Optimization of Finned Tube Oil Cooler by Effective Utilization of Area within the Space Available

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Abstract

Hydro-electric turbines operating all over the world generate enormous amount of heat during their operation due to the friction between turbine shaft and guide bearings. Oil and grease have long been considered as one of the primary methods to lubricate and transfer the heat from oil to water. This thesis work outlines the optimization of finned tube oil cooler for turbine guide bearing application by effective utilization of area so as to extract and deliver more heat as compared to the existing system. The quest is to enhance the cooling capacity of the finned tube oil cooler by marginally increasing the size within the space available retaining the compactness. This complete optimization and development provides the same operational reliability without any risk of accidental or operational damage to the environment.

Keywords: Finned Tube Oil Cooler, Shell and Tube Heat Exchanger, Heat Transfer Enhancement, Optimization

I. INTRODUCTION

A finned tube oil cooler is basically a radiator with oil running through it instead of water, to lubricate the turbine guide bearings and transferring the heat generated in the turbine. They are typical shell-and-tube heat exchanger (STHEs) with an oil chamber being the shell and fins are attached to the water tubes to increase the heat transfer rate. The finned tubes arranged in an in-line manner. Fins are used to increase the effective surface area of heat exchanger tubing. These oil coolers are also called plug-in type because they are plugged into an oil chamber, filled with hydraulic oil to transfer the heat generated in the guide bearing. Oil chamber contains the oil for lubricating as well as for cooling purpose. Generally, it is cylindrical in shape with a circular cross section, although shells of different shape are used in specific applications.

Finned tube oil cooler plays an important role in keeping a well running turbine continuous even when pushed to the limit. Maintaining a proper oil temperature is important to keep the oil in its proper lubricating state and to aid the radiator in keeping the equipment cool. Most of the oil coolers used in industrial areas are of finned tube oil coolers due to their robust geometry construction, easy maintenance and possible upgrades. Also, it suits high pressure application and these coolers can be assembled or replaced without much trouble. These oil coolers are designed to meet the specific duty condition, temperature and pressure of the fluids.

Most of the oil coolers used hydro-turbines are of finned tube oil coolers due to their robust geometry construction, easy maintenance and possible upgrades. Also, it suits high pressure application and these coolers can be assembled or replaced without much trouble. By providing the fins, the transverse velocity of fluid that impinging on tube vertically is augmented greatly which enhances the turbulence and augments the heat transfer on the tube surfaces.

Purposes of lubricating oil include reducing rolling and sliding friction in guide bearings, providing a seal to prevent mass transport from inboard (process) environment to the outboard environment and removing unwanted heat from the guide bearing zone. The suitability of lubricating oil to perform its tasks depends on its thermos physical properties. In practice, temperature control can be challenging. Sufficient heat must be removed so that the returned oil is within a desired temperature range.

The AutoCAD drawing of arrangement of finned tube oil cooler and a model of the finned tube oil cooler used in guide bearing application of a Francis turbine is shown below in Fig.1 and Fig.2, respectively.
The energy balance equations for the lubricating oil and the coolant is given as:

\[ Q = m_h c_{ph} (t_{h1} - t_{h2}) = m_c c_{pc} (t_{c2} - t_{c1}) \]

Lubricating oil systems incorporate the oil coolers for heat rejection and temperature control. Heat is extracted from the lubricating oil through the heat exchanger wall. The heat extraction rate is a function of the heat transfer surface area, the resistance to heat transfer, and the logarithmic mean temperature difference. The generally recognized thermal transport relation is:

\[ Q = UA\Delta T \]

Where \( \Delta T = \text{LMTD} \)
II. CASE HISTORY

A The standard size of oil cooler used in this project is diameter 200mm × 320mm long with an oil tank of diameter 267 mm. The aluminum fins are attached to the Cu-Ni (90/10) water tubes of 19 mm OD and 1 mm thickness. The oil cooler has heat dissipation capacity of 3.7 kW. AutoCAD drawing of this oil cooler is shown below in fig.3.

For a number of reasons, the return oil temperature during periods of high ambient temperature conditions approaches the alarm setting, causing shutdown of the power plant. It is supposed to eliminate this threat to the turbine guide bearings and potential reduced output.

The solution obtained is that the modification should be done on the oil coolers so as to increase its heat dissipation capacity and to reduce the plant down time by optimizing and modifying the oil cooler within the limitation of space available without any risk of accidental or operational oil discharge to the environment.

Fig. 3: AutoCAD Drawing of Standard Oil Cooler (Hydro-electric Francis Turbine).

III. METHODOLOGY

A. Comparative study of different types of coolers

Many different types of coolers have been used for guide bearing oil cooling. The choice of a cooler depends on the heat needed to be dissipated for a specific speed and size of turbine. Enough velocity & pressure head needs to be generated for hot oil flow to take place in case of external coolers. This disadvantage can be overcome by placing the cooler right inside the oil. These plug-in type internal oil coolers which can be assembled or replaced without much trouble.

B. Approach Followed

The standard of the Tubular Exchanger Manufacturers Association (TEMA) describes various components of STHE in detail. The cooler is required to be compact. Therefore, we define the dimension vis-a-vis the cooling capacity of the cooler. Design work is followed by drawing preparation and modelling of the oil cooler.

C. Design development of a new compact cooler

The new cooler under design must be capable of dissipating heat more than the existing oil cooler. Our quest is to enhance the cooling capacity of the plug-in type oil cooler by marginally increasing tube lengths and diameter of the fins, thereby increasing the cooling capacity, while essentially retaining the compactness. The recommended standards (IS: 4503 or TEMA) are followed. The final drawing of the cooler is prepared in AutoCAD software which is shown in fig.4.
D. Design and Operating Parameters

The heat generated in turbine guide bearing is transferred from hot oil to water through a number of fins and water tubes. The design parameters for these fins and water tubes are shown below in Table 1. The oil used for lubrication and heat dissipation purpose is ISO VG 46 hydraulic oil and water is used for cooling purpose. The properties of the hot and cold fluids used in the oil cooler are given below in Table 2. Other descriptions and boundary conditions for the oil coolers are show below in Table 3.

![AutoCAD Drawings of Modified Oil Cooler of a Francis Turbine Guide Bearing](image)

**Table 1**

<table>
<thead>
<tr>
<th>Description</th>
<th>Standard Oil Cooler</th>
<th>Modified Oil Cooler</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer Diameter, D</td>
<td>132 mm</td>
<td>150 mm</td>
</tr>
<tr>
<td>Quantity, in numbers</td>
<td>150</td>
<td>224</td>
</tr>
<tr>
<td>Thickness, t</td>
<td>0.15 mm</td>
<td>0.15 mm</td>
</tr>
<tr>
<td>Length, L</td>
<td>(0.15 × 150)</td>
<td>(0.15 × 224)</td>
</tr>
<tr>
<td>Material</td>
<td>Aluminum</td>
<td>Aluminum</td>
</tr>
<tr>
<td></td>
<td>Cupro-Nickel (Cu-Ni 90/10)</td>
<td>Cupro-Nickel (Cu-Ni 90/10)</td>
</tr>
<tr>
<td>Surface Temperature</td>
<td>25°C</td>
<td>25°C</td>
</tr>
<tr>
<td>Thermal Conductivity, k</td>
<td>205 W/mK</td>
<td>205 W/mK</td>
</tr>
</tbody>
</table>

**Table 2**

<table>
<thead>
<tr>
<th>Description</th>
<th>ISO VG 46 Hydraulic Oil</th>
<th>Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dynamic Viscosity, µ</td>
<td>4.048 × 10⁻² N·s/m²</td>
<td>8.9 × 10⁻⁴ N·s/m²</td>
</tr>
<tr>
<td>Kinematic Viscosity, ν</td>
<td>4.6 × 10⁻⁵ m²/s</td>
<td>8.9 × 10⁻⁷ m²/s</td>
</tr>
<tr>
<td>Density, ρ</td>
<td>880 kg/m³</td>
<td>1000 kg/m³</td>
</tr>
<tr>
<td>Specific Heat, cp</td>
<td>1.760 kJ/kgK</td>
<td>4.186 kJ/kgK</td>
</tr>
<tr>
<td>Thermal conductivity, k</td>
<td>0.134 W/mK</td>
<td>0.606 W/mK</td>
</tr>
<tr>
<td>Flow</td>
<td>18 LPM or 0.264 kg/s</td>
<td>30 LPM or 0.50 kg/s</td>
</tr>
<tr>
<td>Velocity, u</td>
<td>2 m/s</td>
<td>-</td>
</tr>
<tr>
<td>Coefficient of Volumetric Expansion, β</td>
<td>0.007 K⁻¹</td>
<td>0.003 K⁻¹</td>
</tr>
</tbody>
</table>

**Table 3**

<table>
<thead>
<tr>
<th>Description</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall Cooler Size</td>
<td>Standard Oil Cooler Diameter 200mm × 320mm Long</td>
</tr>
<tr>
<td>Diameter of the Central Hole</td>
<td>40 mm</td>
</tr>
<tr>
<td>Pitching of Fins</td>
<td>745±2 Fins per meter</td>
</tr>
<tr>
<td>Maximum Bearing Temperature</td>
<td>75°C</td>
</tr>
<tr>
<td>Oil Inlet Temperature, t₀</td>
<td>50°C</td>
</tr>
<tr>
<td>Water Inlet Temperature, t₂₄</td>
<td>25°C</td>
</tr>
<tr>
<td>Maximum Change in Cooling Water Temperature</td>
<td>4°C</td>
</tr>
</tbody>
</table>

IV. CALCULATIONS

On the basis of given design parameters and boundary conditions, following calculations are carried out:

A. Surface Area of Heat Transfer

Surface area of heat transfer for standard oil cooler is given by,

\[ A_s = \text{Heat transfer area of the aluminium fins} + \text{Heat transfer area of the water tubes} \]
A_f = 3.057 \, m^2 + 0.085 \, m^2 = 3.14 \, m^2

And, Surface area of heat transfer for modified oil cooler is given by.

\[ A_{m} = \text{Heat transfer area of the fins + Heat transfer area of the water tubes} \]

\[ A_{m} = 6.353 \, m^2 + 0.127 \, m^2 = 6.48 \, m^2 \]

**B. Characteristic Length**

For complex figures, characteristic length is given by

\[ L_C = \frac{\text{Volume of the standard oil cooler}}{\text{Surface area of the standard oil cooler}} \]

Volume of the standard oil cooler = \( \frac{\pi}{4} D_s^2 L_s = 2.74 \, m^2 \)

Volume of the modified oil cooler = \( \frac{\pi}{4} D_m^2 L_m = 5.30 \, m^2 \)

So, the characteristic length of standard oil cooler, \( L_{cs} = 0.872 \, m \)

and, the characteristic length of modified oil cooler, \( L_{cm} = 0.817 \, m \)

**C. Outlet Temperature of Hot Fluid**

Since the maximum change in cooling water temperature is 4°C. So, calculating the outlet temperature of hot fluid with respect to 20°C rise in cooling water temperature. The energy balance equation (from eq. 1) is given by,

\[ Q = m_c c_p (t_{c2} - t_{c1}) = 0.264 \times 1.76 \times (50 - t_{h2}) = 0.50 \times 4.186 \times (30 - 32) \]

\[ t_{h2} = 41^\circ C \]

**D. LMTD (Logarithmic Mean Temperature Difference)**

LMTD of Standard Oil Cooler,

\[ \text{LMTD} = \frac{(t_{h1} - t_{c2}) - (t_{h2} - t_{c1})}{\ln \left( \frac{t_{h1} - t_{c2}}{t_{h2} - t_{c1}} \right)} \]

\[ \Delta T_s = \frac{(50 - 32) - (41 - 30)}{\ln \left( \frac{50 - 32}{41 - 30} \right)} \]

\[ \Delta T_s = 14.214^\circ C \]

**E. Overall Heat Transfer Coefficient**

For Standard oil cooler,

Convective heat transfer coefficient for oil side (h_o) is calculated as:

Grashof number, \( Gr = \frac{g \beta_h (T_s - T_w) L_c^3}{\nu_h^2} \)

\[ Gr = \frac{9.81 \times 0.007 \times (50 - 25) \times 0.872^3}{0.000046^2} \]

\[ Gr = 5.38 \times 10^8 \]

Prandtl Number, \( Pr = \frac{\mu_h c_{ph}}{k_h} \)

\[ Pr = \frac{0.04048 \times 1760}{0.134} \]

\[ Pr = 531.68 \]

Rayleigh Number, \( Ra = Gr \times Pr \)

\[ Ra = (5.36 \times 10^8) \times 531.68 \]

\[ Ra = 2.86 \times 10^{11} \]

\( Ra \) = For \( 10^7 \leq Ra \leq 10^{12} \) (turbulent flow)

Nusselt Number, \( Nu = 0.15 \, Ra^{1/3} \)

\[ Nu = 988.30 \]

So, \( h_i = 151.90 \, W/m^2K \)

Now, convective heat transfer coefficient for water side (h_o) is calculated as:
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\[ Gr = \frac{g \beta_c (T_s - T_\infty) L_{cs}^3}{v_h^2} \]

\[ Gr = \frac{9.81 \times 0.003 \times (50 - 25) \times 0.872^3}{0.00000089^2} \]

\[ Gr = 6.16 \times 10^{11} \]

\[ Pr = \frac{\mu c_p c}{k_c} \]

\[ Pr = \frac{0.00089 \times 4186}{0.6065} \]

\[ Pr = 6.145 \]

\[ Ra = Gr \times Pr \]

\[ Ra = 6.14 \times 10^{11} \times 6.145 \]

\[ Ra = 3.78 \times 10^{12} \]

\[ Nu = 0.15 \frac{Ra}{3} = 2337.57 \]

But

\[ Nu = \frac{h_{oi} L_{cs}}{k_c} \]

So, \( h_o = 1624.50 \text{ W/m}^2\text{K} \)

Now, the overall heat transfer coefficient for standard oil cooler is given by,

\[ U_s = 1 \frac{L_c}{h_i + L_c k_c} + \frac{1}{h_o} = 87.32 \text{ W/m}^2 \]

For Modified oil cooler,

Convective heat transfer coefficient for oil side (h_i) is calculated as:

\[ Gr = \frac{g \beta_c (T_s - T_\infty) L_{cm}^3}{v_h^2} \]

\[ Gr = \frac{9.81 \times 0.007 \times (50 - 25) \times 0.817^3}{0.000046^2} \]

\[ Gr = 4.42 \times 10^{8} \]

\[ Pr = \frac{\mu c_p c}{k_h} \]

\[ Pr = \frac{0.04048 \times 1760}{0.134} \]

\[ Pr = 531.68 \]

\[ Ra = Gr \times Pr \]

\[ Ra = (4.42 \times 10^{8}) \times 531.68 = 2.35 \times 10^{14} \]

For \( 10^{7} \leq Ra \leq 10^{12} \) (turbulent flow)

\[ Nu = 0.15 \frac{Ra}{3} = 925.962 \]

Also, \( Nu = \frac{h_{im} L_{cm}}{k_h} \)

So, \( h_{im} = 151.88 \text{ W/m}^2\text{K} \)

Now, the convective heat transfer coefficient for water side (h_o), is calculated as:

\[ Gr = \frac{g \beta_c (T_s - T_\infty) L_{cm}^3}{v_h^2} \]

\[ Gr = \frac{9.81 \times 0.003 \times (50 - 25) \times 0.817^3}{0.00000089^2} \]

\[ Gr = 5.06 \times 10^{11} \]

\[ Pr = \frac{\mu c_p c}{k_c} \]

\[ Pr = \frac{0.00089 \times 4186}{0.6065} \]

\[ Pr = 6.14 \]

\[ Ra = Gr \times Pr \]

\[ Ra = 5.06 \times 10^{11} \times 6.14 = 3.11 \times 10^{12} \]

For \( 10^{7} \leq Ra \leq 10^{12} \) (turbulent flow)

\[ Nu = 0.15 \frac{Ra}{3} = 151.88 \text{ W/m}^2\text{K} \]
Nu = 2189.86

But, \( Nu = \frac{h_{om}L_{cm}}{k_c} \)

So, \( h_{om} = 1625.37 \text{ W/m}^2\text{K} \)

Now, the overall heat transfer coefficient is given by,

\[
U_m = \frac{1}{\frac{1}{h_{im}} + \frac{L_{cm}}{k_{Al}} + \frac{1}{h_{om}}}
\]

\[ U_m = 89.41 \text{ W/m}^2\text{K} \]

**F. Number of Transfer Units**

Number of transfer units is given by,

\[ NTU = \frac{UA}{(mc_p)_{small}} \]

Since, \( m_h c_{ph} = 0.264 \times 1.76 = 0.46464 \)

and \( m_c c_{pc} = 0.50 \times 4.186 = 2.093 \)

Therefore, the number of transfer units for standard and modified cooler will be,

\[
NTU_s = \frac{87.32 \times 3.142}{0.46464} = 590.48
\]

\[
NTU_m = \frac{89.41 \times 6.48}{0.46464} = 1246.94
\]

**V. Result Analysis and Discussion**

The heat transfer rate from the oil cooler is the function of overall heat transfer coefficient, surface area and logarithmic mean temperature difference. The generally recognized thermal transport relation for heat transfer is given by,

For standard oil cooler,

\[
Q_s = U_s A_s \Delta T_s = 87.32 \times 3.142 \times 14.214 = 3899.74 \text{ W} \approx 3.9 \text{ kW}
\]

For the modified oil cooler,

\[
Q_m = Q_s \times \frac{NTU_m}{NTU_s} = 3899.74 \times \frac{1246.94}{590.48} = 8235.23 \text{ W} \approx 8.2 \text{ kW}
\]

**VI. Conclusion**

It is seen from the test results that the heat dissipation capacity of the modified oil cooler is more than the capacity of standard oil cooler at design condition. The calculations are carried out at 30 LPM cooling water flow rate. At the sites of hydro power projects, cooling water flow rate is likely to be higher than 40 LPM, which implies that the heat dissipation capacity of the modified oil cooler is likely to be even more.

The heat transfer rate of optimized oil cooler is also compared with the existing oil cooler. It is found that the heat transfer rate is less in existing oil cooler due to ineffective utilization of heat transfer area. Thus the design can be modified for better heat transfer by increasing the area of heat transfer within the space available.

**Acknowledgments**

This research did not receive any specific grant from funding agencies in the public, commercial, or not-for-profit sectors. However, the research presented here is carried out at Bharat Heavy Electricals Limited, Bhopal, a Maharashtra PSU owned by Govt. of India, for academic purpose only. M/S Patel Heat Exchangers Pvt. Ltd., Bhopal, Madhya Pradesh, helped in the modeling, meshing and carrying out the thermal analysis on the prepared model of modified finned tube oil cooler.

**References**


