Evaluating the Performance of Emergency Diesel Generator Lube Oil Coolers

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Abstract

Nuclear Regulatory Commission (NRC) Generic Letter 89-13 requires a test program to verify the heat transfer capability of safety-related heat exchangers cooled by service water. This paper discusses the design of a test conducted on an emergency diesel generator (EDG) lube oil cooler (LOC). An outline of the test design package (TDP) is presented. The TDP identifies the test method and utilizes an analysis of measurement uncertainty to help specify the number, accuracy, and placement of instruments along with the minimum number of data readings to ensure the acceptability of test results. The EDG LOC heat exchanger presents a challenge in designing a successful test because of the high viscosity and insulating properties of oil. The test data is presented and evaluated to determine the heat transfer capability of the heat exchangers at design basis conditions. The shell-side flow rate was not measured during the test but is calculated based on the heat transfer that is calculated from the flow and temperatures measured on the water side and the temperature difference measured on the shell side. The analysis determines the lube oil convection heat transfer coefficient on the shell side and then computes the heat transfer rate for the heat exchanger at limiting conditions based on measurements collected at test conditions.

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Test Design Package

The goal of the test design package (TDP) is to determine if an acceptable test can be performed on a heat exchanger. A complete TDP specifies heat exchanger and test acceptance criteria and provides a complete test design. The TDP also includes a pre-test uncertainty analysis which helps to specify the number, accuracy, and placement of instruments along with the minimum number of data readings to ensure the acceptability of test results. The design limiting conditions for the emergency diesel generator (EDG) lube oil cooler (LOC) that was tested are as follows:

Entering Cooling Water (CW) Temperature, $T_{c,i}$	132.4 °F
Exiting CW Temperature, $T_{c,o}$	144.6 °F
CW Flow Rate, G_c	350 gpm
Entering Lube Oil (LO) Temperature, $T_{h,i}$	167.5 °F
Exiting LO Temperature, $T_{h,o}$	148.6 °F
LO Flow Rate, G_h	530 gpm
Heat Removal Rate, Q	2.112×10^6 Btu/hr

The test acceptance criteria was an overall heat removal rate of $Q = 1.940 \times 10^6$ Btu/hr, the EDG LOC required heat removal rate at design basis conditions. A target test uncertainty of 25% of the design margin was established by the client. This equated to a maximum uncertainty of 49,000 (Btu/hr) in the projected design capacity. Design margin is defined as the difference in the capacity between the clean and designed fouled conditions of a heat exchanger. The clean condition for a heat exchanger is specified as a fouling resistance of zero. For the LOC the design fouled condition was specified as a fouling resistance of 0.001 hr-ft^2-°F/Btu on the shell and tube side each.

The performance test was conducted by placing the EDG and the EDG LOC in operation at full electrical load long enough for the operating temperatures to stabilize prior to testing. A minimum CW flow rate of 187 gpm was specified to achieve a Reynolds Number of at least 10,000 so that turbulent flow would exist in the tube side of the LOC. To achieve turbulent flow on the shell side, the required LO flow was calculated to be 12,839 gpm. Since this was not feasible, the shell side flow was analyzed in laminar flow. To perform the pre-test uncertainty analysis, the design heat load and fouling were used. The anticipated test conditions are presented in Table I. The high CW inlet temperature is the result of the CW first circulating through the EDG air inter-cooler. A fouling resistance of 0.001 (hr-ft^2-°F/Btu) on the tube side only was assumed for normal test conditions.

Table 1: Expected Test Conditions

Parameter	Expected Condition
Entering CW Temperature, F	107.4
Exiting CW Temperature, F	118.6
CW Flow, gpm	350
Inlet LO Temperature, F	160
Outlet LO Temperature, F	118.1
LO Flow, gpm	224
Heat load, Btu/hr	1,940,000

Analysis of the EDG LOC performance requires the following measurements: CW inlet temperature; CW outlet temperature; CW flow; LO inlet temperature, and LO outlet temperature. Initially, all of the temperatures were to be measured using insulated, surface-mounted resistance temperature detectors (RTDs). However, due to a high temperature bias in the measurements which was the result of the high viscosity and low thermal conductivity of oil, the surface-mounted RTDs were found to be unacceptable for measuring the LO temperatures. It was therefore necessary to specify dual element RTDs and thermowells to measure LO temperatures. Ultrasonic transducers were specified to measure the CW volumetric flow. The LO volumetric flow was computed from an energy balance. A summary of the required number of measurements points based on the specified instruments and the expected test conditions is provided in Table 2.

Table 2: Nominal Test Conditions

Parameter	Number of measurement points
Entering CCW Temperature, F	4
Exiting CCW Temperature, F	6
CCW Flow, gpm	2
Inlet LO Temperature, F	2
Outlet LO Temperature, F	2

Uncertainty of a measured parameter, U, is defined as an interval, about that measured value that has a preassigned probability of containing the true value.

$$\overline{X} \pm U$$

Therefore, the interval, represents a band about the measured mean, $\overline{X}$, within which the true value is confidently expected to lie.

The LOC analysis was based on 95% confidence and specified more than 30 sample degrees of freedom. The pretest uncertainty analysis was based on the methods and principles specified in Reference 1. Measurement uncertainties are composed of bias (systematic) and precision (random) uncertainty. Bias uncertainty terms include instrument bias, data acquisition system biases, and spacial bias. Precision uncertainty includes process trend and random variations and instrument fluctuations. Instrument and data acquisition system biases are presented in Table 3. Conservative spacial biases based on test experience are presented in Table 4. Projected composite measurement uncertainties are presented in Table 5.

Table 3: Instrumental Biases

Instrument	*Instrument Bias*
RdF and HyCal 4-Wire Platinum RTDs, °F	0.2
Controlotron ultrasonic flow meters, % Flow	2.5

Table 4: Spacial Variation

Parameter	*Expected Condition*
Inlet CCW Temperature, °F	0.25
Outlet CCW Temperature, °F	0.75
CCW flow, % Flow	3.0
Inlet LO Temperature, °F	0.1
Outlet LO Temperature, °F	0.5

Table 5: Composite Measurement Uncertainties

Parameter	*Measurement Precision*	*Measurement Bias*	*Composite Uncertainty*
Inlet CCW Temp., °F	.014	.24	0.24
Outlet CCW Temp., °F	.014	.37	0.37
CCW flow, %	0.5	3.9	4.0
Inlet LO Temp., °F	.014	.21	.21
Outlet LO Temp., °F	.014	.41	.41

Sensitivity coefficients were calculated for the design capacity to changes in fouling resistance and the sensitivity of the fouling resistance to changes in each if the measured parameters. For example, the sensitivity of the design capacity to changes in fouling resistance, θ_{Rf}, is calculated for a small change in fouling, $\Delta f = 0.0002$ (hr-ft^2-°F/Btu), as

$$\theta_{Rf} = (Q_{f1} - Q_{f2})/2\Delta f = (2{,}089{,}941 - 2{,}016{,}123)/2\text{x}0.0002 = 1.845\text{x}10^8 \text{ (Btu/hr)/(hr-ft}^2\text{-°F/Btu)}$$

The product of the two sensitivity coefficients along with the uncertainty for the given measured parameters yielded the contribution of the individual parameters to the overall uncertainty. The overall (or composite) uncertainty in design heat capacity was then computed using the square root of the sum of the squares method. Table 6 presents the sensitivity coefficients associated with a change in the computed fouling factor associated with a small change in each parameter and the resulting uncertainty in the design capacity.

Table 6: Uncertainty on Design Capacity

Parameter	Value	Uncertainty	Sensitivity Coefficient	Change in Heat Transfer (Btu/Hr)
Inlet CCW temperature (°F)	132.4	0.24	$1.46\text{x}10^{-4}$	6,466
Outlet CCW temperature (°F)	144.6	0.37	$6.35\text{x}10^{-4}$	43,359
CCW mass flow rate (lbm/hr)	172,920	6,917	$1.95\text{x}10^{-8}$	24,892
Inlet LO temperature (°F)	167.5	0.21	$2.38\text{x}10^{-5}$	922
Outlet LO temperature(°F)	149.0	0.41	$3.78\text{x}10^{-4}$	28,600
Composite				58,000

Due to the insulating property of oil, the design margins are normally small for oil coolers, unless they are purchased for a higher rating as was in this case. With zero fouling resistance, the anticipated maximum heat transfer rate for the LOC was 2,249,000 (Btu/hr). This value is only 309,000 (Btu/hr) greater than the expected heat transfer rate of 1,940,000 (Btu/hr) and only 137,000 (Btu/hr) greater than the required heat transfer rate of 2,112,000 (Btu/hr). The TDP estimated that an uncertainty of 58,000 (Btu/hr) could be obtained assuming that thermowells were use to measure the LO temperature. Although the projected uncertainty was higher than the 25% of design margin it was less than 25% of the expected margin. Since the projected overall uncertainty of 58,000 (Btu/hr) was developed using conservative values and it was believed that the actual test uncertainties would be less, it was determined to go ahead with the testing.

Analysis of Test Data

The analysis of the results uses a general back calculation method for the shell side convection heat transfer coefficient. The limitations associated with using this analytical method are that the shell side test flows and temperatures must be in the same regime as the design conditions flows and temperatures. The analysis of the test data uses the information from the EDG LOC vendor data sheet to determine the corrected shell-side convection heat transfer coefficient and then projects the heat transfer rate for the heat exchanger at limiting conditions based on measurements collected at test conditions. This method eliminates the need for detailed shell-side heat transfer correlations specific to the geometry and configuration of the heat exchanger. Additional EDG LOC design data and physical properties are as follows:

Hot-side fouling resistance, $R_{f,h}$ =0.001 [hr-ft^2-°F/Btu]
Cold-side fouling resistance, $R_{f,c}$ =0.001 [hr-ft^2-°F/Btu]
Cold-side thermal conductivity, k_c = 0.369 [Btu/(hr-ft-°F)]
Hot-side thermal conductivity, k_h = 0.0724 [Btu/(hr-ft-°F)]
Cold-side absolute viscosity, μ_c = 1.14 [lbm/(ft-hr)]
Hot-side absolute viscosity, μ_h = 81.16 [lbm/(ft-hr)]
Cold-side density, ρ = 61.4 [lbm/ft^3]
Cold-side specific heat, $c_{p,c}$ = 0.997 [Btu/(lbm-°F)]
Hot-side specific heat, $c_{p,h}$ = 0.483 [Btu/(lbm-°F)]
Total effective area of finned surface, A_h = 1,962 [ft^2]
Tube length, L = 8 [ft]
Outside tube diameter, d_o = 0.625 [in]
Tube wall thickness, t = 0.049 [in]
Fin efficiency (assumed), $\eta_{f,h}$ = 0.99
Tube material thermal conductivity, k_t = 8.9 [Btu/(hr-ft-°F)]
Number of tubes carrying flow, N_t = 750
Number of fins per foot (assumed), N_f = 240
Fin thickness (assumed), δ = 0.002 [ft]
No. of tube passes, N_p = 4

Corrected log mean temperature difference, $LMTD_c = 18.77$ [°F]

First Compute the Unknown Geometric Parameters.

Outside tube diameter, $D_o = d_o/12 = 0.0521$ [ft]
Inside tube diameter, $d_i = d_o - 2\,t = 0.527$ [in]
Inside tube diameter, $D_i = d_i/12 = 0.0439$ [ft]
Cold-side area, $A_c = \pi\, D_i\, L\, N_t = 828$ [ft^2]
Ratio of hot side to cold area, $A_h/A_c = 2.37$
Outside tube area, $A_o = (D_o/D_i)\, A_c = 981.8$ [ft^2]
Tube wall area, $A_w = (A_o\text{-}A_i)/\ln A_o/A_i = 902.6$ [ft^2]
Flow area of tube, $a_t = (\pi/4)\, D_i^2 = 0.00151$ [ft^2]
Tube wall Resistance, $R_w = (D_o\text{-}D_i)/2\, k_t = 0.00046$ [(hr ft^2 °F)/Btu]

Then compute the Overall Heat Transfer Coefficient.

Greater terminal temperature difference, $\Delta T_1 = T_{h,i}\text{-}T_{c,o} = 22.9$ [°F]
Lesser terminal temperature difference, $\Delta T_2 = T_{h,o}\text{-}T_{c,i} = 16.2$ [°F]
Log mean temperature difference, $LMTD = (\Delta T_1\text{-}\Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 19.36$ [°F]
Log mean temperature correction factor, $F = LMTD_c$ / $LMTD = 0.9697$ [°F]
Effective mean temperature difference, $EMTD = F\,(LMTD) = 18.77$ [°F]
Overall heat transfer coefficient, $U = Q/(\,A_h\, EMTD) = 57.36$ [Btu/(hr ft^2 °F)]

Next, compute the cold-side convection coefficient based on the Petukhov-Kirilllov correlation.

Cold-side flow rate, $m_c = G_c\, \rho\, (60/7.4805) = 172{,}368$ [lbm/hr]
Hot-side flow rate, $m_h = m_c\, (c_{p,c} / c_{p,h})\, [(T_{c,o} - T_{c,i})/(\,T_{h,i} - T_{h,o})] = 229{,}669$ [lbm/hr]
Mass flow rate per tube, $m_t = (N_p/\, N_t)\, m_c = 919.3$ [Lbm/hr]
Volumetric flow rate per tube, $V_t = m_t / \rho = 14.972$ [ft^3/hr]
Tube velocity, $v_t = V_t / a_t = 9884.11$ [ft/hr]
Cold-side Prandtl number, $Pr_c = c_{p,c}\, \mu_c/\, k_c = 3.1$
Cold-side Reynolds number, $Re_c = \rho\, v_t\, D_i / \mu_c = 23{,}400$
Fanning friction factor, $f = (1.58 \ln Re_c - 3.28)^{-2} = 0.00628$
Nusselt number, $Nu = ((f/2)\, Re_c Pr_c)/(1.07+12.7(f/2)^{1/2}\, (Pr_c^{\,2/3}-1)) = 121.28$
Cold-side convection coefficient, $h_c = Nu\, (k_c / D_i) = 1019$ [Btu/(hr ft^2 °F)]

Then compute the Hot-side surface efficiency, η_h, from the area ratios and the fin efficiency.

Area of the finned side per fin, $a_h = (A_h/A_c)\, \pi\, D_i / N_f = 0.00136$ [ft^2/fin]
Area of the prime per fin, $a_p = \pi\, D_o\, \{(1/\, N_f)\text{-}\, \delta\, \} = 0.00035$ [ft^2/fin]
Area of the fin per fin, $a_f = a_h - a_p = 0.00101$ [ft^2/fin]
Surface efficiency $\eta_h = (a_p + a_f \eta_f\,)/a_h = 0.993$

The Hot-side and Cold-side fouling resistances, $R_{f,h}$ and $R_{f,c}$, are given as 0.001 and 0.001 (hr-ft^2-°F)/Btu, respectively. Therefore, the total fouling resistance referenced to the hot side is,

$$R_f = (R_{f,h} / \eta_h) + (A_h/A_c) R_{f,c} = 0.00338 \ \ [(\text{hr ft}^2 \ {}^\circ\text{F})/\text{Btu}]$$

Next, compute the Hot-side film coefficient, h_h , at "Design Point" conditions. Since,

$$1/U = A/(A_h \eta_h h_h) + (A/A_w) R_w + A/(A_c \eta_c h_c) + R_f$$

If one takes the hot side area, A_h ,as the reference area, (e.g. $A = A_h$) and since $\eta_c = 1$, then,

$$1/U = 1/(\eta_h h_h) + (A_h/A_w) R_w + A_h/(A_c h_c) + R_f$$

And

$$h_h = 1/(\eta_h) / \{1/U - R_f - (A_h/A_w) R_w - A_h/(A_c h_c)\} = 94.1 \ \ [\text{Btu}/(\text{hr ft}^2 \ {}^\circ\text{F})]$$

Results of the test were as follows:

$G_c = 479.78$ [gal/min]
$T_{h,i} = 162.63$ [°F]
$T_{c,i} = 98.16$ [°F]
$T_{h,o} = 118.54$ [°F]
$T_{c,o} = 107.4$ [°F]

From these results, the following values may be determined:

$k_c = 0.364$ [Btu/(hr-ft-°F)]
$k_h = 0.0728$ [Btu/(hr-ft-°F)]
$\mu_c = 1.62$ [lbm/(ft-hr)]
$\mu_h = 123.15$ [lbm/(ft-hr)]
$\rho = 61.96$ [lbm/ft^3]
$m_c = G_c \ \rho \ (60/7.4805) = 238{,}437$ lbm/hr
$c_{p,c} = 0.997$ [Btu/(lbm-°F)]
$c_{p,h} = 0.475$ [Btu/(lbm-°F)]
$m_h = m_c \ (c_{p,c} / c_{p,h}) \ [(T_{c,o} - T_{c,i})/(T_{h,i} - T_{h,o})] = 104{,}884$ [lbm/hr]
$\Delta T_1 = T_{h,i} - T_{c,o} = 55.23$ [°F]
$\Delta T_2 = T_{h,o} - T_{c,i} = 20.38$ [°F]
$LMTD = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 34.96$ [°F]
$R = (m_h \ c_{p,h})/(m_c / c_{p,c}) = 0.21$
$P = (T_{h,i} - T_{h,o})/(T_{h,i} - T_{c,i}) = 0.68$

where R is the capacitance ratio and P is the heat exchanger effectiveness. From Reference 2 for a 2-shell, 4 tube pass heat exchanger, F = 0.985.

$EMTD = F\,(LMTD) = 34.43$ [°F]
$Q_{test} = m_c\, c_{p,c}\,(T_{c,o} - T_{c,i}) = 2{,}196{,}500$ [Btu/hr]
$U = Q/(A_h\, EMTD) = 32.5$ [Btu/(hr ft^2 °F)]
$m_t = (N_p/N_t)\, m_c = 1{,}272$ [Lbm/hr]
$V_t = m_t / \rho = 20.52$ [ft^3/hr]
$v_t = V_t / a_t = 13{,}549$ [ft/hr]
$Pr_c = c_{p,c}\, \mu_c / k_c = 4.4$
$Re_c = \rho\, v_t\, D_i / \mu_c = 22{,}800$
$f = (1.58 \ln Re_c - 3.28)^{-2} = 0.00633$
$Nu = ((f/2)\, Re_c Pr_c)/(1.07 + 12.7(f/2)^{1/2}(Pr_c^{2/3} - 1)) = 139.78$
$h_c = Nu\,(k_c / D_i) = 1159$ [Btu/(hr ft^2 °F)]

One must calculate a new value for h_h based on the test conditions on the hot side, since Re and Pr for the hot side will be different than was the case for the "Design Point" conditions. However, one is not required to calculate the absolute values of Re and Pr on the hot side. The value for h_h based on the test conditions on the hot side may be computed based on a correlation by Taborek (Reference 3). If one assumes that the actual value for h_h is a function of some ideal value for the Hot-side thermal conductivity for pure cross flow such that

$$h_h = J_T\, h_{h,ideal}$$

where J_T is a correction factor that is a function of the various leakage and bypass paths and, therefore, remain constant for a given heat exchanger. Therefore,

$$J_T = \left(\frac{h_h}{h_{h,ideal}}\right)_{design} = \left(\frac{h_h}{h_{h,ideal}}\right)_{test}$$

and

$$(h_h)_{test} = \left(\frac{h_{h,test}}{h_{h,design}}\right)_{ideal} h_{h,design}$$

From the Zukauskas correlation (Reference 3),

$$h_h = C\, \mathrm{Re}^m\, \mathrm{Pr}^n \left(\frac{k}{D}\right)$$

For laminar flow across a tube bank where Re_h is less than 500, m=0.4 and n=0.36 (Reference 3). Substituting, C and all geometric parameters cancel out, since they do not change between test and reference conditions.

$$(h_h)_{test} = \frac{[(m_h/\mu_h)^{0.4} (\mu_h c_{p,h}/k_h)^{.36} k_h]_{test}}{[(m_h/\mu_h)^{0.4} (\mu_h c_{p.h}/k_h)^{.36} k_h]_{design}} h_{h,design} = 67.5\,[Btu/hr - ft^2 - {}^{o}F]$$

Therefore, the total apparent fouling resistance as determined by the test may be computed as,

$$R_f = 1/U - 1/(\eta_h h_h) - (A_h/A_w) R_w - A_h/(A_c h_c) = 0.01278 \ [(hr\ ft^2\ {}^oF)/Btu].$$

By assuming that the Hot-side fouling is constant, at 0.001 [(hr ft^2 °F)/Btu] one may imply a Cold-side fouling resistance referenced to the cold side area.

$$R_{f,c} = (A_c/A_h)(R_f - R_{f,h}/\eta_h) = 0.00497 \ [(hr\ ft^2\ {}^oF)/Btu]$$

One should note that this value represents an apparent fouling which also reflects any other deficiencies in the heat exchanger and may be misleading.

In this case, the limiting conditions are the same as the design point conditions. Therefore, the values for fluid properties are the same as those for the design point. The heat transfer at limiting conditions is a function of the heat transfer at test conditions and the following three correction terms (assuming that the change in fin efficiency and tube material thermal conductivity are negligible):

$$h_h' = \frac{1}{\eta_h}\left(\frac{1}{h_h*} - \frac{1}{h_h}\right)$$

$$E' = \frac{EMTD\ *}{EMTD}$$

$$h_c' = \frac{1}{h_c*} - \frac{1}{h_c}$$

So that the heat transfer at limiting conditions, Q*, is

$$Q* = \frac{Q\ E'}{1 + \left(\frac{Q}{EMTD}\right)\left(\frac{h_h'}{A_h} + \frac{h_c'}{A_c}\right)}$$

If the limiting condition is other than the "Design Point", then the physical properties and flow rates must be used to calculate the appropriate values for Re, Pr, h_i and h_o for that particular condition. The final solution is iterative because outlet temperatures and the EMTD are a function of Q*. Therefore, by iteration the heat transfer at limiting conditions is

$$Q^* = 1{,}630{,}00 \text{ [Btu/hr]}$$

and the outlet temperatures are

$$T_{(h,o)} = 152.9 \text{ [°F]}$$
$$T_{(h,c)} = 141.8 \text{ [°F]}$$

Discussion of Results

The test was conducted at a cold-side inlet temperature that was considerably less than the design value, and with a cold-side flow rate that was somewhat greater than design. As a result, the hot-side outlet temperature was considerably below the design value, and the required hot-side flow rate was less than half of the design value. Even though the amount of heat that was transferred during the test was very close to the design value, the test EMTD was considerably greater than the design value, and the U_{test} was considerably less than the design value. The projection of heat transfer from test conditions to limiting conditions, Q*, is a function of the measured heat transfer at test conditions, EMTD Correction, E', the hot side convection coefficient correction, h_h', and the cold side convection coefficient correction, h_c'. Although there is a significant difference between the test and reference hot-side flow rates, this difference is offset somewhat by the difference in the test and reference hot-side viscosities such that the difference between the test and reference Reynolds numbers is fairly small. As a result, the hot side convection coefficient correction is small, and the EMTD correction predominates. The corrections are as follows:

$$E' = 0.6474$$
$$h_h' = -0.00423$$
$$h_c' = 0.00012$$

The total fouling resistance lumped on the hot side is calculated from the test data to be 0.01278 [(hr ft^2 °F)/Btu] compared with the design value of 0.00353 [(hr ft^2 °F)/Btu]. The projected heat transfer at limiting conditions is 1,630,000 Btu/ hr compared with the test acceptance criteria of 1,935,000 Btu/hr. The reason for these discrepancies may include excessive fouling on the hot or cold side, some other heat exchanger performance deficiency. Stainless steel tubing is notorious for fouling in applications with low tube velocity and relatively high temperatures (Reference 4). The test of the EDG LOC implies that the heat exchanger would not meet the acceptance criteria of transferring 1,935,000 Btu/hr at limiting conditions.

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