Coltec Industries

Fairbanks Morse Engine Division

ENGINEERING REPORT

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FILE NUMBER 35-734133

SUBJECT: BG&E Remanufactured Spare Emergency Diesel Genset

DATE: September 13, 1994

PREPARED BY: J. Cooper

REPORT TITLE: Results of 2000 Hr. Test at 385 bhp/cyl. on 38TD8-1/8 O.P. Engine, Report #R-5.08-0236 (9/13/94)

APPROVED BY: Terry Miller

FILE: R-5.08-0476, R-5.08-0236, R-5.08-0515, R-5.08-0530

ENCLOSURES: APPENDIXES A - D
FIGURES 1 - 33

OBJECTIVE

The purpose of this report is to present the results of the 2000 hour test of the subject genset including the disassembly and inspection following the test.

INTRODUCTION

The purpose of this test was to prove the capability to operate the genset at 3300 kilowatts for 2000 hours. This test was to be run in accordance with procedure 10560978. The engine used for this test was Baltimore Gas & Electric engine built as a 38TD8-1/8 Opposed Piston engine with S/N 38D885002TDSM12 whose original rating was 300 hp/cyl.

BACKGROUND

The engine was converted from turbo-blower parallel arrangement to turbo-blower series in order to increase the rating from 300 bhp/cyl to 350 bhp/cyl continuous and 385 bhp/cyl for 2000 hours. A 200 hour test at 3600 kw was run prior to the 2000 hour test. After the 200 hour test, the engine was inspected which included removal and disassembly of all pistons. During the reassembly, all main and rod bearings were replaced on the upper crankline, the rod bearings and selected main bearings were changed on the lower crankline, and the piston rings were replaced on all the pistons. The results of the 200 hour test were discussed in a Qualification Report dated 1/25/94 and filed under 35-734133.

CONCLUSIONS

The engine operated for 2012 hours at the 2000 hour contract rating of 3300 kw. The test was completed with failures of the injection pump erosion sleeves due to cavitational erosion. These failures necessitated two shutdowns of the engine due to low exhaust temperatures. The erosion sleeve failure was not found after the first shutdown. The second shutdown would not have happened if the problem was found after the first shutdown. A new injection pump design is being investigated to eliminate this problem.

The power components did not show distress which can be related to the increased loading. Broken Piston Rings were found in Cylinders 8 and 12 which evidence shows are the result of installation errors. Increased wear was found as a result of the increased loading, but was well within the condemnable limits of the parts.

Some conditions were found with other engine parts, such as air start check valve bodies, an air start valve, injection nozzles, camshafts and flex pump drive gears. These were replaced to bring the engine to an "as-new" condition. The condition of these parts were not severe enough to affect the 2000 hour rating.

Some distress was seen in the torsional damper parts. This distress does not affect the 2000 hour rating, but the damper and its parts should be inspected after running 2000 hours at 3300 kw in lieu of 8700 hours as would be expected at a lower load of 3000 kw.

It should be noted that the engine parts accumulated 2462 hours of operation at which 2219 hours were at increased loads. No wear or distress of engine parts was observed that would cause a reduction in the contract load carrying capacity with the possible exception of the injection pump erosion sleeves.

The data from the 2000 hour test and the inspection show that the test was successfully completed with respect to the engine power components with the exception of the injection pumps. The engine has a new 2000 hour rating of 3300 kw contingent upon the successful testing of the new injection pump design. The engine has a new 200 hour rating of 3600 kw with no limitations.

DISCUSSION

The Test

The endurance test at the 2000 hour rating was commenced on February 2, 1994. The test engine was loaded to a minimum of 4656 bhp at 900 rpm. This load is the horsepower required to generate 3300 kw at a 0.8 lagging power factor.

The testing was uneventful until February 25, 1994 (approximately 550 hours into the test) when a 40 psi drop in firing pressures (from 1360 to 1320 psi) was noted for cylinder 12. Within six days this firing pressure had fallen another 60 psi from 1320 to 1260. By March 15 (1000 hours into the test), the firing pressure had fallen another 100 psi to 1160 psi as shown in Figure 1. It was believed that the piston rings had broken in cylinder #12. Firing pressures in the eleven remaining cylinders stayed at or near their normal values during this time.

On March 29, 1994 (the 1313 hour mark), the operators noted that the engine was under stress. This stress was described as a 1000 kw loss of load, 6 psi in air receiver pressure and reduced exhaust temperatures in cylinders one through four. The engine was shut down and the problems were investigated. The results of the investigation are covered in an Engineering Report written by J. Cooper, dated 3/31/94, and filed under

R-5.08-4306 and R-5.08-0006. During this time, the #12 lower piston and rings were replaced, although this was not the cause of the forced shutdown.

On April 15, 1994, twenty-four hours after the engine was restarted the operators again noted cooling of exhaust temperatures in cylinders one through four and a loss in load. An investigation began as discussed in the Engineering Report written by J. Cooper, dated 4/25/94, and filed under R-5.08-4306. The investigation showed that the injection pump erosion sleeves were breaking due to cavitational erosion during the injection pump stroke.

The test was uneventfully continued from the hour mark 1313 until 2012 hours on June 9, 1994. During the last 700 hours, the engine was shut down at 1542 and 1765 hour meter to inspect the condition of the injection pump erosion sleeve degradation on six injection pumps for cylinders 1, 2 and 3. The results of these inspections are discussed under the section titled inspection.

The performance results of testing are included as Appendix A. Appendix A, Item 2 shows a graph of Average Exhaust and Preturbine Temperatures vs. Air Receiver Temperatures. Note that the Actual Averages follow the Predicted Averages very well. As seen in Appendix A, Items 2-6; engine performance was good during the 2000 hour test.

Lube oil consumption throughout the test is charted in Appendix A. The average lube oil consumption during the 2000 hour test was 5173 hp-hr/gal. Instantaneous lube oil consumption is graphed in Appendix A, Item 7. Lube oil consumption was considered to be within the normal limits throughout the 2000 hours.

Lube oil analyses were taken each week throughout the test. The purpose of the analyses was to trend metal content to detect wear of power parts. As seen in Appendix B, the wear metals did not rise significantly during the test. We did note however that the carbon content rose when the piston rings broke at or near 350 hours. The viscosity of the oil stayed constant throughout the test and did not show lube oil breakdown or thinning.

The Inspection

With the exception of the injection pump erosion sleeve checks at 1542 and 1765 hours, all inspections and parts replacements discussed in this section of the report took place following the completion of the 2000 hour test.

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INJECTION PUMPS

Wear of the erosion sleeves was seen to have started 200 hours after their replacement as shown in Figure 2. The degradation progressed as seen in Figure 3 at 400 plus hours and showed further progression at 700 hours as seen in Figure 4. Injection pumps from cylinders 5 Opposite Control Side (OCS) and 9 Control Side (CS) contained erosion sleeves which had cracked (Figure 5). The chart in Appendix C, shows the condition of all erosion sleeves.

The injection pump plungers showed erosion as well. The erosion was seen as a donut in the middle of the plunger head, and a donut or partial donut near the helix angle with a line attacking and perpendicular to the helix edge. Figures 6 and 7 show the "worst case and the best case" respectively. The plungers were replaced in 14 of the 24 injection pumps due to helix erosion.

Slight erosion seen as a frosted radiusing of the port holes was seen on the injection pump barrel. A small pit was seen in the #7 OCS barrel. This was oriented roughly 20° around the barrel periphery and roughly one centimeter up the barrel from the port hole as seen in Figure 8. Turning the plunger the same amount as it would be turned at rack 8 showed that this pit was on the vertical edge of the helix low end (See Figure 9). This would be the low pressure area during the stroke of the pump making it more susceptible to cavitation. In this case, the mechanics of cavitation caused erosion in the plunger and barrel instead of the erosion sleeve.

INJECTION NOZZLES

All of the nozzles exhibited a poor spray pattern and leaked during a pop test inspection. Cleaning the tip needle a couple of times did not rectify the problem on all tips except one. This required changing 23 out of 24 tips. An inspection of the needles showed that cavitation was taking place on the corner between the tip seat and adjoining shoulder. The erosion caused by cavitation results in a leaky seat and consequently poor spray pattern and tip leakage. This cavitation is a common occurrence in diesel nozzles, and is a function of engine hours and load. The Operating & Maintenance Manual calls for nozzles to be checked every 2160 hours. This engine ran in excess of 2300 hours. Therefore, this cavitation is not considered abnormal. In any case, the engine performance did not suffer as shown by the graphs in Appendix A.

PISTON ASSEMBLIES AND PISTON RINGS

The #2 lower piston was found to have a broken oil drain ring (See Figure 10). The proximity (not near the ring gap or the back - 180° from the ring gap) and the type of break (a piece broken out of one small area) identified this as a piston installation error.

The first and second compression rings of the #8 lower piston were broken as seen in Figure 11. The third ring and the remainder of the second ring were extremely worn. Almost half of the first ring was missing, while the other half was intact although extremely worn. This evidence suggest that the ring broke first at the back due to stresses from opening the ring too far when installing the ring on the piston. The ports caught part of the ring causing it to break up. The second ring was broken very near the ring gap. This implies that the ring caught a burr on the ports raised by the broken first ring. The extreme wear of the remaining rings was caused by the lack of oil on the cylinder walls due to hot blowby gases pushing it away and burning it off; and by the roughened surface (scuff) of the liner wall which occurs when blowby increases sizes of parts and consequently clearances shrink. This piston also showed slight signs of scuffing. A scuff is a common result of broken rings.

The piston ring gaps on all remaining rings increased by as muc[illegible] .046 inches as shown in Appendix D, Item 1. The piston rings we[illegible] [illegible]aced although the wear was well within the condemnable limits.

The piston assemblies were disassembled. All parts were dimensionally and visually inspected. No parts showed significant wear dimensionally as can be seen in Appendix D, Items 2 through 9. Visual distress was seen on cylinders #8 and #12 lower pistons. The distress on the #12 piston is a result of running a new piston in a scuffed liner. This scuff was addressed in the Engineering Report dated 3/31/94. The distress on the #8 piston is the result of recent piston ring breakage. These two pistons were replaced during reassembly. All other parts were reused.

CYLINDER LINERS

The cylinder liners were visually and dimensionally inspected. Scuffing was found in cylinder liners #8 and #12 which necessitated replacement of the liners. As seen in Appendix D, Items 10-19; all undistressed liner bores were within the condemnable limits of 8.145. The highest wear was .004 in the exhaust port area. This wear is over the entire run time of the engine.

MAIN AND CONNECTING ROD BEARINGS

Replacement of all main and connecting rod bearings was required. The crush and freespread of most bearings was below the limit called for by blueprint specification. The loss of crush and freespread is not considered detrimental to operation if it occurs while the bearings are installed. Installing bearings which have lost crush and freespread can lead to assembly errors and a possible bearing failure. If the bearing shell does not snap into the cap or saddle during installation, then the loss of freespread has occurred.

UPPER CRANKSHAFT

Aluminum had transferred from two of the bearings to their corresponding journals. Aluminum was found on the #13 main (thrust) bearing journal as seen in Figure 12 and #5 connecting rod journal as seen in Figure 13. This aluminum transfer is caused by improper crankshaft journal surface finishes. The poor finish also affects the bearing as noted by scratches in Figures 14 and 15. This condition is not considered a failure. The poor finish was a result of a poor finishing operation that took place during the pre-2000 hour inspection.

TORSIONAL DAMPERS

The upper torsional damper was found to have a worn 6th order pin as seen in Figure 16. This wear was denoted by a flat spot on one side of the pin. Tests showed that the case hardness was within specifications. Nine of sixteen upper damper spider bushing showed fretting on the O.D. as seen in Figure 17. Two more had flat spots on the I.D. All upper damper spider bushings were replaced as well as the worn pin. Dimensional inspection of the bushings and the remaining pins did not show in-service overlimits.

Cracks were found in the 3rd order weights of the lower torsional damper shown in Figure 18. The cracking, found by wet magnafluxing, were found to be outside of the limits set by Engineering Instruction 2408FM7. The cracking is believed to be a result of the presence of non-metallic inclusions in the base material.

From the conditions seen during this inspection, there is a concern that at a load of 385 bhp/cyl., the damper is being run near its design limit. It would be good practice to inspect the damper every 2000-2200 hours at 385 bhp/cyl for wear. If wear is evident, the parts should be replaced.

GOVERNOR DRIVE AND FLEX PUMP DRIVE

Pitting of the gear teeth was seen on the Flex Drive Gear in Figure 19. Pitting was also seen on the Governor Drive and the Lube Oil Pump Drive Gears. The backlashes of all gears were within specification. The drive gear flex action, due to its spring loaded coupling to the crankshaft, could cause the drive gear to turn with a rapidly fluctuating angular velocity. The pump driven gears tend to turn at a constant angular velocity. The differences in angular velocities creates a varying gear mesh and results in gear pitting. Gear pitting can be minimized by ensuring gear lineup and tooth contact is correct.

VERTICAL DRIVES

The wear plates and two bushings were worn in the vertical drive center section. Replacement of these items is normal as they are considered to be wear items. One of the bushings that was replaced showed fretting. This was found to be a bushing that was manufactured before the bore size was increased. A new bushing will have less tendency to fret due to increased clearances.

CAMSHAFTS

Upon visual inspection of the camshafts, two lobes showed signs of distress. The #10 CS lobe showed wear as seen in Figure 20. This wear extends across the lobe on the pumping ramp. The hardness was 48 Rockwell C in the damaged area and 58 to 61 on the rest of the lobe. The drawing calls for 60 to 65 Rockwell C. The #8 OCS lobe showed a line of wear along the pumping ramp of the lobe as seen in Figure 21. The hardness of the lobe was 58 to 61 Rockwell C. The worn camshaft sections were replaced.

Typically, wear across the cam lobe means that lobe or tappet roller hardness is incorrect. Wear along the cam lobe means that the tappet - cam lobe alignment is incorrect. Tappet alignment is to be performed according to Fairbanks Morse Engineering Instruction 2207FM3 to ensure that the tappet roller runs true to the cam lobe.

TAPPET ROLLERS

Visual inspection showed two tappet rollers with signs of distress. The #10 CS roller and the #11 OCS roller displayed wear across the roller as seen in Figure 22. The hardness of the rollers was 47 and 54 Rockwell C, respectively. The blueprint hardness is 59 Rockwell C minimum.

AIR START DISTRIBUTOR

The air start valve for the #7 cylinder in the air start distributor was flattened about .025" on the tappet end. A hardness check at the worn tip resulted in a Rockwell C hardness of 64. At a spot on the lapped surface 3/8" away from the tip, the hardness was 55 HRC and on the small cross section of the shaft the hardness was 49. The drawing calls for a hardness of 47-59 Rockwell C. Presumably, the valve must have been stuck in the guide by a burr or a tangled spring and rode the cam. The increased contact stresses due to the riding action work hardened the valve as it was flattening it. Due to the large ratio of cam to valve contact area, the cam had not worn.

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AIR START CHECK VALVES

Eight of the twelve air start check valve bodies were found to be worn upon disassembly. This wear occurred in the "balance piston" bore and was noted by grooving. The bores of the bodies were found to .004 to .013" out of round. Two valve stems were found with runout over the maximum of .002", but these stems corresponded to the valve bodies that were the least out of round. These two valve stems were replaced. The Balance Pistons were not found to have excessive runout or be out of round.

EXHAUST MANIFOLDS

The flanges of the exhaust manifolds were checked for flatness. Warped manifold - exhaust extension flanges were found on all the exhaust manifolds. The range of warpage was from .006-.060 inches. The manifold flanges were remachined to a flatness tolerance of .005". On any flange where the groove depth dropped below .095 inches the groove was recut.

All flanges of the exhaust extensions were found to be flat with .005". All manifold - belt flanges on the manifolds were found to be flat within .005".

IN GENERAL

Much has been said in this report of parts that have "conditions". Figures 23 inclusive are some of the photographs which were available to show acceptable engine parts.

Note how little the piston rings wore in 2000 hours in Figure 23, or the excellent appearance of the piston pin and bushings in Figure 24.

Figures 25 and 26 are photos of turbine blades as viewed through the turbocharger exhaust casings. Figures 27 and 28 are of the nozzle rings. Note how little carbon has collected in these areas.
Figures 29 through 33 are photos of upper and lower rod and main bearings. Notice how clean and free of deep scratches they are. Overheating cannot be seen on them.

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FLORIDA POWER CORPORATION

CRYSTAL RIVER UNIT 3

DOCKET NUMBER 50-302/LICENSE NUMBER DPR-72

ATTACHMENT H

UPDATE OF COMMITMENTS MADE IN TSCRN 210

ATTACHMENT H
UPDATE OF COMMITMENTS MADE IN TSCRN 210

In FPC's letter dated June 14, 1997 (3F0697-10), FPC provided the NRC with Technical Specification Change Request Notice (TSCRN) 210. That letter contained the following commitments. Those commitments completed have been marked accordingly.

1. Calculations

- By September 15, 1997, FPC will confirm to the NRC that the expected maximum steady state accident loads on the EDGs are bounded by the lower limit of the EDG refueling interval surveillance test.

Due: September 15, 1997
Status: Complete

- The remainder of the calculations will be completed prior to implementation of the license amendment resulting from TSCRN 210.

Due: Prior to implementation of the license amendment resulting from TSCRN 210

- By September 15, 1997, FPC will confirm that the calculations involving EFW block valve cycling and Control Complex Cooling are complete and their conclusions support TSCRN 210.

Due: September 15, 1997
Status: Complete

2. Modifications

Prior to NRC approval of the license amendment resulting from TSCRN 210, FPC will confirm that the modifications do not involve an unreviewed safety question, and that no changes were made in the proposed modifications which would alter the proposed Technical Specifications or Bases.

Due: September 15, 1997
Status: Complete

3. Procedures

Prior to NRC approval of the license amendment resulting from TSCRN 210, FPC will confirm that the necessary procedure changes do not involve an unreviewed safety question, and that no changes were made to the proposed procedures which would alter the proposed Technical Specifications or Bases.

Due: September 15, 1997
Status: Complete

4. FSAR

FPC will complete and submit FSAR Revision 24 prior to restart to address these changes associated with the SBLOCA solution sets.

Due: Prior to restart
Status: In progress

5. FWP-7

FPC also will have available Auxiliary Feedwater Pump 7 (FWP-7) which will be powered by a dedicated diesel generator installed during the current outage. The use, maintenance, and testing of FWP-7 will be controlled by plant procedures that will be approved prior to CR-3 restart to ensure that availability and reliability is appropriately addressed commensurate with its importance.

Due: Prior to restart
Status: FWP-7 Diesel Generator power supply installed and tested. Procedures are being developed.

6. Permanent Modifications

Prior to the beginning of Cycle 12, FPC will implement the permanent actions to address EDG capacity limitations. Presently, the two primary options under consideration are to (1) modify the existing EDGs, further increasing their capacity or (2) install a diesel-driven emergency feedwater pump. Included with either of these options is the removal of the automatic Emergency Feedwater Initiation and Control System trip of the motor driven EFW pump.

Due: Prior to the beginning of Cycle 12
Status: In planning

7. Interim Technical Specification Measures

Prior to the beginning of Cycle 12, an additional TSCRN will be submitted to reflect the resolution of the EDG capacity limitations and to remove the interim measures proposed by TSCRN 210.

Due: 12 months prior to the beginning of Cycle 12

Status: Being tracked

8. Engineering Evaluation of Decay Heat Removal in Mode 4

An engineering evaluation is being performed on issues concerning decay heat removal in Mode 4. This evaluation does not affect the SBLOCA analyses as presented in this submittal.

Due: Prior to restart
Status: In progress

9. EDG Testing

FPC will successfully complete testing in accordance with its written EDG test plan and obtain vendor certification to demonstrate that the Emergency Diesel Generators are qualified to perform at their new service ratings specified by TSCRN 210.

Due: Prior to entering Mode 4 from the forced outage initiated on September 2, 1996
Status: In progress

FLORIDA POWER CORPORATION

CRYSTAL RIVER UNIT 3

DOCKET NUMBER 50-302/LICENSE NUMBER DPR-72

ATTACHMENT I

LIST OF ACRONYMS AND ABBREVIATIONS USED

ATTACHMENT I

LIST OF ACRONYMS AND ABBREVIATIONS USED

AHF ------------------Air Handling Fan
ASV-------------------Auxiliary Steam Valve
BG&E ----------------Baltimore Gas & Electric
CFM ------------------Cubic Feet per Minute
CFR-------------------Code of Federal Regulations
CR-3 ------------------Crystal River Unit 3
CREVS---------------Control Room Emergency Ventilation System
CT---------------------Current Transformer
DC Electrical--------Direct Current Electrical
DHR ------------------Decay Heat Removal
DLHE ----------------Diesel Lube Oil Heat Exchanger
EDG ------------------Emergency Diesel Generator
EFIC------------------Emergency Feedwater Initiation and Control System
EFP -------------------Emergency Feedwater Pump
EFV-------------------Emergency Feedwater Valve
EFW ------------------Emergency Feedwater
EOP-------------------Emergency Operating Procedure
ES --------------------Engineered Safeguards
FMEA ----------------Failure Modes and Effects Analysis
FWP ------------------Auxiliary Feedwater Pump
HP---------------------Horsepower
HPI--------------------High Pressure Injection
IEEE------------------Institute of Electronic and Electrical Engineers
ISCM -----------------Inadequate Subcooling Margin
kW --------------------kilowatts
LOCA ----------------Loss of Coolant Accident
LOOP-----------------Loss of Offsite Power
LPI--------------------Low Pressure Injection
MAR------------------Modification Approval Record
MOV -----------------Motor Operated Valve
MSIV -----------------Main Steam Isolation Valve
MUV -----------------Makeup Valve
NPSH-----------------Net Positive Suction Head
OA --------------------Operator Action
OTSG-----------------Once Through Steam Generator
psig -------------------pounds per square inch gauge
PORV ----------------Pilot Operated Relief Valve
PPO-------------------Primary Plant Operator
PSV -------------------Pressurizer Safety Valve
RB---------------------Reactor Building
RBCU --- ----------Reactor Building Cooling Units

RBIC------------------Reactor Building Isolation and Cooling
RCP-------------------Reactor Coolant Pump
RCS-------------------Reactor Coolant System
RPM ------------------Revolutions per Minute
RWP------------------Nuclear Services Seawater Pump
SAG-------------------CR-3 Safety Assessment Group
SBLOCA-------------Small Break Loss of Coolant Accident
SCM ------------------Subcooling Margin
SPDS -----------------Safety Parameter Display System
SWP -------------------Nuclear Services Closed Cycle Cooling Pump
TSCRN---------------Technical Specification Change Request Notice
USQ ------------------Unreviewed Safety Question