are plotted from data contained in Reference No. 3 and No. 4 and show excellent agreement with fuel rates as predicted by the Builder. Utilizing fuel flow and various

Figure 2.5

Comparison of M.V. COLUMBIA STBD Engine BSFCs, 3/25/83, with Original Test Stand Data



power data sources obtained during the sea trials, curves representative of actual in-service performance have also been plotted in Figure 2.5. Two curves based on fuel consumption from the test meters and TDI fuel consumption as predicted by rack setting (Reference No. 3) and TDI test de-rated engine brake horsepower from rack and ERPM (Figure 2.1) were also plotted. These curves are the upper and lower most lines on Figure 2.5. They show virtually no agreement between either the predicted fuel flow or the flow as measured, one being 45 percent higher and the other 14.7 percent lower than original test stand BSFC values shown in Figure 2.5. The fifth and final curve plots BSFC for the vessel's starboard engine from data obtained from the test fuel oil meters and from power as measured by the torsionmeter installed for the trial. This curve shows a much greater slope than the test stand data, with fuel rate decreasing with increasing engine load. From this curve, at 5815 BHP, the difference between the test stand performance and the observed BSFC for the starboard engine is 18%. A plot of BSFC based on power from the torsionmeter and fuel from the rack setting would, in fact, result in a curve that would fall well below the abscisa of Figure 2.5. As in the case of the TDI data plot, resultant fuel rates in the range of 0.233 to 0.30 LBS/BHP-HR, equating from 45% to 62% thermal efficiency, are well outside the range of the most efficient medium speed diesel engine capability and therefore, are unacceptable. BSFC data derived utilizing fuel meter flow rates in both instances (TDI predicted and torsionmeter power outputs) show brake specific fuel rates considerably in excess of the factory test stand rates. As stated in the introduction to Section 2.0, the absolute magnitude of fuel flow values recorded by the test meters may be open to challenge. Based on the past history of these engines, it would seem reasonable to assume that they are in fact consuming fuel at a rate considerably in excess of original and design predicted performance, perhaps by as much as 10%. Potential sources of this increase may include operation at reduced ERPM, cylinder load imbalance, improper fuel injection timing, lack of an optimized fuel metering system (nozzle, injector, pump) for low load operation, increased cylinder liner/piston clearances, reduced BMEP and less than anticipated turbocharger efficiencies.

All fuel rates shown have been corrected to design on the basis of lower heating value content of the fuel actually burned to the design lower fuel heating value content of 18,190 BTU/LB assumed for design predicted performance calculations. Sample fuel rate calculations are contained in Appendix E, along with a laboratory analysis of the fuel actually burned during the trial.

2.2 Turbocharger Performance

Turbocharger performance was reviewed quantitively and qualitatively based on data obtained for the starboard engine from the March 25, 1983 trials. Comparisons have also been made with the original Elliot turbochargers, based on data contained in Reference No. 3.

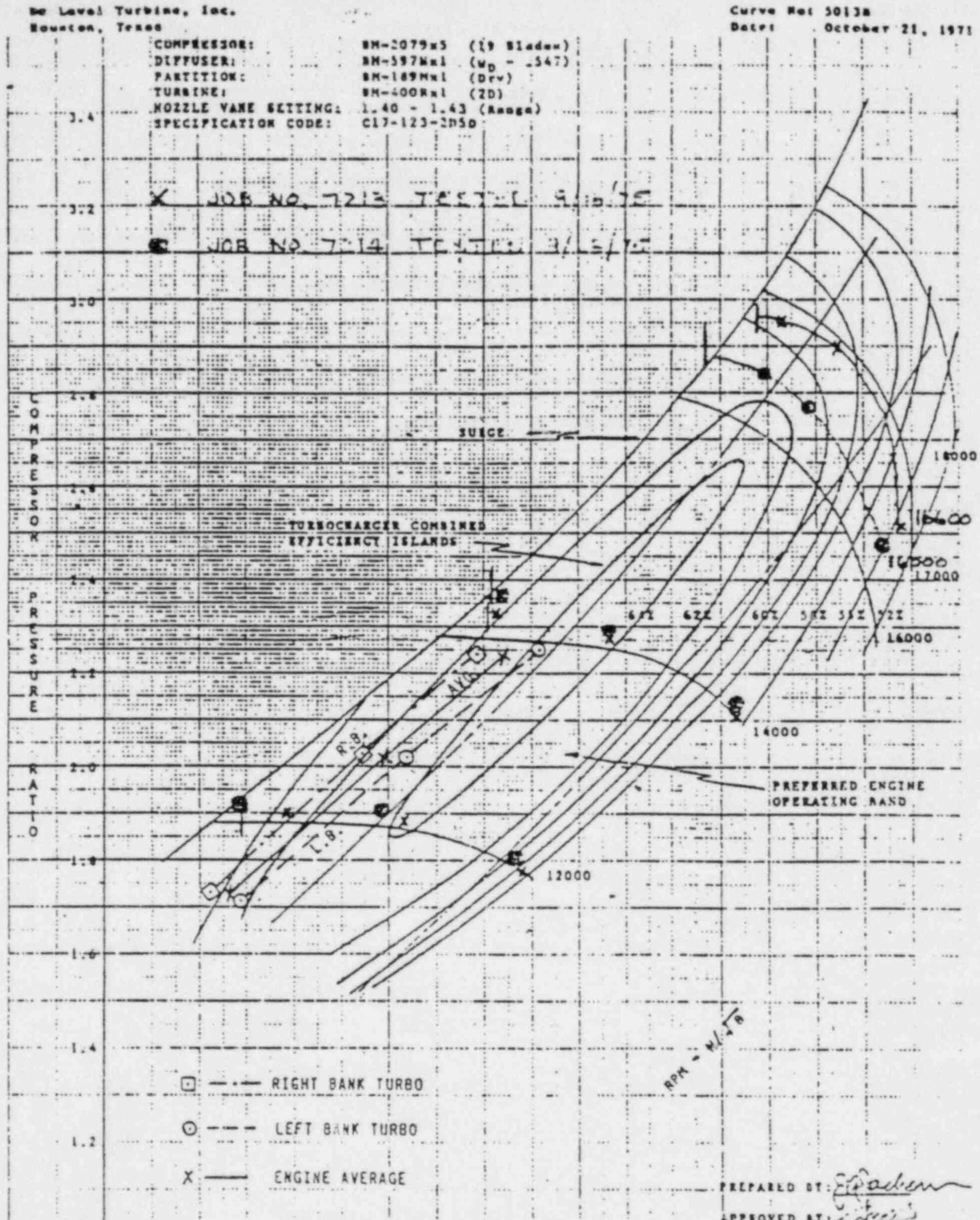
<u>Air Delivered</u>: Table 2.6 presents the results of turbo and engine air and gas computations which quantify the observed air flows delivered by the new DE C-17-123 turbos. Data, formulae and sample calculations are contained in Appendices B and E.

TABLE 2.6 TURBO AIR FLOW CALCULATIONS RESULTS					
ERPM	AVG. TURBO RPM	ENGINE A/F RATIO	AVG. COMPRESSOR PRESSURE RATIO	AVG. TOTAL ACFM PER TURBO	AVG. TOTAL SCFM PER TURBO
363	12,240	28.67	1.725	7100.8	6771.7
385	13,608	30.54	2.03	8956.5	8483.3
403	15,100	30.71	2.245	10273.4	9747.3

The results of this tabulation have also been presented graphically in Figure 2.6 in which corrected air flows in SFCM have been plotted versus compressor (boost) pressure ratio for the average values shown in Table 2.6 and for the individual right and left bank blower outputs. The correction from actual to standard flow (SCFM) was made to take into account the compressor inlet temperature and pressure difference between the conditions observed on the vessel and the design standards on which the unit's design performance is based, as shown in Figure 2.6. The data, as plotted in Figure 2.6, would indicate that the compressors are operating considerably closer to the predicted surge line than would be desired as shown by the relation of the lines of observed performance which fall to the outside

Figure 2.6

DE-C-17-123 Turbo Performance, 3/25/83 M.V. Columbia



and to the left of the preferred engine operating band also shown on Figure 2.6. The right bank blower pressure ratio/total air flow curve is closer to the predicted surge line than the left bank plot. Subsequent to the sea trials and during initial voyages, a turbo on the starboard engine was observed going into surge. It is most likely that this is the same turbo identified as the right bank turbo by the sea trial data, as indicated by this unit's plotted performance falling closest to the theoretical surge line in Figure 2.6. Therefore, it must be concluded from this data that the turbos as supplied by TDI are not properly matched to the engines' new de-rated output.

Another observation on sea trials relative to the turbos and verified during data reduction is the disagreement in plotted flows and pressure ratios in Figure 2.6 and apparent turbo RPM. The turbo RPM's logged in Appendix B by TDI, even after correcting for observed temperature, do not correlate at all with predicted RPM's on Figure 2.6. At the time of trials there was some question as to the accuracy of the turbo tachometers supplied as part of the de-rating workscope. (Two (2) tachometers failed during the trials.) It would appear that the turbo tachometer readings are in error.

Combined Efficiency: In an attempt to provide an additional correlative data point for the turbo compressor plots shown in Figure 2.6, a combined turbo/compressor efficiency was computed for each ERPM test point. The results are presented below in Table 2.7. Data, formulae and sample calculations are contained in Appendices B, F and G.

TABLE 2.7

COMPARISON OF COMPUTED AND TDI PREDICTED COMBINED TURBO/COMPRESSOR EFFICIENCY

ERPM	LEFT BANK TC (Calc[1]/Pred[2]/Δ,%[3])	RIGHT BANK TC (Calc/Pred/Δ,%)	AVERAGE TC (Calc/Pred/Δ,%)
363	60.87%/61.5%/1.02%	58.39%/60.0%/2.7%	59.63%/60.75%/1.86%
385	62.55%/63%/.55%	61.27%/61.8%/.85%	61.96%/62.4%/.7%
403	63.60%/64%/.625%	61.11%/62.2%/1.75%	62.36%/63.1%/1.19%

1 Calculated from test data
2 From Figure 2.6
3 $\Delta,\% = \frac{(\text{Pred-Calc})}{(\text{Pred})}(100)$

The results presented in Table 2.7 show a very good correlation between computed values of combined turbocharger efficiency and predicted efficiency based on the operating lines plotted for the right, left and average turbocharger values of compressor pressure ratio and corrected air flows in Figure 2.6. Due to the lack of accurate turbo RPM values, these data become significant in that they provide a well established third reference point which supports the location of the turbo operating lines, as plotted in Figure 2.6, closer than would be desired to the theoretical surge line for the DE C-17-123 compressors.

<u>Comparison of Other Performance Data</u>: Other engine and turbo data was reviewed and compared with starboard engine performance with the original Elliot turbochargers at similar loads. The results of these investigations which also indicate that a significant increase in airflow has occurred, are summarized below.

* Cylinder Exhaust Temperatures: A comparison of pre and post DE turbo installation cylinder temperatures, based on the March 25, 1983 sea trial data and data contained in Reference No. 3, indicates average temperature reductions in the range of 75 to 125°F per cylinder at similar loads.

* Charge Air/Exhaust Manifold Pressure Differentials: In a gross sense, if the engine is considered as an orifice, then the pressure drop across the engine from charge air to exhaust manifold is approximately indicative of air flow through the engine. This pressure differential, after installation of the DE turbos, increased by as much as 7.5 times at similar engine loads.

* Firing Pressures: A comparison of the March 25, 1983 sea trial data and similar information from Reference No. 3 shows little or no change in peak cyclinder firing pressure and continued unbalance from cylinder to cylinder. The TDI representatives onboard at the time indicated that these were lower than anticipated and that correction of this problem by advancing the fuel injection timing and balancing the cylinder pressures in the starboard engine would likely improve overall operating efficiency.

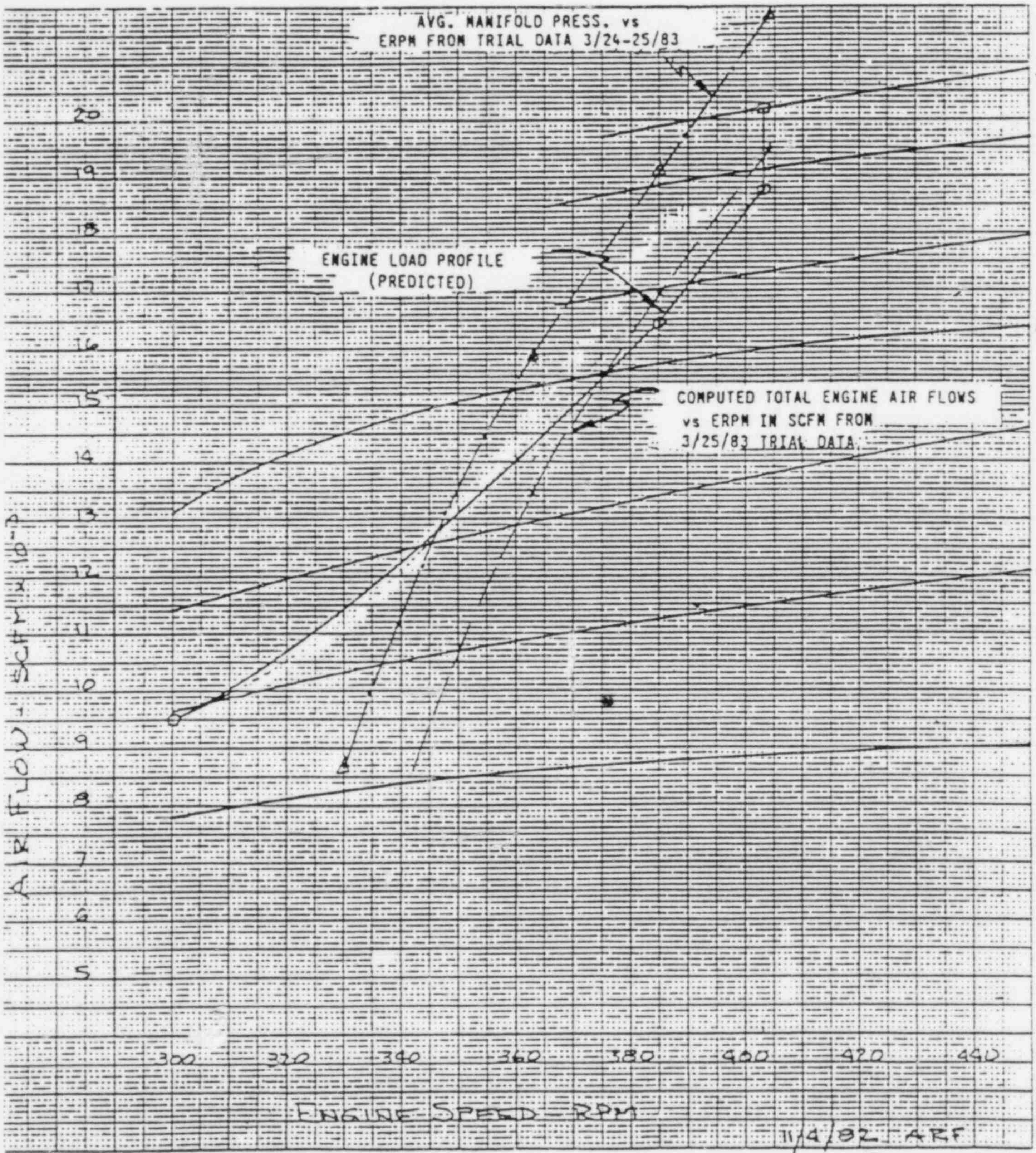
* Charge Air Manifold Pressures: Figure 2.7 presents a plot of anticipated charge air manifold pressures versus ERPM provided by TDI for the new turbos. Overlaid on this graph are additional curves which plot actual manifold pressure observed versus ERPM (dash-dot line) and computed total engine air flows in SCFM versus ERPM (dashed line). The

Figure 2.7

Predicted and Observed Air Flow and Manifold Pressure

vs

ERPM for DE -C-17-123 Turbos on M.V. Columbia



predicted air flows corresponding to the observed manifold pressures are significantly higher than the computed values shown. Referring to Figure 2.6, this would have been the case had the observed operating line for the turbos fallen within the preferred engine operating envelope with correspondingly higher turbo efficiencies. Also, in the case of both curves plotted from the March 25, 1983 sea trial data in Figure 2.7, more total air flow at 385 and 403 ERPM's is indicated than from the TDI plot of predicted performance. However, both plots of the observed data indicate that air flow from the turbos appears to fall off much more rapidly than predicted at lower engine loads. This performance may account for the observed surging during trials after rapid application and removal of propeller pitch (engine load) during response testing.

Turbo Response: On March 25, 1983, brief qualatative tests of turbo response to rapidly increasing and decreasing engine load commands were conducted. These consisted primarily of bridge control initiated crash astern and crash ahead maneuvers. On one such maneuver, the port engine stalled and dropped off the line completely. At various times under severe load application or removal, all turbos were heard squealing or barking back. Some squealing, indicative of turbo surge, was also noted during steady state operation at the 385 ERPM test point. Additionally, a very high pitched noise was also determined as eminating from the discharge side of the turbos during the 385 ERPM test run. It is speculated that this may be the result of a harmonic or resonant frequency condition for the turbos occurring at this engine speed, as it seemed to decrease when the engine was operated above or below this point. It was also noted, especially by those familiar with COLUMBIA's past response characteristics, that the current load control program added pitch to the propellers (increased engine load) at a rate much higher than ever noted previously. At the time it was felt that the rapid pitch application by the control was the major causitive problem for the engine/turbo response difficulties previously described.

2.3 Post Trial Performance

Throughout the report preparation period, and up to April 18, 1983, Seaworthy has been made aware of various problems and conditions in the COLUMBIA after entering service on April 1, 1983, which collaborate and expand on much of the trial data and discussions already presented in this section.

Turbo Surging: The frequency of observed turbo surging increased during the initial voyage, primarily on the port engine. Turbo surge is defined simplistically as the range of unstable operation which occurs when air flow through the compressor is reduced while the compressor pressure ratio (pressure at discharge divided by pressure at the suction) remains constant, shifting the operating point on the compressor map (Figure 2.6) to the left of the surge line. In severe cases, a flow reversal in the compressor may occur. Turbo surging results in unstable engine operation, air starvation, poor combustion and reduction or fluctuations in engine speed and power. Surging can also cause mechanical damage to the compressor as a result of increased mechanical stresses which occur during surge. To relieve this situation, pitch (and ERPM) were reduced. Based on the engine operating lines plotted in Figure 2.6, the problem of continued and more frequent surging is not surprising given the closeness of the engine operating line to the theoretical surge line. Also, various changes in actual ambient conditions such as temperature, barometric pressure, humidity, intake filter cleanliness, etc. can cause the surge line for the turbo to shift further to the right, encroaching even more on the actual engine operating line. This is further supported by the fact that hard ship turns also caused the turbos to go into surge, giving additional credence to the closeness of the surge line to the engine operating line. However, difficulties with the load control portion of the engine control system may have also contributed to this situation.

Another contributing factor is the match of the DE-17-123 turbos capable of an output that would satisfy the air requirements of the original 9200 BHP rating of the engine. If, in fact, these are the same units in terms of capacity, they have ended up operating in a situation depicted by Figures 2.8 and 2.9, taken from Reference No. 5. Figure 2.8 shows the speed, pitch and power relationship for a generic four-cycle engine fitted with a CRP wheel. Extreme pitch seating is applied from point

Figure 2.8

Propeller Law Power Curves with a Controllable Pitch Propeller

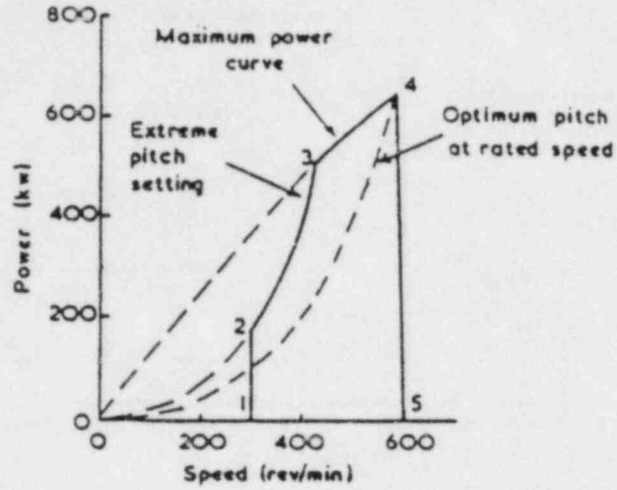
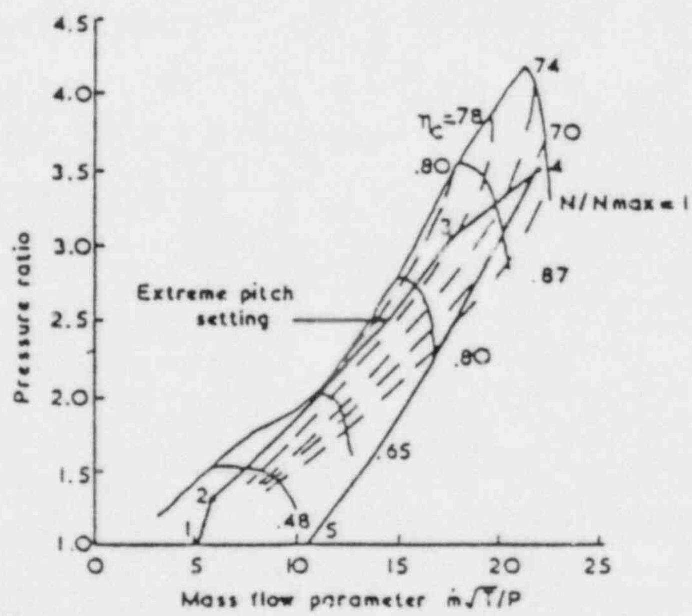


Figure 2.9

Compressor Matching with a Controllable Pitch Propeller



(2) to (3), building up to the maximum power portion of the curve from (3) to (4) as speed is increased. If this same pitch curve is overlaid on a compressor map for a unit matched for the engine's maximum output, as shown in Figure 2.9, the following can occur. Operation at maximum engine output, points (3) to (4), places the compressor well away from surge and close to maximum turbo efficiency. However, as load is reduced, essentially by lowering speed, while maintaining a maximum pitch setting, points (3) to (2), the extreme pitch setting line comes very close to the surge line. This situation is further aggrevated by the "waist" or dip in the surge line characteristic of operating a highly rated turbo at lower outputs, as shown. Thus, for an engine fitted with a CRP, it is the extreme pitch setting curve and not maximum engine/propeller speed which determines the surge margin and related matching requirements. Referring back to the data plotted in Figure 2.6, extrapolation of these operating lines to a higher compressor output shows the slope carrying them into a more stable (further from surge) and efficient area on the compressor map. As indicated by the dotted lines of decreasing pitch setting in Figure 2.9, a reduction in pitch setting will move the engine curve away from surge which is exactly the experience reported on the initial voyage of the COLUMBIA.

As a final point requiring clarification by TDI, relative to turbo surge, it is noted that an increase in charge air manifold temperatures up to 50°F was desired by TDI to improve the combustion process at the lower operating outputs for the de-rated engines. It appears that this has been partially achieved. However, a review of test stand data for a similar DMRV-16-4 engine fitted with DE C-17-123 turbos (Reference No. 4) indicated that on two occasions, at outputs of 6027 BHP/300 ERPM and 8450 BHP/390 ERPM, charge air manifold temperatures were reduced from 150 $^{\circ}$F to 125°F to eliminate turbo surge. Given the current surge problem and the test stand data, it would appear that the desired increase in charge air manifold temperature for improved combustion quality is a possible contributory cause of turbo surging. As a minimum, reduction in charge air temperature to reduce surging, if viable, cannot be accomplished without some negative impact on the low load combustion process.

Trabon System: At the time of report writing it was understood that while the system was operational, certain components required for proper system function, including a micro-switch, had failed. Proper dosage rates and frequencies had also not been provided.

Structural Items: Difficulties in this area centered around leaks in the compressor discharge transition piece/inter-cooler plenimum, specifically on the port engine outboard turbo. TDI had admitted that these structures have caused considerable problems as a result of cracking and leaking in similar applications.

Waste Heat Recovery System: It was reported that during the initial voyage the oil-fired boiler operated continuously as a supplement to the waste heat boilers' steam output for auxiliary and hotel loads. This was not the case prior to de-rating.

The short fall of waste heat generated steam results from a combination of factors. Operating the engines at a lower output will reduce exhaust gas mass flow, although this is offset somewhat by improved turbo air delivery. More significantly, the cylinder exhaust and turbo exhaust temperatures have been substantially reduced. Thus, each pound of exhaust gas carries less heat with it up the stack to be recouped in the waste heat boiler. Because the exhaust flow after the turbo on each engine splits and flows through a silencer/spark arrester and a waste heat boiler, this situation can be remedied to some degree by diverting a greater flow of exhaust gas through the boiler by restricting flow through the silencer on each engine.

Cooling System: The increased jacket water temperatures desired by TDI to enhance part load or de-rated engine performance, has not been obtained. Operation in colder air and sea temperatures in Alaskan waters on the first voyage of the season resulted in a reduction in charge air temperatures to 145°F versus the 155°F values observed during the trial. The automatic temperature (AMOT) control valves were noted as functioning and closed at this time.

Control System: Mathers Controls has been working steadily on resolving the pitch control program difficulties as reported during the sea trials and subsequent voyages. The rapid application of pitch has surely aggrevated the surging and response problems observed to date. Conversely, had TDI's predicted performance, relative to power output, fuel rack and engine RPM more closely conformed to what was actually obtained after de-rating and had the turbo matching been further from the surge region, the control difficulties presently being experienced would probably not have been as severe or as persistent.

3.0 HISTORICAL REVIEW AND ANALYSIS OF ENGINE COMPONENT FAILURES

3.1 Introduction

An indepth investigation of the M/V COLUMBIA's engines' operating, maintenance and repair history was performed. The primary objectives of this investigation were:

- The identification and tabulation of significant engine component failures
- The tabulation of major maintenance actions performed to either prevent catastrophic failures or to maintain the engine in an operational condition to meet vessel schedule requirements.
- The identification of the causes of the failures and excessive maintenance requirements and of the resultant corrective actions taken either by the owner, if any, or TDI.

The results of this investigation are presented in the following sections with detailed supporting technical data.

3.1.1. Data Sources

The following listing identifies the data sources used:

- Chief Engineer's voyage reports
- Port Engineer's weekly reports, memos and correspondence files
- Ship's Logs
- American Bureau of Shipping (ABS) surveys and files
- U.S.C.G. Reports
- State of Alaska, Department of Transportation files
- Transamerica DeLaval, Inc. (TDI) reports, correspondence and invoices

Metallurgical Reports

Lubricant Analysis Reports

Consultants' Reports and Analyses

The documents reviewed approximated 10,000 pages of data.

3.1.2. Time Period

The engines' history was analyzed from the time of delivery, June 1974 until the 1982/83 overhaul, ending March 25, 1983. The pre-delivery, shipyard engine tear-down was not included in the historical review.

3.1.3. Chronological Methodology

The methodology used in collecting, tabulating and summarizing the data was consistent with the manner in which the Alaska Marine Highway System (AMHS) keeps their files and records. Specifically, time frames were categorized as either warm weather "operational" periods, or "overhaul" periods when more work is typically performed. All data was evaluated on a chronological basis. The details of the chronological investigation are presented in Appendix B.

3.2 Maintenance History Tabulations

Tables 3.1A and 3.1B provide final summaries of the data collected and analyzed. Table 3.1A provides a totalization of all significant maintenance occurrences by engine component over the operating life of the engines to date - 30,000 hours. This table is set up to provide a direct comparison of actual component life between corrective maintenance actions and the scheduled or

TABLE 3.1A

SUMMARY M/V COLUMBIA
ENTERPRISE DMRV-16-4 MAINTENANCE/FAILURE HISTORY - 30,000 HRS/PER ENGINE

COMPONENT	CORRECTIVE MAINTENANCE				COMPONENT LIFE				Remarks
	Total Removed or No. of Occurrences for Corrective Maintenance	Total or Average Hours Between Occurrences	Projected IDI Min. Corrective Maintenance Time	Typical Medium Speed Diesel Manufacturers' TBO	Total Units Scrapped or Unuseable	Average Hours of Component Life	IDI Estimated Component Life	Typical Medium Speed Diesel Manufacturers' Estimated Component Life	
CYLINDER HEADS	287	2,900	8,000	12,000	20	13,000		N/A	No. of Rebuilds Unknown
CYLINDER LINERS	138	5,400	24,000	20,000	20	19,600		100,000	
PISTONS	149	4,390	24,000	20,000				100,000	
PISTON RINGS (SETS)	142	5,200	24,000	20,000	142	5,200		20,000	
MASTER & LINK CONNECTING ROD	97	4,390			7	16,360		Engine Life	
MASTER ROD BEARING REPLACEMENT	50	8,035	No Listing	30,000	50	8,035	NOT AVAILABLE	60,000	
CAM SHAFT					4	24,000		No Listing	
MAIN BEARINGS	ALL	16,000	24,000	20,000				45,000	Bearings Rolled Top to Bottom
MAJOR OVERHAULS	4	6,025	24,000	20,000					Head, Piston, Liner Bearing, Simultaneous Removal, Rebuild, Repair
TURBOCHARGER	40	2,340	8,000	12,000	16*	2,830			*Bearings, Castings Seals
TURBOCHARGER OVERHAUL	40	2,480	8,000	12,000				20,000**	**Bearings

TABLE 3.1B

SUMMARY M/V COLUMBIA
DOCUMENTED COMPONENT FAILURE MODES

ENTERPRISE DMRV-16-4 - 30,000 HRS/PER ENGINE

COMPONENT	Cause or Occurrence	No. of Occurrences for Corrective Maintenance	Total or Average Hours Between Occurrences	Remarks
CYLINDER HEADS	Cracked	58	3,200	Rebuildable
	Manufacturing Defects	24	3,000	Rebuildable
	Warped Head - Fire Ring Failure	77	2,890	
PISTONS	Corrective Maintenance or Modifications to Crown/Skirt	92	4,870	
MASTER & LINK CONNECTING ROD	Articulated Bearing/Pin	40	10,080	
	Fastener Mechanism	49	6,475	

expected intervals specified by the engine manufacturer, TDI, in their maintenance handbook. Additionally, data is provided for other typical medium speed engines' maintenance intervals and parts life. This data is provided to supplement TDI information in certain areas where it was lacking and to provide additional appropriate comparative data. The information presented is based on typical medium speed diesels of equal or higher power and speeds, operating on MDO.

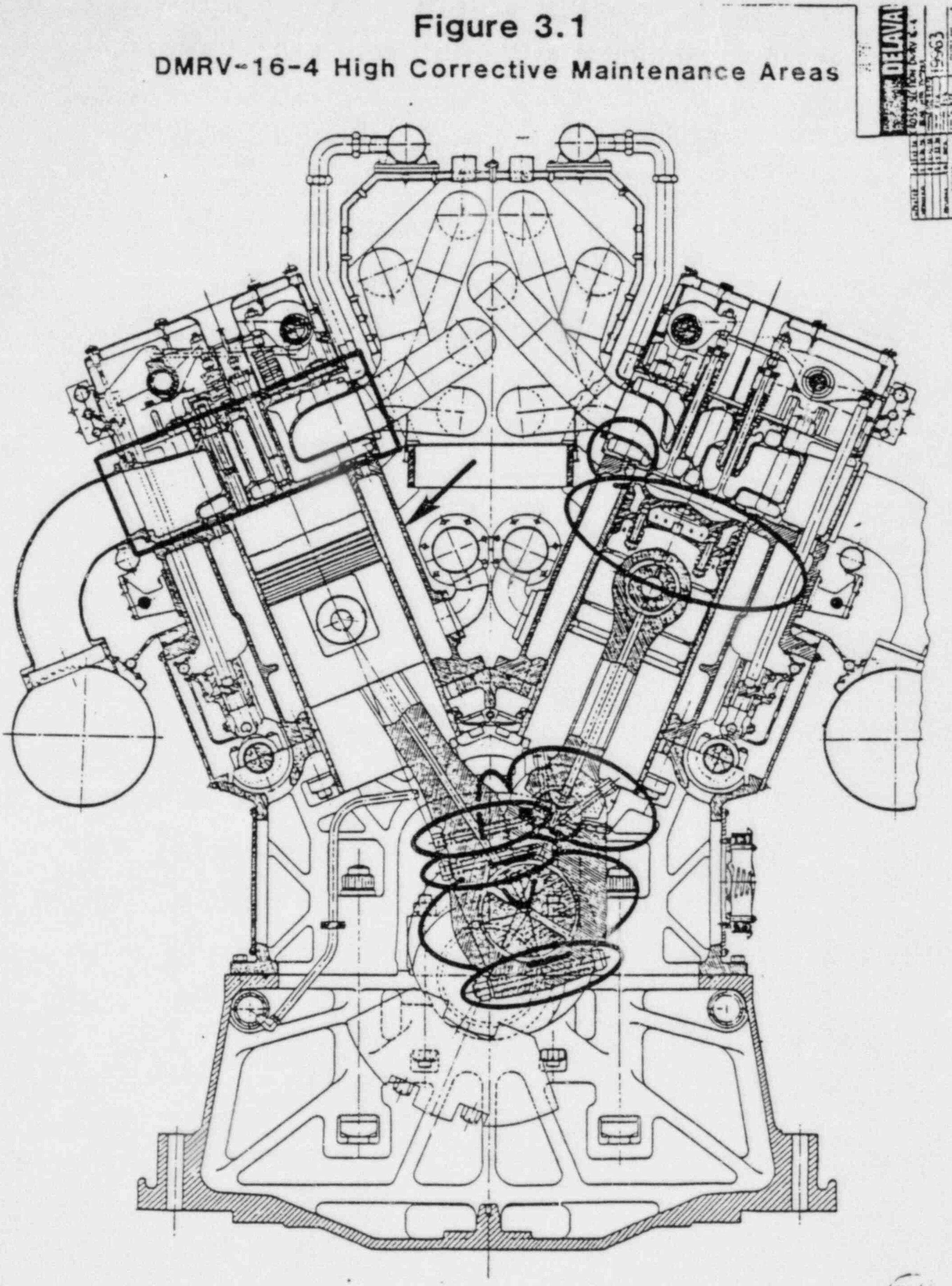
Finally, Table 3.1A identifies the number of components scrapped and the average life of that component. Again, a comparison of actual average component life to expected component life can be readily made.

Table 3.1B provides a final summary of the causes for the component corrective maintenance actions present in Table 3.1A. This table identifies the documented causes which were obtained as a result of the detailed data investigation. As can be seen, there are obvious differences in the values between the total occurrences (all causes) and the documented cause or reason totals. This is essentially due to the absence of detailed, extensive and accurate record keeping and documentation practices of the operator.

Figure 3.1 provides an illustration of the engine areas experiencing recurring failures and inordinately high maintenance actions. These areas are identified by dark outlining.

Figure 3.1

DMRV-16-4 High Corrective Maintenance Areas



3.3 Summary of Maintenance/Failure History

The following section refers to Tables 3.1A and 3.1B and provides the narrative and analyzation of the data presented. It should be noted that component design is not analyzed, instead the results of the existing engine design and its impact on component life is presented.

3.3.1 Cylinder Heads

Cylinder head removal and failure rate are very high. Numerically, 287 heads were removed for corrective maintenance with an average time between removals of 2900 hours as shown in Table 3.1A.

This equates to every head on both of the engines (32 heads) being changed nine (9) times during their operating life to date. Comparing this to a TDI suggested reconditioning cycle of 8000 hours per head, or approximately four (4) times in 32,000 hours of engine operation, this means that the heads have been removed in excess of twice the scheduled maintenance frequency.

The reason for the head removals were varied, as is shown in Table 3.1B. However, the types of failures could generally be described as being integral to the head and its construction and/or reflective of the head materials. The use of cast steel for a head material gives the manufacturer a superior material relative to the mechanical and metallurgical properties, particularly where the manufacturer uses a welding deposition technique (hard facing) for the valve seats. However, the detrimental feature of cast steel is the poorer castability of cast steel (versus cast iron) and the requirement for different and more closely controlled foundry casting techniques. The results of some of these casting problems have been representative of the types of failures observed in the cylinder heads. Specific failures are head cracking and fire deck warping from high stress areas, and porosity from gas and contaminant inclusions. Additional casting technique problems which have been observed in the heads have been core shifting which has resulted in thin cross-sections and misaligned cooling passages.

Two additional problems which have plagued the head construction and interface areas are the exhaust valve guides and head warping along the 3-9 o'clock axis. In the case of the valve guide problems, insufficient documentation was available to reflect the number of occurrences chargeable to guide failures or valve guide induced failures such as carbon build-up on the valve stems which resulted in stuck valves or guide damage. The 1975, '75 and '76 files contain reports of pieces of valve guides breaking off and causing foreign object damaged (FOD) to turbochargers but insufficient numerical data has resulted in this type of failure being omitted from the historical summary. However, the head/valve guide area has been subject to continuous modifications starting with the 1976/77 overhaul when all the guides were machined flush with the exhaust gas passage and continuing to the 1982/83 overhaul when additional valve guide length was removed and a valve guide oiling/sealing (Trabon) system was added to control the rocker box sooting problem.

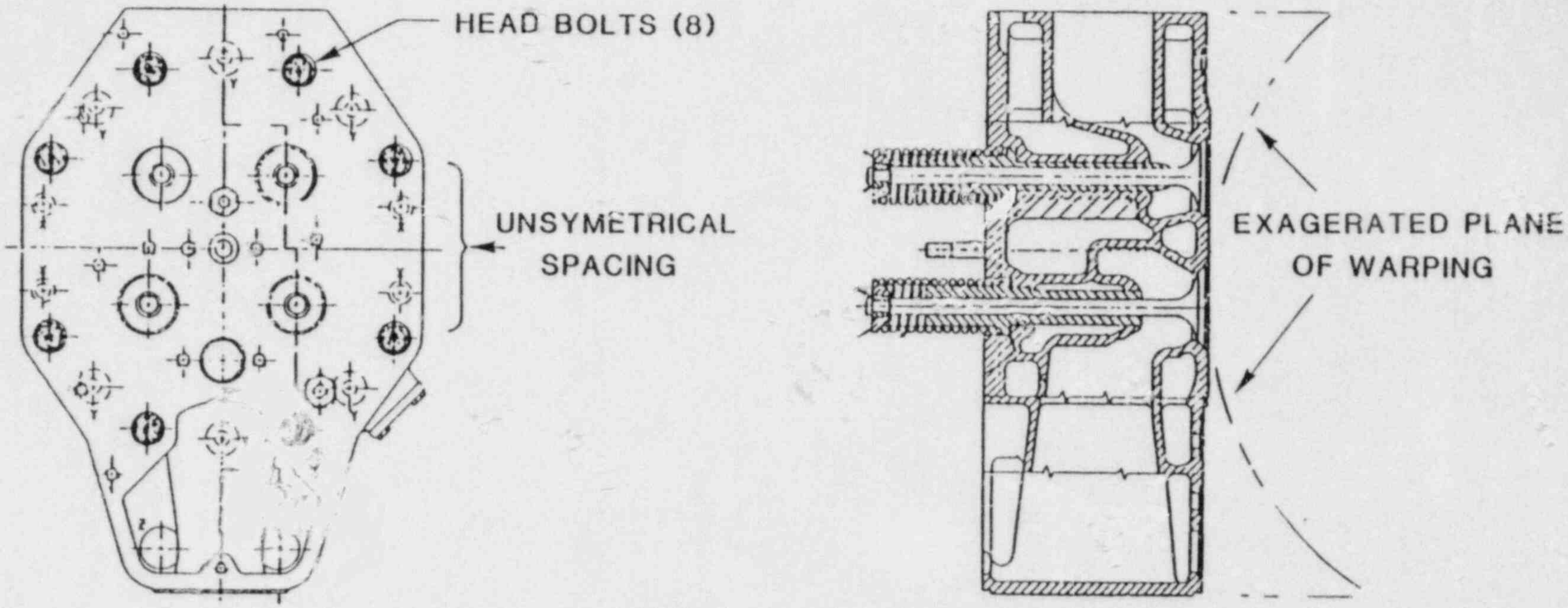


Figure 3.2

DMRV-16-4 Cylinder Head and Valves

The head warping problem caused two secondary modes of failure. One mode is the fire deck warping which has resulted in internal cracking while the other is excessive metal removal during head reconditioning which has resulted in a shortened head life. The second mode of failure due to head warping is the premature unloading of the fire ring gasket which results in the 3-9 o'clock fire ring burn out. Reference to Table 3.1B lists 77 head warping/fire ring failures which does not coincide with an observation made by the Chief Engineer (M/V COLUMBIA) where he estimated that 70-80% of the heads removed showed fire ring distress (brown streaking in 3-9 o'clock) or fire ring failure (black streaking in 3-9 o'clock position). TDI has attributed the cause for this type of failure to be the unsymmetrical head bolting pattern around the 3-9 o'clock axis, as illustrated in Figure 3.2. This unsymmetrical pattern results because of the nearness of the adjoining heads which does not physically allow a head bolt to be placed on a regularly spaced circumferential bolting pattern. The subsequent bolt tightening results in a bending moment to be formed (or hogging) perpendicular to the 3-9 o'clock axis. TDI has reinforced the interior head area perpendicular to the 3-9 o'clock axis with a "strong-back". Heads with this design modification are presently in service for a total of approximately 6500 hours with reported failures of three (3) heads in that period. The scrapping of twenty (20) heads with an average life of 13,300 hours represents a high rate of failure when it is considered that this represents 62% of the total heads in service. The reasons for scrapping have been thin fire decks (due to repeated machining cuts during reconditioning and/or shifted casting cores) or unrepairable interior cracks or porosities or cracks between valve seats, or non-concentric valve stem to valve seat diameters. Some of the heads have been scrapped during TDI factory reconditioning due to valve bridge cracking during valve seat welding deposition.

The data presented in Table 3.1A indicates a high failure rate for reasons of both design and material selection.

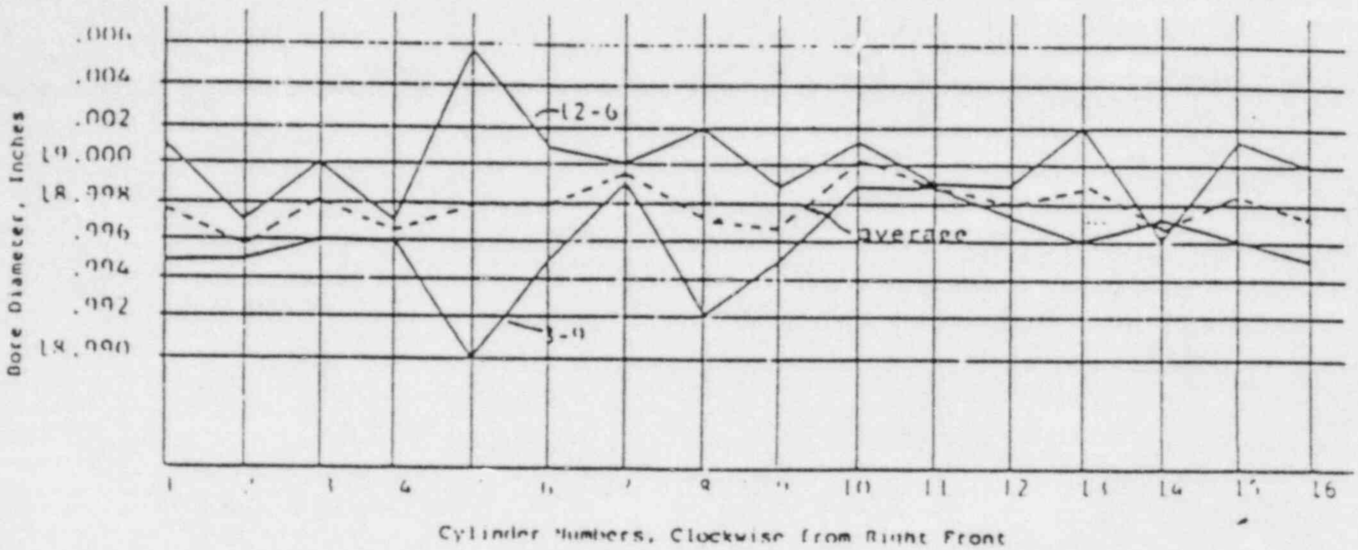
3.3.2 Cylinder Liners

The cylinder liner removal and failure rate is very high. The most common reason for liner removal is the necessity for honing to restore liner roundness or surface quality when piston rings were changed. Other reasons for liner removal are attributed to the lower liner to block seal failures which occurred during the first two (2) years of service. Table 3.1A lists 138 liner removals for corrective maintenance with an average time between removals of 5400 hours.

The failure mechanism may be attributed to several coincidental factors. The first factor is gauling and scoring of the liners due to embedded materials between the moving surfaces. In some cases this has been from foreign matter, or chrome from the ring surfaces that has been spaulled or flaked from the compression ring wearing surface which has become embedded in the piston crown, other rings or the piston skirt with the resultant scoring. The second factor is the premature wear caused by the unburned

Figure 3.3

Permanent Liner Deformation, Bore Diameter



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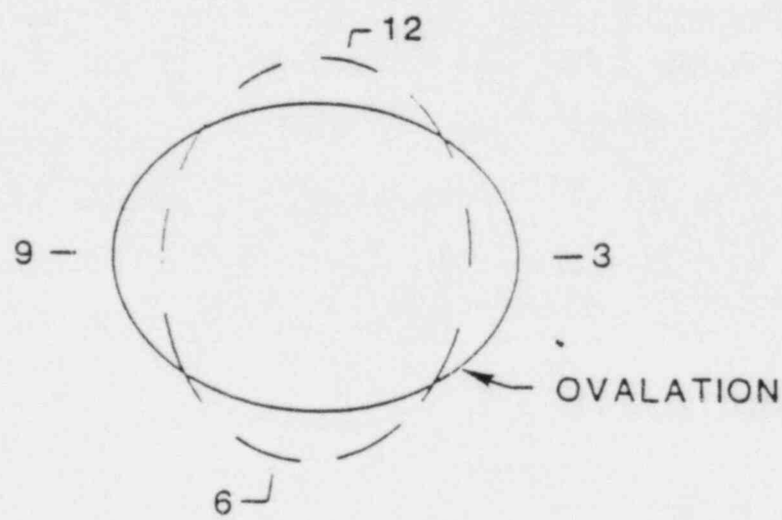
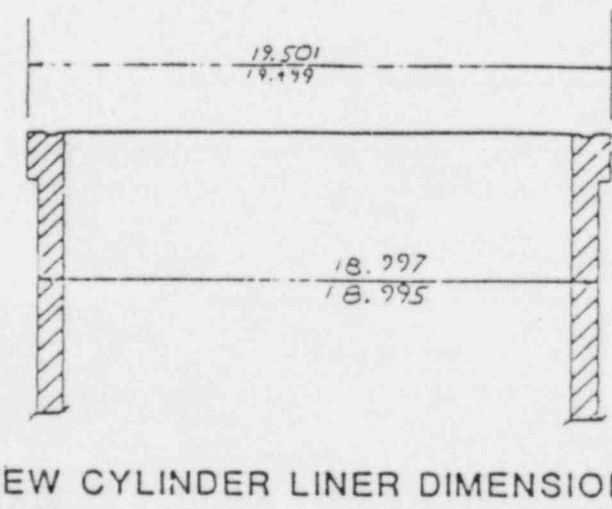




NEW CYLINDER LINER DIMENSIONS

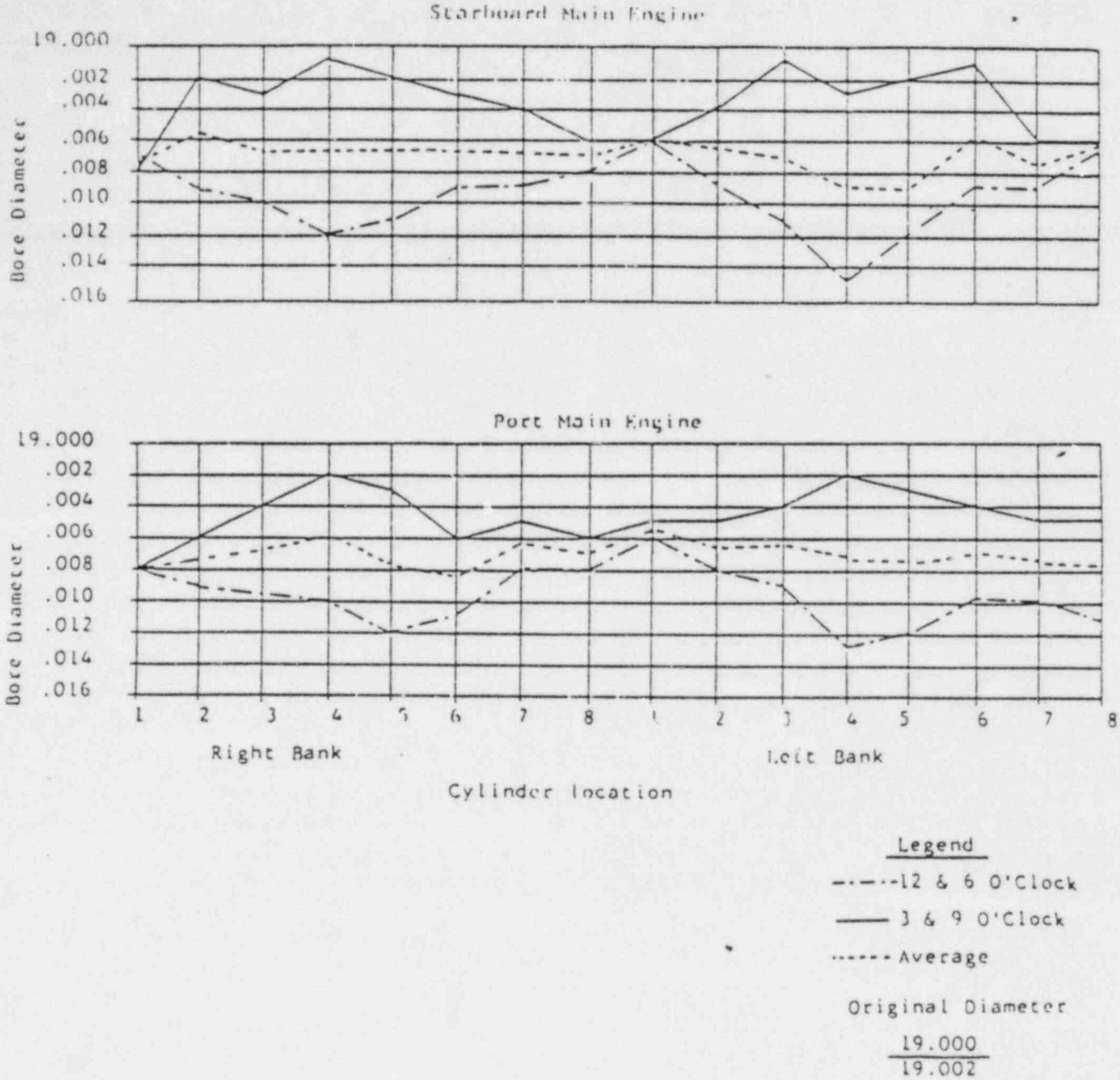


Figure 3.4

Bore Diameter, Engine Blocks M/V Columbia

(From Fig. 3 Jon Jacobson Report, March 31, 1981.)

carbon due to the incomplete combustion in the cylinder. Here, many factors are at work, including raw fuel impingement on the cylinder walls, abrasive carbon wear between the moving surfaces and potential hot spots from the partial combustion process. Improvement of the combustion process should help to extend the cylinder life due to reduced carbon generation and abrasion.

Another persistent type of failure attributing to a premature liner scrapping has occurred as the result of liner ovalation as reported in Reference No. 6 and No. 7 during the 1980/81 overhaul. In this instance liner deformation has been observed as a direct result of block deformation. That is, the liner ovalates to an increasing dimension in the 6-12 o'clock position (athwart ships) with a decreasing dimension in the 3-9 o'clock position (fore/aft) where the liner is clamped in the counter-bored block lip area. The observed measurements from Reference No. 6 is reproduced in Figure 3.3 to graphically illustrate the observed change in the liner dimensions. The result of this liner deformation is the ensuing ring/piston wear distortion and premature liner wear that occurs as the moving parts try to conform to the dimension changes. The magnitude of this problem is more graphically presented in Figure 3.4 from Reference No. 6, showing the block deformation which ultimately deforms the liners.

Of particular importance is the repeatable dimensional change in cylinder number 4 (mid block area) for both engine blocks in the port and starboard engines. The short liner life of 19,600 hours, as opposed to a projected 50,000-100,000 hours, could be attributed to the high wear rate that has been accelerated by incomplete combustion and by the mechanical forces that cause the cylinder to ovalate.

3.3.3 Pistons

The number of pistons removed (149) has been influenced by some of the other component corrective maintenance actions, such as worn liners and failed connecting rods. However, there were several impending failures of bolting mechanisms and crown to skirt oil seals that, upon piston removal, were detected and corrected before a catastrophic casualty occurred. Reference to Table 3.1B lists ninety-two (92) piston removals specifically for maintenance or modifications to the piston crowns or skirts. The type of modifications made to the piston consisted of decreasing crown diameter, modifying lube oil passages and seals, and machining modifications to ring grooves and piston skirts. Modifications of this nature are often considered a product improvement, but in many cases are really design corrections.

Piston crown fastener problems have been observed at various intervals. Records indicate that several crown to skirt bolts have broken, or in the case of several overhauls, these same bolts have been found loosened from the specified torque level. This problem continues to manifest itself by the observation of fretting (metal-to-metal movement and wear) under the bolted surfaces and bolt washers. The fact that this occurs indicates that there is surface movement under high stress conditions.

Piston rings have required frequent change out and scrapping due to accelerated wear and, in the case of the compression rings, due to chrome overlay chipping or flaking from the surface which has become embedded in the crown, other rings, the tining of the piston skirt or in the liner. Where these flakes have been embedded for some time, deep scoring usually results, as shown in Table 3.1A. One hundred forty-two (142) piston ring sets have been replaced on the thirty-two pistons over the 30,000 hours of running time. This equates to approximately four (4) renewals of rings during a time interval when only one renewal should have been necessary. An improvement in cylinder combustion should decrease this frequency as outlined in Section 3.3.2, "Cylinder Liners".

3.3.4 Master and Link Connecting Rods

The articulated connecting rod has been the location of failures which could have been catastrophic if the cracked rods had not been discovered when they were. Prior to 1979, there had been one failure by cracking of a connecting rod, coupled with a link rod bearing failure. This same type of failure was later discovered at approximately 21,000 hours of operation in the Fall of 1979. At that time, approximately 25% of the link rod boxes were found fractured in the link pin area between the link pin bushing and the serrated joint. Figure 3.5 shows the orientation of two typical crack propagations (crack A & B) which were found in two (2) separate rod boxes. These fractures occurred in high stress areas in a high cycle fatigue mode, as illustrated, following.

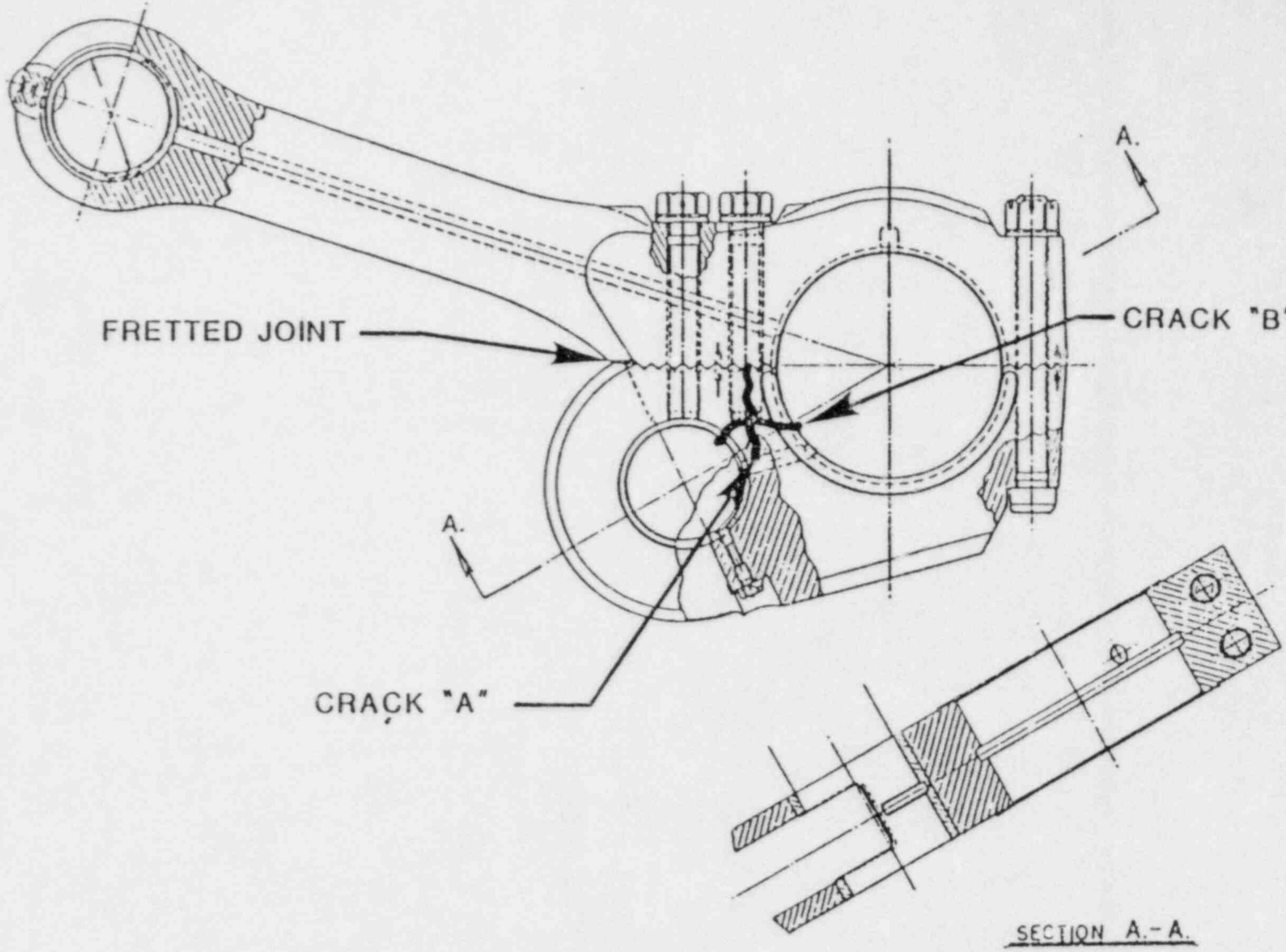


Figure 3.5

Master Rod and Connecting Rod Box Assembly

Assume:

$$\text{Average Engine Speed}_{Max} = 380 \text{ Rev/Min}$$

Therefore:

At 21,000 Hours

$$= 21{,}000 \text{ HRS} \times 60 \frac{\text{Min}}{\text{HR}} \times 380 \text{ Rev/Min} \times 1 \frac{\text{Cycle}}{\text{Rev}}$$

$$= 4.8 \times 10^8 \text{ Cycles}$$

Which, by definition, is above low cycle fatigue 10^4 cycles and below infinite life where cycles exceed 10^9 cycles.

The high stress area was reduced by increasing the cross-sectional area when TDI decreased the connecting rod bolt size from 1-7/8" to 1-1/2" and changed bolt configuration and materials. Additional modifications were made to the link rod box external contours by increasing radii to decrease stress concentration areas.

The rod box has exhibited other signs of distress in the link pin bushing. This has been addressed by a change in bushing materials. The rod box also houses part of the connecting rod bearings which have been subject to failure. The 1979/80 engine overhaul disclosed one broken and failed bearing. Upon inspection, many of the other bearings were observed to be showing signs of distress in the form of fretting and carbonized oil deposits. Typically, these observations are associated with excessive temperature and/or high loading. Confirmation of this phenomenon was provided by Northwest Laboratories[17] when a metallurgical examination was made of the No. 6 rod bearing. Table 3.1A lists fifty (50) connecting rod bearing change outs in 30,000 hours of engine operation which represents three (3) complete renewals in a time interval when only one renewal should be expected.

Another area observed to be a point of high loading forces is the serrated joint between master connecting rod and connecting rod box. The serrated surfaces have shown signs of stress in the "V" of the serrations in the form of fretting. This phenomenon is illustrative of metal-against-metal movement under high loading conditions. The only corrective action initiated to date to control this problem, is the action taken by the ship's engineers; wherein, upon component tear-down, they will hand dress and polish these surfaces to effect the best bearing surface possible to distribute the loading. It should be noted that the ship's crew typically will improve the surface finish relative to the machined surface "as received" from the factory. However, even with the care that is exercised by the crew to effect a good load bearing surface, there is still fretting observed upon component disassembly. This would indicate that relative movement may be induced by either a partial relaxation of the bolting forces due to uneven torquing or vibratory forces induced from the cylinder firing loads and/or crankshaft. Additional supporting evidence which indicates that a problem exists in this area is the fretting and gauling observed between the bolt head and washer surfaces, and washer surfaces and connecting rod surfaces. TDI has made washer material changes in an effort to control the fretting; however, subsequent examinations have shown that this problem still exists.

A conclusion which can be reached by the number of failures and the foregoing discussion is that the articulated rod and its components experience complex and highly loaded surfaces due to the various modes of failure and distress that have been observed in both the structural parts and bearing surfaces.

3.3.5 Camshafts

Reference to Table 3.1A lists a total renewal of four camshafts (two (2) per engine) for the engines at 24,000 hours. This numerical figure could be misleading if it is interpreted as a total failure of the camshaft. In this case, a number of cam lobes were worn beyond acceptable limits and renewal of these lobes was necessary. However, due to the design of the camshaft, the cost of a new shaft was less than the repair cost of the old shaft. Althrough this is a design decision made by the manufacturer, it is considered a premature corrective maintenance item relative to the total life expectancy of the component in this application.

3.3.6 Main Bearings

Table 3.1A shows a total bearing replacement at approximately 16,000 hours. The action that was actually taken was to swap the lower main bearing for the top main bearing because the bottom main bearing had worn beyond maximum allowable limits. TDI's preventative maintenance schedule lists 24,000 hours as the first time interval when main bearings should be inspected and replaced, if required. If a comparison is made between other typical medium speed diesels and their anticipated component life, it may be realized that the COLUMBIA's main bearings experienced premature wear.

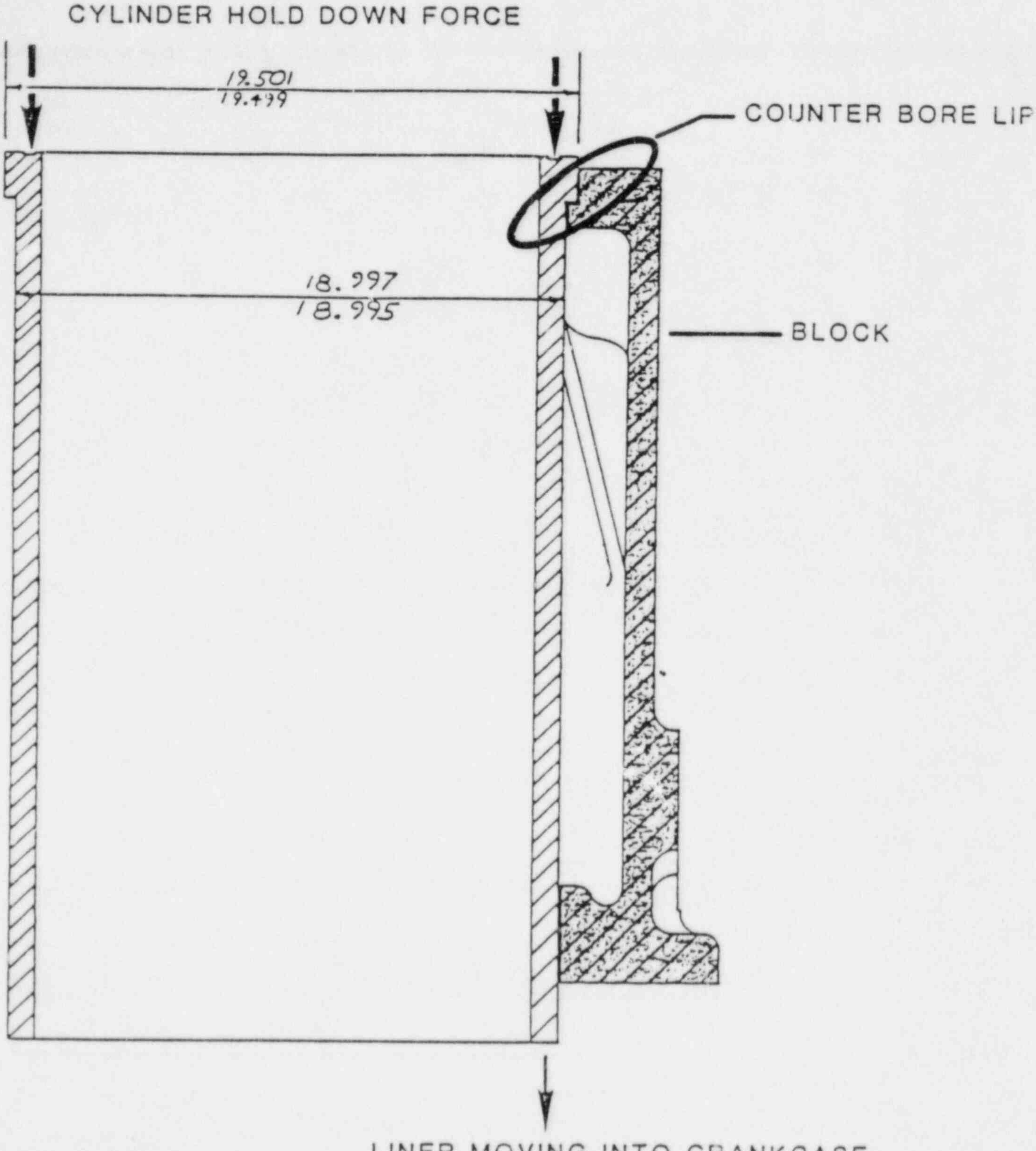
One condition which may have contributed to this is the incomplete combustion experienced with the Elliott turbochargers and the resultant high carbon loading imposed on the lube oil. The high carbon loading in the lube oil was further compounded because of the inability of the lube oil system to continuously purify the lube oil and remove the carbon particles. This is a function of the existing lube oil system design where a single purifier is shared between the two engine lube oil sumps on a rotated basis. The addition of another purifier would permit the lube oil systems to have individual dedicated purifiers, filters and hence, continuous contaminant removal for each engine. This would result in better lube oil quality.

3.3.7 Cylinder Block

Table 3.1A lists the scrapping of four (4) cylinder blocks. The reasons for replacing these blocks were based primarily on two basic documented observations. The following narrative is a summary from the "Engine Rebuild Report", by Jon O. Jacobson, March 31, 1981. The first observation was the deformation or lowering of the cylinder liner block counterbore lip, as illustrated in Figures 3.6 and 3.7. The mechanism by which this was happening is illustrated in Figures 3.8 and 3.9, wherein the counterbore lip was cracking under the high stress of the cylinder head hold down force. Non-destructive testing was employed to determine the extent of the cracki g in both engines. The results are presented in Figures 3.10 and 3.11. The magnitude of the cracking, the extent of the cracking, and the potential for the liner "dropping" into the crankcase, with the ensuing catastrophic results, provided a strong case for block renewal.

Figure 3.6

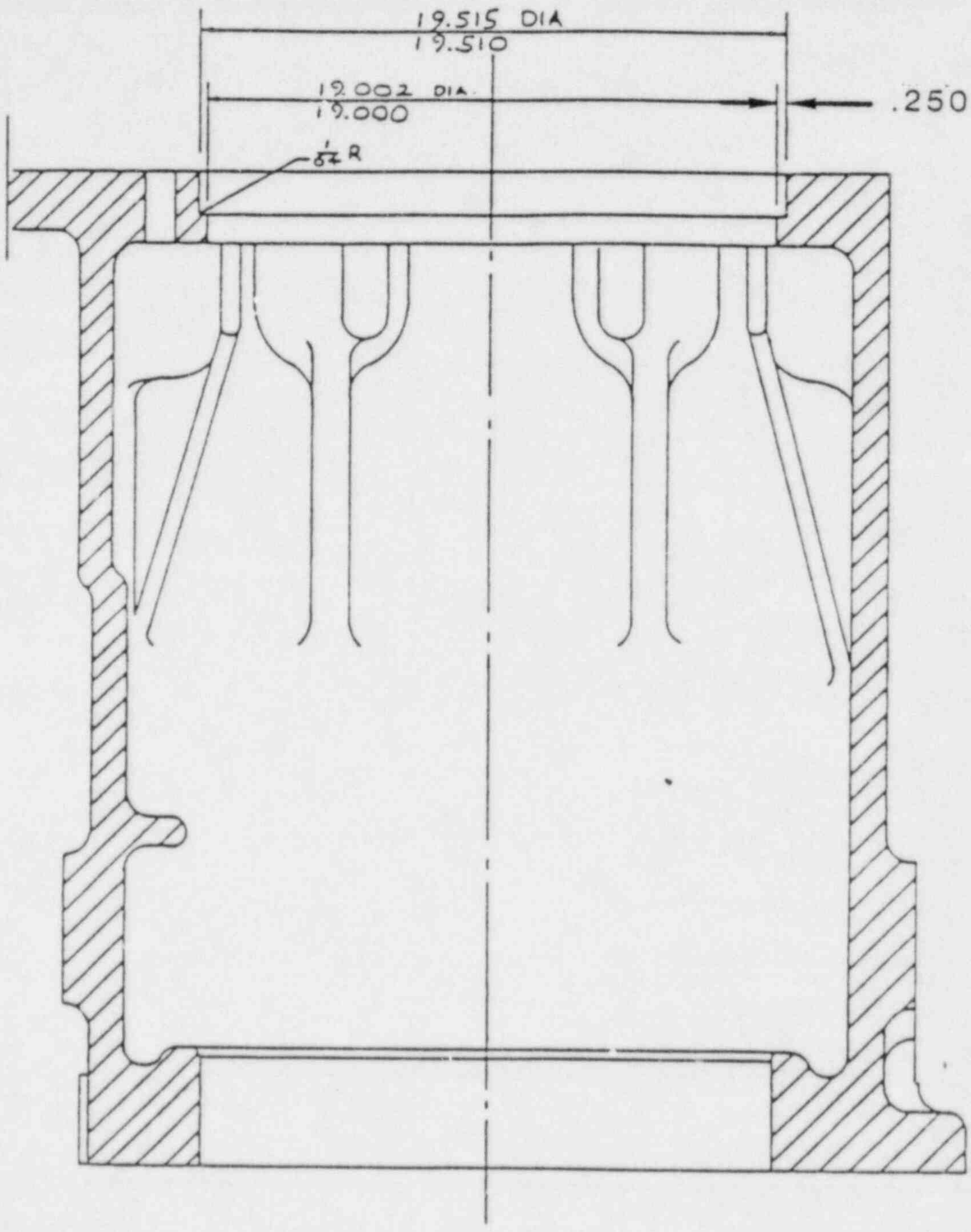
Upper Cylinder Liner & Block Section



*Reprinted from: "Engine Rebuild Report-M.V. COLUMBIA"
Jon O. Jacobson, March 31, 1981

Figure 3.7

Cylinder Configuration, Engine Block

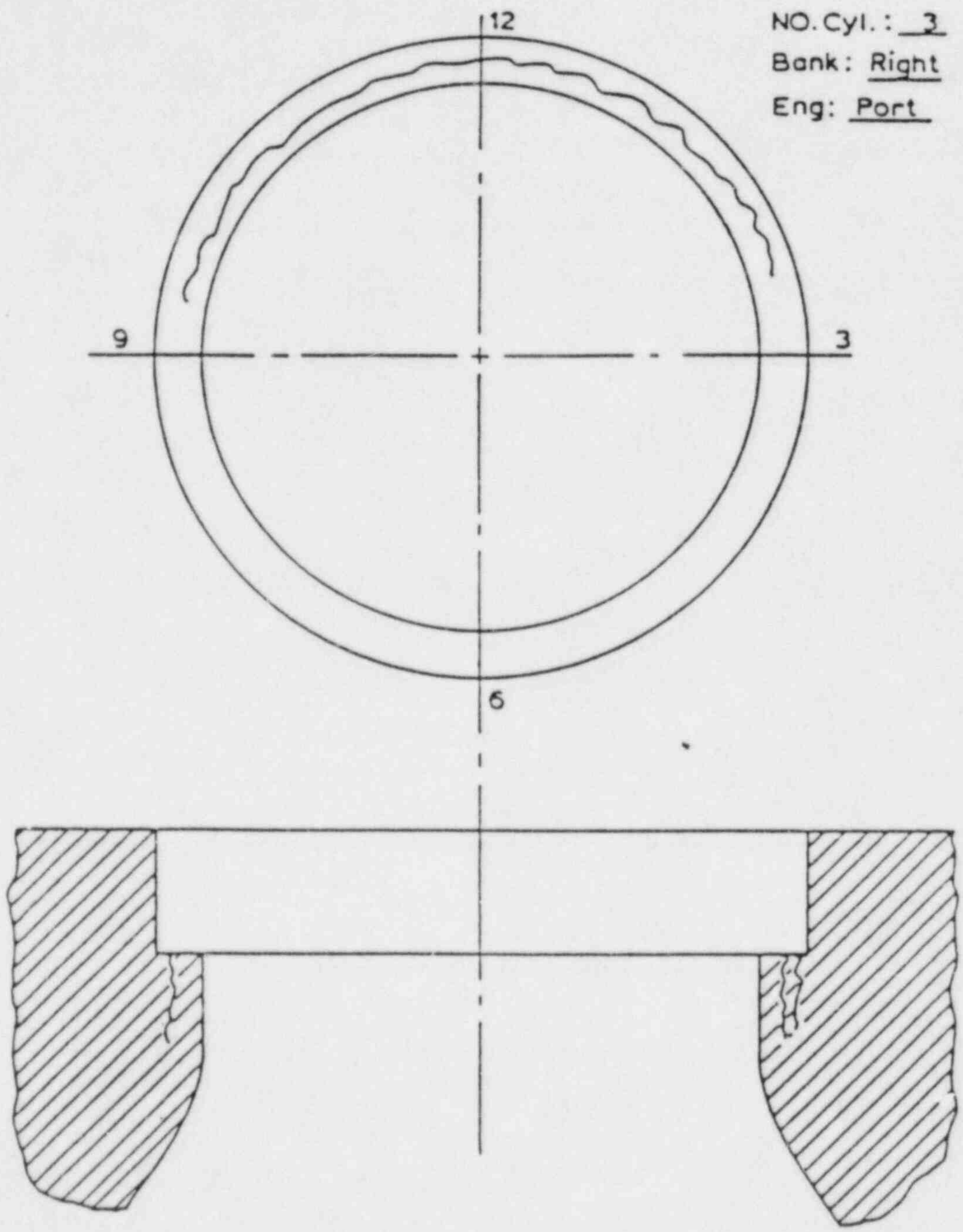


*Reprinted from: "Engine Rebuild Report-M.V. COLUMBIA"
Jon O. Jacobson, March 31, 1981

Figure 3.8

Nondestructive Testing,

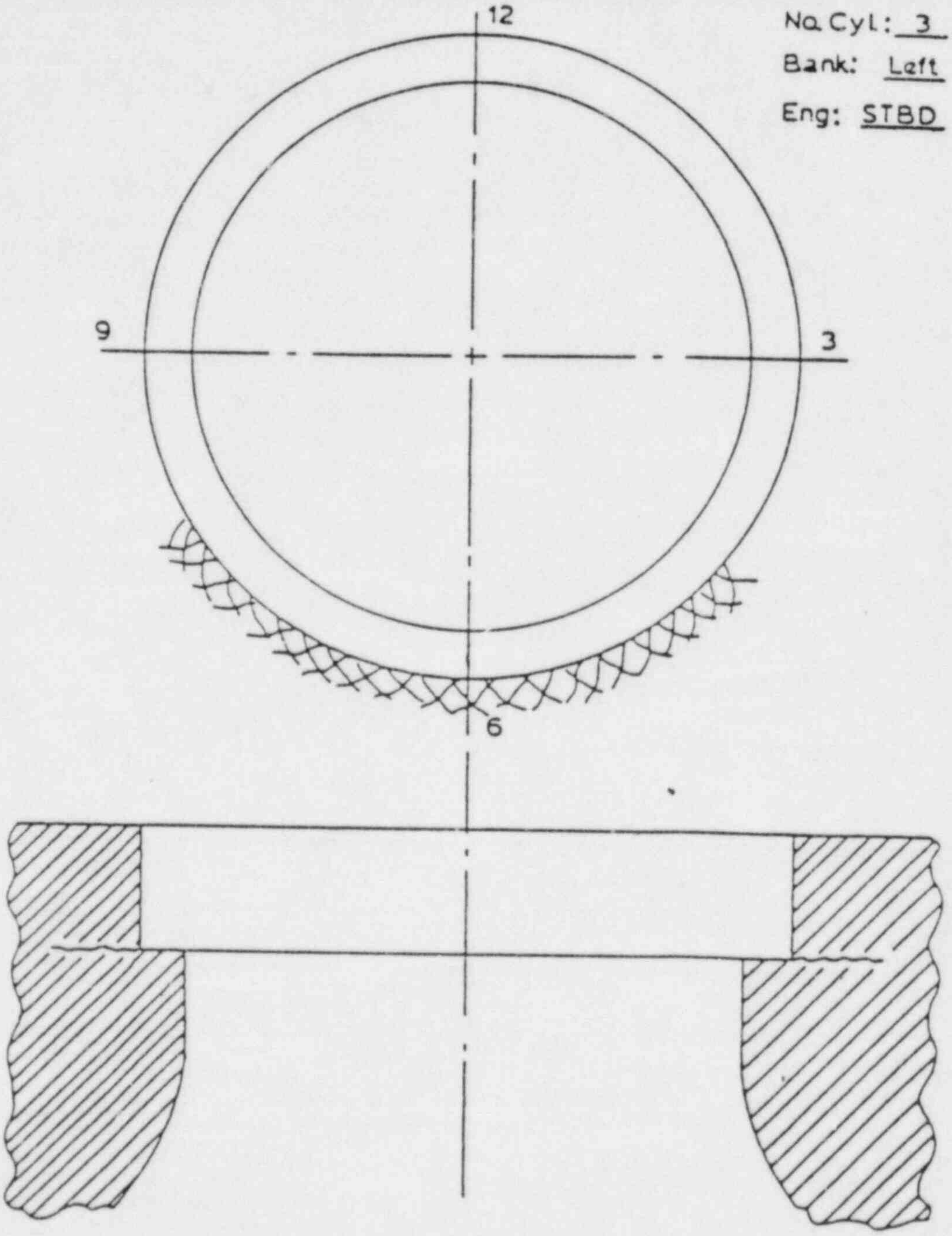
Cylinder Block, Shear Cracks, Counterbore Lip



*Reprinted from: "Engine Rebuild Report-M.V. COLUMBIA"
Jon O. Jacobson, March 31, 1981

Figure 3.9

Nondestructive Testing, Cylinder Block, Delamination Cracks



*Reprinted from: "Engine Rebuild Report-M.V. COLUMBIA"
Jon O. Jacobson, March 31, 1981

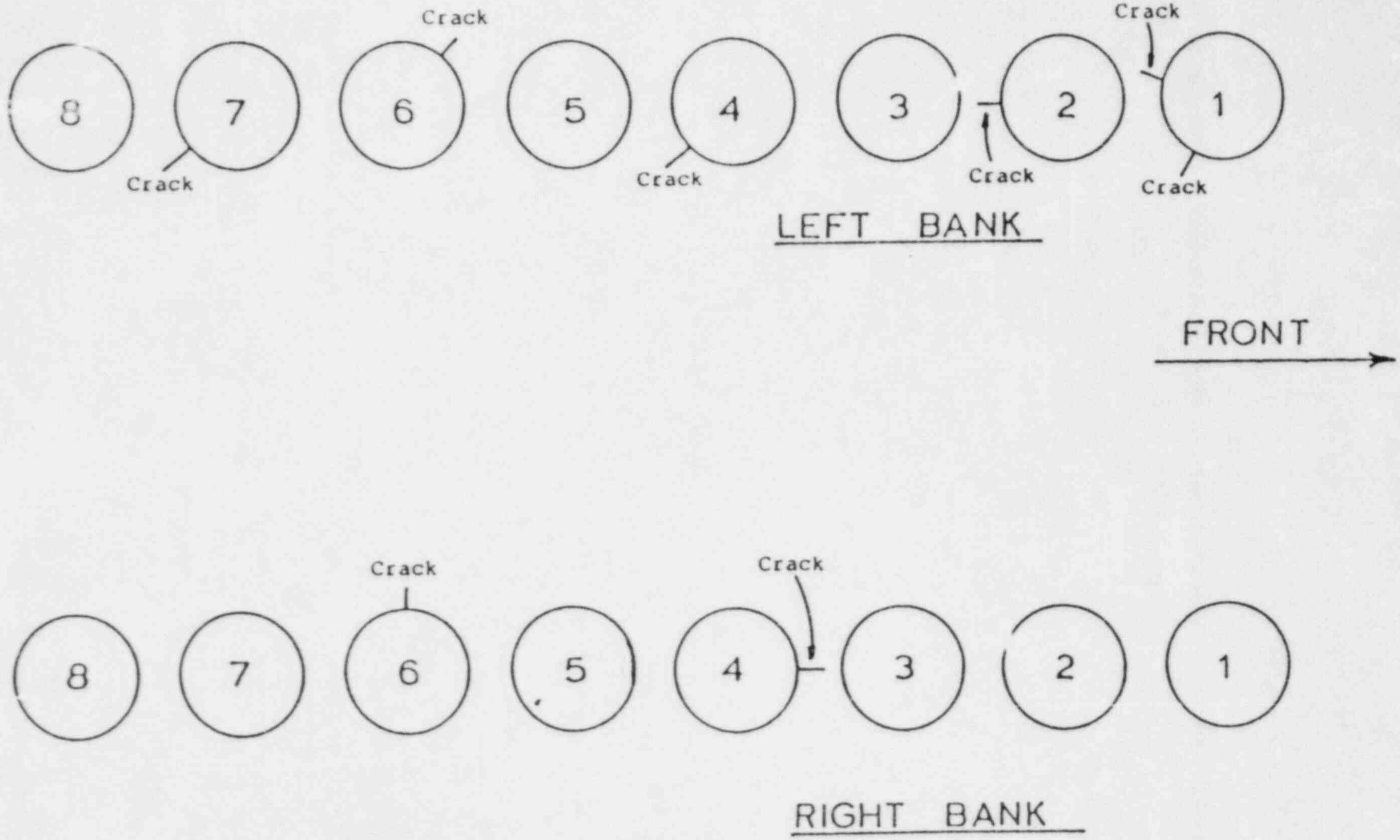


Figure 3.10

Nondestructive Testing, Port Main Engine

*Reprinted from: "Engine Rebuild Report-M.V. COLUMBIA"
Jon O. Jacobson, March 31, 1981

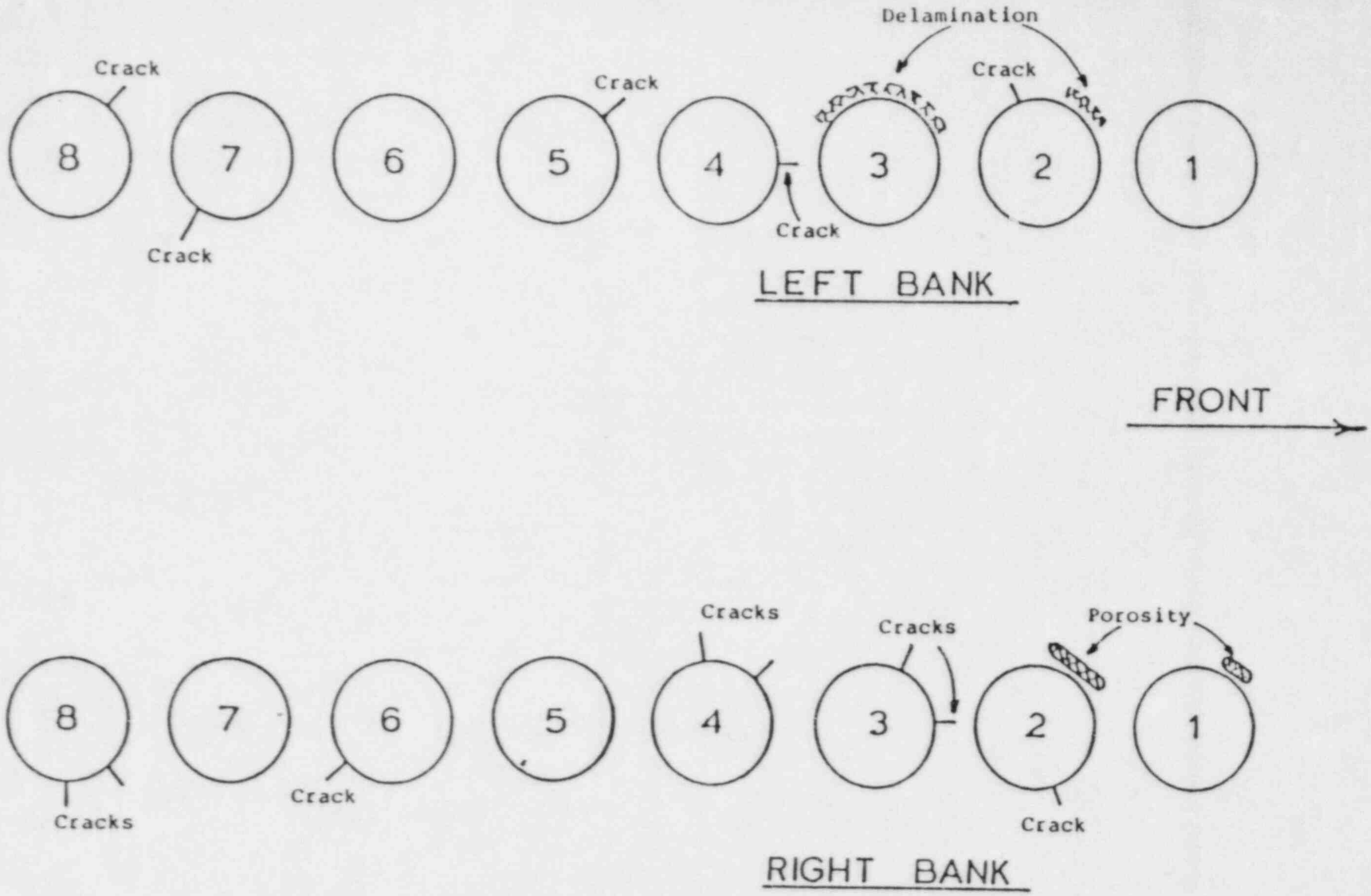


Figure 3.11

Nondestructive Testing, Starboard Main Engine

*Reprinted from: "Engine Rebuild Report-M.V. COLUMBIA"
Jon O. Jacobson, March 31, 198

A second factor which contributed to the ultimate decision to replace the cylinder blocks was the continuing cylinder liner counterbore diametral distortion which was maximized at the number four (4) cylinder locations (mid block) on all the blocks. A summary of these measurements was previously presented in Figure 3.4. The significance of this non-symmetrical dimensional change was the effect it was having on the cylinder bores and the cylinder liners relative to a time base of 24,000 hours. If no improvements are made by the manufacturer, it can be predicted that the blocks would have to be replaced at least two (2) more times in the twenty (20) year life of the vessel.

The "Engine Rebuild Report" also investigates the inadequate and irregular block to crankcase bolt torque values documented at the time of engine overhaul. An additional observation is made of the fretting or apparent movement which took place between the cylinder block and crankcase base surfaces. Two problem areas arise here. The first is the implication that correct bolt tensioning was used at the time of manufacture and assembly, and/or that thermal or cyclic loading contributed to the relaxing of the tension which contributed to the relative surface movement; or, that improper tensioning occurred at the time of assembly and that the surface fretting was the result.

The basic conclusion that can be drawn from the preceding is that the block had to be either replaced or repaired due to the dimensional changes and the casting cracking that was developing around the cylinder lip counterbore. The fact that an acceptable repair was not presented which addressed the cracking problem left AMHS with the only option of replacing the blocks. The fact that multiple cracking did occur indicates that the manufacturer has a design problem in this area of the engine.

3.3.8 Major Overhauls

Reference to Table 3.1A shows four (4) major overhauls at 6,000 hour intervals during 30,000 hours of engine running time. A major overhaul was charged against the engines as a unit (two engines, four times) any time it was necessary to conduct corrective maintenance, which included liner removal, piston ring replacement and replacement of multiple bearings, either connecting rod, main or articulated link bearings or pins. Coinciding with this work was the routine turbo cleaning and head reconditioning. Major overhauls occurred during the overhaul periods of 1975/76, 1978/79, 1979/80 and 1980/81.

An overhaul interval of 6,000 hours represents a rate four (4) times faster than expected between anticipated overhauls for either TDI or other diesel manufacturers.

3.3.9 Turbochargers

The turbochargers have historically been an item of high maintenance with multiple types of failures, including leaking oil/air seals, bearings, nozzles, rotors/cracked casings and fasteners. Reference to Table 3.1A lists forty (40) removals which coincide with forty (40) overhauls for the four (4) turbochargers on the two engines (two turbos/engine). The 2,300 hours between corrective maintenance actions means that at least one turbo would require removal sometime before the annual overhaul. Table 3.1A notes a TDI recommended scheduled maintenance cycle of 8,000 hours with other diesel manufacturers listing a TBO of 12,000 hours.

The turbocharger and engine performance with the Elliott Turbochargers was covered in Section 2.0. The reason for the recent retrofit of the turbochargers was based on the analysis of the data as previously recorded, including the turbochargers' inability to deliver a high enough quantity of air at an acceptable manifold pressure.

3.4 Summary of Findings

It can be seen from the foregoing Section that major moving components of the engine failed or required an inordinate amount of corrective maintenance at a significantly higher rate as compared to either TDI's recommended scheduled maintenance or other typical diesel manufacturers' TBOs. The types of failures, and number of failures of some of the major components indicates design deficiencies in these components. Two critical components which have been subject to failure, which are not typically expected to routinely fail, were the articulated connecting rods and cylinder blocks.

Another item of significance is the extended overhaul time required due to the greater amount of corrective maintenance which impacted the operational schedule as illustrated in Figure 3.12. The Figure shows that actual maintenance periods exceeded

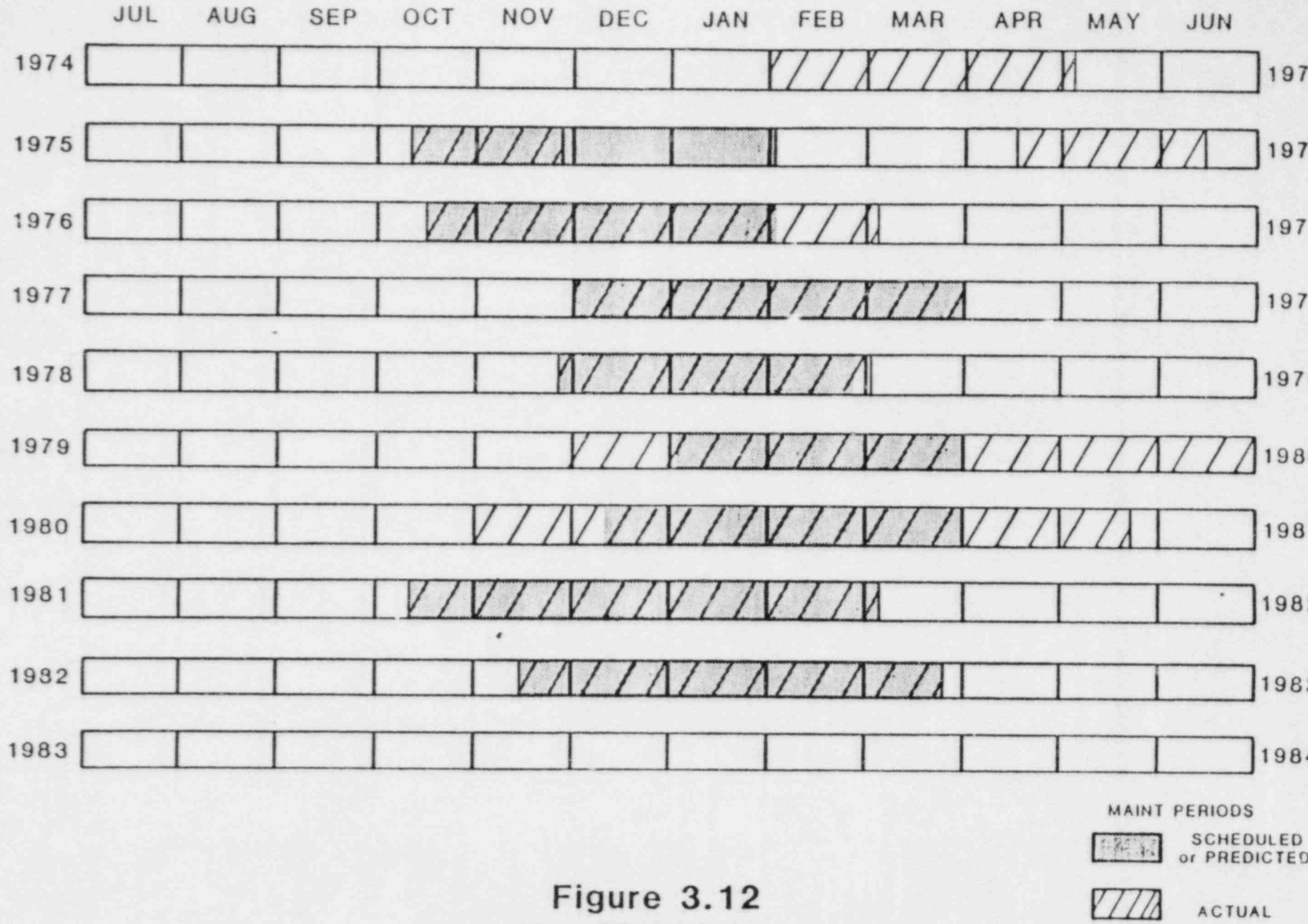


Figure 3.12
M/V COLUMBIA
Operational & Maintenance Periods

projected periods by 30%, resulting in a loss of thirteen (13) months of potential revenue. Original planning records show generating time that for the five (5) years preceding 1980, maintenance periods of three (3) months were allocated for the entire ship. Post-1980 overhaul periods witnessed a consistent lengthening of the maintenance cycle as dictated by the engines' requirements. Finally, in 1982 and 1983, the maintenance period had increased by a total of one month and the in-service dates were being set to accommodate the engine overhauls.

One item which has not been addressed previously is the operation of the engines on heavy fuel oil (HFO). Original contract specifications required these engines to operate on HFO. Demonstration of HFO operation was accomplished upon delivery of the ship and for approximately eight (8) months after that. However, the HFO operation was discontinued after the initial eight (8) months. The records did not disclose why HFO operation was suspended but the following observation can be made relative to what the impact would have been on these engines if HFO had been burned. A survey of other medium speed diesel manufacturers' TBO schedules typically reduce length between overhauls by approximately fifty percent (50%). It can be projected that the impact of continuous HFO operation on these engines would have resulted in considerably higher wear and failure rates than those recorded in Table 3.1.

4.0 ESTIMATED POTENTIAL REDUCTION OF COMPONENT FAILURES AFTER ENGINE DE-RATING

4.1 Introduction

M/V COLUMBIA had been operating at reduced power levels, approximately 7,000 HP, for approximately three or four years prior to the 1982/83 overhaul and de-rating. The recent de-rating, inclusive of the C-17 turbo modification with a projected engine rating of 6164 HP @ 403 RPM Maximum Continuous Rating (MCR), represents an additional 14% reduction of power from the preceding reduced operational power levels. Therefore, the lower power level does not represent a radical change from the previous years' operating scenario. A lower BMEP should mean lower transmitted forces to the various engine components. Likewise, the replacement turbocharger with greater air delivery capability should enable the more complete combustion of the fuel thereby reducing the stress and wear rate on combustion related components, such as piston rings and cylinder liners.

4.2 Projected Corrective Maintenance and Expected Component Life

The following subsections present a review, and where possible, analyses on the components which would be affected or which were subjects of early failure and replacement, as presented in Table 3.1.

4.2.1 Cylinder Heads

The de-rated engine will reduce thermal and mechanical stresses induced by combustion. The improved air flow to the cylinders with the resultant lowering of the exhaust gas temperature should be beneficial to combustion gas path components such as valves and valve seats.

The installation of the Trabon system was initiated to reduce the carbon/soot loading in the lube oil via the rocker boxes. The greatest contribution to soot reduction will be the improved combustion offered by the turbocharger modification. However, based on the premise that the Trabon system (oil injection around the valve stem in the valve guide) will eliminate the valve stem/guide blowby, it is in our opinion, of marginal value. This judgment is based on the continued level of exhaust gas pressure in the form of dynamic head that results from the expansion of gas from the cylinder at exhaust valve opening where typically the gas reaches sonic velocities.

Head cracking may be lessened due to the reduced thermal stresses experienced during operation. However, internal head cracking and porosity leaks and core shifting due to manufacturing problems will probably remain at the same level as witnessed by the cooling outlet problems which were experienced with the 16 new heads.

Head failures due to warping in the 3-9 o'clock position has been addressed by the manufacturers by the addition of a "strong back" (reinforcing perpendicular to the 3-9 o'clock axis). Theoretically, this should contribute to the solution of the problem. A review of the records indicates that in 6,800 hours of operation, two (2) heads have experienced removal because of water leaks and one (1) head has been removed with no reason given. No observations were recorded relative to the fire ring gasket or fire deck warping. The average removal rate is one unit for approximtely every 2,300 hours of operation (all heads were removed from the S.M.E. which had the sixteen (16) new heads installed).

4.2.2 Cylinder Liners

Cylinder liner removal rate should decrease and life expectancy should increase from improved combustion with the resultant decrease in soot generation and hence, reduction in abrasive particles. The lacquering problem which has been observed in the liners should also be reduced because of the improved combustion.

Lacquer is actually a combination of resins, soot, oxygenates, oil and water produced by oxidation at combustion temperatures. The increased presence of soot acts as an increased nucleus site wherein the soot precipitates on the cooler liner walls and the resin-like substance concentrates around and between the soot particles. Reduction of the soot particles should therefore reduce the lacquer accumulation.

Liner removals influenced by the dimensional change of the cylinder block are not expected to change due to the mechanical deformations imposed on the liners by the block. Therefore, it can be projected that some liners will reach the end of their useful life at approximately 20,000 to 30,000 hours.

4.2.3 Pistons

Piston removals for ring replacement should decrease for the same related combustion improvement reasons discussed in the cylinder liner Section 4.2.2. However, the same dimensional change problems which affected the liner roundness will also adversely affect the ring wear rate and life expectancy.

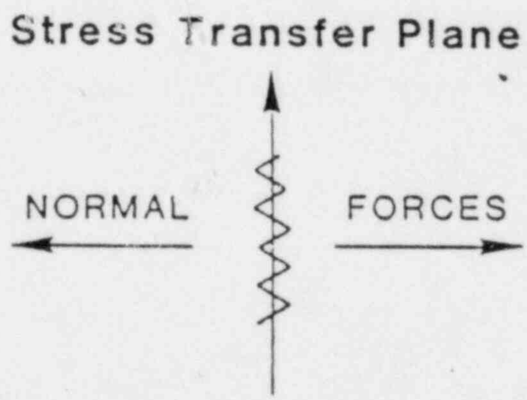
Piston removals for modifications to crowns, skirts and fastening mechanisms should be finalized as this model of engine has been in existence for approximately ten years and therefore the manufacturer should have incorporated, by now, all the related design corrections gained through experience on existing engines.

4.2.4 Master Link and Connecting Rods

The articulated rod removal rate frequency can be expected to decrease due to the decreased loading on that assembly with a subsequent anticipated increase in component life. However, the approximate 14% reduction in engine de-rating is not a significant reduction when the previous operational power levels of 7,000 HP are considered and therefore improvement may be marginal.

The link rod bearing and pin useful life expectancy should increase due to the overall decreased loadings. However, this whole assembly is considered a highly loaded part relative to the dynamic forces induced on these components. Past corrective maintenance procedures have been concerned with failures induced by poor quality control of the pin and bushing. If these manufacturing problems are resolved, then the remaining problem of bearing wear should be minimized.

The fasteners and link rod box cracking phenomenon was addressed by TDI as previously discussed in Section 3.3.4. The decrease in bolt diameter from 1-7/8" to 1-1/2" resulted in a net width gain of .75 inches on the total width of the connecting rod assembly. This modification plus the radius changes should increase the expected life of the component. It should be realized, however, that the shorter connecting rod for an equal stroke and crank throw radius has the greater angular swing and greater side thrust, hence greater loading on the piston and link pin and bushing. Intergral with the fastening mechanism is the serrated joint which has been the site of repeated observations of fretting and therefore relative movement. The employment of the serrated joint is recognized for its value in transferring stress loads in the plane perpendicular to the serrations, however, it would appear that the resultant forces acting normal to the serration plane are contributing to the induced movement.



In a four-cycle engine this is especially pronounced when it is considered that there is a reversal of loading on each connecting link rod from compression to tension on each cycle along the legs of the "V" which is formed by the centerline of the connecting rod axis. The serrated joint surface then is constantly being subjected to the normal and parallel forces induced by the two pistons. This type of a joint is also subject to controversy among manufacturers because of the argument that bearing distortion is more easily induced because of variation in serrated surface irregularities and bolt tightening. Two of these factors have been observed to date. This is further reinforced by the failure mechanisms noted in the Northwest Laboratories Report (Reference No. 17) which concluded that the connecting rod bearing was damaged by localized over heating. This finding reinforces the observations made during the 1979/80 overhaul of distressed connecting rod bearings with localized loading spots, fretting damage and carbonized oil deposits.

4.2.5 Camshafts

The camshaft wear rate should be decreased slightly due to the reduction of carbon in the lube oil. However, because the cam shaft is unaffected by engine de-rating except for the lower engine speeds, it is expected that the useful component life will remain approximately the same.

4.2.6 Main Bearings

The main bearing wear rate should be decreased slightly due to the reduction of carbon in the lube oil and the slightly lower crankshaft loads due to lower BMEP. It is our opinion that the bearing will experience approximately the same 16,000-20,000 hours before requiring renewal.

4.2.7 Cylinder Block

The engine de-rating will lessen the thermal stresses due to the reduced heat rejection to the jacket cooling water and will lessen the mechanical stresses due to reduced BMEP.

The two major items of concern are the cylinder liner counterbore dimensional symmetry and reoccurrence of counterbore lip cracking. In the instance of counterbore symmetry, we would expect a reoccurrence of this phenomenon unless structural and design changes have been made to the cylinder blocks. In the case of the cracked cylinder liner counterbore lips, we would expect this to reoccur unless design changes are made to the block in that area. In the absence of formal notification by TDI that a change has been made and is the subject of a retrofit, we would anticipate block lip cracking in the future. It should be noted that unsymmetrical oval counterbores can be repaired if that is the only problem with the block at that time. However, a structural weakness with the potential for catostrophic failure should still be considered as a replacement item.

The cylinder block to engine base fretting that was observed during the 1980/81 overhaul has also been observed in other manufacturers engines. The relaxed cylinder block to engine base tie rods contributed to the severity of the fretting and as Mr. Jacobson recommended, this can be minimized by periodic retorquing of the tie rods as availability dictates. If cylinder block to engine base fretting does occur, this may be repaired by various resurfacing techniques. It should be realized that once fretting occurs to the extent that a minute relaxation of tensioning occurs, the fretting effect accelerates at an increasing rate with the subsequent increasing relaxation of the tie rod tensioning.

4.2.8 Major Overhauls

The total effect of engine de-rating will result in an improved level of component quality where those components are exposed to the combustion gas path. However, we do not feel that a quantum improvement will result which will approach the TDI suggested maintenance time intervals.

4.2.9 Turbochargers

There is no turbocharger history to draw upon to speculate on a corrective maintenance interval; however, the TDI suggested maintenance schedule lists 8,000 hours as the TBO cycle.

There are several comments that are germane to the turbocharger and exhaust system. The C-17 turbos do not have the capability for localized blocking or rotor restraint devices in the event of failure. This is considered a drawback because the entire engine must be shutdown in the event of a failure. Most turbo manufacturers have a method for blocking the rotor so that the engine may be run naturally aspirated at reduced power.

The second item of concern is the cantilevered mounting method which was used to intergrate the turbos into the engine exhaust and manifold air systems. A cursory examination of the foundation and bracing assembly would raise doubts as to the long term integrity of the structure due to the amount of weight which is cantilevered over the front of the engines. The weakest point of the assembly appears to be the point of attachment at the engine block.

The final comment concerns the exhaust system which has been the object of repeated repairs, replacements, and design configurations. Historically, this has been an area of high maintenance. It was not reviewed in Section 3.0 because it is not one of the moving parts within the engine. Therefore, this system should be subject to less stress due to the lowered exhaust temperatures and hence, require less corrective maintenance. However, based on historical data, we would still expect to see this system receive a higher than usual amount of service.

4.3 Additional Modifications and Corrections of Problems Created By Engine De-Rating

4.3.1 Lube Oil Systems

The lube oil systems current configuration utilizes a common purifier which is switched between engines; however, each engine is equipped with a full flow filter for constant filtering capability. A change should be made in the lube oil system so that another purifier is procured to effect a dedicated lube oil system. The inclusion of a dedicated system would significantly improve the lube oil quality and reduce the carbon loading on the system. The piping modifications to the lube oil system should definitely include the repair/replacement of the duplex full flow lube filter diverter valve to permit change over during operation instead of the present requirement to shut the engine down.

An optional addition to the lube oil system would be the installation of a polishing system which would take suction from the lube oil sump and return to the sump.

4.3.2. Cooling Water System

Recent modifications to the cooling water system have resulted in the inability to utilize the stand-by jacket water cooling pump. Failure of this pump in the present configuration would require engine shutdown. The stand-by jacket water pump is required to ensure continuous engine operation.

The temperature control system may also require modification to ensure sufficient temperature in the jacket water system and turbocharger after cooler. Initial voyage results indicate that a deficiency exists in this system.

4.3.3 Turbochargers

In Section 2.0 it was shown that the turbochargers were not matched to the engines. It is imperative, therefore, that a proper match be made of turbocharger to engine to ensure future engine reliability and efficient operation.

4.3.4 Waste Heat Boiler

Subsequent sea trial feedback has indicated that insufficient exhaust gas is being routed through the waste heat boiler with the consequential requirement that the auxiliary boiler make up the difference in steam shortfall at increased cost due to the additional fuel consumption of the boiler. This will require the installation of an equalizer baffle or orifice to ensure sufficient heat to the boiler.

4.3.5 Engine Performance Optimization

The sea trial data indicates that engine power output is lower than what is required by contract and that the engine is not at its most efficient de-rated operating level. Additional problems are seen with the lower than expected firing pressures. Problems of this nature are usually associated with fuel timing and metering systems.

5.0 RE-ENGINING ECONOMIC ANALYSIS AND COMPARISON OF HISTORICAL MAIN ENGINE OPERATING COSTS AND EXPECTED DE-RATED ENGINE OPERATING COSTS

The cost analyses presented in this section deal with three (3) major subject areas. The first is the quantification of an average annual main engine maintenance and repair cost and the estimated reduction in this expenditure which can reasonably be expected to result from derating. Also addressed are other capital expenditures which, in Seaworthy's opinion, must be made to ensure the reliability and efficient performance of the de-rated engines or which must be made as a result of additional problems resulting from the de-rating project based on the status and results of this effort to date. The third and final area of discussion presented in this section is an economic trade-off analysis which compares continued operation of the de-rated engines, including the additional capital expenditures required for reliable and efficient operation, against the estimated cost associated with the re-engining of the M.V. COLUMBIA.

5.1 Historical Main Engine Related Cost Review and Development

As a basis for establishing an estimated main engine maintenance and repair average annual cost, various operating cost records were reviewed, dating from 1976 through 1982. These included, for the most part, purchase order type documents, major main engine overhaul cost breakdown reports and AMHS Fiscal Year Expense and Revenue Statements for the COLUMBIA. Costs associated with the current de-rating project were not included. The bulk of these records provided a gross or macro view of engine/engine department costs for each year. Because the majority of these records lacked a detailed itemized breakdown into such areas as individual labor category, spare parts by component, cosumables, contractor or repair facility which clearly identified the associated expenditure as being main engine related, the following approach was taken. Fifty (50) percent of all cost obtained and identified as accruing during each annual propulsion plant/ engine room overhaul period when the vessel was out of service, were assumed to represent that portion of the total annual power plant overhaul period costs directly related to main engine maintenance and repair. Taking a similar approach for the operating portion (and associated costs) of each year, twenty (20) percent of the identical cost categories were taken as being representative of main engine related maintenance and repair costs while the vessel was in service. The cost categories and typical associated elements are listed below.

<u>OVERHAUL PERIOD</u>

1. Labor:
   - Base Wages
   - Overtime
2. Commodities:
   - Spare Parts
   - Consumables
3. Contractual:
   - Shipyard
   - Service Reps.
   - Other contractors

<u>OPERATING SEASON</u>

1. Labor:
   - Base Wages
   - Overtime
2. Commodities:
   - Consumables less fuel and lube oil
   - Spare Parts
3. Contractual:
   - Riding crews
   - Service Reps.
   - Others

4. Equipment:
   - Tools

4. Equipment:
   - Tools

Table 5.1 presents the estimated annual operating season and overhaul period costs derived for main engine maintenance and repair.

TABLE 5-1

SUMMARY OF ESTIMATED ANNUAL MAIN ENGINE RELATED M&R COST, 1976 to 1982

Year	Operating Season, $	Overhaul Period, $	Yearly Total, $
1976	$ 242,041	$ 87,710	$ 329,751
1977	34,433	64,540	98,973
1978	218,935	242,221	461,156
1979	155,674	449,048	604,722
1980	179,994	344,055	524,049
1981	169,184	433,546	602,730
1982	201,598	269,228	470,826
Estimated Average Annual Main Engine M&R Cost: $441,740/Year, Historical			

Discounting present problems associated with the engine derating project, it is reasonable to anticipate that the average annual main engine maintenance and repair costs shown above, after resolution of the current difficulties, would be reduced. While the exact value of this expenditure reduction requires considerable speculation, it is felt that an improvement of 25% is a reasonable approximation. This is based primarily on Seaworthy's past experience in performing similar analyses and a correlation of the historical main engine maintenance and repair data including component failure analysis, overhaul reports, maintenance and repair related costs and ABS surveyor reports as summarized in Sections 3.0 and 4.0 of this report. Also taken into consideration was TDI's performance record relative to providing cost effective, permanent and sound engineering solutions to numerous design, production and (to a lesser extent) operating based engine component failures which have significantly increased this annual expenditure. Specifically, it is believed that this reduction in M&R costs will accrue from engine de-rating and new turbos as a result of minimal improvements in component life and time between repair and/or overhaul for the following components based on the discussions presented in Section 4.0.

1. Cylinder Heads
2. Cylinder Liners
3. Piston Rings
4. Articulated Connecting Rod Assembly
5. Main Bearings
6. Exhaust Manifold/Cylinder Head Jumpers
7. Lube Oil Life (Carbon loading reduction)

5.2 Propulsion System Modifications Required in Addition To or As A Result of Main Engine De-Rating

As part of the workscope which is addressed by this report an evaluation as to the adequacy of component and systems modifications made as a result of the engine de-rating was conducted. The intent of this investigation was to identify and quantify, in terms of time and cost, additional work felt necessary to ensure the future operating reliability and efficiency of the COLUMBIA's de-rated propulsion plant. Additional work, some of which is major, has also been identified and quantified as a result of the performance of the main engines during the March 24-25, 1983 sea trials and subsequent voyages. These modifications, along with supporting rationale, estimates of the time required to accomplish them relative to the scope of the work and Rough Order of Magnitude, ROM, cost estimates for each are provided in Table 5.2. The ROM cost estimates include components/hardware and necessary installation materials and labor.

5.3 Re-Engining Economic Trade-Off Analysis

An economic comparison has been made which evaluates the continued operation of the existing main engines after de-rating versus the installation and operation of new Heavy Fuel Oil (HFO) capable engines identical to the types specified in Reference No. 2, based on varying values of the assumed remaining useful life of the vessel. The various cost elements, methodologies applied and results are presented in the following paragraphs.

5.3.1 Cost Elements

The cost elements established for this analysis have been categorized in two (2) main areas, that of acquisition costs and annual operating costs.

Acquisition Costs: An associated capital expenditure for each

TABLE 5.2

ADDITIONAL PROPULSION SYSTEM MODIFICATIONS REQUIRED FOR M.V. COLUMBIA
AFTER DE-RATING AS OF APRIL 1, 1983

Modification/Alteration	Documentation/Supporting Rational	Time To Complete	Rough Order of Magnitude (ROM) Cost
1. Lubricating Oil System	Installation of 2nd L.O. purifier to provide simultaneous L.O. purification for both main engines, polishing filter for each engine and modification of existing filter valving for improved operation.	Overhaul Period	$150,000
2. Combustion Improvements	It is anticipated that actions including F.O. injection timing advancement, cylinder firing pressure balancing and fuel metering components (pumps, injector, nozzles) may have to be modified/replaced to restore original design fuel consumption at the de-rated output.	Operating Season (No loss of service)	$60,000
3. Cooling System	Modification of engine cooling loops to increase jacket water temperature as part of de-rating process and to restore jacket water/fresh water pump redundant service capability.	Operating Season (No loss of service)	$30,000
4. Exhaust Gas Pyrometer System	Replacement of existing exhaust gas pyrometer system with a more accurate, reliable and useful system.	Overhaul Period	$75,000

TABLE 5.2 CONTINUED

Modification/Alteration	Documentation/Supporting Rational	Time To Complete	Rough Order of Magnitude (ROM) Cost
5. Turbochargers	Based on data and discussions presented in Section 2.0, the turbos are not matched to de-rated engine load profile. Replacement with properly matched units is felt to be the most prudent and reliable fix for this program.	Overhaul Period	$470,000 (Price includes: 4 turbos, spares, tools, spare rotating element, new tachometers, transisition ducting, foundation engineering & labor)
6. Waste Heat Recovered Steam Generating System	As a result of de-rating there is a short fall in steam available for auxiliary and hotel loads. This has cuased a noticable increase in fuel consumption as a result of continuous operation of the auxiliary oil-fired boiler. This situation may be rectified by diverting more exhaust gas away from the silencer and into the waste heat boiler on each engine.	Operating Season (No loss of service)	$20,000
7. Control System	Pitch schedule and load control portion of the main engine control system has not been properly set up for the new de-rated engine operating profile.	Operating Season (No loss of service)	$30,000
8. Structural	Leaks in the compressor discharge/ manifold inlet transition pieces have been noted and can be expected to increase. Installation of flexiable transition pieces would relieve this situation.	Operating Season (No loss of service)	$20,000
		TOTAL	$855,000.00

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alternative, continued operation of the de-rated main engines and re-engining of the COLUMBIA with HFO capable diesels, was established. These values were assumed to include costs for purchase, installation labor, installation materials, rip-out and other typical activities associated with this type of work. For the continued operation of the main engines, an acquisition cost of $855,000.00, established in paragraph 5.2 was utilized. Values of $6, 7, 8 and 9 million dollars have been assumed as a range of acquisition costs, representative of a potential re-engining cost spread for the COLUMBIA, in order to test the sensitivity of the analysis to this potential variable as described in paragraph 5.3.3.

Annual Operating Costs: Because this evaluation is limited in its consideration of only operating side economics, the annual operating expenditures considered were those felt to be directly attributable to main engine operation, fuel oil consumption, lubricating oil consumption and maintenance and repair. While new engines might arguably increase the operating seasons for the vessel over continued operation of the existing units as a result of improved reliability and reduced maintenance, the impact of this possibility was not factored as it implies revenue-side analysis which was beyond the scope of this evaluation. In deriving the operating cost elements for each option, an optimum operating year of 5400 hours at an average output of 10,500 BHP was assumed for both alternatives so that the analysis could be conducted on an equivalent basis.

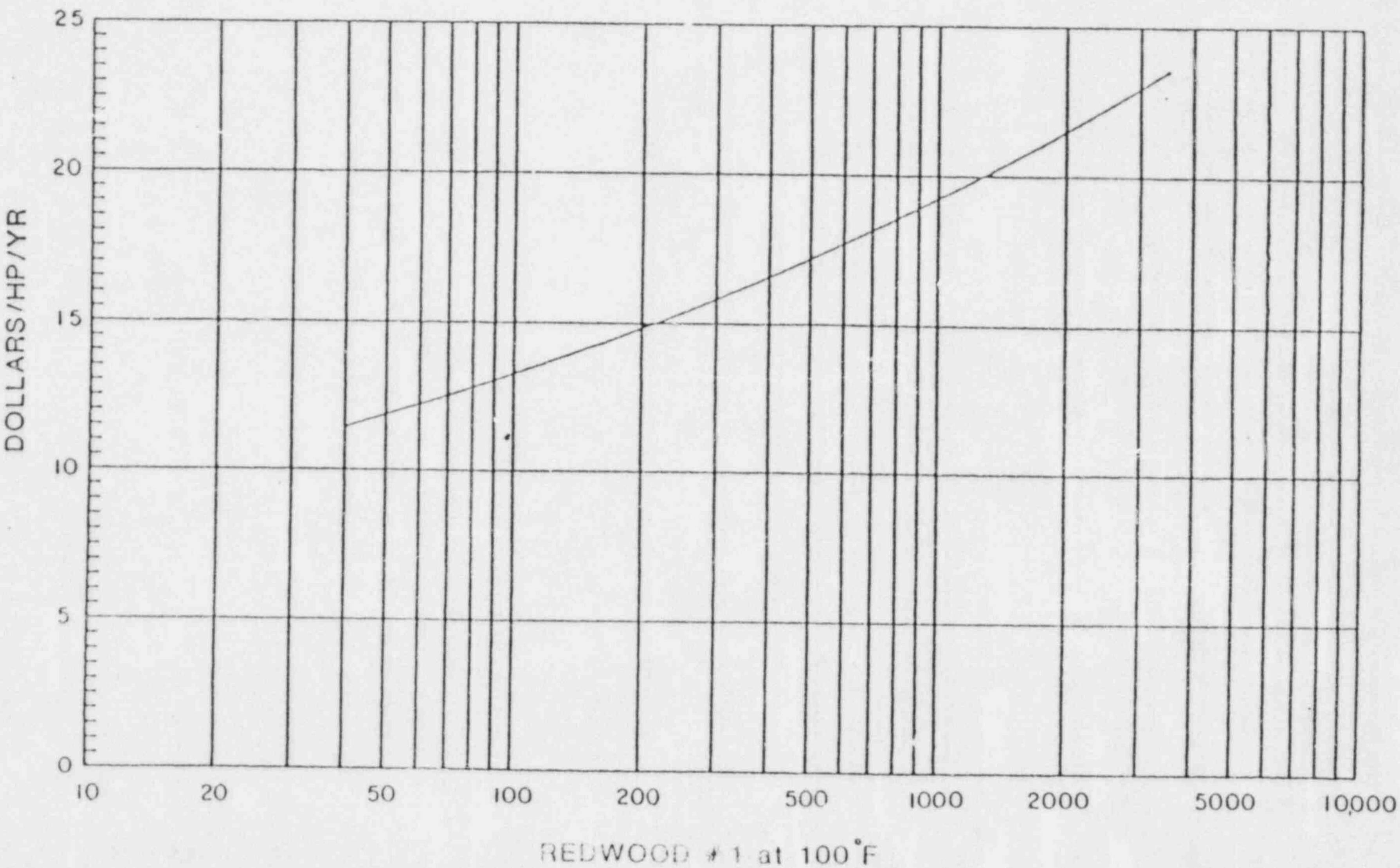
Fuel Cost: Annual fuel costs were computed for each alternative based on at-sea operation for 5400 hours per year at 10,500 BHP. For each alternative quoted, manufacturer's fuel rates were utilized and adjusted in the following manner. The existing DMRV-16-4 engines' design quoted fuel rate was increased by 10% to account for a 3% guarantee margin in addition to a 7% increase which is felt to be representative of the deterioration in performance that has occurred based on the historical failure analyses conducted in Section 3.0 and 4.0. A final upward adjustment was made to account for the difference in typical Marine Diesel Oil (MDO) heating value versus the heating value of Marine Gas Oil (MGO) on which design quoted fuel rate is based. Future operation on Heavy Fuel Oil was not considered as a viable alternative for the existing engines based on the documented poor past performance and reliability experienced with the engines while operating on MDO. The new engines' design quoted fuel rate was adjusted in an identical fashion to that described previously including a quoted 5% guarantee margin and an adjustment for heating value differences for MDO and operation on HFO. The HFO to be utilized was assumed as 180 CST (1500 Second Redwood No. 1, SR1). No increase in fuel rate for the new engines was assumed. Fuel pricing utilized per metric ton was that posted during March 1983 at the Port of Seattle for MDO and 180 CST fuels. Sample consumption fuel calculations are contained in Appendix I .

Lube Oil: Lubricating oil cost were derived utilizing manufacturers quoted lube oil rates of one (1) gallon/5000 BHP hours for the Enterprise engine and 1.5 grams/BHP-HR for the new engine. For operation on MDO a lube oil with a total base number of TBN-10 was utilized due to the low sulfur content in this fuel. For new engine operation on HFO a lube oil with a TBN of 30 was utilized as a result of the increased sulfur content of HFO. The TBN designation basically is an indicator of a higher content of various chemical addatives put in the oil to neutralize the potential of increased acid corrosion attack of engine internals when operating on HFO. Lube oil prices utilized were those posted in Seattle as of March 1983. Sample lube oil costs calculations are contained in Appendix I.

Maintenance and Repair Costs: The annual maintenance and repair costs utilized for the existing engines was the historic rate derived in paragraph 5.1 and adjusted downward, based on future improvements expected from the engine de-rating. Costs for the new engine were derived utilizing Figure 5.1, a curve of maintenance costs for typical medium speed diesel engines in \$/HP-Yr versus fuel oil viscosity. This curve was initially generated by Seaworthy as a result of two research projects performed for the U.S. Maritime Administration dealing with the influence of fuel quality on the maintenance and repair of marine diesel engines and has been updated on a frequent basis in published papers and presentations given by Seaworthy personnel. It is felt to reasonably account for the well-documented and universally accepted fact that engine M&R costs increase as the quality of fuel supplied (using increasing viscosity

Figure 5.1

Impact of Varying Fuel Quality on Engine Maintenance, Total Spares, Consumables, and Labor



as an indicator) decreases. Appendix I contains calculations of estimated annual maintenance and repair costs for the new engines for MDO and HFO operation. The components of the cost derived utilizing this curve are essentially identical to those utilized in developing the historical cost for the existing engines; labor, spare parts, consumables, tools, etc.

Table 5.3 and 5.4 present unit costs utilized for fuel and lube oil in the analysis and a summary of acquisition and first year annual operating costs, respectively.

5.3.2 Economic Analysis Methodology

The approach taken in establishing the potential economic benefit associated with re-engining of the COLUMBIA versus continued operation of the existing engines is best summarized in the following manner. It is the determination of whether the annual operating cost differentials (existing engine less new engine annual costs) justify the initial non-recurring acquisition cost differential (re-engining less de-rating modification costs) over the anticipated remaining useful vessel operating life as determined by utilizing the annual cash flow differentials to calculate the following financial indicators: Net Present Value, NPV, Internal Rate of Return, IRR, Simple Payback, SPB and total Life Cycle Costs, LCC. These computations have been performed by micro-computer. The actual computer output data is contained in Appendix H . These and other terms, as applied in this analysis, are defined in Table 5.5.

To take into account the influence of inflation over the investment

TABLE 5.3

FUEL AND LUBE OIL UNIT COSTS

Fuel & Lube Oil Types	Unit Price
1. MDO	$276/Metric Ton
2. HFO (180 CST)	$181/Metric Ton
3. TBN-10 Lube Oil	$3.86/Gallon
4. TBN-30 Lube Oil	$4.26/Gallon

TABLE 5.4

SUMMARY OF ACQUISITION AND
FIRST YEAR ANNUAL OPERATING COST ESTIMATES

Cost Category	Existing Engines	New Engines
1. Acquisition Costs:	$ 855,000.00	$6,7,8 & 9 Million (See Paragraph 5.3.3)
2. Annual Operating Cost:		
Fuel Oil:		
MDO	$2,666,440.00	$2,416,380.00
HFO	- - - -	$1,655,970.00
Lube Oil:		
MDO	$60,630.00	$100,090.00
HFO	- - - -	$122,600.00
Maintenance & Repair:		
MDO	$331,310.00	$120,750.00
HFO	- - - -	$218,400.00

TABLE 5.5

DEFINITION OF ECONOMIC ANALYSIS TERMINOLOGY

TERM	DEFINITION
1. Acquisition Cost	Total value in dollars of all cost associated with the acquisition and installation/modification of each alternative.
2. Investment Period (Remaining Vessel Life)	Period of time in years over which the vessel is expected to operate and produce the anticipated savings, normally, the remaining useful vessel life after conversion or upgrading.
3. Method of Financing	Source of capital to cover the associated acquisition cost, assumed here to be 100% equity by the State of Alaska. (Other sources may include external financing or combinations of part equity and part external financing).
4. Discount Rate	The minimum rate selected by an organization which a prospective investment must return, assumed here as 10%.
5. Fuel Price Escalation Rate	The annual rate in percent at which fuel price is estimated to increase throughout the remaining vessel life.
6. General Economic Inflation Rate	The annual rate in percent at which the cost of non-fuel related goods and services is anticipated to increase throughout the remaining vessel life.
7. Salvage Value	An estimate of the market value of machinery components associated with the conversion or upgrading at the end of the remaining vessel life. In most instances the only real future value of this equipment is that of scrap which is usually quite small in comparison to the original acquisition cost (assumed here as 0.)
8. Annual Operating Cost Elements	For each propulsion plant alternative considered, a first year's operating cost must be quantified. Three elements have been assumed to make up the total annual operating costs associated with each alternative: 1. Fuel Costs 2. Lubricating Oil Costs 3. Maintenance and Repair Costs

TABLE 5.5 CONTINUED

TERMS	DEFINITION
9. Net Present Values, NPV	The total value in today's dollars of all future annual cash flow differentials discounted back at the discount rate selected.
10. Internal Rate of Return, IRR	The rate of interest yielded when the future values of all annual cash flow differentials are assumed to be invested so as to equal the acquisition cost differential for the alternative considered.
11. Simple Payback, SPB	The break-even point of the investment in years when the future values of accrued annual cash flow differentials equals the acquisition cost differential for the alternative considered.
12. Life Cycle Cost, LCC	The total projected cost of an alternative over its expected investment life, including acquisition and annual operating costs.

life (remaining vessel life), the computational procedure escalated projected annual operating expenditures at a rate of 6% per year for maintenance and repair costs and 8% per year for fuel and lube oil costs. This is based on recent historical trends which indicate that annual escalation rates for petroleum related products and by-products have grown at a more rapid pace than non-petroleum based goods and services which would emcompass maintenance and repair cost components. The investment periods utilized here were assumed to be a range of values, from ten to twenty years, equating to the life remaining for the COLUMBIA. A starting date of June 1, 1984 was assumed, which would allow for time required to re-engine if such a decision were made at this time. Because pricing and cost data was estimated in 1983 dollars, these elements were escalated at previously mentioned rates of inflation to reflect costs as of the theoretical June 1, 1984 start date. These and other pertinent economic analysis input data are summarized in Table 5.6.

TABLE 5.6

ECONOMIC ANALYSIS IN-PUT DATA AND ASSUMPTIONS

1. Investment Period (Remaining vessel life)	10,15 and 20 years (See Paragraph 5.3.3)
2. Method of Financing	100% Equity
3. Discount Rate	10%
4. Escalation Rates:	
Maintenance & Repair	6%/Year
Fuel & Lube Oil	8%/Year
5. Salvage Value	-0-

5.3.3 Sensitivity Analysis

To provide a broader scope for the previous described economic analysis, the sensitivity of decision to re-engine versus continued operation of the existing engines was tested relative to the impact of varying investment period/remaining vessel life and acquisition cost estimates for the re-engining alternative. These sensitivity analyses were performed as an integral part of the computer calculations referenced earlier, the results of which are contained in Appendix I.

5.4 Discussion of Results

The results of the computer-based economic analysis calculations are presented in Tables 5.7 and 5.8, for both alternatives operating on MDO and for the existing engines running on MDO and the new engines on HFO, respectively. Addressing Table 5.7, first, it can be seen that, for the re-engining alternative operating on MDO, it is only when the acquisition cost is assumed to be $6,000,000 and the investment period ten (10) years that all financial indicators support the decision to re-engine. Any increase beyond $6,000,000 as an assumed acquisition cost while maintaining the investment period of time available to recover the capital expenditure required for engine change out at ten (10) years, results in various indicators failing to support re-engining. These are manifested as a decrease in IRR to zero, or as shown, "No Return" which indicates that the investment would not be recouped in the basis of annual savings over continued operation of the existing engine within

TABLE 5.7

SUMMARY OF RE-ENGINING ECONOMIC ANALYSIS RESULTS FOR NEW ENGINE OPERATION ON MDO

ASSUMED RE-ENGINING ACQUISITION COST	ASSUMED REMAINING VESSEL LIFE 10 YEARS	ASSUMED REMAINING VESSEL LIFE 15 YEARS	ASSUMED REMAINING VESSEL LIFE 20 YEARS
$6,000,000	NPV: -$1,255,124 IRR: 4.5% LCC Δ-$1,531,916 SPB: 8.25 Years	NPV: $331,499 IRR: 10.9% LCC Δ-$7,031,500 SPB: 8.25 Years	NPV: $1,722,831 IRR: 13.5% LCC Δ-$14,799,070 SPB: 8.25 Years
$7,000,000	NPV: - $2,255,124 IRR: 1.4% LCC Δ-$531,916 SPB: 9.4 Years	NPV: -$668,500 IRR: 8.3% LCC Δ-$6,031,500 SPB: 9.4 Years	NPV: $722,831 IRR: 11.3% LCC Δ-$13,799,070 SPB: 9.4 Years
$8,000,000	NPV: - $3,255,124 IRR: No Return LCC Δ $478,054 SPB: 10.5 Years	NPV: - $1,668,501 IRR: 6.3% LCC Δ-$5,031,500 SPB: 10.5 Years	NPV: - $277,169 IRR: 9.55% LCC Δ-$12,799,070 SPB: 10.5 Years
$9,000,000	NPV: - $4,255,124 IRR: No Return LCC Δ $1,468,084 SPB: 11.5 Years	NPV: - $2,668,501 IRR: 4.7% LCC Δ-$4,031,500 SPB: 11.5 Years	NPV: -$1,277,169 IRR: 8.1% LCC Δ-$11,799,070 SPB: 11.5 Years

TABLE 5.8

SUMMARY OF RE-ENGINING ECONOMIC ANALYSIS RESULTS FOR NEW ENGINE OPERATION ON HFO

ASSUMED RE-ENGINING ACQUISITION COST	ASSUMED REMAINING VESSEL LIFE		
	10 YEAR	15 YEAR	20 YEAR
$6,000,000	NPV: $5,110,098 IRR: 27.2% LCC△ -$12,554,115 SPB: 3.75 Years	NPV: $9,516,611 IRR: 30.4% LCC△ -$27,847,131 SPB: 3.75 Years	NPV: $13,506,697 IRR: 31.3% LCC△ -$50,149,680 SPB: 3.75 Years
$7,000,000	NPV: $4,110,098 IRR: 22.1% LCC△ -$11,544,115 SPB: 4.4 Years	NPV: $5,516,611 IRR: 25.9% LCC△ -$26,847,131 SPB: 4.4 Years	NPV: $12,506,697 IRR: 27.1% LCC△ -$49,149,680 SPB: 4.4 Years
$8,000,000	NPV: $3,110,098 IRR: 18.2% LCC△ -$10,554,115 SPB: 5.0 Years	NPV: $7,516,611 IRR: 22.5% LCC△ -$25,847,131 SPB: 5.0 Years	NPV: $18,173,831 IRR: 24.0T LCC△ -$48,149,680 SPB: 5.0 Years
$9,000,000	NPV: $2,110,098 IRR: 15.0 % LCC△ -$9,554,115 SPB: 5.6 Years	NPV: $6,516,611 IRR: 19.9% LCC△ -$24,847,131 SPB: 5.6 Years	NPV: $10,506,697 IRR: 21.6% LCC△ -$47,149,680 SPB: 5.6 Years

the given investment period. Also, a change in LCCΔ from negative to a positive dollar value for a given combination of new engine acquisition cost and remaining vessel life does not support re-engining. This is because LCCΔ has been set up mathematically to equal the remaining dollar value when total life cycle costs associated with continued operation of the existing engines is subtracted from the equivalent costs associated with re-engining. Thus, as long as life cycle costs for the re-engining alternative are less than those for continued Enterprise engine operation, the LCCΔ will be negative. A change to positive indicates that this life cycle cost relationship has been reversed. Also, between the three (3) financial indicators, NPV, IRR and LCCΔ, for certain acquisition cost/time scenarios in Table 5.7, an apparent conflict seems to occur, that being the fact that the IRR and LCCΔ values tend to suport re-engining while the NPV value shown is negative. This is due to the fact that IRR and LCCΔ are computed on the basis of future inflated cash flow differentials while NPV represents the sum of all cash flow differentials during the investment period in todays dollars discounted back at 10%. Thus, a negative NPV may not in itself mean that the re-engining should not be undertaken, but that it does not begin to payback until late in the investment period in terms of accuring positive annual cash flow differentials. This relationship can be easily determined by inspection of predicted annual cash flow data for the alternative in question contained in Appendix 1. Simple payback, SPB, speaks for itself in that the shorter this period is in years, generally, the more attractive the investment. The results shown in both Tables 5.7 and 5.8 are absolute but do not show a clear optimum scenario in that a minimum time period required to recoup the investment or that a certain rate of return has been obtained. These criteria, it is assumed, will be established and factored by AMHS.

Referring to Table 5.8, it becomes immediately obvious that re-engining for HFO operation is an economically superior alternative to re-engining for continued MDO operation. All that remains to be identified is a satisfactory rate of return on investment and how long in years this return should take, and a range of satisfactory capital cost/investment period scenarios for re-engining for HFO operation can be selected.

6.0 CONCLUSIONS AND RECOMMENDATIONS

Based on the results of the detailed review and analysis of current performance data and main engine maintenance and repair history and related cost information as presented in previous report sections, numerous conclusions and resultant recommendations have been made which are presented in this section of the report. For the sake of organizational clarity and brevity, the attendant conclusions and recommendations have been divided into the following relevant categories:

- Sea Trial Performance
- Adequacy of the Engine De-Rating
- Additional Modifications
- Economic Evaluation of Re-engining

* <u>Sea Trial Performance</u>:

1. The engines as de-rated by TDI failed to develop the required power outputs as specified in the work scope of the contract authorizing this work.

2. The turbochargers, as indicated by surge problems observed during the trials and on subsequent voyages are not properly matched to the new de-rated engine operating profile. Emperical data presented in Section 2.0 further supports this conclusion.

3. Numerous other problems of a smaller magnitude also identified in Section 2.0, have developed as a result

of the de-rating work and for the most part are unresolved.

4. Adequate air flow appears to have been provided to the engines by the new turbochargers. Brake Mean Effective Pressures at the new operating outputs are equal to, or less than, those specified in the de-rating contract.

5. It is possible that some minor portion of the turbocharger surge problem is related to the difficulties being encountered with the pitch scheduling portion of the main engine control system. TDI should be required to assist and work closely with Mathers Controls to establish responsibility for and correct this situation.

6. Based on the above described performance, TDI should be put on notice that the de-rating work to date is unacceptable and payment withheld.

* Adequacy of the Engine De-Rating:

1. Based on a review of main engine historical maintenance and repair data and a comparison of engine component failure frequency and mode with the modification accomplished as a result of the de-rating effort, it is anticipated that only minimal overall improvement in failure rates and time between failures c. overhauls will occur. The most significant portion of this

improvement will occur for those components directly impacted by the improved combustion process which results from the increased availability of air blown for combustion.

2. It is believed that for the remainder of the engine component failures identified in Sections 3.0 and 4.0 those not directly influenced by increased air flow, little or no change in failure rate, and probably no more than would be obtained by simply running the original engines at a reduced output without officially de-rating will occur. These component failures include:
   - Cyliner heads - design and manufacturing defects
   - Cylinder liner distortion and wear - due to block distortion
   - Piston ring distortion and wear - due to block distortion
   - Cylinder blocks - distortion and cracking
   - Connecting rod bearings - design of articulated connecting rod assembly
   - Main bearings - premature wear, high loading
   - Cam shafts - premature wear

3. It is estimated that when equated to dollars, the reduction in main engine maintenance and repair historical average annual cost resulting from de-rating may approach twenty-five percent (25%).

4. The existing de-rated engines after incorporation of the additional modification identified in this report, can be kept running almost indefinitely if AMSH is willing to continue to maintain them at the same comparatively high rate in terms of time and dollars.

* Additional Modifications:

Numerous additional modifications have been identified in Section 5.0 and should be incorporated to enhance the future reliable and efficient operation of the de-rated engines. Some of the more important of these modifications are a result of, and not in addition to, the de-rating effort. The most significant of these is the turbocharger mismatch which should be rectified by TDI by installing new matched turbochargers at no additional cost to the de-rating contract.

* Economic Evaluation of Re-engining of the M/V COLUMBIA:

1. Re-engining of the COLUMBIA for operation on Marine Diesel Oil, MDO, depending on the acquisition cost estimate/remaining vessel life combination considered, can offer a significant economic advantage over continued operation of the existing de-rated engines on MDO.

2. Re-engining of the vessel to operate on Heavy Fuel Oil, HFO, is a clearly superior economic alternative compared to both re-engining for MDO operation or

continued operation of the de-rated engines on MDO, regardless of the acquisition cost/investment period combination considered in the economic analysis presented in Section 5.0.

Based on the technical analysis and evaluation conducted and documented in this report and the results derived for the range of estimated re-engining acquisition cost/remaining vessel life combinations considered as part of the economic analysis, it is recommended that the M/V COLUMBIA be re-engined for HFO operation at the earliest opportunity.

REFERENCES

1. Delivery Order D.O. 707573 from the State of Alaska TDI for main engine derating new turbocharger installation and Trabon lubricating system in the M/V COLUMBIA, 10/15/82.

2. M. Guralnick Associates, Inc., report, "Performance Predictions and Engine Selection Criteria for the M/V COLUMBIA, June 1982, S.O.A. Contract No. X61744.

3. TDI Report, Shipboard Test, M/V COLUMBIA, Starboard engine, S/N 72033, August 31, 1981.

4. TDI to Seaworthy Engine Systems, Inc., Transmittal dated 3/31/83, Typical C-17-123 compressor performance map and Halter Marine DMRV-16-4 (DE C-17-173 turbos) test stand log sheet performance data.

5. Jarota, M.S. and Watson, M., Turbocharging the Internal Combustion Engine, 1982.

6. Jon O. Jacobson, Engine Rebuild Report - M/V COLUMBIA, March 31, 1981

7. G. Beshouri, J. Siegal, Inspection and Maintenance Report - Port and Starboard Main Engines, M/V COLUMBIA, Transamerica DeLaval, Inc., November 1980.

8. Handbook for Selection of Marine Diesels, DeLaval Engine & Compressor Division, 550 85th Avenue, Oakland, California 94621.

9. Stinson, K.W., M.E., "Diesel Engineering Handbook", 12th Edition, Business Journals, Inc., Stamford, Connecticut 1976.

10. Marine Diesel Standard Practices, Diesel Engine Manufacturers Association, New York, New York, 1971.

11. Lichty, L.C., "Internal Combustion Engines", McGraw-Hill, New York, New York, 1951.

12. Modern Marine Engineers Manual, Vol. II, A. Osbourne, Editor, Cornell Maritime Press, Maryland, 1943.

13. Lamb, J., "The Running and Maintenance of the Marine Diesel Engine", 6th Edition, Charles Griffin & Company, Ltd., London, 1976.

14. Pounder, C.C., "Marine Diesel Engines", Newnes-Butterworthys London, 1972.

15. Standard Handbook for Mechanical Engineers, T. Baumeister & L.S. Marks, Editors, 7th Edition, 1967.

References (continued)

16. Zinner, K., "Supercharging of Internal Combustion Engines", Springer-Verlag, Berlin, 1978.

17. Northwest Laboratories, "Damaged Bearing from Columbia", Report No. E18022, Seattle, Washington, 7 April 1980.

GLOSSARY OF ENGINE RELATED TERMS AND COMMONLY USED FORMULAE

Piston Displacement - The cylinder volume in cubic inches swept by the pistons of an engine. It is equal to the number of cylinders times the area of each piston in square inches times the stroke in inches.

Piston Speed - The total number of feet traveled by a piston in a given time interval, usually expressed in feed per minute. It is sometimes called piston travel.

Horsepower (hp) - A time rate of doing work. One U.S. (and British) horsepower is equal to 33,000 foot-pounds per minute. One horsepower (metric) is equal to 75.0 kilogrameters per second. The relationship between U.S. and metric horsepower is:

One U.S. horsepower equals 1.014 metric horsepower

One metric horsepower equals 0.9863 U.S. horsepower

Indicated Horsepower (ihp) - The horsepower developed in the cylinder. It can be determined from the mean indicated pressure, the engine speed and cylinder dimensions. The formula is shown in the Formula Appendix.

Mean Indicated Pressure (mip)- A defined, constant, hypothetical pressure which would deliver to the top of the piston in one stroke the same work as is actually delivered to the top of the piston by the working fluid in one cycle. The formula is shown in the Formula Appendix.

Brake Horsepower (bhp) - The horsepower delivered by the engine shaft at the output end. The name is derived from the fact that it was originally measured by a brake device. The formula is shown in the Formula Appendix.

Shaft Horsepower - The net power available at the output coupling of a transmission system, such as propulsion gearing, electric propulsion system, slip coupling, etc. It differs from the brake horsepower of the engine by the amount of losses in the transmission device or system.

Brake Mean Effective Pressure (bmep) - A derived factor represented by "P" when the PLAN formula is equated to BHP. It is also equal to the meand indicated pressure (MIP) multiplied by the mechanical efficiency expressed decimally. It cannot be measured directly. See the Formula Appendix.

Torque - A moment which tends to produce rotation. It is the product of force and radius, expressed in pound-feet or pound-inches. See the Formula Appendix.

Indicated Thermal Efficiency - The ratio of the heat equivalent of one horsepower-hour to the number of heat units actually supplied per indicated horsepower-hour. This may be calculated from either the high or low heat value of the fuel, with proper designations as to which value is used. See the Formula Appendix.

Brake Thermal Efficiency - The ratio of the heat equivalent of one horsepower-hour to the number of heat units actually supplied per brake horsepower-hour. This may be calculated from either the high or low heat value of the fuel, with proper designation as to which value is used. See the Formula Appendix.

Mechanical Efficiency - The ratio of brake horsepower to indicated horsepower.

Turbocharger Surging - Thephenomena arising during surging appear from the blower characteristic. Because of the pulsating consumption of air, variations in pressure occur in the scavenging air receiver. The resulting pulsations act back through the air cooler and discharge pipe, which means that the impeller does not work against a uniform pressure, the result being that the amount of air from the blower will vary.

For example, if the air suction filter becomes contaminated, the amount of air through the blowers is reduced. This means that the operation point will move to the left on the blower characteristic, because the effect is the same as an increased resistance to the flow through the blower system and the curve for the flow resistance will then be higher. The result of the above-mentioned pulsations can be that the upper point on the blower characteristic will be exceeded, and the blower ceases to deliver air. The effect is that the flow resistance is reduced and the blower will again deliver air, and this alternating effect will continue, i.e., the blower will not work in a stable manner and will surge.

With engines having more than one turbocharger delivering to the same scavenging air receiver, surging conditions will result in air being pressed backwards through the surging turbocharger by the remaining turbochargers.

The symptoms of surging are:

1. Unusual noise at the suction side of the turbocharger - can be a muffled but violent boom.

2. The amount of air sucked in by the turbocharger can vary a great deal - can be confirmed by placing a piece of paper against the suction filter.

3. The pressure of the scavenging air in the receiver is considerably lower than normal and varies widely.

4. Sharp fluctuations in the air pressure drop during passage through the air filter.

Surging can often be prevented by lifting the safety valve on the scavenging air receiver, and at the same time, reducing the power of the main engine. The turbocharger system must be cleaned at the earliest opportunity.

Surging can be caused by:

1. Contamination of elements in the turbocharger system.

2. Failure in the supply of energy to the turbocharger, for example, due o one or more of the engine cylinders not providing full power.

<u>Bosch Smoke Number</u> - Bosch Smoke Number is an indication of the opacity (clarity) of the exhaust gases existing from a diesel engine as determined by the Bosch Smoke Meter. This is a filtering type smoke meter, usually portable, in which a primary sensor is used to collect a specific volume of exhaust gas by having it flow through a tab of filter paper. Any soot

is trapped by the filter paper. This paper is then put into a photo-electric type reflection meter to determine the Bosch Smoke Number. Generally, a reading of 1.0 indicates a slight hazing of the exhaust gas. Thus, readings falling well below 1.0 indicate a very clean, clear exhaust gas condition which is indicative of more than adequate air flow into the engine.

FORMULAE

1. Horsepower per cylinder (any reciprocating engine) is:

$$hp = \frac{P \times L \times A \times N}{33,000}$$

where

P = Indicated mean effective pressure, psi; or brake mean effective pressure, psi (corresponds with ihp or bhp)

L = Stroke of piston in feet

A = Net piston area sq. in.

N = Number of power strokes per cylinder per minute

2. Brake Horsepower (test stand) is:

$$bhp = \frac{2 \times \pi \times r \times rpm \times W}{33,000}$$

where

r = Distance between the shaft center and the point of application of the weight to the brake arm, in feet

W = Effective weight on the brake arm in pounds

rpm = Revolutions per minute of the brake shaft

π = 3.1416

3. Horsepower per cylinder (any single-acting internal combustion engine) is:

$$hp = \frac{P \times D^2 \times L \times rpm}{C}$$

where

hp = Horsepower per cylinder (bhp or ihp)
P = Mep in psi, Bmep or imep corresponds with bhp or ihp
D = Diameter of cylinder bore in inches
L = Length of stroke in inches
*C = 1,010,000 for four-cycle engines
*C = 505,000 for two-cycle engines

4. Brake Mean Effective Pressure is:

$$\text{bmep} = \frac{\text{bhp} \times 33{,}000}{L \times A \times N}$$

where

bhp = Brake horsepower per cylinder and L, A and N are the same as mentioned in formula 1 for horsepower per cylinder.

5. Mean Indicated Effective Pressure is:

$$\text{imep} = \frac{\text{ihp} \times 33{,}000}{L \times A \times N}$$

where

ihp = Indicated horsepower per cylinder and L, A and N are as mentioned in formula 1 for horsepower per cylinder.

6. Mean Effective Pressure is:

$$P = \frac{\text{hp} \times C}{D^2 \times L \times \text{rpm}}$$

where

hp = Horsepower per cylinder (bhp or ihp)
P = Mep in psi, Bmep or imep corresponds with bhp or ihp
D = Diameter of cylinder bore in inches
L = Length of stroke in inches
*C = 1,010,000 for four-cycle engines
*C = 505,000 for two-cycle engines

Formulae (3) and (5) may be used for engines having cylinder dimensions in metric units, with modification of constants as follows: (The hp will still be in British or U.S. units of 33,000 ft-lbs per min.)

P = Psi as before
D = Diameter of cylinder bore in centimeters
L = Length of stroke in cencimeters
*C = 16,500,000 for four-cycle engines
*C = 8,250,000 for two-cycle engines

7. Brake Mean Effective Pressure is:

$$\text{bmep} = \begin{bmatrix}\text{indicated mean effective} \\ \text{pressure (imep)}\end{bmatrix} \times \begin{bmatrix}\text{mechanical efficiency} \\ \text{expressed decimally}\end{bmatrix}$$

8. Torque in ft-lbs = $Q = \dfrac{5252 \times hp}{rpm}$

where

hp = Transmitted horsepower
rpm = Rotational speed or shaft in revolutions per minute

9. Piston Speed = fpm = length of stroke in feet x rpm x 2

10. Indicated Thermal Efficiency = $E_i = \dfrac{2544}{H \times w_i}$

where, for oil Diesel engines

H = High heat value of fuel used
w_i = Fuel consumption in lb/ihp/hr

or, for gas and dual fuel Diesel enginès

H = Heat value of fuel used (hhv for fuel and lhv for gas fuel)
w_i = Fuel consumption/ihp/hr (consumption in lb for fuel oil and cu ft for gas)

11. Brake Thermal Efficiency = $E_b = \dfrac{2544}{H \times w_b}$

where, for oil Diesel engines

H = High heat value of fuel used
w_b = Fuel consumption in lb/bhp/hr

or, for gas and dual fuel Diesel engines

H = Heat value of fuel used (hhv for fuel and lhv for gas fuel)
w_b = Fuel conspumption/bhp/hr (consumption in lb for fuel oil and cu ft for gas)

12. Mechanical Efficiency in per cent $= \frac{bhp}{ihp} \times 100$

13. Horsepower Requirements of Pumps:

(a) Circulating water pumps, for jacket water or raw water systems, when total dynamic head is specified in feet of water:

$$\text{hp input} = \frac{\text{gpm} \times H}{C \times e}$$

where

H = Total dynamic head expressed in feet of water
C = 3960 for fresh water (62.4 lb/cu ft)
C = 3855 for salt water (64 lb/cu ft)
e = Pump efficiency, expressed decimally

(b) Lubricating oil or fuel oil pumps:

$$\text{hp input} = \frac{\text{gpm} \times p}{1720 \times e}$$

where

p = Discharge pressure, psi
e = Pump efficiency, expressed decimally

(with the discharge head expressed in psi, the constant 1720 is independent of variations in density of the liquid pumped. Horsepower capacities of oil pump mechanical drives or electric motors must be sufficient to start the pump with cold oil, usually assumed to have a maximum viscosity of 3000 SSU. The pump size must be selected to give the required capacity with hot oil, having a viscosity assumed to be 100 SSU.)

14. Specific fuel consumption correction factor for fuels of various high heat value:

$$\text{Factor} = \frac{\text{btu (hhv)}}{19{,}350}$$

where

(hhv) = the high heat value of the fuel used

---

* Approximate values acceptable for computation. Note that the constant is based on fps system. Bore and stroke are given in inches and hp is British or U.S. of 33,000 ft-lbs per min.

APPENDIX A

TRIAL AGENDA
M/V COLUMBIA, MARCH 24, 1983

SEA TRIAL AGENDA
M/V COLUMBIA - MARCH 24, 1983

Leave Pier 48 @ 0700 HRS.

Enterprise Break-In:

	Eng. RPM	Shaft RPM	% Rated Pwr	Approx. BHP	Time
	300	166.94	40%	2500	1 HR
Mid Power Cruise	330	183.63	53%	3300	1 HR

Engine Performance Test:

	Eng. RPM	Shaft RPM	% Rated Pwr	Approx. BHP	Time
	360	200.33	69%	4300	1 HR
Design Svc Cruise	385	214.24	85%	5248	1 HR
MCR @ 100%	403	224.26	100%	6164	4 HR
10% Overload	403	224.26	110%	6791	1 HR

Controls/Turbocharger Response Test:

(Mather Controls to supply)

Min. to include -	
Bridge/ER/Local Control Test/Transfer	1 HR
(Slow) Ahead/Stop/Astern	1 HR
Full Ahead/Full Astern	1 HR
	12 HRS

Estimate Return Pier 48 - 1900 HRS.

APPENDIX B

M/V COLUMBIA
MARCH 24-25, 1983 TRIAL DATA

[illegible]

<u>M.V. COLUMBIA</u>, STBD MAIN ENGINE

TUNING PARAMETERS	ENGINE RPM									
	330		363		385		405			
DATE	3/24/83		3/25/83		3/25/83		3/25/83			
RUNNING TIME, MINS.	56		60		25		100			
*SHAFT RPM	182		194.5		214		221.2			
*TORQUE FT-LB x 10^3	89		93.9		112.6		143.65			
*SHP	3109		3567.5		4558.5		6054			
F.O. SUPPLIED, GALS	203		352		167		731			
F.O. RETURNED, GALS	105		100		44		169			
F.O. CONSUMED, GALS.	98		252		123		562			
% O_2 AFTER TURBO	12		11.5		11.5		13.6			
SEA TEMP °F	56		49.5		55		53			
SHIP SPEED, KNOTS	11.3		17.0		18.2		19.65			
BAROMETER, INS. Hg A	29.45		29.86		29.86		29.86			
PROPELLER PITCH	.475		24°		25°		25°			
TURBO RPM OB/INB.	460/435		755/703.5		861/871		944.7/950.7			
BLOWER PRESS. PSIG	3.3/3.2		11.15/11.1		15.1/15.1		15.60/15.47			
L.O. PUMP RPM	1750		1750		1750		1750			
J.W. PUMP RPM	1750		1750		1750		1750			
F.W. PUMP RPM	1750		1750		1750		1750			
SW PUMP RPM	1750		INTERMITTENT OPERATION DUE TO FAILURE OF PILOT VALVES							
LO PUMP ΔP	55		54		56		53			
JW PUMP ΔP	35		16		17		19			
FW PUMP ΔP	18		34		35.5		36			
SW PUMP ΔP	16.5		INTERMITTENT OPERATION DUE TO FAILURE OF PILOT VALVES							
RED. GEAR L.O. PRESS	35.5		34.5		34.5		34.5			
RED. GEAR L.O. IN TEMP, °F	109		113		118		123			
DRAFT: F 13'5", A 16'6", MH 8'5"										

* FROM INSTANTANEOUS READINGS FROM TORSIONMETER.

Power Engine Factory Test Log

LOG B

CUSTOMER ______ MODEL ______ TYPE SERVICE ______ JOB NO ______ ENGINE NO ______ DIRECTION OF ROTATION (AS VIEWED FROM FLYWHEEL END) ______

LB RB

EXHAUST TEMPERATURES - °F

Dual Converter — TURBINE INLET (T - TOP B - BOTTOM)

	Line	LFTL	LFTB	LFB	RFT	RFB	LRT	LRB	RRTL	RRTB	RRB	Turbine Exhaust LF	RF	LB	RB
1	1	5423	6216									5727	5618		
2	2	7326	7507									6194	665.7	Bonus	
1	3	7716	[illegible]									6464	6155		
2R	4	7062	530.3									6295	6272	[illegible]	
3	5	[illegible]	537.6									6987	616.2		
4	6	7661	[illegible]									5972	6455		
	7														
	8														
	9														
	10														

EXHAUST ELBOWS

Line	1L	2L	3L	4L	5L	6L	7L	8L	9L	10L	1R	2R	3R	4R	5R	6R	7R	8R	9R	10R	AVG	Line
1	570	560	565	550	555	510	510	575			555	530	570	570	540	575	575	570			554	1
2	640	620	600	618	645	675	445	675			620	626	670	590	680	725	613	670			662	2
3	[illegible]				[illegible]								[illegible]		[illegible]							3
4	[illegible]				[illegible]								[illegible]		[illegible]							4
5	[illegible]				[illegible]								[illegible]		[illegible]							5
6	[illegible]				[illegible]								[illegible]		[illegible]							6
7																						7
8																						8
9																						9
10																						10

	Line	Engine Water PSIG to Engine	To Engine (°F)	From Engine (°F)	Turbocharger Water Outlet (°F) LF	RF	LR	RR	Aftercooler Water Inlet (°F) F	R	Outlet (°F) F	R
1	1	42.5		167								
2	2	33		167								
1	3											
	4								Run INTERRUPTED			
	5											
2R	6											
3	7											
4	8											
	9											
	10											

FIRING PRESSURES - PSIG \ [illegible]

Line	1L	2L	3L	4L	5L	6L	7L	8L	9L	10L	1R	2R	3R	4R	5R	6R	7R	8R	9R	10R	AVG	Line
1	610	550	552	520	530	520	520	570			570	550	540	580	570	570	570	580			548	1
2	860	770	740	700	700	700	760	810			700	800	780	710	760	770	790	800			755	2
3	930	920	870	850	550	810	550	910			840	860	860	880	820	830	810	800				3
4	1050	1010	990	920	973	920	920	1010			1010	1050	960	1050	965	910	960	910				4
5					1070	970	1030	1070														5
6	1230	1100	1050	950	970	900	965	1000			920	960	950	900	910	940	930	810				6
7	1220	1200	1110	1180	1110	1100	1100	1110			1120	1050	1050	1050	1050	1050	1010	1150			26.1	7
8	1170	1180	1105	1060	1050	1100	1150	1130			1040	1080	1070	1060	1050	1120	1050	1020			25.6	8
9																						9
10																						10

Line	Lubricating Oil Pressure - PSIG From Pump	To Engine	Cam Header	To Turbocharger LF	RF	LR	RR	Lubricating Oil Temperature °F To Engine	From Engine	Turbocharger Drain LF	RF	LR	RR	Crank Case Press (in H2O)
1			57	16	16				167					
2			49	17	17				167					
3														
4														
5														
6														
7														
8														
9														
10														

REMARKS

1. PITCH 1.0 TURBO 4760 & 4400
2. 1.3 6560 & 6740
3. 3.8 ALL FOUR TURBO HOWLING HEAVILY
4. – RUN (2) REPEAT - RUN 2 INTERRUPTED DUE TO SPEED CHANGE
5. LOST FUEL RETURN LINE PORT MAIN
6. START 11:22 - FINISHED 11:37
7. START 13:05 - FINISHED 13:20
8. START 14:05 - FINISHED 14:20

LEGEND L, LB - LEFT BANK F - ENGINE FRONT AT NO. 1 CYLINDER
R, RB - RIGHT BANK R - ENGINE REAR FLYWHEEL END
(WHEN VIEWED FROM FLYWHEEL END)

Transamerica Delaval
Transamerica Delaval Inc
Engine and Compressor Division

TURBOCHARGER MAKE & MODEL
LF ______ RF ______
LR ______ RR ______

OPERATOR ______ DATE 5/25/83
ENGINEER ______ LOG B
WITNESS ______ JOB NO ______

Power Engine Factory Test Log

LOG A

CUSTOMER ______ MODEL ______ TEST STAND ______ JOB NO ______ ENGINE NO ______ FULL LOAD RATING ______ BHP @ ______ RPM ______ BMEP

	Line	Date	Wall Clock Time On Load	Wall Clock Time This Log	% Load (Nom.)	Dyno RPM	Scale (N.M)	BHP (Corr.)	BMEP	Disc Revs	Time (sec)	Power Factor	A.C. Volts	A.C. Amps	KW	Gen Eff	BHP (Corr.)	BMEP	Gov Load Limit
1	1	3/24				510		1405											
2	2	3/24				520		3175											
1	3	3/25				500		4460											
2R	4	3/25				537		4450											
3	5	3/25				503		5050											
4	6	3/25				503		4750											
	7							▲											
	8							BHP PER RACK & SPEED											
	9																		
	10																		

Fuel Pump Rack Position (MM)

Line	1L	2L	3L	4L	5L	6L	7L	8L	9L	10L	1R	2R	3R	4R	5R	6R	7R	8R	9R	Avg	Line
1	16	17	17	16.5	17	16.5	17	16			12.5	12.5	12.5	—	18	17	18	17		17	1
2	21	22	22	22	22.5	22	22	22			21	23	21.5	22	23	21.5	23	21.5		22.25	2
3	23	25	24	24	24	24	23	23			24	24	25	25	25	24	24	25		24.1	3
4	24	25	25	25	25	25	25	24			26	25	25	25	25	25	25	25		24.9	4
5	26	27	26	26	26	26	26	25			27	26	25	26	27	26	25	25		26.1	5
6	25	26	26	25	25	25	25	25			25	26	26	26	26	26	26	26		25.6	6
7																					7
8																					8
9																					9
10																					10

Turbocharger Pressures

	Line	Blower Inlet (in. H2O) LF	RF	LR	RR	Blower Discharge (in. Hg) LF	RF	LR	RR	Air Cooler LR	RR	Turbine Inlet (in. Hg) LF	RF	LR	RR	Turbine Exhaust (in. H2O) LF	RF	LR	RR
1	1	6.5	6.5			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5.4	[illegible]	1.5	[illegible]		
2	2	1.4	1.4			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	12.25	[illegible]	[illegible]	0.6	1.6		
1	3	1.7	1.7			[illegible]	21.7	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.1	[illegible]		
2R	4	2.4	2.4			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.5	[illegible]		
3	5	2.9	2.4			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	26.7	[illegible]	[illegible]	0.8	[illegible]		
A	6	2.2	2.3			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.5	[illegible]		
	7																		
	8										* SYSTEM SURGING SLIGHTLY								
	9																		
	10																		

Turbocharger Temperatures / Air Flow Nozzle Size / Ambient

Line	Blower Inlet (°F) LF	RF	LR	RR	Blower Discharge (°F) LF	RF	LR	RR	Air Manif (°F) LB	RB	Turbocharger RPM LF	RF	LR	RR	Nozzle ΔP (in. H2O) LF	RF	LR	RR	Air Flow SCFM Total	Baro (Corr.) in. Hg	Air (°F)	Line
1	[illegible]	[illegible]			[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]							[illegible]	1
2	[illegible]	[illegible]		75	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]									[illegible]	2
3	[illegible]	[illegible]			[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]							[illegible]	3
4	[illegible]	[illegible]			[illegible]	[illegible]			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]							[illegible]	4
5	[illegible]	[illegible]			[illegible]	[illegible]			118	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]							[illegible]	5
6	[illegible]	[illegible]			[illegible]	[illegible]			118	[illegible]	[illegible]	[illegible]	931	[illegible]							115.5	6
7																						7
8									* ERROR													8
9																						9
10																						10

Fuel Oil Data / Diesel BSFC / Fuel Gas Data / Gas BSFC / Dual Fuel BSFC / Remarks

	Line	Manif (psi)	API Gravity As Read @ °F	API Corrected To 60°F	LHV (Btu/lb)	Lbs Meas	Time (sec)	lb/hr	Btu/bhp-hr	(Corr.) lb/bhp-hr	Remarks
											SMOKE RDG 0.4/0.5/0.4 BOSCH SMOKE NO.
1	1										0.2/0.4/0.5
2	2										0.45/0.35/0.1
1	3		36.3	91							0.2/0.25/0.2
2R	4										0.45/0.45/0.2
3	5										0.3/0.3/0.3
4	6										
	7										
	8										
	9										
	10										

INJECTION/IGNITION TIMING °BTDC ______ LB ______ RB

FUEL OIL SPEC ______

LUBE OIL SPEC ______

Transamerica Delaval
Transamerica Delaval Inc
Engine and Compressor Division

AIR FLOW NOZZLE COEFFICIENT
LF ______ RF ______
LR ______ RR ______
(FLYWHEEL END)

OPERATOR ______ DATE 3/25/83
ENGINEER ______ LOG NO A
WITNESS ______ JOB NO ______

APPENDIX C

M/V COLUMBIA SHAFT HORSEPOWER MEASUREMENT
SEA TRIALS MARCH 24-25, 1983