ENCLOSURE 3

MPR Associates, Inc.
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"Emergency Diesel Generator
Fan & Radiator Performance Evaluation"

MPR ASSOCIATES INC. ENGINEERS

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Revision 1
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Emergency Diesel Generator Fan & Radiator Performance Evaluation

Prepared for

Oyster Creek Nuclear Station
Amergen Energy Co., LLC
Forked River, NJ 08731

MPR
ASSOCIATES INC.
ENGINEERS

Emergency Diesel Generator Fan & Radiator Performance Evaluation

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Executive Summary

To assist Amergen Energy evaluate the EDG #1 loose fan drive shaft bearing pillow block event on May 17, 2004 at Oyster Creek Nuclear Station, MPR Associates completed a review of event reports and the Joliet, Illinois MP36 diesel generator test plan and test results. MPR also prepared fan belt slippage calculations and diesel engine jacket water and lube oil temperature rise as a result of reduced air flow through the radiator.

Based on:

- the reported good condition of the fan belt and its subsequent reinstallation on EDG #1;
- the lack of extensive belt squealing noise as contrasted to the banging noise from the pillow block hitting the support pedestal during the event, and;
- most importantly, our analysis of the effects of drive shaft movement during the event;

we conclude that during the time the pillow block was loose and fan belt tension was reduced the EDG #1 fan speed was approximately 573 rpm as compared to the design rated speed of 637 rpm. The lower fan speed (90% of design fan speed) results in a cooling air flow rate through the engine's radiator of 112,400 SCFM as compared to a design air flow rate of 125,000 SCFM.

Further, our analysis concludes the fan belt would have had a belt life of nearly one month if it had been required to continue to operate with slippage at 573 rpm fan speed. Note that as the belt approached its end of life, fan speed would likely degrade further.

The resulting air flow of 112,400 SCFM (90% of design air flow) is sufficient to ensure the EDG #1 EMD Model 645 20-cylinder diesel engine <u>will not</u> overheat when operating at a steady-state load of 2600 kWe. With the ambient air temperature of 70°F, the resulting steady-state engine inlet lube oil temperature is approximately 206°F and the inlet jacket water temperature is approximately 187°F. The plots of engine heat up rate for fan speeds ranging from 100% to 55% of design rated speed are provided in Appendix B.

MPR Associates conclude that the EDG #1 operation would not have been limited by increasing engine jacket water and lube oil temperatures. EDG #1 would not have prematurely been shut down by a high jacket water temperature trip.

Contents

Tables

Figures

1 Introduction

1.1 EDG #1 Cooling Fan Event Background

During a normally scheduled monthly surveillance run of EDG #1 at Oyster Creek Nuclear Station early on May 17, 2004, the plant operating personnel reported hearing an unusual noise coming from the EDG room. The noise was not noticed when the surveillance test began more than an hour earlier. The operator reported a banging noise was coming from the radiator fan drive shaft and belt area. After operating a total of 1 hour and 23 minutes on May 17th, EDG #1 was shutdown (Ref. 1). Inspection of the fan drive assembly determined the pillow block assembly (Item 2 on Figure 1-1) that supports the lower drive shaft from the front of the diesel engine was loose.

EDG #1 at Oyster Creek Nuclear Station is known as an MP45A diesel generator set and consists of a 20-cylinder EMD Model 645 diesel engine driving a generator with a design rating of 2,600 kWe. As shown in Figure 1-1, the MP45A is cooled by a radiator with a belt driven fan to exhaust heat from the engine's jacket water system to the atmosphere. The pillow block is mounted to its support pedestal by two ¾"-10 hex head bolts, nuts and washers (Ref. 2). The lower bolt, washer and nut were found on the floor and the upper bolt was loose. The nut had backed off about 5 full turns or a distance of ½". As noted in Reference 1, the fan belt had been replaced during a scheduled maintenance outage and EDG #1 had been returned to service on April 30, 2004. From the time the fan belt was replaced to the start of the surveillance run on May 17, 2004, EDG #1 had operated at idle speed or rated speed for approximately 7 ½ hours (Ref. 1). The USNRC inspection report (Ref. 3) stated that the event was the result of failure to implement appropriate procedural requirements for maintenance in that "Technicians failed to follow written procedures to torque the cooling fan shaft bearing bolts following fan belt replacement as prescribed by Technical Specification 6.8.1."

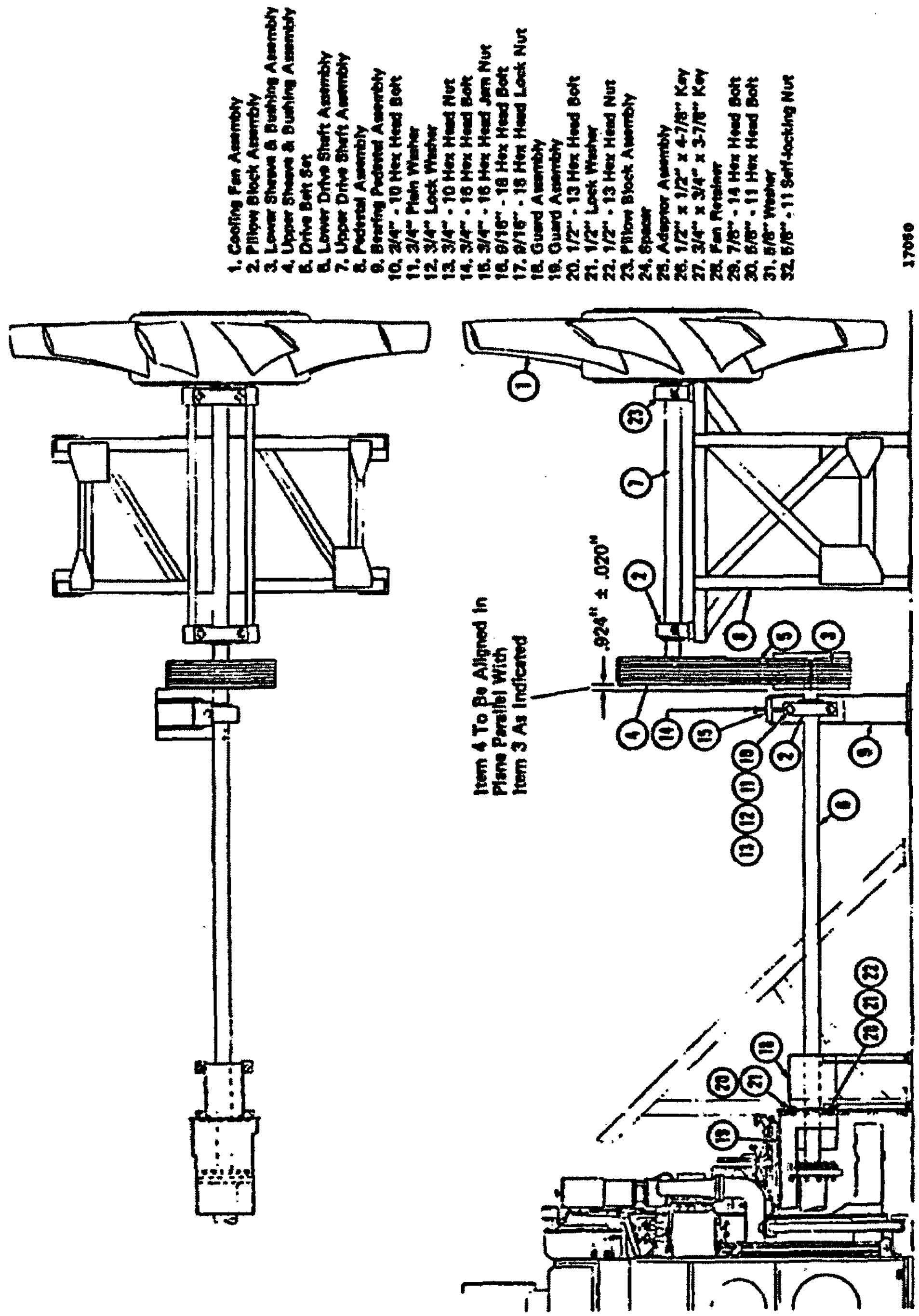

Figure 1-1 MP45A Cooling Fan and Drive Assembly

(From Reference 2)

1.2 Off-Site Diesel Engine Testing to Demonstrate EDG Operability

In an attempt to determine the effect of a loosened drive shaft support bearing pillow block, a test was conducted using a similar diesel generator set. A test plan (Ref. 4) to simulate the degraded shaft support condition of EDG #1 was developed by Engine Systems, Inc. (ESI) the nuclear power industry distributor for EMD diesel generators. The test was performed on a similar EMD diesel generator at a power plant in Joliet, Illinois. The diesel generator at Joliet is an MP36 unit which has a 16-cylinder EMD 645 diesel engine as the prime mover. It is radiator cooled with most of the fan shaft support components being similar, if not identical, to the components in the MP45A at Oyster Creek. The differences in fan components between the two units are summarized in Reference 5. The most significant differences are the fan belt sheave diameters, both upper and lower, in the MP36 unit are smaller than used in the MP45A. Thus while both engines having an operating speed of 900 rpm, the resulting fan speed (571 rpm) of the MP36 is slower than the MP45A (637 rpm). The fan belts are the same Goodyear part.

As described in Reference 6, the loose pillow block test completed on the MP36 diesel engine was partially successful. The test of the MP36 diesel generator was also recorded on video tape (Ref.7). A number of interesting observations can be developed from the report and the video tape. Specifically, even with degraded support condition for the lower fan shaft bearing pillow block, the following conditions were demonstrated by the test:

- More than six hours of operation were achieved before the engine shut down due to high jacket water inlet temperature (> 195– 200°F).
- The ¾" upper bolt and nut were observed to be erratically oscillating during the test, but they never backed completely out of the pillow block. The pillow block continuously pivoted on the end of the fan belt tension adjusting bolt (Item 14 on Figure 1-1) during the test.
- The fan belt lost a lot of tension during the pillow block movement and began slipping and skipping such that the fan speed degraded to about 56% of design speed (Ref. 6).
- The fan speed became very erratic in the later stages of the test, but the fan never stopped.

1.3 Analyses of Diesel Engine Operation at Reduced Fan Speed

As another method of considering the ability of the EDG #1 to operate for an extended period of time with a loosened drive shaft bearing pillow block, Amergen Energy tasked MPR Associates to calculate the extent of fan belt slippage and the resulting change in fan speed of the EDG. From the reduced airflow that results from reduced fan speed, MPR also calculated the rate at which the EMD 645 20-cylinder diesel engine jacket water and lube oil temperatures would increase. As discussed in Section 2.2, the fan belt was skipping and jumping so much on the MP36 at Joliet as to not be representative of what was likely the case at Oyster Creek. Thus, MPR calculated belt slippage, fan speed and engine heat up rate for EDG #1.

2 Technical Evaluation

2.1 Independent Technical Evaluation

As tasked, MPR Associates performed an independent evaluation of the reports related to and the information available from the EDG #1 loose fan drive shaft bearing pillow block event on May 17, 2004 at Oyster Creek Nuclear Station. Our review of the technical information was necessary to obtain much of the design information and operating data associated with the MP45A diesel generator as compared to the MP36 diesel generator used in the test at Joliet, Illinois.

Once we had a good understanding of the facts from those sources, MPR prepared two calculations:

- to determine an estimate of the fan speed resulting from the loose bearing pillow block, and
- to determine the steady-state jacket water and lube oil temperatures within the 20-cylinder EMD Model 645 diesel engine operating at rated load of 2,600 kWe with an ambient air temperature of 70°F.

Section 2.2 provides a summary of the major factors affecting the MP45A fan performance based on our review of the available information. Section 2.3 summarizes assumptions made in the fan speed calculation. Section 2.4 summarizes the results of MPR's calculations of fan speed resulting from the loose pillow block and the resulting EMD 645 diesel engine jacket water and lube oil temperatures when operating at a reduced fan speed. Finally, Appendix A contains the completed EDG #1 fan speed calculations and Appendix B contains the EDG #1 heat up rate calculations.

2.2 Factors Relevant to EDG #1 Operation

From the technical reports and data associated with the loose pillow block event, there are a number of significant factors or conditions related to the operation of EDG #1 that suggest it would have been capable of operating for an extended period of time without suffering any damage or unusual wear. The following is a brief summary of those factors:

1. Even after the Oyster Creek EDG operators reported unusual noises during the May 17th surveillance run, EDG #1 completed more than one hour of operation. The operators were able to collect standard log sheet data before shutting down the EDG. All recorded diesel engine operating parameters were "normal" at the time of shut down (Ref. 5).

2. After EDG #1 stopped, the reported belt deflection was about 3/8 inch when pushed by a person's fingers (Ref. 5) versus the design value of 7/16 inch when a force of 80 to 104 lbs is applied. Assuming a person can push on the belt with a 45 lb force, that 3/8 inch

deflection suggests the belt tension was approximately 800 lbs. This is slightly less than the calculated belt tension required to drive the fan at its rated speed of 637 rpm. This suggests the EDG #1 fan belt was not slipping much during the time the upper pillow block bolt was loose and the lower bolt was missing.

3. It was also reported that the lower drive shaft could not be moved laterally back against the support pedestal without extreme force (Ref 5). This also suggests the fan belt was still under significant tension.

4. When EDG #1 was being restored to service on May 17th, mechanics re-tensioned the fan belt, torqued all the pillow block mounting bolts, and took vibrations readings while the engine was operating. No abnormal vibration readings were recorded and EDG #1 was restored to service (Ref. 1).

5. The Goodyear fan belt was newly installed on EDG #1 during the planned maintenance outage in April 2004. According to Reference 1, the fan belt had supported EDG#1 for about 10 ½ hours of operation through May 17th. The belt was inspected and found to be in good condition (no indication of stretch or burning or overheating). The fan belt was re-installed on EDG #1 (Ref. 3). This is in contrast to the fan belt from the Joliet engine test which showed signs of glazing on the running surfaces (an indication of excessive slipping) and one edge showed signs of distress in some fibers. The belt actually had fewer hours of operating time during the tests than the belt from EDG #1.

6. There was a measured difference in the centerline alignment of the upper and lower drive sheaves on the MP36 test engine in Joliet, Illinois as compared to EDG #1 at Oyster Creek (Ref.5). These minor differences made during generator set field fabrication may explain some of the greater movement of the fan belt and lower drive sheave during the MP36 test.

7. On EDG #1 the radiator exhaust louvers regularly cycle from closed to open during normal operation to hold the jacket water coolant nominal temperature at 175°F. The changing louver position results in a cyclic torque load on the drive shaft and fan belt since the fan torque reduces to a very small value when the louvers are closed. However, when the engine jacket water temperature exceeds 175°F, in our calculations (Appendix A), we assume the louvers remain fully open and the fan shaft torque remains stable.

8. The lower drive sheave on the MP45A EDG is larger in diameter (15 inch versus 12.5 inch) and therefore heavier than the drive sheave on the MP36 diesel generator set in Joliet, Illinois. This greater weight would tend to increase the tension on the fan belt on EDG #1 and likely helped to further stabilize and maintain the belt tension when the pillow block was loose.

9. When the 16-cylinder engine at Joliet was starting to overheat toward the end of the 6-hour run/test due to the extensive fan belt slipping/skipping, the jacket water temperature was observed to increase at the rate of about 1°F per minute (Ref. 6). This is a very rapid rate of temperature rise after once achieving steady-state conditions that was never observed at Oyster Creek.

10. According to Reference 6, during initial test setup, the pillow block on the MP36 in Joliet, Illinois lost contact with the belt tensioning bolt (Item 14 on Figure 1-1) when the engine was shut down. The fan belt tensioning was again adjusted and during the formal test the pillow block was always bearing on the belt tensioning bolt. This appears to be the case with EDG #1, namely that the tensioning bolt always remained in contact with the top of the pillow block.

11. When the pillow block mounting bolts were both loose, but the lower one still providing some restraining force, there was no reduction of fan speed during the MP36 test at Joliet, Illinois (Ref. 6). It is reasonable to expect that the same condition existed in EDG #1 at Oyster Creek before the lower bolt dropped out on May 17th.

Based on the above factors, MPR concludes the drive shaft bearing pillow block movement on EDG #1 was less than occurred during the test of the MP36 diesel generator set.

2.3 Calculation Assumptions

MPR had to make a number of assumptions in the calculations we prepared to determine fan speed, air flow rate and engine heat up rate and final operating temperatures. Most of these assumptions had to be made because much of the required design data was not available from the following sources:

- General Motors Model 645 diesel engine technical manual for MP45A diesel generator sets
- Engine Systems Incorporated, the nuclear power industry equipment distributor
- The Young Radiator Company, the lube oil cooler and radiator manufacturer
- Koppers, presumed to be the fan manufacturer

Much of the design data needed to very accurately calculate fan and engine operating conditions was either unavailable or considered to be proprietary. Note that MPR did obtain useful technical information from the Goodyear Company, the fan belt manufacturer.

Some specific assumptions made in the fan speed calculations of conditions for EDG #1 include:

- At the fan's rotating speed of 637 rpm, the assumed air flow rate is 125,000 SCFM through the radiator (from Ref. 5).
- As noted above, as the engine heats up above its normal operating temperature, we assumed the louvers on the radiator air flow discharge remained fully open.
- To consider likely ambient conditions at Oyster Creek Nuclear Station during April and May, we assumed the ambient air temperature was 70°F.
- Based on the inspection of the EDG #1 fan belt after the event (Ref. 6), we assumed there was no measurable residual belt stretch.
- Belt tension loss due to centrifugal force as it passes the sheaves is negligible. (This assumption was subsequently confirmed by another calculation.)
- The coefficient of friction between the belt and the sheaves is constant.

- The impact of alignment differences of the upper and lower sheaves on fan belt tension is negligible.
- The fan belt was installed and tensioned in accordance with the guidance in Reference 2 except that the pillow block mounting bolts were not properly torqued. The deflection force assumed on the belt was in the middle of the specified/acceptable range.

The detailed list of assumptions made regarding the EMD Model 645 diesel engine heat up rate are listed on Pages 4 and 5 of the calculation, Appendix B.

2.4 SUMMARY OF CALCULATION RESULTS

2.4.1 Calculated Fan Speed

Using the assumptions noted in Section 2.3, fan belt and mechanical design handbooks, and technical information obtained from Goodyear, MPR calculated the fan speed for EDG #1 as a result of the loss of belt tension due to the loose pillow block. As shown in Appendix A, the calculated fan speed during the EDG #1 surveillance run on May 17, 2004 was 573 rpm. The design fan speed is 637 rpm, thus the calculated 573 rpm represents a 10% reduction. The axial fan performance at 90% of design rated speed is determined to be about 90% of its design value or approximately 112,400 SCFM. Finally, we calculated the fan belt operating under these conditions would have an estimated service life of nearly one month of continuous operation.

2.4.2 Calculated EMD Diesel Engine Heat Up Rate

As demonstrated during the Joliet test, a fan speed of 307 to 320 rpm is unacceptable in that the 16-cylinder EMD 645 diesel engine overheated after about six hours. However, at a calculated fan speed of 573 rpm versus the design speed of 637 rpm, the EDG #1 fan is still providing sufficient air flow (approximately 90% of design air flow) through the radiator that the 20-cylinder EMD 645 diesel engine operating at rated load <u>will not</u> overheat. Based on our calculations in Appendix B, assuming an ambient temperature of 70°F, the engine jacket water inlet temperature reaches a steady-state value of approximately 187°F and the engine lube oil inlet temperature reaches a steady-state value of approximately 206°F. Both of these temperatures are higher than measured for EDG #1 (Ref. 8), however they are lower than the EMD Model 645 diesel engine shutdown temperature limits (Ref. 9).

The maximum steady-state engine temperature conditions were also calculated at an ambient air temperature of 70°F to ensure the calculations in Appendices A and B accurately represented the actual conditions before the fan belt loss-of-tension event. With the fan running at the design rated speed of 637 rpm, the calculated maximum engine jacket water inlet temperature is 181°F and the maximum engine lube oil temperature is 196°F. On April 30, 2004, the recorded ambient temperature was 70°F, thus the recorded data from that date shows very good agreement, within 3°F, with the calculated results in Table 2-1 below. As shown in Table 2-1, these temperature values are very consistent with the calculated engine jacket water and lube oil inlet temperatures in Appendix B. There is only one area where we have made an adjustment in the results and that is regarding lube oil temperature rise within the engine. Based on information provided in Reference 10, we were provided an estimated ratio between the engine heat discharged through the radiator and the heat discharged through the lube oil cooler. This guideline information is for sizing radiators and coolers for a number of engines and is an estimate. As noted in Appendix B, our calculated engine lube oil temperature rise is 8.6°F more than the maximum actually measured during six different EDG #1 surveillance runs recorded in Reference 8. These recorded data cover EDG #1 surveillance runs with ambient air temperatures ranging from 32°F to 70°F. In all six cases with the engine operating at rated load, the recorded lube oil temperature rise within the engine is between 11°F and 15°F. Thus, we have included that lube oil temperature rise of 15°F in the results summarized below. This adjustment does not have a significant effect on the calculated engine heat up rate and the inlet jacket water and lube oil temperatures calculated in Appendix B.

Table 2-1 Summary of Actual and Calculated Diesel Engine Operating Temperatures

Engine Operating Conditions (Rated Load, 2,600 kWe)	Air Flow Conditions Across Radiator	Engine Jacket Water Inlet Temp, °F	Engine Jacket Water Outlet Temp, °F	Engine Lube Oil Inlet Temp, °F	Engine Lube Oil Outlet Temp, °F
Ambient air temperature, 70°F (on April 30, 2004; from Ref. 8)	Actual flow during the EDG #1 surveillance run **before** bolts became loose.	178	-- Not recorded	194	208
Ambient air temperature, 70°F	**Calculated** at **100%** design flow rate; 637 rpm (Appendix B)	181	190.5	196	211
Ambient air temperature, 70°	**Calculated** at **90%** design flow rate; 573 rpm (Appendix B)	187	196	204	219
Ambient air temperature, 70°F	**Calculated** at **55%** design flow rate; 350 rpm (Appendix B)	223	232.5	239	254

3 Conclusions

The Joliet MP36 degraded fan drive shaft bearing support test served the purpose of bounding the event and suggesting that even in the degraded condition the engine was able to operate for more than six hours without over heating.

Based on our review of event reports, the MP36 test plan and test results, and our independent fan belt slippage calculations, we conclude that the fan speed would have been reduced during the EDG #1 surveillance test on May 17, 2004. Based on:

- the reported good condition of the fan belt and its subsequent reinstallation on EDG #1;
- the lack of extensive belt squealing noise as contrasted to the banging noise from the pillow block hitting the support pedestal during the event; and;
- most importantly, our analysis of the effects of drive shaft movement during the event;

we conclude that during the time the pillow block was loose and fan belt tension was reduced the EDG #1 fan speed was approximately 573 rpm as compared to the design rated speed of 637 rpm. The lower fan speed (90% of design fan speed) results in a cooling air flow rate through the engine's radiator of 112,400 SCFM as compared to a design air flow rate of 125,000 SCFM.

Further, our analysis concludes the fan belt would have had a belt life of nearly one month if it had been required to continue to operate with slippage at 573 rpm fan speed. Note that as the belt approached its end of life, fan speed would likely degrade further.

The resulting air flow of 112,400 SCFM (90% of design air flow) is sufficient to ensure the EDG #1 EMD Model 645 20-cylinder diesel engine <u>will not</u> overheat when operating at steady-state load of 2600 kWe. With the ambient air temperature of 70°F, the resulting steady-state lube oil inlet temperature is approximately 206°F and the jacket water inlet temperature is approximately 187°F. The plots of EMD Model 645 diesel engine heat up rate and steady-state temperatures for fan speeds ranging from 100% to 55% of design rated speed are provided in Appendix B.

MPR Associates conclude that the EDG #1 operation would not have been limited by increasing engine jacket water and lube oil temperatures. EDG #1 would not have prematurely been shut down by a high jacket water temperature trip.

4 References

1. Prompt Investigation Report, Oyster Creek, subject: "Failure of The #1 EDG Cooling Fan," date of event May 17, 2004.

2. General Motors Electro-Motive Division Maintenance Instruction 1200, Rev A, dated February 1979, subject: "MP45 Cooling Fan and Related Drive Train Assembly."

3. USNRC Report EA-04-142 dated August 12, 2004, subject: "Oyster Creek Generating Station – NRC Inspection Report 050000219/2004003; Preliminary Greater Than Green Finding."

4. ESI Document Number 6012458-TP-1 Revision 0 July 6, 2004, Subject: "Test Plan of an EMD MP Radiator Fan Drive with Degraded Lower Pillow Block Bearing Mounting Bolts."

5. "Oyster Creek EDG Cooling Fan Drive Test, Technical Background and Basis," forwarded by e-mail dated August 18, 2004.

6. ESI Document Number 6012458-TR-1 Revision 1 August 25, 2004, Subject: "Test Report of an EMD MP Radiator Fan Drive with Degraded Lower Pillow Block Bearing Mounting Bolts."

7. Unified Engineering, Inc. Video Tape dated August 2, 2004, subject: "Condensed Video of all Bracket Testing at Joliet Station on 7/28/04 and 7/29/04."

8. E-mail from Amergen Energy (Dave Jones) to MPR (Art Killinger) on August 27, 2004, subject: Log Sheet Data of EDG #1 Surveillance Parameters.

9. General Motors Electro-Motive Division Application and Installation Data, GM Series 645 Diesel Power Units dated December 1983 (data for turbocharged 16- and 20-cylinder engines).

10. Memorandum of Telecon between MPR and Exelon, Subject: EMD Model 645 Diesel Engine Performance Details/Guidelines, dated August 25, 2004.

A EDG Radiator Fan Speed Calculation

MPR

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CALCULATION TITLE PAGE

Client: Exelon Amergen Energy	
Project: Diesel Generator	Task No. 0083-0401-0314-01
Title: Engine Driven Radiator Fan Speed Calculation	Calculation No. 0083-0314-CZ

Preparer / Date	Checker / Date	Reviewer & Approver / Date	Rev. No.
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RECORD OF REVISIONS

Calculation No.	Prepared By	Checked By	Page: 2
0083-0314-CZ			

Revision	Affected Pages	Description
0	All	Initial Issue

Note: *The revision number found on each individual page of the calculation carries the revision level of the calculation in effect at the time that page was last revised.*

Table of Contents

1.0 PURPOSE

The purpose of this calculation is to determine the rotational speed of the engine driven radiator fan on EDG No. 1 at Oyster Creek Nuclear Generating Station. Following maintenance work, the radiator fan drive belt tension degraded because the mounting bolts on the fan drive shaft pillow block assembly were not properly installed. The lower bolt fell out and the upper bolt loosened during a monthly surveillance test. The calculated fan speed will be used to determine the radiator fan air flow rate which will be used in an evaluation of the EDG system temperatures. The service life of the belt in the degraded tension condition is also estimated.

2.0 SUMMARY OF RESULTS

The calculated fan speed is 573 rpm. The air flow rate provided by the fan is 112,400 cfm. The estimated belt life in the degraded tension condition is 618 hours.

3.0 DISCUSSION

3.1 Assumptions

1. Belt does not stretch during the event. The heating and stretch of the Goodyear Flexten belt is considered minimal during the period of time that the engine needs to run (maximum stretch of the belt is 1.5% of its original length over the 99,000-hour design life of the belt, References 7 and 1).

2. Tension due to centrifugal force is negligible.

3. Coefficient of friction between belt and sheaves is constant.

4. The belt was installed and tensioned in accordance with the guidance in Reference 5 except that the pillow block bolts were not tightened. The deflection force applied was in the middle of the specified range.

5. The impact of misalignment of the upper and lower sheaves on belt pretension is negligible. The effect of vertical alignment of the sheaves and the lateral alignment of the lower fan shaft on the belt slippage is included in the center distance calculation.

3.2 Parameters of Radiator Fan Belt Drive for Oyster Creek EDG No. 1

The parameters of the belt drive for Oyster Creek EDG No. 1 are summarized in Table 1.

Table 1. Belt Drive Parameter for MP45A

Parameters		References
Lower Sheave Effective Diameter	d = 15.0"	Reference 8
Upper Sheave Effective Diameter	D = 21.2"	Reference 8
Fan Belt Assembly	Torque Team Plus 5VF1120	Reference 9
Effective Belt Length	L = 112"	Reference 1, 8
Driving Shaft Rated Speed	N_1 = 900 rpm	Reference 8
Fan Rated Speed	N_2 = 637 rpm	Reference 8
Number of Ribs	N = 8	Reference 1
Maximum Deflection Force by GM	F_L = 13 x 8 = 104 lbs	Reference 5
Belt Modulus Factor	K = 4.88	Reference 2
Belt Weight/belt or rib	W = 0.16 lb/ft	Reference 2

3.3 Torque and Tension Equation Derivation

The torque transmitted by the belt is:

$\tau = D/2\,(T_1 - T_2) = D/2 \cdot T_e$ (Reference 3)

where:

D = driven sheave effective diameter, in = 21.2 in
T_e= effective belt tension, lbs = $(T_1 - T_2)$
T_1= tight side belt tension, lbs
T_2= slack side belt tension, lbs

For the same belt drive in two different tension conditions, the torque transmitted by the drive is directly proportional to the effective tension. Therefore,

$$\frac{\tau_1}{\tau_2} = \frac{T_{e1}}{T_{e2}} \qquad \text{(Eq. 1)}$$

Installation tension: $T_i = (T_1 + T_2)/2$ (Reference 3) (Eq. 2) which is the same as the belt pretension: $T_i = T_S\, N$ (Eq. 3)

where:

T_s = installation strand tension, lbs (per belt or rib)

N = number of belts or ribs

For a V-belt drive, assuming that the tension created by centrifugal force of the belt element is negligible, the equation of belt tension and coefficient of friction is:

$$\frac{T_1}{T_2} = e^{\frac{f\phi}{\sin\beta}} = R$$ (Reference 3)

where:

f = coefficient of friction

φ = the angle of contact in radians

β = "wedge" angle of the V-belt

R is used to determine the tension ratio factor = $\frac{R}{R-1}$

$$\frac{R}{R-1} = \frac{\frac{T_1}{T_2}}{\frac{T_1}{T_2}-1} = \frac{T_1}{T_1 - T_2} = \frac{T_1}{T_e}$$

Therefore, $T_1 = T_e \frac{R}{R-1}$, $\quad T_2 = \frac{T_1}{R} = T_e \frac{1}{R-1}$

Substitute T_1 and T_2 into Eq. 2: $\quad T_i = \frac{T_e}{2}\frac{R+1}{R-1}$

Since f, φ and β are constants, R is a constant, $\frac{T_{i1}}{T_{i2}} = \frac{T_{e1}}{T_{e2}}$

Combine with Eq. 1 and Eq. 3, $\quad \frac{\tau_1}{\tau_2} = \frac{(T_S N)_1}{(T_S N)_2}$ (Eq. 4)

3.4 Required Pretension

A minimum installation strand tension needs to be applied on the belt to transmit the required power. The installation tension or "static strand tension, Ts" (per belt or rib) can be determined with the following equation:

$$T_S = \frac{63030 \cdot HP \cdot (2A_R - 1)}{N \cdot d \cdot N_1} + \frac{W \cdot d^2 N_1^{\,2}}{1.69 \times 10^6}$$ (Reference 2)

Where:

HP = Fan horsepower = 100 hp (Reference 8)

A_R = Tension ratio factor = 1.2852 at $\frac{D-d}{C} = 0.226$

by interpolation of factors in Table 41 of Reference 2,

$$\frac{1.30-1.28}{0.30-0.20} = \frac{x-1.28}{0.226-0.20} \qquad x = \frac{0.02}{0.1} \times 0.026 + 1.28 = 1.2852 = A_R$$

N = number of ribs = 8
d = smaller sheave diameter, in = 15.0 in
N_1 = smaller sheave speed, rpm = 900 rpm
W = belt weight per rib, lb/ft = 0.16 lb/ft

$$T_S = \frac{63030 \times 100\ (2 \times 1.2852 - 1)}{8 \times 15 \times 900} + \frac{0.16 \times 15^2 \times 900^2}{1.69 \times 10^6} = 108.9\ lb$$

The required tension in the belt is $T_S \cdot N$ = 108.9 x 8 = 871.2 lb

3.5 Pretension Created by the Tensioning Procedure Specified in GM Maintenance Instruction (M.I. 1200)

It is specified on page 10 of General Motors Maintenance Instruction (M.I. 1200) (Reference 5) that, after installation of the belts on the sheaves, the belts need to be adjusted to the following condition. Each belt should deflect 7/16 in when the applied force is 10 to 13 pounds for each belt in a set of 8 belts. When the one 8-rib belt installed, the same procedure was followed. The average of 10 lb and 13 lb, or 11.5 lb, deflection force is used to calculate the belt pretension. The total deflection force on the set of 8 belts is assumed to be 8 x 11.5 lb = 92 lb.

The deflection force for a V-belt is determined with the following equation:

$$F_L = \frac{T_S N}{16} + \frac{NKP}{L} \qquad \text{(Reference 2)} \qquad \text{(Eq. 5)}$$

where:

F_L = belt deflection force, lbs
K = belt modulus factor
P = span length, in
L = belt length, in

$$P = C\left[1 - 0.125\left(\frac{D-d}{C}\right)^2\right]$$ (Reference 2)

$$C = \frac{k + \sqrt{k^2 - 32\ (D-d)^2}}{16}$$ (Reference 4, p8-55) (Eq. 6)

$k = 4\ L_p - 6.28\ (D + d)$ (Reference 4, p8-55) (Eq. 7)

where:

C = center distance between the two sheaves, in

L_p = belt pitch length, i.e. the effective length, in = 112 in

k = 4 x 112 – 6.28 (21.2 + 15.0) = 220.66 in

$$C = \frac{220.66 + \sqrt{220.66^2 - 32\ (21.2 - 15.0)^2}}{16} = 27.41\ \text{in}$$

$$P = 27.41\left[1 - 0.125\left(\frac{21.2 - 15.0}{27.41}\right)^2\right] = 27.23\ \text{in}$$

From Eq. 5, $$T_S N = 16\left[F_L - \frac{NKP}{L}\right]$$ (Eq. 8)

Assume that the impact of misalignment of the upper and lower sheaves on the pretension is minimal. The assumed installation tension is:

$$T_S N = 16\left[92 - \frac{8 \times 4.88 \times 27.23}{112}\right] = 1320\ \text{lbs}$$

Thus, the belt pretension of 1320 lbs exceeds the tension of 871.2 lbs, calculated in Section 3.4, required to transmit the engine drive horsepower to the fan shaft.

3.6 Tension Change from Pillow Block Movement

<u>Initial Belt Length Calculation:</u>

By Hooke's Law, $F = k \cdot \Delta L / L_i$

Where:

k = the belt modulus = 160,000 lb per 100% elongation per belt or rib (Reference 10)
ΔL = the belt length change

Under the belt pretension of F = 1320 lbs,

$$\Delta L = F \cdot L_i/k = 1320 \text{ x } 112 / (160{,}000 \text{ x } 8) = 0.1155 \text{ in}$$

The initial belt length $L_i = L + \Delta L = 112.1155$ in

<u>Minimum Center Distance Calculation:</u>

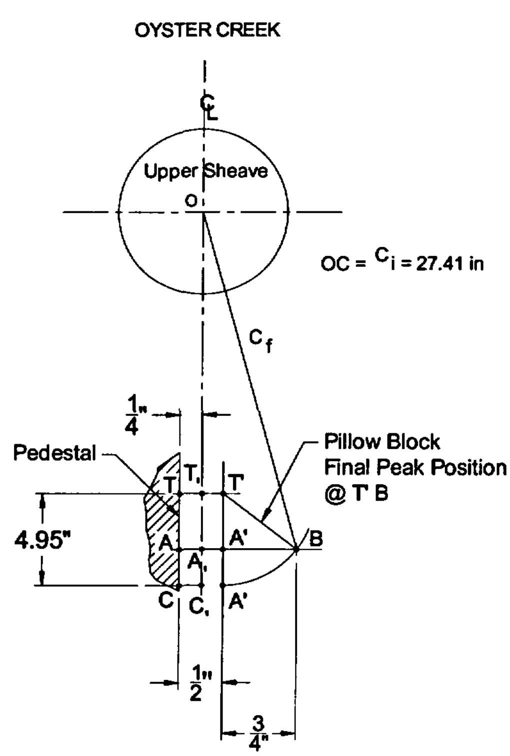

Figure 1. Illustration of the Belt Drive Misalignment and the Pillow Block Vibration Peak Position

As shown in Figure 1, TC is the pillow block original installation position. T'C' is the pillow block position after the lower bearing bolt was lost and the upper bolt backed out ½ inch as found on EDG 1 at Oyster Creek. T'B is the pillow block vibration peak position where the center distance between the upper and lower sheaves is the smallest.

After the belt is pretensioned, the center distance is calculated with Eq. 6 and Eq. 7:

$$k = 4 \times 112.1155 - 6.28\,(21.2 + 15.0) = 221.126 \text{ in}$$

$$C_i = \frac{221.126 + \sqrt{221.126^2 - 32\ (21.2 - 15.0)^2}}{16} = 27.466 \text{ in}$$

By scaling the distance from the pillow block top edge to the bearing center line on page 3 of M.I. 1200 (Reference 5), $TC = 4.95$ in.

From Reference 8, $A_1B = AA' - AA_1 + A'B = \frac{1}{2} - \frac{1}{4} + \frac{3}{4} = 1.00\ in$

$$TA = T'A' = \sqrt{T'B^2 - A'B^2} = \sqrt{TC^2 - A'B^2} = \sqrt{4.95^2 - 0.75^2} = 4.89\ in$$

$$OT_1 \approx OC - TC = C_i - TC = 27.466 - 4.95 = 22.516\ in$$

$$OA_1 = OT_1 + T_1A_1 = 22.516 + 4.89 = 27.406\ in$$

$$C_f = OB = \sqrt{OA_1^2 + A_1B^2} = \sqrt{27.406^2 + 1.00^2} = 27.424\ in$$

<u>Belt Length Relaxation</u>

The belt length is calculated with:

$$L = 2 \cdot C \cdot \cos\theta + \pi\,(D + d)/2 + \pi \cdot \theta\,(D - d)/180 \qquad \text{(Reference 6)}$$

Where: $\theta = \sin^{-1}(\frac{D-d}{2C})$

The initial belt length: $L_i = 112.1155$ in

$$\theta = \sin^{-1}(\frac{D-d}{2C}) = \sin^{-1}(\frac{21.2 - 15.0}{2 \times 27.424}) = 6.4906°$$

The length after the lower sheave centerline moved:

L_f = 2 x 27.424 x cos 6.4906° + π (21.2 + 15.0)/2 + π x 6.4906° (21.2 – 15.0)/180 = 111.0616 in

The belt has relaxed: ($L_i - L_f$) = 0.0539 in

Tension Loss Because of Belt Length Relaxation

$\Delta T_s N$ = 8 x 160,000 x 0.0539/112.1155 = 615.0 lbs

3.7 *Calculation of Fan Speed on MP45A*

From Eq. 5.3 of Reference 11, the fan horsepower varies as the cube of the shaft speed:

$$\frac{P_1}{P_2} = (\frac{N_1}{N_2})^3 \qquad \text{(Eq. 9)}$$

where: P = fan horsepower = $\tau \cdot \omega = \tau \cdot 2\pi N$

Substitute into Eq. 9 and reorganize: $\frac{\tau_1}{\tau_2} = \frac{N_1^2}{N_2^2}$

Combine with Eq.4: $\frac{(T_S N)_1}{(T_S N)_2} = \frac{N_1^2}{N_2^2}$

The belt tension in the fan belt drive: $(T_s N)_1$ = 1320 – 615 = 705 lb

The required belt tension: $(T_s N)_2$ = 871.2 lb

$$\frac{N_1^2}{N_2^2} = \frac{705.0}{871.2} = 80.92\% \qquad \frac{N_1}{N_2} = \sqrt{80.92\%} = 89.95\%$$

The fan shaft speed with belt slippage on MP45A is

$$N_1 = 89.95\ \% \times 637\ rpm = 573\ rpm$$

The required tension due to the centrifugal force of the belt element is by Reference 2

$\frac{Wd^2(rpm)^2}{1.69\times10^6} = \frac{0.16\times21.2^2(573)^2}{1.69\times10^6} = 13.97$ lb, which is small enough to be omitted. This confirms the assumption made on page 4.

3.8 Calculation of Air Flow at Lowered Fan Speed

From Eq. 5.1 of Reference 11, $\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$

From Reference 8, the air flow at rated speed (637 rpm) is 125,000 cfm. The air flow at lowered fan speed of 573 rpm, the air flow is

$$Q_1 = \frac{N_1}{N_2} Q_2 = \frac{573}{637} \times 125000 = 112{,}400 \text{ cfm}$$

3.9 Belt Life Estimation

As calculated in Section 3.7, the belt tension decreased to 80.92% of required tension. From tension vs belt life curve (Reference 6) from Goodyear, at 20% decrease in tension, the belt life decrease percentage is:

80% + 31/32 x 20% = 99.375%

The belt life is estimated as (1 – 99.375%) (Performance Index) = 618.75 hr

Where the performance index = belt life = 99,000 hr (Reference 1)

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B EDG Heat up Rate Calculation

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CALCULATION TITLE PAGE

Client: Amergen Energy	Page 1 of 20, Plus Append. A-D (44 total)
Project: Oyster Creek EDG Radiator	Task No. 0083-0314
Title: EDG Heatup With Reduced Radiator Heat Removal Capability	Calculation No. 0083-0314-RCS01

Preparer / Date	Checker / Date	Reviewer & Approver / Date	Rev. No.
R. C. Sanders	S. Kiffer	A. Killinger	0
R. C. Sanders 09-13-04	S. Kiffer 09-13-04	A. Killinger 9/13/04	
R. C. Sanders	S. Kiffer	A. Killinger	1

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RECORD OF REVISIONS

Calculation No.	Prepared By	Checked By	Page: 2
0083-0314-RCS01			

Revision	Affected Pages	Description
0	All	Initial Issue
1	All	1. Increased lube oil flow rate to be consistent with pages 2-4 and 6-4 of Reference 3. 2. Deleted heatup curves for 90 °F ambient air temperature. 3. Added several air flow rates to heatup curves for 70 °F ambient air temperature.

Note: *The revision number found on each individual page of the calculation carries the revision level of the calculation in effect at the time that page was last revised.*

Table of Contents

PURPOSE

The purpose of this calculation is to determine the heatup rate for the Oyster Creek EDG with reduced radiator heat removal capability.

BACKGROUND

Oyster Creek recently experienced an incident in which the drive pulley for the EDG radiator fan became loose, resulting in a loss of tension in the fan drive belts and reduced air flow through the radiator. This calculation determines the effect of reduced radiator air flow on the heatup rate of the EDG.

RESULTS

The results of this calculation are shown in Figures 2 and 3.

Figure 2 shows the temperature of the coolant entering the engine assuming the EDG is operating at its rated power of 2600 KWe with an ambient air temperature of 70 °F. Coolant inlet temperature is shown for air flow rates ranging from design flow (125,000 scfm) to 55% of design flow (68,750 scfm).

Figure 3 shows the temperature of the lube oil entering the engine assuming the EDG is operating at its rated power of 2600 KWe, with an ambient air temperature of 70 °F. Lube oil inlet temperature is shown for air flow rates ranging from design flow (125,000 scfm) to 55% of design flow (68,750 scfm).

Also shown in Figures 2 and 3 are temperatures measured on Oyster Creek EDG 1 on 4/30/04 (Reference 10), when the ambient air temperature was about 70 °F and the EDG was operating at about 2700 KWe. These data show that the analytical model developed in this calculation agrees well with measured operating data.

ASSUMPTIONS

1. Radiant and convective heat losses from the surfaces of the EDG and associated components are neglected. All of the heat removed from the EDG is through the radiator. This is conservative since radiant and convective heat losses will reduce the heatup rate.

2. The heat added by the cooling water pump and lube oil pump is negligible.

3. The physical properties of the coolant are constant throughout the cooling water system. That is, effects of temperature on the coolant physical properties are neglected.

4. The physical properties of the lube oil are constant throughout the lube oil system. That is, effects of temperature on the lube oil physical properties are neglected.

5. The ratio of the temperature rise of the coolant in the lube oil cooler to the temperature rise of the coolant in the engine is constant and equal to the ratio of the heat added in the lube oil cooler to the heat added in the engine at steady state conditions.

6. The heat absorbing capacity [(mass)X(specific heat)] of the air in the radiator is negligible.

7. The efficiency of the fins in the radiator remains constant at the calculated design value of 0.40

8. The efficiency of the fins in the lube oil cooler remains constant at the calculated design value of 0.93

9. The initial temperatures of the coolant and lube oil are equal to the ambient air temperature.

10. The ratio of the heat removal capability [(heat transfer coefficient)X(heat transfer area)] of the radiator for the test EDG (Model MP-36) to the heat removal capability of the radiator for the Oyster Creek EDG (Model MP-45) is equal to the ratio of the physical sizes of the engines (16 cylinders compared to 20 cylinders). This assumption used to benchmark the analytical heatup model.

11. The ratio of the heat absorbing capacity [(mass)X(specific heat)] of the cooling system for the test EDG (Model MP-36) to the heat absorbing capacity of the cooling system for the Oyster Creek EDG (Model MP-45) is equal to the ratio of the physical sizes of the engines (16 cylinders compared to 20 cylinders). This assumption used to benchmark the analytical heatup model.

ANALYSIS

A simplified schematic of the Oyster Creek EDG radiator cooling system is shown in Figure 1. The heatup rates for the temperature of the cooling water entering the engine (T_{C1}) and the temperature of the lube oil entering the engine (T_{L1}) are calculated in the following sections.

Cooling Water Heatup Rate

Define,

$$T_{Cav} = \frac{1}{2}(T_{C2} + T_{C3}) \qquad \text{Equation 1}$$

$$T_{Aav} = \frac{1}{2}(T_{Ai} + T_{Ao}) \qquad \text{Equation 2}$$

Where,

T_{Cav} = the average coolant temperature in the cooling system, °F

T_{C2} = the coolant temperature exiting the engine (entering the radiator), °F

T_{C3} = the coolant temperature entering the lube oil cooler (exiting the radiator), °F

T_{Aav} = the average air temperature in the radiator, °F

T_{Ai} = the air temperature entering the radiator, °F

T_{Ao} = the air temperature exiting the radiator, °F

A transient heat balance on the air in the radiator can be approximated with the following equation,

$$(MC_p)_A\left(\frac{dT_{Aav}}{dt}\right) = (UA)_R(T_{Cav} - T_{Aav}) - C_A(T_{Ao} - T_{Ai}) \qquad \text{Equation 3}$$

Where,

$(MC_p)_A$ = heat absorbing capacity [(mass)X(specific heat)] of the air in the radiator, Btu/°F

$(UA)_R$ = effective product of heat transfer coefficient times heat transfer area for the radiator, Btu/hr °F

C_A = heat transport capacity [(mass flow rate)X(specific heat)] of air flowing through the radiator, Btu/hr °F

Assuming that $(MC_p)_A$ is negligible, Equation 3 becomes

$0 = (UA)_R(T_{Cav} - T_{Aav}) - C_A(T_{Ao} - T_{Ai})$, or

$$T_{Ao} = T_{Ai} + [(UA)_R / C_A](T_{Cav} - T_{Aav}) \qquad \text{Equation 4}$$

From Equation 2,

$$T_{Ao} = 2T_{Aav} - T_{Ai} \qquad \text{Equation 5}$$

Setting Equation 4 equal to Equation 5 and solving for T_{Aav} gives,

$$T_{Aav} = \frac{T_{Ai} + [(UA)_R / 2C_A]T_{Cav}}{1 + [(UA)_R / 2C_A]} \qquad \text{Equation 6}$$

Neglecting radiant and convective heat losses from the coolant system, a transient heat balance on the coolant in the engine and lube oil cooler can be approximated with the following equation,

$$(MC_p)_{EL}\left(\frac{dT_{Cav}}{dt}\right) = Q_T - C_C(T_{C2} - T_{C3}) \qquad \text{Equation 7}$$

Where,

$(MC_p)_{EL}$ = effective heat absorbing capacity [(mass)X(specific heat)] of the coolant in the engine and lube oil cooler (including associated metal), Btu/°F

Q_T = total heat added to the coolant (engine plus lube oil cooler), Btu/hr

C_C = heat transport capacity [(mass flow rate)X(specific heat)] of coolant flowing through the coolant system, Btu/hr °F

A transient heat balance on the coolant in the radiator can be approximated with the following equation,

$$\left(MC_p\right)_R\left(\frac{dT_{Cav}}{dt}\right) = C_C\left(T_{C2} - T_{C3}\right) - \left(UA\right)_R\left(T_{Cav} - T_{Aav}\right)$$ Equation 8

Where,

$(MC_p)_R$ = effective heat absorbing capacity [(mass)X(specific heat)] of the coolant in the radiator (including associated metal), Btu/°F

Adding Equations 7 and 8 gives,

$$\left(MC_p\right)_{CT}\left(\frac{dT_{Cav}}{dt}\right) = Q_T - \left(UA\right)_R\left(T_{Cav} - T_{Aav}\right)$$ Equation 9

Where,

$$\left(MC_p\right)_{CT} = \left(MC_p\right)_{EL} + \left(MC_p\right)_R$$ Equation 10

Substituting Equation 6 into Equation 9 and rearranging terms gives,

$$\left(MC_p\right)_{CT}\left(\frac{dT_{Cav}}{dt}\right) + \left[\frac{(UA)_R}{1+(UA)_R/2C_A}\right]T_{Cav} = Q_T + \left[\frac{(UA)_R}{1+(UA)_R/2C_A}\right]T_{Ai}$$ Equation 11

The solution to Equation 11 is

$$T_{Cav} = T_{Ai} + \left[\frac{1+(UA)_R/2C_A}{(UA)_R}\right]Q_T + K_1 e^{-\lambda_c t}$$ Equation 12

Where,

K_1 = constant of integration

$$\lambda_c = \frac{(UA)_R}{\left(MC_p\right)_{CT}\left[1+(UA)_R/2C_A\right]}$$ Equation 13

From Equation 7,

$$T_{C3} = T_{C2} + \frac{(MC_p)_{EL}}{C_C}\left(\frac{dT_{Cav}}{dt}\right) - \frac{Q_T}{C_C}$$ Equation 14

Substituting Equation 12 into Equation 14 and rearranging terms gives,

$$T_{C3} = T_{C2} - \frac{Q_T}{C_C} - \frac{(\lambda_c)(MC_p)_{EL}}{C_C}\left(K_1 e^{-\lambda_c t}\right)$$ Equation 15

From Equation 1,

$$T_{C3} = 2T_{Cav} - T_{C2}$$ Equation 16

Substituting Equation 12 into Equation 16 gives,

$$T_{C3} = 2T_{Ai} + 2\left[\frac{1 + (UA)_R/2C_A}{(UA)_R}\right]Q_T + 2K_1 e^{-\lambda_c t} - T_{C2}$$ Equation 17

Setting Equation 15 equal to Equation 17 and solving for T_{C2} gives,

$$T_{C2} = T_{Ai} + \left[\frac{1}{2C_A} + \frac{1}{(UA)_R} + \frac{1}{2C_C}\right]Q_T + \left[1 + \frac{(\lambda_C)(MC_p)_{EL}}{2C_C}\right]K_1 e^{-\lambda_c t}$$ Equation 18

Substituting Equation 18 into Equation 15 and rearranging terms gives,

$$T_{C3} = T_{Ai} + \left[\frac{1}{2C_A} + \frac{1}{(UA)_R} - \frac{1}{2C_C}\right]Q_T + \left[1 - \frac{(\lambda_C)(MC_p)_{EL}}{2C_C}\right]K_1 e^{-\lambda_c t}$$ Equation 19

Assume the ratio of coolant temperature rise in the lube oil cooler ($T_{C1} - T_{C3}$) to coolant temperature rise in the engine ($T_{C2} - T_{C1}$) remains constant and is equal to the ratio of the heat added in the lube oil cooler (Q_L) to the heat added in the engine (Q_E) at steady state conditions. That is,

$$\frac{T_{C1} - T_{C3}}{T_{C2} - T_{C1}} = \frac{Q_L}{Q_E}, \text{ or}$$

$$T_{C1} = \frac{(Q_L)T_{C2} + (Q_E)T_{C3}}{Q_T}$$ Equation 20

Where,

$$Q_T = Q_E + Q_L \qquad \text{Equation 21}$$

Substituting Equations 18 and 19 into Equation 20 and rearranging terms gives,

$$T_{C1} = T_{Ai} + \left[\frac{1}{2C_A} + \frac{1}{(UA)_R} - \frac{(Q_E - Q_L)}{2C_C Q_T}\right] Q_T + \left[1 - \frac{(Q_E - Q_L)(\lambda_C)(MC_p)_{EL}}{2C_C Q_T}\right] K_1 e^{-\lambda_c t} \qquad \text{Equation 22}$$

As t goes to infinity, T_{C1} goes to T_{C1f}, where

T_{C1f} = the final steady state value of T_{C1}, °F

$$T_{C1f} = T_{Ai} + \left[\frac{1}{2C_A} + \frac{1}{(UA)_R} - \frac{(Q_E - Q_L)}{2C_C Q_T}\right] Q_T \qquad \text{Equation 23}$$

Therefore,

$$T_{C1} = T_{C1f} + \left[1 - \frac{(Q_E - Q_L)(\lambda_C)(MC_p)_{EL}}{2C_C Q_T}\right] K_1 e^{-\lambda_c t} \qquad \text{Equation 24}$$

At t equals zero, T_{C1} equals T_{C1i}, where

T_{C1i} = the initial value of T_{C1}, °F

Therefore,

$$T_{C1f} + \left[1 - \frac{(Q_E - Q_L)(\lambda_C)(MC_p)_{EL}}{2C_C Q_T}\right] K_1 = T_{C1i}\text{, or}$$

$$K_1 = \frac{-2C_C Q_T (T_{C1f} - T_{C1i})}{2C_C Q_T - (Q_E - Q_L)(\lambda_C)(MC_p)_{EL}} \qquad \text{Equation 25}$$

Therefore,

$$T_{C1} = T_{C1f} - \left(T_{C1f} - T_{C1i}\right)e^{-\lambda_c t}$$ Equation 26

From Appendices A through D,

$$Q_E = 5.123X10^6 \ Btu/hr$$

$$Q_L = 1.986X10^6 \ Btu/hr$$

$$Q_T = 7.109X10^6 \ Btu/hr$$

$$C_C = 5.394X10^5 \ Btu/hr\,^\circ F$$

$$C_L = 8.402X10^4 \ Btu/hr\,^\circ F$$

$C_A = 1.073G_A \ Btu/hr\,^\circ F$, G_A in scfm

$(UA)_R = \dfrac{1.838X10^5}{\left[1+\dfrac{1426.22}{G_A^{0.60}}\right]} \ Btu/hr\ ^\circ F$, G_A in scfm

$(UA)_C = 6.729X10^4$ Btu/hr °F

$$\left(MC_p\right)_{CT} = 1.182X10^4 \ Btu/^\circ F$$

Using these values along with Equations 13, 23 and 26, T_{C1} has been calculated as a function of time for several air flow rates with an ambient air temperature of 70 °F. The results are shown in Figure 2.

Lube Oil Heatup Rate.

Define,

$$T_{CLav} = \frac{1}{2}\left(T_{C1} + T_{C3}\right)$$ Equation 27

$$T_{Lav} = \frac{1}{2}\left(T_{L1} + T_{L2}\right)$$ Equation 28

Where,

T_{CLav} = the average coolant temperature in the lube oil cooler, °F

T_{C1} = the coolant temperature exiting the lube oil cooler (entering the engine), °F

T_{C3} = the coolant temperature entering the lube oil cooler (exiting the radiator), °F

T_{Lav} = the average lube oil temperature, °F

T_{L1} = the lube oil temperature entering the engine (exiting the lube oil cooler), °F

T_{L2} = the lube oil temperature exiting the engine (entering the lube oil cooler), °F

Substituting Equations 19 and 22 into Equation 27, rearranging terms and using Equations 23 and 25 gives,

$$T_{CLav} = T_{C1f} - \left(\frac{Q_L}{2C_C}\right) - B\left(T_{C1f} - T_{C1i}\right)e^{-\lambda_c t} \qquad \text{Equation 29}$$

Where,

$$B = \frac{2C_C Q_T - Q_E \lambda_c \left(MC_p\right)_{EL}}{2C_C Q_T - \left(Q_E - Q_L\right)\lambda_c \left(MC_p\right)_{EL}} \qquad \text{Equation 30}$$

A transient heat balance on the lube oil in the engine can be approximated with the following equation,

$$\left(MC_p\right)_{LOE}\left(\frac{dT_{Lav}}{dt}\right) = Q_L - C_L\left(T_{L2} - T_{L1}\right) \qquad \text{Equation 31}$$

Where,

$(MC_p)_{LOE}$ = effective heat absorbing capacity [(mass)X(specific heat)] of the lube oil in the engine (including associated metal), Btu/°F

C_L = heat transport capacity [(mass flow rate)X(specific heat)] of lube oil flowing through the lube oil system, Btu/hr °F

A transient heat balance on the lube oil in the lube oil cooler can be approximated with the following equation,

$$(MC_p)_{LOC}\left(\frac{dT_{Lav}}{dt}\right) = C_L(T_{L2} - T_{L1}) - (UA)_C(T_{Lav} - T_{CLav}) \qquad \text{Equation 32}$$

Where,

$(MC_p)_{LOC}$ = effective heat absorbing capacity [(mass)X(specific heat)] of the lube oil in the lube oil cooler (including associated metal), Btu/°F

$(UA)_C$ = effective product of heat transfer coefficient times heat transfer area for the lube oil cooler, Btu/hr °F

Adding Equations 31 and 32 gives,

$$(MC_p)_{LOT}\left(\frac{dT_{Lav}}{dt}\right) = Q_L - (UA)_C(T_{Lav} - T_{CLav}) \qquad \text{Equation 33}$$

Where,

$$(MC_p)_{LOT} = (MC_p)_{LOE} + (MC_p)_{LOC} \qquad \text{Equation 34}$$

Substituting Equation 29 into Equation 33 and rearranging terms gives,

$$(MC_p)_{LOT}\left(\frac{dT_{Lav}}{dt}\right) + (UA)_C T_{Lav} = (UA)_C T_{C1f} + (UA)_C\left[\frac{1}{(UA)_C} - \frac{1}{2C_C}\right]Q_L - B(UA)_C(T_{C1f} - T_{C1i})e^{-\lambda_c t} \qquad \text{Equation 35}$$

The solution to Equation 35 is

$$T_{Lav} = T_{C1f} + \left[\frac{1}{(UA)_C} - \frac{1}{2C_C}\right]Q_L - B\left(\frac{\lambda_L}{\lambda_L - \lambda_C}\right)(T_{C1f} - T_{C1i})e^{-\lambda_c t} + K_2 e^{-\lambda_L t} \qquad \text{Equation 36}$$

Where,
K_2 = constant of integration

$$\lambda_L = \frac{(UA)_C}{(MC_p)_{LOT}}$$ Equation 37

From Equation 31,

$$T_{L2} = T_{L1} + \frac{Q_L}{C_L} - \frac{(MC_p)_{LOE}}{C_L}\left(\frac{dT_{Lav}}{dt}\right)$$ Equation 38

Substituting Equation 36 into Equation 38 and rearranging terms gives,

$$T_{L2} = T_{L1} + \frac{Q_L}{C_L} - B\left[\frac{(MC_p)_{LOE}}{C_L}\right]\left(\frac{\lambda_L \lambda_C}{\lambda_L - \lambda_C}\right)(T_{C1f} - T_{C1i})e^{-\lambda_c t} + \frac{\lambda_L (MC_p)_{LOE}}{C_L}\left(K_2 e^{-\lambda_L t}\right)$$ Equation 39

From Equation 28,

$$T_{L2} = 2T_{Lav} - T_{L1}$$ Equation 40

Substituting Equation 36 into Equation 40 gives,

$$T_{L2} = 2T_{C1f} + 2\left[\frac{1}{(UA)_C} - \frac{1}{2C_C}\right]Q_L - 2B\left(\frac{\lambda_L}{\lambda_L - \lambda_C}\right)(T_{C1f} - T_{C1i})e^{-\lambda_c t} + 2K_2 e^{-\lambda_L t} - T_{L1}$$ Equation 41

Setting Equation 39 equal to Equation 41 and solving for T_{L1} gives,

$$T_{L1} = T_{C1f} + \left[\frac{1}{(UA)_C} - \frac{1}{2C_C} - \frac{1}{2C_L}\right]Q_L - B\left(\frac{\lambda_L}{\lambda_L - \lambda_C}\right)\left[1 - \frac{\lambda_C (MC_p)_{LOE}}{2C_L}\right](T_{C1f} - T_{C1i})e^{-\lambda_c t} + K_2\left[1 - \frac{\lambda_L (MC_p)_{LOE}}{2C_L}\right]e^{-\lambda_L t}$$ Equation 42

As t goes to infinity, T_{L1} goes to T_{L1f}, where

T_{L1f} = the final steady state value of T_{L1}, °F

$$T_{L1f} = T_{C1f} + \left[\frac{1}{(UA)_C} - \frac{1}{2C_C} - \frac{1}{2C_L}\right]Q_L \qquad \text{Equation 43}$$

Therefore,

$$T_{L1} = T_{L1f} - B\left(\frac{\lambda_L}{\lambda_L - \lambda_C}\right)\left[1 - \frac{\lambda_C (MC_p)_{LOE}}{2C_L}\right](T_{C1f} - T_{C1i})e^{-\lambda_c t} + K_2\left[1 - \frac{\lambda_L (MC_p)_{LOE}}{2C_L}\right]e^{-\lambda_L t} \qquad \text{Equation 44}$$

At t equals zero, T_{L1} equals T_{L1i}, where

T_{L1i} = the initial value of T_{L1}, °F

Therefore,

$$T_{L1f} - B\left(\frac{\lambda_L}{\lambda_L - \lambda_C}\right)\left[1 - \frac{\lambda_C (MC_p)_{LOE}}{2C_L}\right](T_{C1f} - T_{C1i}) + K_2\left[1 - \frac{\lambda_L (MC_p)_{LOE}}{2C_L}\right] = T_{L1i}, \text{ or}$$

$$K_2 = \frac{B\left(\frac{\lambda_L}{\lambda_L - \lambda_C}\right)\left[2C_L - \lambda_C (MC_p)_{LOE}\right](T_{C1f} - T_{C1i}) - 2C_L(T_{L1f} - T_{L1i})}{2C_L - \lambda_L (MC_p)_{LOE}} \qquad \text{Equation 45}$$

Substituting Equation 45 into Equation 44 and rearranging terms gives,

$$T_{L1} = T_{L1f} + \left(\frac{B}{2C_L}\right)\left(\frac{\lambda_L}{\lambda_L - \lambda_C}\right)\left[2C_L - \lambda_c (MC_P)_{LOE}\right](T_{C1f} - T_{C1i})\left[e^{-\lambda_L t} - e^{-\lambda_c t}\right] - (T_{L1f} - T_{L1i})e^{-\lambda_L t} \qquad \text{Equation 46}$$

In Appendix D, it is shown that λ_L is approximately equal to λ_C. In this case, Equation 46 becomes,

$$T_{L1} = T_{L1f} - (T_{L1f} - T_{L1i})e^{-\lambda_c t} \qquad \text{Equation 47}$$

From Appendices A through D,

$$Q_E = 5.123X10^6\ Btu/hr$$

$$Q_L = 1.986X10^6\ Btu/hr$$

$$Q_T = 7.109X10^6\ Btu/hr$$

$$C_C = 5.394X10^5\ Btu/hr\ ^\circ F$$

$$C_L = 8.402X10^4\ Btu/hr\ ^\circ F$$

$C_A = 1.073G_A\ Btu/hr\ ^\circ F$, G_A in scfm

$(UA)_R = \dfrac{1.838X10^5}{\left[1+\dfrac{1426.22}{G_A^{0.60}}\right]}\ Btu/hr\ ^\circ F$, G_A in scfm

$(UA)_C = 6.729X10^4$ Btu/hr °F

$$(MC_p)_{CT} = 1.182X10^4\ Btu/^\circ F$$

Using these values along with Equations 13, 23, 26, 43 and 47, T_{L1} has been calculated as a function of time for several air flow rates with an ambient air temperature of 70 °F. The results are shown in Figure 3.

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10. Email from Amergen Energy (D. Jones) to MPR (A. Killinger), Subject: EDG1 Surveillance Parameters, dated August 26, 2004.

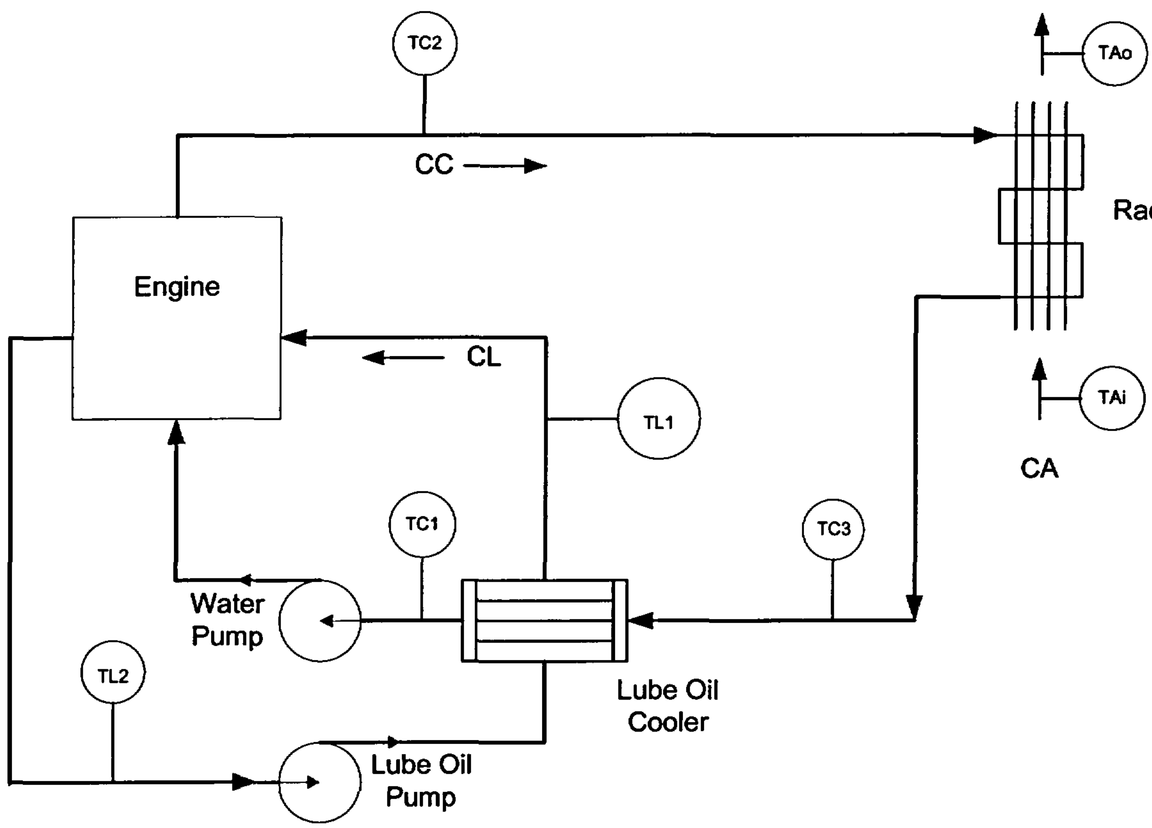
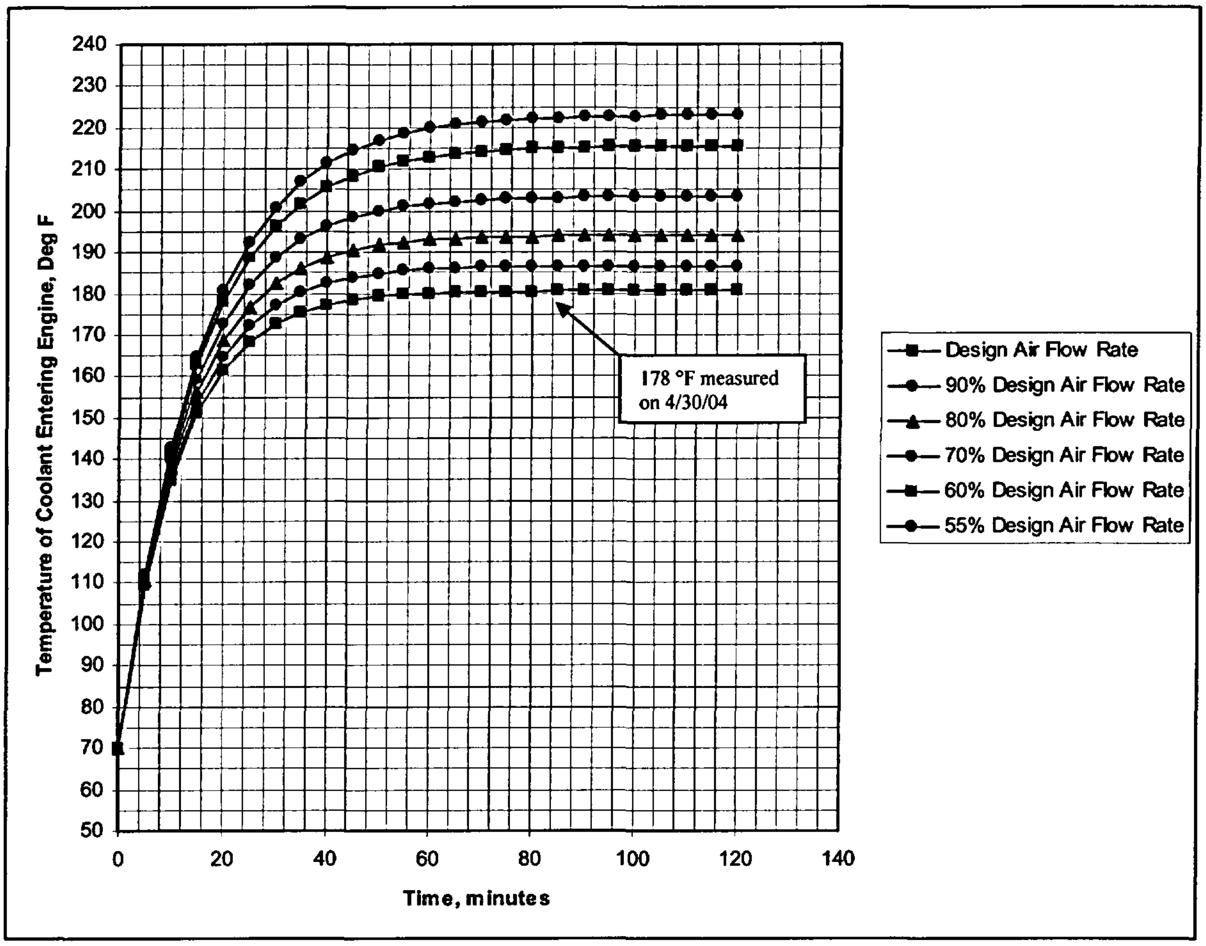


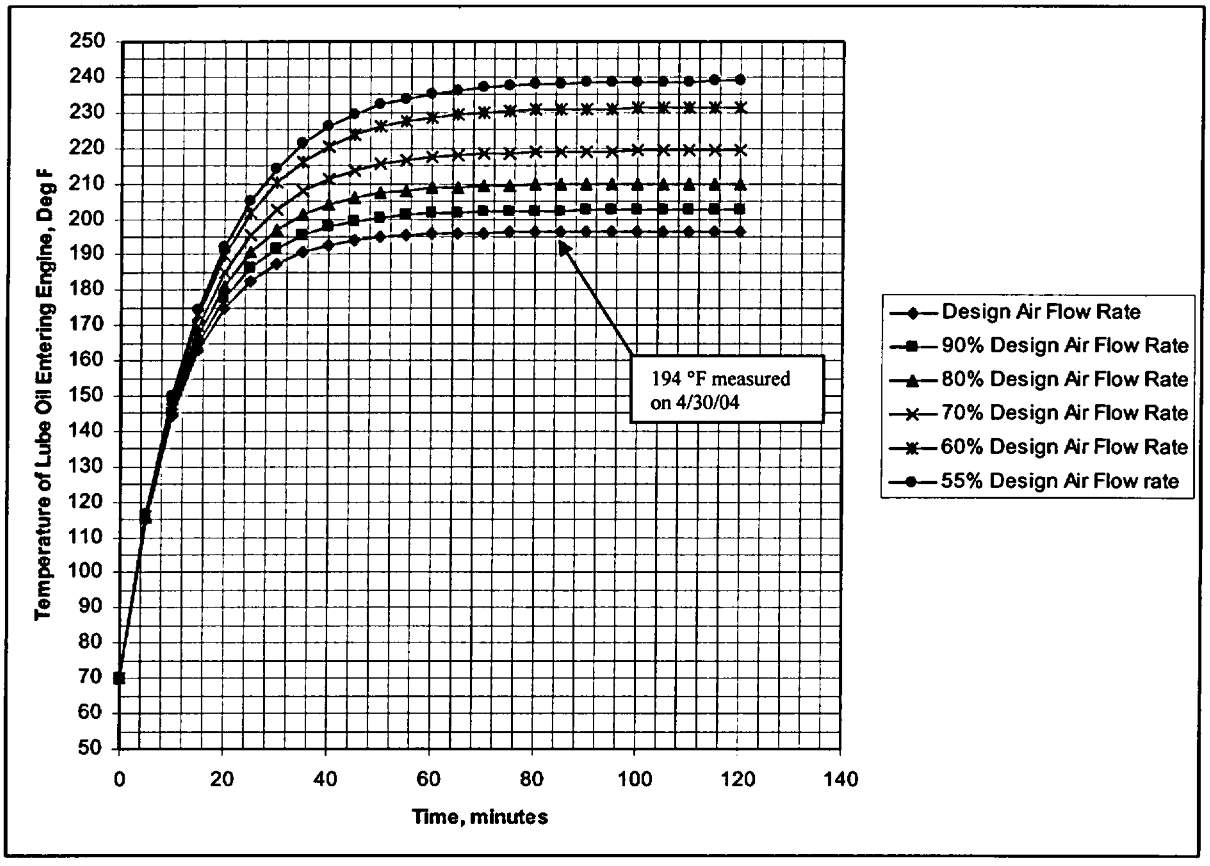
Figure 1. Radiator Cooling System for Oyster Creek EDG



Design Air Flow Rate = 125,000 scfm

Steady State Coolant Outlet Temperature = Steady State Coolant Inlet Temperature + 9.5 °F

Figure 2. Temperature of Cooling Water Entering Oyster Creek EDG Engine 70°F Ambient Air Temperature



Design Air Flow Rate = 125,000 scfm

Steady State Lube Oil Outlet Temperature = Steady State Lube Oil Inlet Temperature + 23.6 °F

Figure 3. Temperature of Lube Oil Entering Oyster Creek EDG Engine
70°F Ambient Air Temperature

A
Heat Loads

PURPOSE

The purpose of this appendix is to determine the heat loads (Q_E, Q_L, Q_T) for the Oyster Creek EDG cooling system. The references used in this appendix are listed in the Reference section of the main body of this calculation.

ANALYSIS

From References 1 and 2, the Oyster Creek EDGs are General Motors Model MP-45 units with 20 cylinder diesel engines rated at 3485 bhp. From Reference 2, the heat rejected to the jacket water is 24.5 Btu/min per bhp and the heat rejected to the lube oil is 9.5 Btu/min per bhp. Therefore,

$$Q_E = \left(\frac{24.5\,Btu}{\min bhp}\right)\left(\frac{60\,\min}{hr}\right)(3485\,bhp)$$

$$Q_E = 5.123X10^6\ Btu/hr \qquad \text{Equation A-1}$$

$$Q_L = \left(\frac{9.5\,Btu}{\min bhp}\right)\left(\frac{60\,\min}{hr}\right)(3485\,bhp)$$

$$Q_L = 1.986X10^6\ Btu/hr \qquad \text{Equation A-2}$$

$$Q_T = Q_E + Q_L = 5.123X10^6 + 1.986X10^6$$

$$Q_T = 7.109X10^6\ Btu/hr \qquad \text{Equation A-3}$$

B
Heat Transport Capacities

PURPOSE

The purpose of this appendix is to determine the heat transport capacities, C_C, C_L, C_A, [(mass flow rate)X(specific heat)] for the fluids (cooling water, lube oil, air) used in the Oyster Creek EDG cooling system. The references used in this appendix are listed in the Reference section of the main body of this calculation.

ANALYSIS

C_C

From page 2-4 of Reference 3, the engine cooling water flow rate is 1100 gpm. From Reference 1, Oyster Creek does not use anti-freeze; therefore, this analysis will use the physical properties of fresh water. From Reference 2, the radiators were sized for a maximum engine water outlet temperature of 210 °F with an ambient temperature of 90 °F. Therefore, assume an average temperature of about 150 °F. From Table 13 (page A14) of Reference 4,

$\rho_C = 61.2\ \text{lb/ft}^3$

$(C_p)_C = 0.999\ \text{Btu/lb °F}$

$$C_C = \left(\frac{1100\ gal}{\min}\right)\left(\frac{60\ \min}{hr}\right)\left(\frac{1\ ft^3}{7.4805\ gal}\right)\left(\frac{61.2\ lb}{ft^3}\right)\left(\frac{0.999\ Btu}{lb\,^\circ F}\right)$$

$C_C = 5.394X10^5\ Btu/hr\,^\circ F$ Equation B-1

C_L

From pages 2-4 and 6-4 of Reference 3, the engine lube oil flow rate is 390 gpm (scavenger oil pump flow rate). From Reference 2, the lube oil coolers were sized for a maximum engine lube oil outlet temperature of 240 °F with an ambient temperature of 90 °F. Therefore, assume an average temperature of about 165 °F. From Table 16 (page A18) of Reference 4,

$\rho_L = 53.4$ lb/ft^3

$(C_p)_L = 0.503$ Btu/lb °F

$$C_L = \left(\frac{390\ gal}{\min}\right)\left(\frac{60\ \min}{hr}\right)\left(\frac{1\ ft^3}{7.4805\ gal}\right)\left(\frac{53.4\ lb}{ft^3}\right)\left(\frac{0.503\ Btu}{lb\,^\circ F}\right)$$

$C_L = 8.402X10^4\ Btu/hr\,^\circ F$ Equation B-2

C_A

From Reference 1, the design air flow rate is 125,000 scfm. From Table 27 (page A26) of Reference 4, at standard conditions (14.7 psia, 60 °F),

$\rho_A = 0.0739$ lb/ft^3

$(C_p)_A = 0.242$ Btu/lb °F

$$C_A = \left(\frac{G_A\ ft^3}{\min}\right)\left(\frac{60\ \min}{hr}\right)\left(\frac{0.0739\ lb}{ft^3}\right)\left(\frac{0.242\ Btu}{lb\,^\circ F}\right)$$

$C_A = 1.073 G_A\ Btu/hr\,^\circ F$, G_A in scfm Equation B-3

$C_A = 1.341X10^5\ Btu/hr\,^\circ F$, at design flow

C
Heat Exchanger Characteristics

PURPOSE

The purpose of this appendix is to determine the heat exchanger characteristics [$(UA)_R$, $(UA)_C$] for the radiator and lube oil cooler used in the Oyster Creek EDG cooling system. The references used in this appendix are listed in the Reference section of the main body of this calculation.

ANALYSIS

(UA)R

The radiator design information used in the following analysis is from Reference 5.

The radiator is a finned tube cross flow heat exchanger with the cooling water flowing on the insides of the tubes and the air flowing over the finned outside surfaces of the tubes. From page 124 of Reference 6, the overall heat transfer coefficient for the radiator is given by

$$U = \left[\frac{1}{E_f h_o} + \frac{r_o}{E_f} + r_w + \left(\frac{A_o}{A_i} \right) \left(\frac{1}{h_i} + r_i \right) \right]^{-1} \qquad \text{Equation C-1}$$

Where,

U = the overall heat transfer coefficient, Btu/hr ft^2 °F

h_o = film heat transfer coefficient on outside surfaces of tubes, Btu/hr ft^2 °F

h_i = film heat transfer coefficient on inside surfaces of tubes, Btu/hr ft^2 °F

r_o = thermal fouling resistance on outside surfaces of tubes, hr ft^2 °F/Btu

r_i = thermal fouling resistance on inside surfaces of tubes, hr ft^2 °F/Btu

r_w = thermal resistance of tube walls, hr ft^2 °F/Btu

E_f = fin efficiency

A_o = total outside heat transfer surface area, ft^2

A_i = total inside heat transfer surface area, ft^2

From pages 286 and 290 of Reference 6,

r_o = 0.001 hr ft^2 °F/Btu, compressed air

r_i = 0.001 hr ft^2 °F/Btu, engine jacket water

The radiator contains 460 red brass tubes with an OD of 0.500 inch and a wall thickness of 0.025 inch. Therefore, the ID of each tube is 0.450 inch (0.500-2x0.025). From page 125 of Reference 6, the thermal resistance of the tube walls is,

$$r_w = \left(\frac{OD}{24k_t}\right)\ln\left(\frac{OD}{ID}\right)$$

Where,

k_t = thermal conductivity of the tube material, Btu/hr ft °F

From Table 10 (page A9) of Reference 4,

k_t = 35.2 Btu/hr ft °F, red brass

Therefore,

$$r_w = \left(\frac{0.500}{24(35.2)}\right)\ln\left(\frac{0.500}{0.450}\right)$$

r_w = 0.00006 hr ft^2 °F/Btu

The fins are parallel perforated aluminum plates with the tubes passing through the perforations.

Each radiator contains 1060 plates, each with a height of 78.0 inches, a width of 15.75 inches and a thickness of 0.010 inch. The available face area of each plate is

$$A_{FP} = (78.0)(15.75) - (460)\left(\frac{\pi}{4}\right)(0.500)^2$$

$$A_{FP} = 1138.18\ in^2 = 7.904\ ft^2$$

The total fin area is

$$A_f = (2)(1060)(7.904)$$

$$A_f = 16{,}757\ ft^2$$

The effective length of the tubes is 106.125 inches. The bare length of the tubes is

$$L_b = 106.125 - (1060)(0.010) = 95.525\ inches$$

The bare surface area of the tubes is

$$A_b = (460)(\pi)(0.500)(95.525)$$

$$A_b = 69{,}023\ in^2 = 479\ ft^2$$

Therefore,

$$A_o = 16{,}757 + 479 = 17{,}236\ ft^2$$

The inside surface area of the tubes is

$$A_i = (460)(\pi)(0.450)(106.125)$$

$$A_i = 69{,}014\ in^2 = 479\ ft^2$$

From page 2-4 of Reference 3, the engine cooling water flow rate is 1100 gpm. From Reference 1, Oyster Creek does not use anti-freeze; therefore, this analysis will use the physical properties of fresh water. From Reference 2, the radiator was sized for a maximum engine water outlet temperature of 210 °F with an ambient temperature of 90 °F. Therefore, assume an average temperature of about 150 °F. From Table 13 (page A14) of Reference 4,

$\rho_C = 61.2$ lb/ft^3

$(C_p)_C = 0.999$ Btu/lb °F

$k_C = 0.378$ Btu/hr ft °F

$\mu_C = 2.98X10^{-4}$ lb/ft s

$Pr_C = 2.73$

The velocity of the coolant flowing through the tubes is

$$V_C = \left[\frac{1100\ gal/\min}{(460)}\right]\left(\frac{1\ \min}{60\ s}\right)\left(\frac{1\ ft^3}{7.4805\ gal}\right)\left[\frac{4}{\pi(0.450/12\ ft)^2}\right] = 4.824\ ft/s$$

The Reynolds number is

$$\text{Re}_C = \frac{(61.2\ lb/ft^3)(4.824\ ft/s)(0.450/12\ ft)}{2.98X10^{-4}\ lb/ft\,s} = 3.715X10^4$$

The inside heat transfer coefficient is given by Equation 6.63 (page 413) of Reference 4

$$h_i = (0.023)\left(\frac{k_C}{ID}\right)(\text{Re}_C)^{0.8}(\text{Pr}_C)^{0.3} = (0.023)\left(\frac{0.378}{0.450/12}\right)(3.715X10^4)^{0.8}(2.73)^{0.3}$$

$$h_i = 1419\ Btu/hr\ ft^2\ {}^\circ F$$

From Reference 1, the design air flow rate is 125,000 scfm. From Table 27 (page A26) of Reference 4, at standard conditions (14.7 psia, 60 °F),

$\rho_A = 0.0739$ lb/ft^3

$(C_p)_A = 0.242$ Btu/lb °F

$k_A = 0.0143$ Btu/hr ft °F

$\mu_A = 1.214 \text{X} 10^{-5}$ lb/ft s

$Pr_A = 0.71$
The face area of the radiator is

$$A_F = (78.0)(106.125) = 8277.75\ in^2 = 57.484\ ft^2$$

The face velocity of the air is

$$V_\infty = \left[\frac{G_A\ ft^3/\min}{(57.484\ ft^2)}\right]\left(\frac{1\ \min}{60\ s}\right)$$

$$V_\infty = \frac{G_A}{3449.04}\ ft/s,\ G_A \text{ in scfm}$$

The tubes are arranged in a staggered triangular array, as shown in Figure C-1. S_T and S_L are estimated as follows:

$$S_T = \frac{39\ inches}{26} = 1.50\ inches$$

$$S_L = \frac{15.75\ inches}{9} = 1.75\ inches$$

$$a = \frac{1}{2}(S_T + OD) = \frac{1}{2}(1.50 + 0.50) = 1.00\ inch$$

$$b = \sqrt{\left(\frac{S_T}{2}\right)^2 + S_L^2} = \sqrt{\left(\frac{1.50}{2}\right)^2 + (1.75)^2} = 1.90\ inches$$

Since a is less than b, the maximum velocity of the air flowing over the tubes is given by (pages 473 and 474 of Reference 4)

$$V_{\max} = \left(\frac{S_T}{S_T - OD}\right)V_\infty = \left(\frac{1.50}{1.50 - 0.50}\right)\left(\frac{G_A}{3449.04}\right)$$

$$V_{\max} = \frac{G_A}{2299.36}\ ft/s,\ G_A \text{ in scfm}$$

$V_{\max} = 54.36\ ft/s$, at design flow of 125,000 scfm

The Reynolds number is given by

$$\mathrm{Re}_A = \frac{\rho_A V_{\max}(OD)}{\mu_A} = \frac{\left(0.0739\,lb/ft^3\right)\left(\frac{G_A}{2299.36}\,ft/s\right)\left(0.50/12\,ft\right)}{1.214X10^{-5}\,lb/ft\,s}$$

$\mathrm{Re}_A = (0.11031)G_A$, G_A in scfm

$\mathrm{Re}_A = 1.379X10^4$, at design flow of 125,000 scfm

Neglecting the correction factor for film temperature, the outside heat transfer coefficient is given by Equation 7.30 on page 475 of Reference 4,

$$h_o = (0.35)\left(\frac{k_A}{OD}\right)\left(\frac{S_T}{S_L}\right)^{0.2}(\mathrm{Re}_A)^{0.60}(\mathrm{Pr}_A)^{0.36} = (0.35)\left(\frac{0.0143}{0.50/12}\right)\left(\frac{1.5}{1.75}\right)^{0.2}(0.11031G_A)^{0.60}(0.71)^{0.36}$$

$h_o = \left(2.7431X10^{-2}\right)G_A^{0.60}\ Btu/hr\,ft^2\,{}^\circ F$, G_A in scfm

$h_o = 31.36\ Btu/hr\,ft^2\,{}^\circ F$, at design flow of 125,000 scfm

Fin efficiency will be calculated assuming equivalent circular fins, as follows,

$$R_i = \frac{OD}{2} = \frac{0.500}{2} = 0.250\ in$$

$t = 0.010$ inch

The fin face area associated with each tube is

$$A_{fF} = \frac{A_{FP}}{460}$$

This area is given by

$\pi\left(R_o^2 - R_i^2\right) = \frac{A_{FP}}{460}$, or

$$R_o = \left[\frac{A_{FP}}{460\pi} + R_i^2\right]^{0.5} = \left[\frac{1138.18}{460\pi} + (0.250)^2\right]^{0.5} = 0.922\ in$$

From Table 10 (page A9) of Reference 4,

k = 94.7 Btu/hr ft °F, aluminum

$$P = \left(R_o + \frac{t}{2} - R_i\right)^{3/2} \sqrt{\frac{2h_o}{kt(R_o - R_i)}}$$

$$P = \left(0.922 + \frac{0.010}{2} - 0.250\right)^{3/2} \sqrt{\frac{2(31.36/12)}{(94.7)(0.010)(0.922 - 0.250)}}$$

$$P = 1.60$$

$$Z = \frac{R_o + \left(\frac{t}{2}\right)}{R_i} = \frac{0.922 + \left(\frac{0.010}{2}\right)}{0.250}$$

$$Z = 3.7$$

From Figure 2.19 (page 107) of Reference 4, the fin efficiency is estimated to be

$E_f = 0.40$

This is the fin efficiency at design air flow (125,000 scfm). For this analysis it will be assumed that the fin efficiency remains constant at the design value during the heat up transient. Therefore,

$$U = \left[\frac{1}{(0.40)\left(2.7431X10^{-2}G_A^{0.60}\right)} + \frac{0.001}{0.40} + 0.00006 + \left(\frac{17{,}236}{479}\right)\left(\frac{1}{1419} + 0.001\right)\right]^{-1}$$

$$U = \frac{15.649}{\left[1 + \dfrac{1426.22}{G_A^{0.60}}\right]} \; Btu/hr\,ft^2\,{}^\circ F,\ G_A \text{ in scfm}$$

$U = 6.963\ Btu/hr\,ft^2\,{}^\circ F$, at design flow of 125,000 scfm

Therefore,

$$(UA)_R = (17{,}236)\frac{15.649}{\left[1 + \dfrac{1426.22}{G_A^{0.60}}\right]},$$

$$(UA)_R = \frac{2.697X10^5}{\left[1 + \dfrac{1426.22}{G_A^{0.60}}\right]} \; Btu/hr\ {}^\circ F,\ G_A \text{ in scfm} \qquad \text{Equation C-2}$$

$(UA)_R = 1.200X10^5\ Btu/hr\ {}^\circ F$, at design flow of 125,000 scfm

From Equation 18, the final steady state temperature of the coolant exiting the engine (T_{C2}) is

$$T_{C2f} = T_{Ai} + \left[\frac{1}{2C_A} + \frac{1}{(UA)_R} + \frac{1}{2C_C}\right]Q_T \text{; or}$$

$$\frac{1}{(UA)_R} = \frac{T_{C2f} - T_{Ai}}{Q_T} - \frac{1}{2}\left[\frac{1}{C_A} + \frac{1}{C_C}\right]$$

From Reference 2, the radiator was sized for a maximum engine water outlet temperature of 210 °F with an ambient temperature of 90 °F. From Appendices A and B,

$Q_T = 7.109X10^6\ Btu/hr$

$C_C = 5.394X10^5\ Btu/hr\,{}^\circ F$

$C_A = 1.341X10^5\ Btu/hr\,{}^\circ F$, at design flow

$$\frac{1}{(UA)_R} = \frac{210-90}{7.109X10^6} - \frac{1}{2}\left[\frac{1}{1.341X10^5} + \frac{1}{5.394X10^5}\right]$$

$(UA)_R = 8.180X10^4$ Btu/hr °F

Therefore, Equation C-2 will be adjusted as follows,

$$(UA)_R = \left(\frac{8.180X10^4}{1.20X10^5}\right)\frac{2.697X10^5}{\left[1+\frac{1426.22}{G_A^{0.60}}\right]}\ Btu/hr\ ^\circ F$$

$$(UA)_R = \frac{1.838X10^5}{\left[1+\frac{1426.22}{G_A^{0.60}}\right]}\ Btu/hr\ ^\circ F,\ G_A \text{ in scfm} \qquad \text{Equation C-3}$$

$(UA)_R = 8.180X10^4\ Btu/hr\ ^\circ F$, at design flow of 125,000 scfm

This adjustment (about 32%) is due to the fact that Equation 18 is based on the approximation that the heat transferred in the radiator is equal to $(UA)_R$ times the average temperature difference between the coolant and the air. In reality, the heat transferred is equal to $(UA)_R$ times the corrected log mean temperature difference.

From Equation 23, the final steady state temperature of the coolant entering the engine (T_{C1}) is

$$T_{C1f} = T_{Ai} + \left[\frac{1}{2C_A} + \frac{1}{(UA)_R} - \frac{(Q_E - Q_L)}{2C_C Q_T}\right]Q_T$$

From Appendix A,

$Q_E = 5.123X10^6\ Btu/hr$

$Q_L = 1.986X10^6\ Btu/hr$

$Q_T = 7.109X10^6\ Btu/hr$

$$T_{C1f} = 90.0 + \left[\frac{1}{2(1.341X10^5)} + \frac{1}{8.180X10^4} - \frac{(5.123X10^6 - 1.986X10^6)}{2(5.394X10^5)(7.109X10^6)}\right](7.109X10^6)$$

$T_{C1f} = 200.51\ ^\circ F$

(UA)c

The lube oil cooler design information used in the following analysis is from Reference 7.

The lube oil cooler is a finned tube cross flow heat exchanger with the cooling water flowing on the insides of the tubes and the lube oil flowing over the finned outside surfaces of the tubes. From page 124 of Reference 6, the overall heat transfer coefficient for the cooler is given by

$$U = \left[\frac{1}{E_f h_o} + \frac{r_o}{E_f} + r_w + \left(\frac{A_o}{A_i} \right) \left(\frac{1}{h_i} + r_i \right) \right]^{-1} \qquad \text{Equation C-1}$$

Where,

U = the overall heat transfer coefficient, Btu/hr ft^2 °F

h_o = film heat transfer coefficient on outside surfaces of tubes, Btu/hr ft^2 °F

h_i = film heat transfer coefficient on inside surfaces of tubes, Btu/hr ft^2 °F

r_o = thermal fouling resistance on outside surfaces of tubes, hr ft^2 °F/Btu

r_i = thermal fouling resistance on inside surfaces of tubes, hr ft^2 °F/Btu

r_w = thermal resistance of tube walls, hr ft^2 °F/Btu

E_f = fin efficiency

A_o = total outside heat transfer surface area, ft^2

A_i = total inside heat transfer surface area, ft^2

From pages 286 and 290 of Reference 6,

r_o = 0.001 hr ft^2 °F/Btu, engine lube oil

$r_i = 0.001$ hr ft^2 °F/Btu, engine jacket water

The cooler contains 1384 red brass tubes with an OD of 0.250 inch and a wall thickness of 0.017 inch. Therefore, the ID of each tube is 0.216 inch (0.250-2x0.017). From page 125 of Reference 6, the thermal resistance of the tube walls is,

$$r_w = \left(\frac{OD}{24k_t}\right)\ln\left(\frac{OD}{ID}\right)$$

Where,

k_t = thermal conductivity of the tube material, Btu/hr ft °F

From Table 10 (page A9) of Reference 4,

$k_t = 35.2$ Btu/hr ft °F, red brass

Therefore,

$$r_w = \left(\frac{0.250}{24(35.2)}\right)\ln\left(\frac{0.250}{0.216}\right)$$

$r_w = 0.00004$ hr ft^2 °F/Btu

The fins are parallel perforated aluminum plates with the tubes passing through the perforations. The cooler contains 784 plates, each with a height of 32.0 inches, a width of 6.0 inches and a thickness of 0.010 inch. The available face area of each plate is

$$A_{FP} = (32.0)(6.0) - (1384)\left(\frac{\pi}{4}\right)(0.250)^2$$

$$A_{FP} = 124.06\ in^2 = 0.8616\ ft^2$$

The total fin area is

$$A_f = (2)(784)(0.8616)$$

$$A_f = 1351\ ft^2$$

The effective length of the tubes is 26.66 inches. The bare length of the tubes is

$$L_b = 26.66 - (784)(0.010) = 18.82 \ inches$$

The bare surface area of the tubes is

$$A_b = (1384)(\pi)(0.250)(18.82)$$

$$A_b = 20{,}457 \ in^2 = 142 \ ft^2$$

Therefore,

$$A_o = 1351 + 142 = 1493 \ ft^2$$

The inside surface area of the tubes is

$$A_i = (1384)(\pi)(0.216)(26.66)$$

$$A_i = 25{,}038 \ in^2 = 174 \ ft^2$$

The velocity of the coolant flowing through the tubes is

$$V_C = \left[\frac{1100 \ gal/\min}{(1384)}\right]\left(\frac{1 \min}{60 \ s}\right)\left(\frac{1 \ ft^3}{7.4805 \ gal}\right)\left[\frac{4}{\pi(0.216/12 \ ft)^2}\right] = 6.959 \ ft/s$$

The Reynolds number is

$$\mathrm{Re}_C = \frac{(61.2 \ lb/ft^3)(6.959 \ ft/s)(0.216/12 \ ft)}{2.98X10^{-4} \ lb/ft\,s} = 2.5725X10^4$$

The inside heat transfer coefficient is given by Equation 6.63 (page 413) of Reference 4

$$h_i = (0.023)\left(\frac{k_C}{ID}\right)(\mathrm{Re}_C)^{0.8}(\mathrm{Pr}_C)^{0.4} = (0.023)\left(\frac{0.378}{0.216/12}\right)(2.5725X10^4)^{0.8}(2.73)^{0.4}$$

$$h_i = 2436.1 \ Btu/hr \ ft^2 \ {}^\circ F$$

From pages 2-4 and 6-4 of Reference 3, the engine lube oil flow rate is 390 gpm (scavenger oil pump flow rate). From Reference 2, the lube oil coolers were sized for a maximum engine lube oil outlet temperature of 240 °F with an ambient temperature of 90 °F. Therefore, assume an average temperature of about 165 °F. From Table 16 (page A18) of Reference 4,
$\rho_L = 53.4$ lb/ft^3

$(C_p)_L = 0.503$ Btu/lb °F

$k_L = 0.0801$ Btu/hr ft °F

$\mu_L = 2.982 \times 10^{-2}$ lb/ft s

$Pr_L = 661$

The face area of the cooler is

$$A_F = (32.0)(26.66) = 853.12\ in^2 = 5.924\ ft^2$$

The face velocity of the lube oil is

$$V_\infty = \left[\frac{390\ gal/\min}{(5.924\ ft^2)}\right]\left(\frac{1\ \min}{60\ s}\right)\left(\frac{1\ ft^3}{7.4805\ gal}\right)$$

$$V_\infty = 0.1467\ ft/s$$

The tubes are arranged in a staggered triangular array, as shown in Figure C-1. S_T and S_L are estimated as follows:

$$S_T = \frac{32\ inches}{87} = 0.3678\ inch$$

$$S_L = \frac{6}{16} = 0.3750\ inch$$

$$a = \frac{1}{2}(S_T + OD) = \frac{1}{2}(0.3678 + 0.250) = 0.3089\ inch$$

$$b = \sqrt{\left(\frac{S_T}{2}\right)^2 + S_L^2} = \sqrt{\left(\frac{0.3678}{2}\right)^2 + (0.3750)^2} = 0.4177\ inches$$

Since a is less than b, the maximum velocity of the lube oil flowing over the tubes is given by (pages 473 and 474 of Reference 4)

$$V_{\max} = \left(\frac{S_T}{S_T - OD}\right)V_\infty = \left(\frac{0.3678}{0.3678 - 0.250}\right)(0.1467)$$

$$V_{\max} = 0.4580\ ft/s$$

The Reynolds number is given by

$$\mathrm{Re}_L = \frac{\rho_L V_{\max}(OD)}{\mu_L} = \frac{(53.4\,lb/ft^3)(0.4580\,ft/s)(0.250/12\,ft)}{2.982X10^{-2}\ lb/ft\,s}$$

$$\mathrm{Re}_L = 17.087$$

Neglecting the correction factor for film temperature, the outside heat transfer coefficient is given by Equation 7.28 on page 475 of Reference 4,

$$h_o = (0.9)\left(\frac{k_L}{OD}\right)(\mathrm{Re}_L)^{0.4}(\mathrm{Pr}_L)^{0.36} = (0.9)\left(\frac{0.0801}{0.250/12}\right)(17.087)^{0.40}(661)^{0.36}$$

$$h_o = 111.55\ Btu/hr\,ft^2\,{}^\circ F$$

Fin efficiency will be calculated assuming equivalent circular fins, as follows,

$$R_i = \frac{OD}{2} = \frac{0.2500}{2} = 0.1250\ in$$

t = 0.010 inch

The fin face area associated with each tube is

$$A_{fF} = \frac{A_{FP}}{1384}$$

This area is given by

$$\pi\left(R_o^2 - R_i^2\right) = \frac{A_{FP}}{1384}, \text{ or}$$

$$R_o = \left[\frac{A_{FP}}{1384\pi} + R_i^2\right]^{0.5} = \left[\frac{124.06}{1384\pi} + (0.1250)^2\right]^{0.5} = 0.210\ in$$

From Table 10 (page A9) of Reference 4,

k = 94.7 Btu/hr ft °F, aluminum

$$P = \left(R_o + \frac{t}{2} - R_i\right)^{3/2} \sqrt{\frac{2h_o}{kt(R_o - R_i)}}$$

$$P = \left(0.210 + \frac{0.010}{2} - 0.1250\right)^{3/2} \sqrt{\frac{2(111.55/12)}{(94.7)(0.010)(0.210 - 0.1250)}}$$

$$P = 0.41$$

$$Z = \frac{R_o + \left(\frac{t}{2}\right)}{R_i} = \frac{0.210 + \left(\frac{0.010}{2}\right)}{0.1250}$$

$$Z = 1.72$$

From Figure 2.19 (page 107) of Reference 4, the fin efficiency is estimated to be

$E_f = 0.93$

$$U = \left[\frac{1}{(0.93)(111.55)} + \frac{0.001}{0.93} + 0.00004 + \left(\frac{1493}{174}\right)\left(\frac{1}{2436.1} + 0.001\right)\right]^{-1}$$

$$U = 43.824\ Btu/hr\ ft^2\ {}^\circ F$$

$$(UA)_C = (43.824)(1493)$$

$$(UA)_C = 6.543X10^4\ Btu/hr\ {}^\circ F \qquad \text{Equation C-4}$$

From Equations 41 and 42, the final steady state temperature of the lube oil exiting the engine (T_{L2}) is

$$T_{L2f} = T_{C1f} + \left[\frac{1}{2C_L} + \frac{1}{(UA)_C} - \frac{1}{2C_C}\right]Q_L \text{, or}$$

$$\frac{1}{(UA)_C} = \frac{T_{L2f} - T_{C1f}}{Q_L} + \frac{1}{2}\left[\frac{1}{C_C} - \frac{1}{C_L}\right]$$

From Reference 2, the cooler was sized for a maximum engine lube oil outlet temperature of 240 °F with an ambient temperature of 90 °F. From Appendices A and B,

$$Q_L = 1.986X10^6 \ Btu/hr$$

$$C_C = 5.394X10^5 \ Btu/hr\,^\circ F$$

$$C_L = 8.402X10^4 \ Btu/hr\,^\circ F$$

$$\frac{1}{(UA)_C} = \frac{240.0 - 200.51}{1.986X10^6} + \frac{1}{2}\left[\frac{1}{5.394X10^5} - \frac{1}{8.402X10^4}\right]$$

$(UA)_C = 6.729X10^4$ Btu/hr °F

Therefore, $(UA)_C$ will be adjusted to,

$(UA)_C = 6.729X10^4$ Btu/hr °F Equation C-5

This adjustment (about 3%) is due to the fact that Equations 41 and 42 are based on the approximation that the heat transferred in the cooler is equal to $(UA)_C$ times the average temperature difference between the coolant and the lube oil. In reality, the heat transferred is equal to $(UA)_C$ times the corrected log mean temperature difference.

From Equation 43, the final steady state temperature of the lube oil entering the engine (T_{L1}) is

$$T_{L1f} = T_{C1f} + \left[\frac{1}{(UA)_C} - \frac{1}{2}\left(\frac{1}{C_C} + \frac{1}{C_L}\right)\right]Q_L$$

$$T_{L1f} = 200.51 + \left[\frac{1}{6.729X10^4} - \frac{1}{2}\left(\frac{1}{5.394X10^5} + \frac{1}{8.402X10^4}\right)\right]\left(1.986X10^6\right)$$

$$T_{L1f} = 216.36\ ^\circ F$$

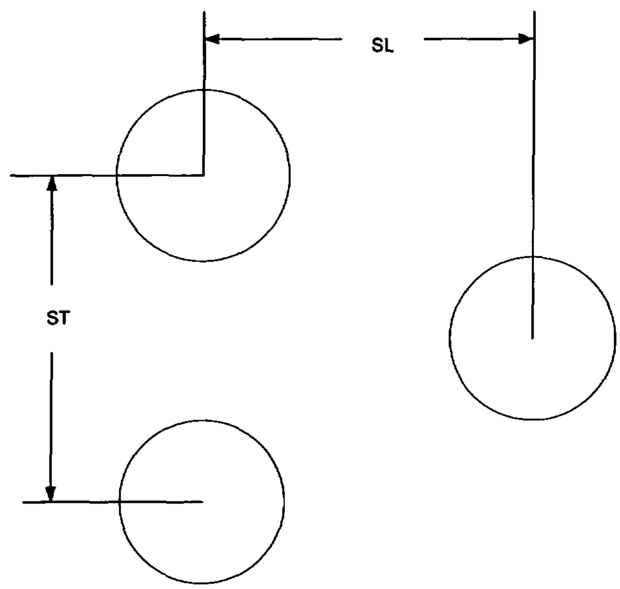


Figure C-1. Oyster Creek EDG Radiator and Lube Oil Cooler Tube Arrangement

D
Cooling System Heat Absorbing Capacity

PURPOSE

The purpose of this appendix is to determine the heat absorbing capacity, $(MC_p)_{CT}$, [(mass)X(specific heat)] for the cooling water system used in the Oyster Creek EDG cooling system. The references used in this appendix are listed in the Reference section of the main body of this calculation.

ANALYSIS

Reference 8 provides heat up data for a General Motors Model MP-36 EDG unit with a 16 cylinder diesel engine. Similar heat up data are not available for the Model MP-45 (20 cylinder) engine used at Oyster Creek. Therefore, the Model MP-36 data will be used to estimate $(MC_p)_{CT}$ for the Model MP-45 engine.

The heat up data for the coolant and lube oil are shown in Figure D-1. From Equation 26 the coolant temperatures shown in Figure D-1 can be represented by the following equation,

$$T_C = T_{Cf} - (T_{Cf} - T_{Ci})e^{-\lambda_{CT} t} \qquad \text{Equation D-1}$$

Where, λ_{CT} is the time decay constant for the coolant system.

Assuming the time decay constant (λ_{LT}) for the lube oil system is approximately equal to λ_{CT}, the lube oil temperatures shown in Figure D-1 can be represented by the following equation,

$$T_L = T_{Lf} - (T_{Lf} - T_{Li})e^{-\lambda_{LT} t} \qquad \text{Equation D-2}$$

In Equations D-1 and D-2, the subscript f refers to final conditions, the subscript i refers to initial conditions and the subscript T refers to test conditions.

Equations D-1 and D-2 can also be written as,

$$\ln\left(\frac{T_{Cf}-T_{Ci}}{T_{Cf}-T_{C}}\right)=\lambda_{CT}t$$ Equation D-3

$$\ln\left(\frac{T_{Lf}-T_{Li}}{T_{Lf}-T_{L}}\right)=\lambda_{LT}t$$ Equation D-4

From the test data,

T_{Ci} = 141 °F

T_{Li} = 135 °F

From the test data, it appears that the final steady state temperatures are about 187 °F and 231 °F for the coolant and lube oil, respectively. That is,

T_{Cf} = 187 °F

T_{Lf} = 231 °F

Therefore, Equations D-3 and D-4 become,

$$\ln\left(\frac{46}{187-T_{C}}\right)=\lambda_{CT}t$$ Equation D-5

$$\ln\left(\frac{96}{231-T_{L}}\right)=\lambda_{LT}t$$ Equation D-6

The measured values of T_C and T_L have been substituted into Equations D-5 and D-6, and the results plotted as a function of time, see Figure D-2.

From Figure D-2, it can be seen that for the first 30 minutes, the coolant data and the lube oil data fall on essentially the same line, indicating that λ_{LT} is approximately equal to λ_{CT}. For times greater than 30 minutes, there is considerable scatter in the data. This is believed to be due to the thermostatic temperature control valve regulating the flow rate of coolant through the radiator to maintain the temperatures at their final steady state values.

The coolant and lube oil data shown in Figure D-2 for the first 30 minutes are combined to determine an effective value for λ_{CT}. A linear regression analysis of these data gives,
$\lambda_{CT} = 8.836\text{X}10^{-2}$/min

From Equation 13,

$$\lambda_{CT} = \frac{(UA)_{RT}}{(MC_p)_{CTT}\left[1+(UA)_{RT}/2C_{AT}\right]}, \text{ or}$$

$$(MC_p)_{CTT} = \frac{(UA)_{RT}}{\lambda_{CT}\left[1+(UA)_{RT}/2C_{AT}\right]} \qquad \text{Equation D-7}$$

Reference 9 provides design information for the MP-36 radiator; however, sufficient information is not provided to calculate $(UA)_{RT}$ from first principles; therefore, $(UA)_{RT}$ and C_{AT} will be estimated from $(UA)_R$ and C_A given in Appendices B and C (at design air flow rate of 125,000 scfm) as follows.

$$(UA)_{RT} = \left(\frac{16}{20}\right)(UA)_R = \left(\frac{16}{20}\right)\left(8.180X10^4\right) = 6.544X10^4 \ Btu/hr\,^\circ F$$

$$C_{AT} = \left(\frac{16}{20}\right)C_A = \left(\frac{16}{20}\right)\left(1.341X10^5\right) = 1.073X10^5 \ Btu/hr\,^\circ F$$

Therefore,

$$(MC_p)_{CTT} = \frac{6.544X10^4}{\left(8.836X10^{-2}\right)\left[1+\frac{6.544X10^4}{2\left(1.073X10^5\right)}\right]}\left(\frac{1\,hr}{60\,\text{min}}\right) = 9.459X10^3 \ Btu/^\circ F$$

Assuming (MC_p) is proportional to engine size,

$$(MC_p)_{CT} = \left(\frac{20}{16}\right)(MC_p)_{CTT} = \left(\frac{20}{16}\right)\left(9.459X10^3\right) = 1.182X10^4 \ Btu/^\circ F$$

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Calculation No. 0083-0314-RCS01	Prepared By	Checked By	Page: D-4 Revision: 1

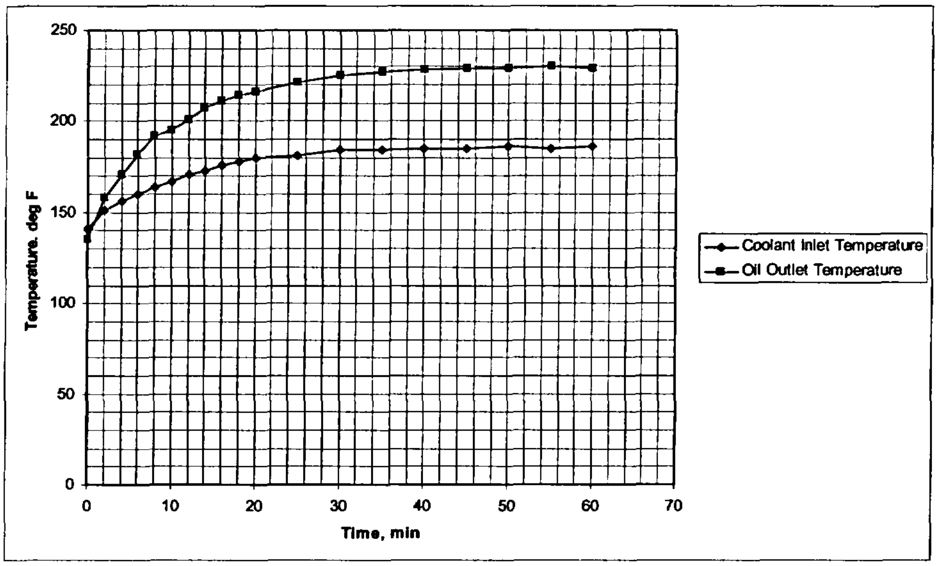

Figure D-1. Heat Up Data for Model MP-36 Diesel Engine

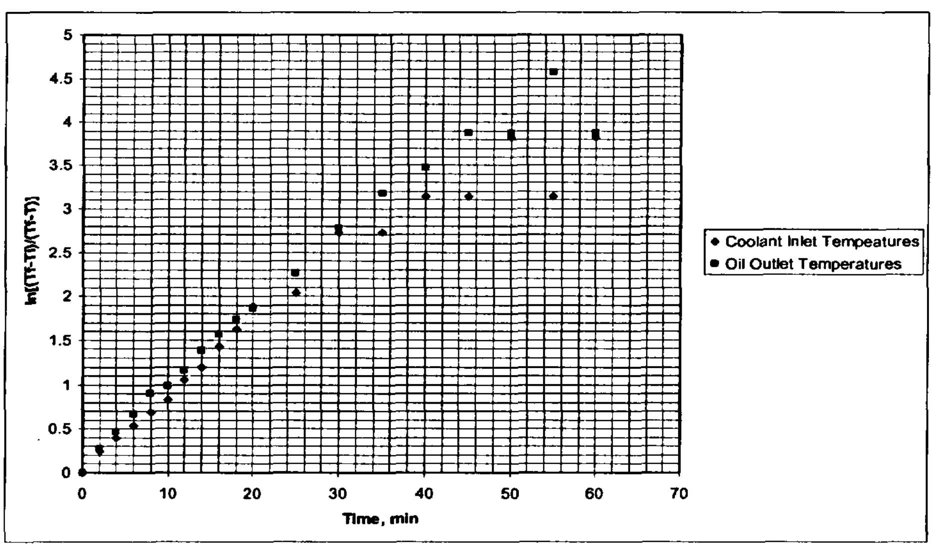

Figure D-2. Heat Up Data for Model MP-36 Diesel Engine