THERMAL TEST ANALYSIS OF BEARING FRAME INLINE PUMPS ON HIGH TEMPERATURE SERVICE

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ABSTRACT

Testing of an API Standard 610 [1] bearing frame inline pump at pumpage temperatures up to 650°F is presented. The thermal response throughout the pump, and specifically in the bearing frame area, is quantified. Steady state bearing operating temperatures are measured for various lubrication options and cooling configurations.

The experimental results are used to correlate an analytical thermal model previously developed to analyze response of an API 610 horizontal overhung process pump. The predictive capability of the model for bearing frame inline pumps is described.

Conclusions are developed regarding lubrication and cooling requirements for bearing frame inline pumps to meet the operating temperatures required in refinery processing units.

INTRODUCTION

Background of Inline Pump Designs

Inline pumps have been applied in refinery services for many years, both in process applications and offsite. API inlines have typically been available in two designs—a close-coupled version (Figure 1) and a rigid-coupled version (Figure 2). Much application experience has been developed over the years with these designs, and preferences have developed based on mechanical reliability under various process conditions. While a number of users experienced success applying these designs, other users imposed strict application limits based on their experience, limiting use to noncritical offsite and utility services such as tank farms and transfer.

The advantages of an inline pump design are becoming more significant today as debottlenecking and capacity expansion projects are developed. Available space and cost are always issues, and inline pumps are potentially capable of easing these constraints. Specifically, plot size requirements for inline pumps are smaller than horizontal pumps and piping configurations can be simplified for inlines if space is limited. Installed cost is typically lower for an inline design since baseplate leveling and grouting procedures are not required and cold/hot shaft alignment procedures are not necessary.

However, the advantages of inline pumps have not outweighed the mechanical reliability issues with many users. Application on demanding high temperature process services is a key issue with both designs. Close-coupled designs quickly developed a reputation for reliability problems due to motor bearing and mechanical seal failures, particularly on hot process applications. Shaft conducted heat reduced critical motor bearing clearances creating
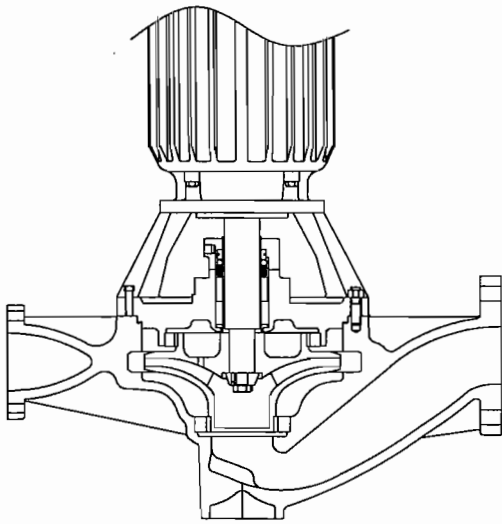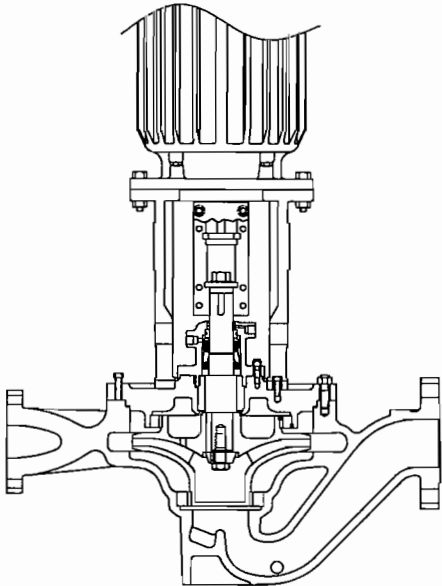
Figure 1. Close-Coupled Inline Pump.

Figure 2. Rigid-Coupled Inline Pump.

abnormally high bearing loading along with premature deterioration of the lubricant. Rigid coupled designs better isolate the hot process temperature from the driver bearings, but mechanical reliability suffers due to worn clearances within the hydrodynamic bearing at the impeller and deterioration of the precision fits required in the rigid spacer coupling. These problems impact mechanical seal and motor bearing reliability and resulting MTBF has been less than satisfactory for some users.

Bearing Frame Inline Designs

The bearing frame inline design shown in Figure 3 was developed in response to these reliability issues. Proper design incorporates the mechanical integrity of an API horizontal overhung process pump with the space and installation cost savings of an inline. Shaft stiffness and predictable rotordynamics assure seal life equal to horizontals. The bearing frame takes all the hydraulic loads, eliminating substantial driver bearing loading, and controlling vibration. The application of the inline bearing frame design
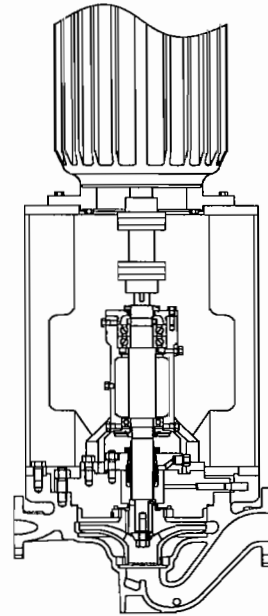

Figure 3. Bearing Frame Inline Pump.

in typical process applications alongside horizontal designs is justifiable.

But how does the bearing frame inline design respond to hot process applications, a key issue with previous inline designs? Inline pumps normally cannot have cooling applied to the bearing oil sump to address higher anticipated bearing operating temperature, as can a horizontal. Additionally, ambient temperatures in the immediate vicinity of the bearings may be higher than a horizontal due to convection within the motor support from the hot casing. ANSI bearing frame inline pumps, designed to ASME B73.2 [2], have proven themselves for 20 years in chemical processing applications. However, API services demand stretching the proven technology envelope with higher pressure and temperature requirements (Figure 4).
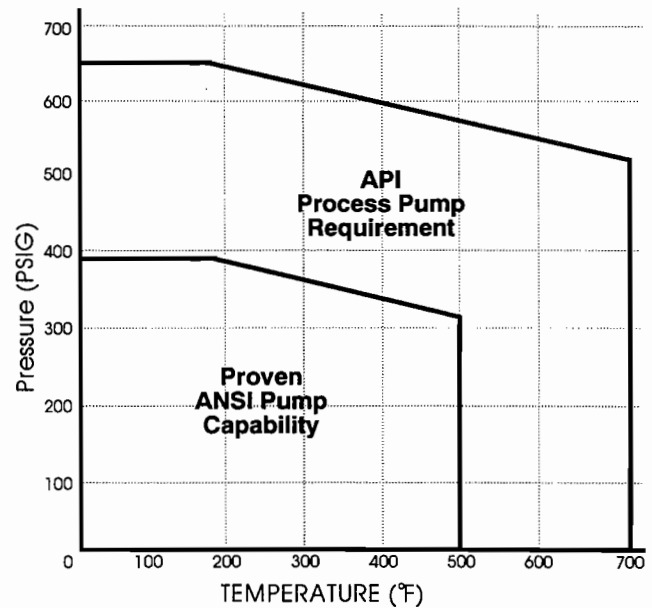

Figure 4. Pressure-Temperature Envelope Comparison.

API bearing design must be more rigorous to handle the loads, and lubrication and cooling designs must maintain bearings within acceptable operating limits at the elevated pumpage temperatures normally encountered in hydrocarbon processing applications.

The Need for a Thermal Model

Due to these unique design constraints, it becomes necessary to thoroughly evaluate lubrication designs and the impact of cooling devices on bearing operating temperatures. An analytical model which will closely predict thermal response based on pumpage, ambient, cooling and flush temperatures is required to assure that the necessary cooling devices are applied for the specified lubrication method. High temperature testing is required to verify design assumptions and validate the analytical model.

Lubrication Designs and Cooling Devices

Grease lubrication is the traditional method of lubricating vertically mounted antifriction bearings (Figure 5), having been successfully applied in a vast majority of low to intermediate temperature services. As process temperatures rise over 250°F, however, special consideration must be given to limiting conductive heat transfer through the shaft and bearing frame structure, along with convective heat transfer from the casing to the atmosphere adjacent to the bearing frame.
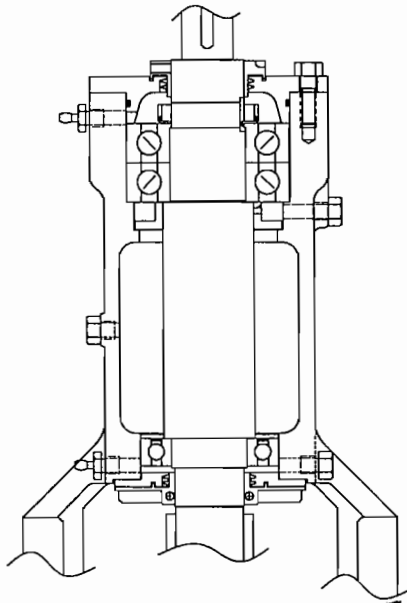

Figure 6. Oil Sump Lubricated Bearing Frame.

Figure 5. Grease Lubricated Bearing Frame.

Figure 7. Pure Oil Mist Lubricated Bearing Frame.

Oil sump lubrication, shown in Figure 6, in various configurations, is currently being supplied on bearing frame inline pumps. Field experience is limited in some cases and MTBF will ultimately differentiate the better designs. Bearing operating temperatures are highly dependent on design of the oil sump, pumping device, oil delivery system, and cooling mechanisms. A properly engineered unit will achieve bearing operating temperatures comparable to a ring-oiled horizontal pump and be equally as reliable.

Pure oil mist lubrication (Figure 7) is the traditional lubrication method for elevated process temperature service due to the cool, aerated, controlled misting of the bearing operating environment. It is particularly important at these higher pumpage temperatures to control heat conduction through the shaft and bearing frame to avoid thermally induced bearing loads.
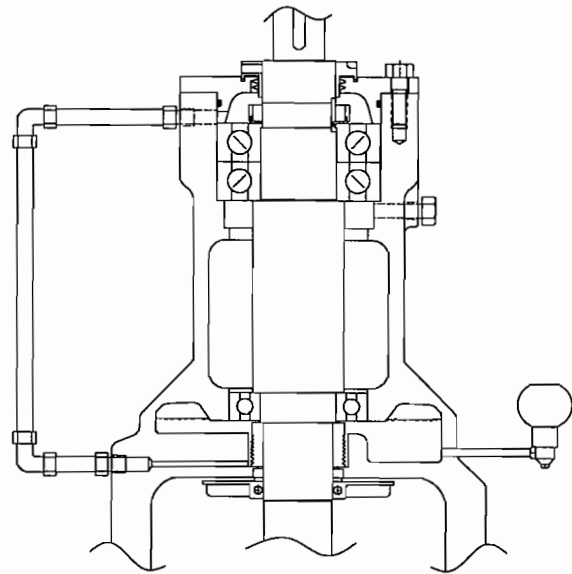
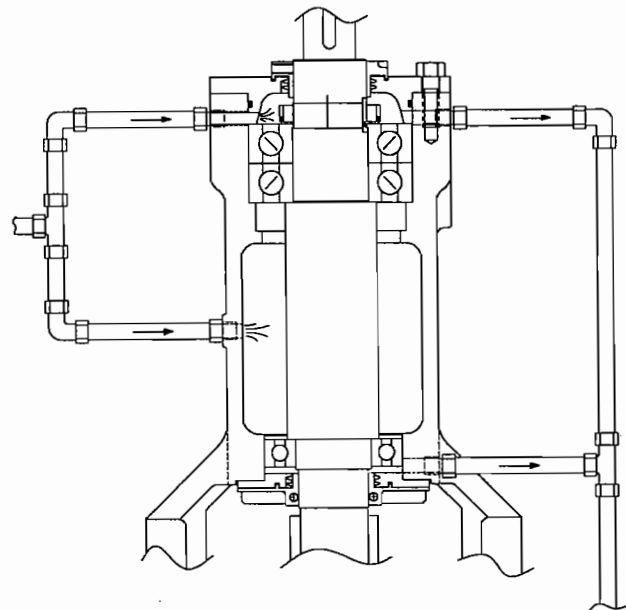Cooling devices are applied to bearing frame inline pumps to control heat flow to the bearings at elevated pumping temperatures. The application of these devices is dependent on service conditions such as pumpage temperature, seal flush system design, ambient temperature, and bearing loading. All impact how devices are applied since balancing the heat flow through both the shaft and frame is significant in assuring proper bearing operating temperatures. Devices that can be effectively applied to bearing frame inlines are shown in Figure 8 and include a shaft mounted heat flinger, an outboard fan, and jacket cooling of the seal chamber cover.

EXPERIMENTAL TESTING

Various lubrication designs and cooling devices were tested to verify thermal response. Temperature was measured at various
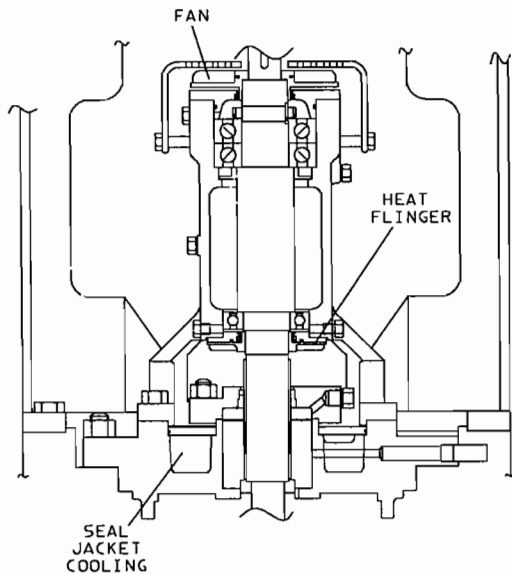
Figure 8. Typical Cooling Devices.

Figure 9. High Temperature Test Facility.

points on the test unit and observations developed. Many factors impact heat generation and heat flow that are dependent on design and conditions of service. Such factors include bearing design and loading, lubricant characteristics, mechanical seal flushing systems, ambient temperature, air flow conditions, cooling jacket design, cooling water characteristics, bearing mounting design, and lubrication system design. The experimental testing presented here represents temperature response of a specific design and set of test conditions. A bearing operating temperature range is shown on the data plots to address the variability related to these factors. The data is applicable to bearing frame inline designs operating at 3600 rpm, 60°F to 100°F ambient temperatures and power levels to 200 hp. This testing does not apply to close-coupled or rigid-coupled inline designs. Each has unique design characteristics which impact thermal response. However, a large majority of pumps of these designs are supplied with grease lubricated motor bearings and general conclusions regarding operating limits may be inferred based on analysis of the particular design and reference to the grease lubricated bearing frame test results from this program.

The data developed during this experimental testing was used to validate the prediction model as described in the following section.

High Temperature Test Facility

A high temperature test facility (Figure 9) was developed to provide test capability at pumpage temperatures up to 650°F. Due to the hazards involved, the facility was designed to properly safeguard personnel in the event of system upset or failure of a sealing component. The test facility is totally enclosed with environmental control, remote process control and remote data acquisition. Pumpage used in the facility is a heat transfer fluid designed to operate at 200°F to 700°F. The closed loop test stand incorporates flow, pressure, and temperature control and monitoring.

Test Pump

The pump utilized for testing was a size 2×3–13 (N_s = 600) API 610 bearing frame inline with a 150 hp C-face vertical motor operating at 3600 rpm. This design generates heads to 700 ft and flows to 400 gpm. The pump bearings are the same as those applied on the horizontal overhung pump—7312 duplex angular contact thrust and 212 deep groove ball radial. An API Plan 11 system was used to flush the mechanical seal.
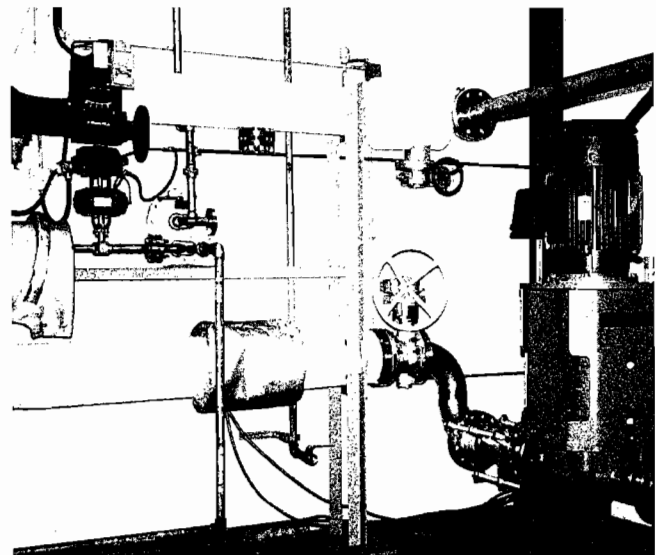
Instrumentation

Data acquisition was accomplished with thermocouples for temperature measurement on the pump and thermowell mounted thermocouples for the pumpage. The location of the thermocouple mounting is shown in Figure 10. All data were simultaneously sampled and plotted from pump startup through stabilization of all temperatures for each pumping condition.
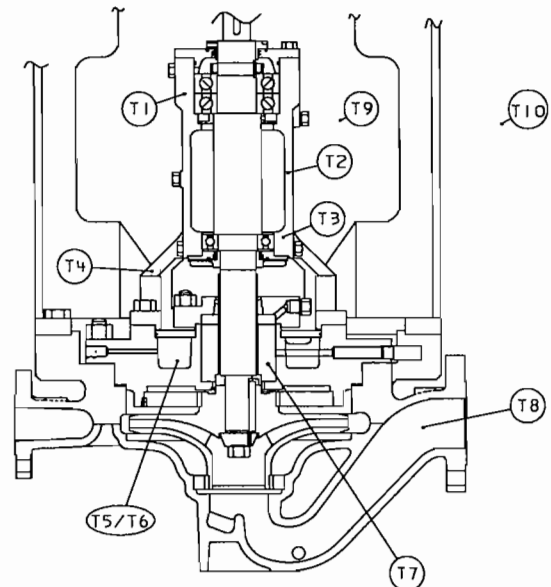

Figure 10. Thermocouple Locations Used for Experimental Testing.

TEST RESULTS

Baseline Thermal Profile

As a baseline, a grease lubricated configuration was tested at pumpage temperatures of 170°F and 350°F to establish profiles of steady state temperature throughout the bearing frame structure.

The maps shown in Figure 11 and Figure 12 define relative temperature distribution for the unit with no cooling supplied to the seal chamber jacket. Note that at the 170°F pumpage temperature, convection cools the lower half of the bearing frame as expected, but the heat generated by the grease lubricated thrust bearing maintains temperature at the upper end of the frame at a steady state level of 165°F.
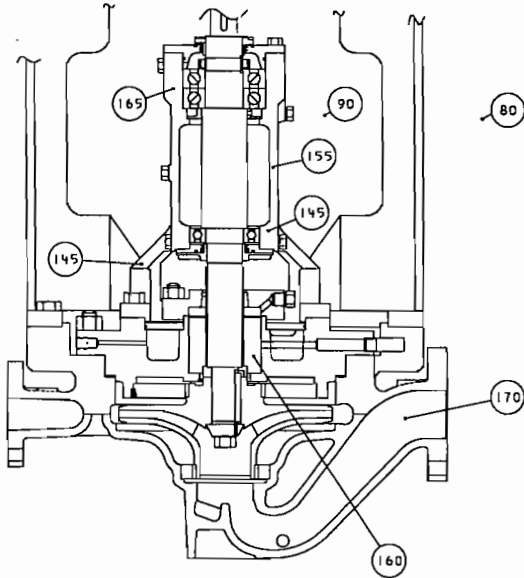

Figure 11. Steady State Temperature Profile for Grease Lubrication with 170°F Pumpage.
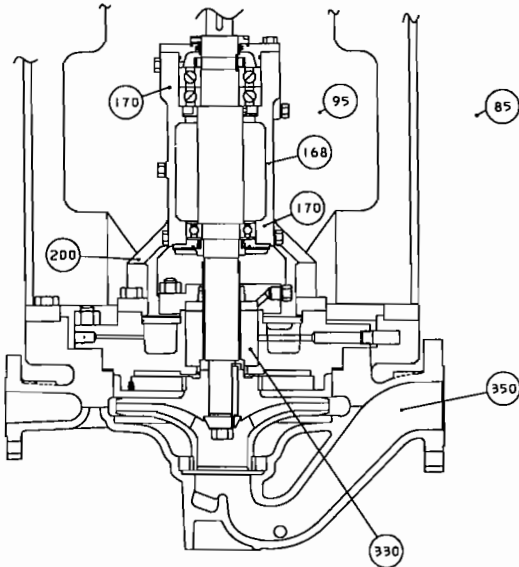

Figure 12. Steady State Temperature Profile for Grease Lubrication with 350°F Pumpage.

At the 350°F pumpage temperature, heat conducted through the shaft significantly increases radial bearing operating temperature while the thrust bearing operates at a slightly increased level indicating an incremental heat addition over that generated by the bearing. Note that air temperature within the motor support is about 10°F hotter than outside due to heat convected from the seal

chamber cover and confined within the support. Effective design of the motor support and application of cooling devices is required to minimize the confined air temperature since it has a direct impact on bearing operating temperature.

Heat Flinger and Fan Testing

Defining the impact of these cooling devices in controlling specific heat flows was a key objective of the experimental testing. The function of the heat flinger is primarily to dissipate shaft conducted heat. Proper design will also reduce heat conducted through frame arms and heat convected off the seal chamber cover by inducing necessary air flow. The fan's primary function is to induce air flow along the bearing frame walls, effectively increasing the convective heat transfer coefficient. The impact of these cooling devices is described in the following sections.

Grease Lubrication Testing

The grease lubricated configuration was tested at pumpage temperatures to 500°F with various cooling devices applied to determine effects. Thrust and radial bearing operating temperatures are presented in Figure 13, with the following observations:

• The thrust bearing generates heat due to imposed load and viscous churning of the grease lubricant. This is the controlling factor in bearing operating temperature at pumpage temperatures less than 250°F.

• The fan reduces bearing operating temperature throughout the pumpage temperature range.

• The heat flinger and jacket cooling have minimal affect on bearing operating temperatures at lower temperatures.

• The radial bearing operating temperature is highly dependent on pumpage temperature due to heat conduction through the shaft. The application of cooling devices, particularly a heat flinger, is critical at higher pumpage temperatures.

• Bearing operating temperatures can be controlled within the 180°F API 610 limit, with proper application of cooling devices, at pumpage temperatures to 400°F.

Oil Sump Lubrication Testing

Oil sump lubrication was tested at pumpage temperature to 600°F and the impact of cooling devices defined. Thrust and radial
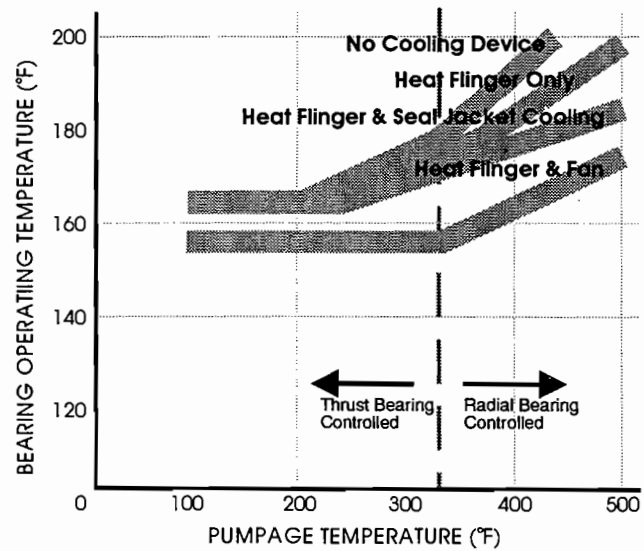

Figure 13. Grease Lubrication Test Results with Cooling Device Impact.

bearing operating temperatures are presented in Figure 14. Observations are as follows:

• The reduced thrust bearing operating temperature at lower pumpage temperatures is indicative of better transfer of thrust bearing generated heat by the oil system, using the oil sump to convect heat to atmosphere.

• Bearing operating temperatures rise linearly as pumpage temperature increases.

• Heat flinger and jacket cooling impact is minimal at lower temperatures but very effective as pumpage temperature increases.

• A fan is effective throughout the temperature range as heat is removed from the bearing frame walls and the oil sump.

• Bearing operating temperature can be controlled within the 180°F API 610 limit, with proper application of cooling devices, at pumpage temperatures to 600°F.
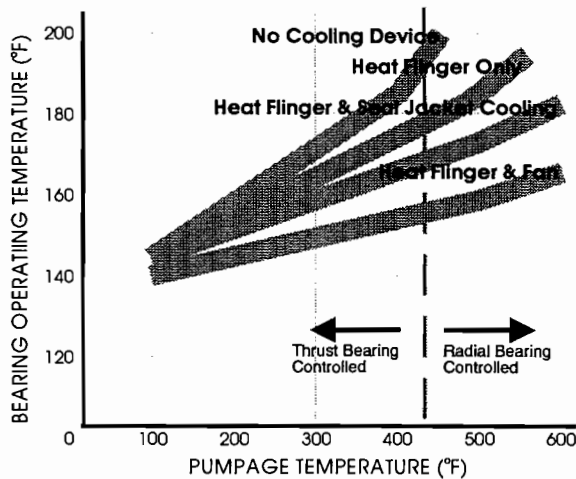

Figure 14. Oil Sump Lubrication Test Results with Cooling Device Impact.

Oil Mist Lubrication Testing

The oil mist configuration was tested at pumpage temperatures up to 650°F with the effect of cooling devices evaluated. The data are presented in Figure 15, with the following observations:

• The heat generated by both bearings is effectively removed by use of oil mist lubrication. The mist minimizes heat generated due to lubricant viscous churning and effectively removes load dependent generated heat, reducing bearing temperatures significantly and permitting operation at higher pumpage temperatures.

• Seal chamber jacket cooling and the heat flinger effectively control heat transfer to the bearings at higher temperature, with similar temperature reductions on both radial and thrust bearings.

• Operation at pumpage temperatures exceeding 700°F is acceptable with oil mist lubrication, maintaining bearing temperatures below the API 610 limit of 180°F assuming proper application of cooling mechanisms.

ADAPTATION OF THE EXISTING THERMAL MODEL

Background

The thermal model analysis was first developed in 1983 to predict thermal behavior in horizontal, ring oil lubricated API-610
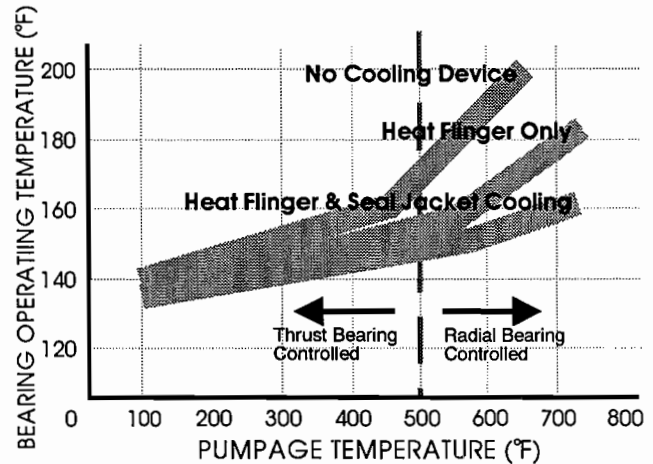

Figure 15. Oil Mist Lubrication Test Results with Cooling Device Impact.

overhung process pumps [3]. A significant amount of testing was done at that time to develop and validate the analytical tool. The model is computerized and is used today to predict bearing housing and oil temperatures for unique applications as well as to evaluate design options. The model analyzes bearing heat generation, shaft heat conduction, convective heat loss and heat loss to various cooling mechanisms.

A heat energy balance was used as the foundation for the horizontal pump analysis. For a steady state oil sump temperature to be reached, the heat energy entering the oil sump must equal the heat energy leaving the oil sump. This balance and its components are described by the following Equation (1) and shown in Figure 16.

$$
\begin{array}{l}
\left[\begin{array}{c}\text{Rate of heat}\\\text{generated by}\\\text{bearings}\end{array}\right] + \left[\begin{array}{c}\text{Rate of heat}\\\text{conducted}\\\text{from the}\\\text{pumpage}\end{array}\right] =
\end{array}
$$

$$
\left[\begin{array}{c}\text{Rate of heat}\\\text{convected \&}\\\text{radiated to}\\\text{the cooling}\\\text{environment}\end{array}\right] + \left[\begin{array}{c}\text{Rate of heat}\\\text{removed by}\\\text{the oil cooler}\end{array}\right] + \left[\begin{array}{c}\text{Rate of heat}\\\text{removed by}\\\text{seal jacket}\end{array}\right]
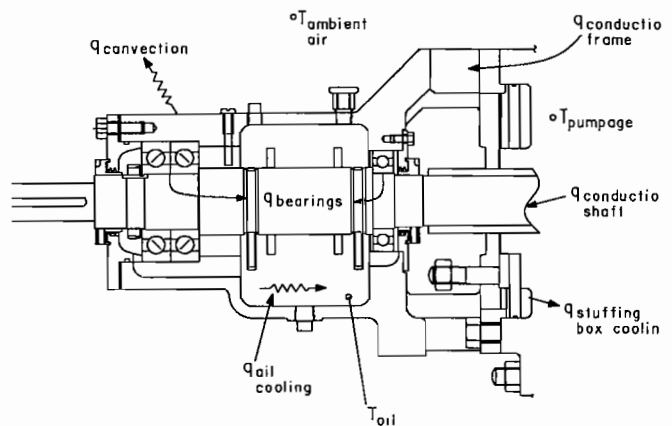$$

(1)

Figure 16. Heat Transfer Paths Used for the Horizontal Model.

This successful model was used as the basis for the bearing frame inline thermal model with adjustments applied as described in the following section.

Adjustments to the Model

The energy balance can be applied to a bearing frame inline thermal analysis with only basic adjustments. Specifically, oil coolers are normally not employed and heat flingers are utilized as a shaft heat sink. The effect of a fan is analyzed by increasing the convection heat transfer coefficient for the bearing frame as defined in the literature [4, 5] and verified with experimental test. Additionally, expanding the model to analyze both grease and pure oil mist can be accomplished by adjusting the bearing heat generated by viscous churning of the lubricant.

As adapted for bearing frame inline analysis, the heat energy balance is described in Equation (2) and shown pictorially in Figure 17.

$$
\begin{array}{l}
\text{Rate of heat} \\
\text{generated by} \\
\text{bearings}
\end{array}
+
\begin{array}{l}
\text{Rate of heat} \\
\text{conducted} \\
\text{from the} \\
\text{pumpage}
\end{array}
=
$$

(2)

$$
\begin{array}{l}
\text{Rate of heat} \\
\text{convected \&} \\
\text{radiated to} \\
\text{the cooling} \\
\text{environment}
\end{array}
+
\begin{array}{l}
\text{Rate of heat} \\
\text{removed by} \\
\text{the heat} \\
\text{flinger}
\end{array}
+
\begin{array}{l}
\text{Rate of heat} \\
\text{removed by} \\
\text{seal jacket}
\end{array}
$$

$q_{bearings} + (q_{shaft} + q_{casing}) = q_{frame} + q_{flinger} + q_{jacket}$
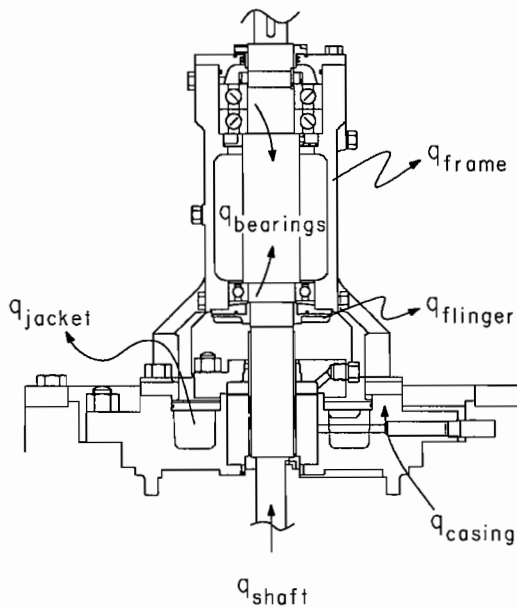

Figure 17. Heat Transfer Paths Used for the Bearing Frame Inline Model.

The equation detail for the heat flows that are common to the horizontal heat balance in Equation (1) are presented in the referenced paper and will not be reviewed here. Only alterations and additions to the equation detail will be discussed.

Bearing generated heat, as used in the model, is comprised of load dependent heat generation and viscous churning or load

independent heat generation [6]. The viscous churning component is controlled by the frictional moment M_o:

$$
M_o = 10^{-7} f_o (vn)^{2/3} d_m^3
$$

where

f_o = viscous torque factor
v = lubricant kinematic viscosity
n = bearing rpm
d_m = bearing mean diameter

The viscous torque factor is selected based on lubricant type and method, allowing the frictional moment, and therefore bearing heat generation, to be adjusted for analysis of grease and oil mist lubrication in bearing frame inline pumps.

The heat flinger removes heat by conducting heat from the shaft and using forced convection to remove heat from the flinger. This heat flow is added to the model in the following form:

$$
q_{flinger} = h_f A_f (T_{pumpage} - T_{air})
$$

where

h_f = empirically derived overall heat transfer coefficient
A_f = surface area of the flinger

These adjustments adapt the existing horizontal thermal model for use on bearing frame inline pumps.

Model Correlation

Correlation of the data and model was examined for many test conditions with error to actual test detailed for several conditions in Table 1. This validates the model as a valuable tool for use in predicting the thermal behavior of the inline pump at elevated pumpage temperatures with different lubrication systems and various cooling devices.

Table 1. Comparison of Experimental Test Date to Model Predictions.

Lubrication Method	Cooling Method	Pumpage Temperature (°F)	Bearing Operating Temperature		
			Actual (°F)	Predicted (°F)	Absolute Error (°F)
Grease	Heat Flinger	175	166	172	
Oil Sump	Heat Flinger	424	178	173	-5
Oil Mist	Heat Flinger & Seal Jacket Cooling	650	156	150	-6

Technology Development

The computerized model is being further refined by adding finite element analysis capability. The fundamental work of developing a solid model (Figure 18) and a three dimensional FEA mesh (Figure 19) has been completed for the horizontal model and analysis efforts continue to finalize the FEA model and adjust it for the inline design. The finite element model will clearly add another level of sophistication to the model's capability, allowing better analysis of thermal transients, unique application criteria, and unique lubrication properties.

CONCLUSIONS

The experimental testing and its adaptation to the existing predictive model have provided the tools to properly evaluate the impact of high temperature operation with API 610 bearing frame
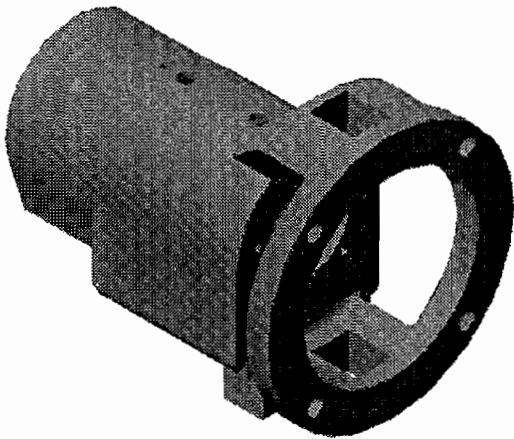
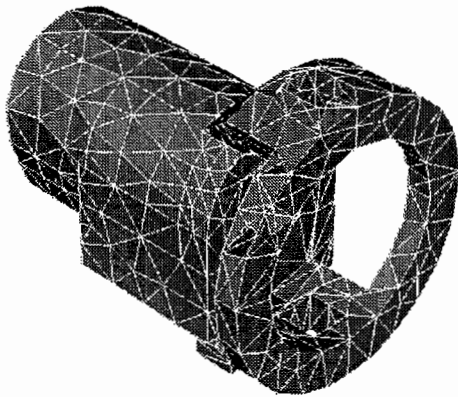*Figure 18. Solid Model of a Horizontal Bearing Frame.*

Figure 19. Three Dimensional Finite Element Mesh Developed for Thermal Model.

inline pumps. The use of this model clearly defines lubrication and cooling requirements necessary to assure acceptable bearing operating temperatures for any condition of service.

The testing and the model define limits of operation for grease, oil sump, and oil mist lubrication as well as cooling device requirements to meet specific application conditions. Conservative limits are presented in Table 2.

Lubrication options can be applied that will result in bearing operating temperatures that meet the requirements of API 610 at

Table 2. General Pumpage Temperature Limits for Various Lubrication Systems and Cooling Devices Applied to Bearing Frame Inline Pumps.

Cooling Device Applied	Pumpage Temperature Limit (°F)		
	Grease Lubrication	Oil Sump Lubrication	Oil Mist Lubrication
None	250	300	450
Heat Flinger	300	350	600
Heat Flinger & Seal Jacket Cooling	350	450	750
Heat Flinger & Fan	450	600	

Note that data applies as follows: 3600 RPM operation; ambient temperature range of 60°F - 100°F; horsepower range to 200 HP; uncooled seal flush.

the elevated pumpage temperatures commonly specified for refinery process. This provides the user with the option to specify a bearing frame inline pump for process services and be confident that shaft stiffness, vibration and lubrication will provide mechanical seal and bearing service life comparable to a standard horizontal process pump. As an aid to the user, API 610 Eighth Edition will incorporate requirements for bearing frame inline designs, specifying that they meet the same mechanical design requirements as the horizontal overhung process pump.

REFERENCES

1. American Petroleum Institute, "Centrifugal Pumps for General Refinery Services," API Standard 610, Seventh Edition (1989).

2. American Society of Mechanical Engineers, "Specification for Vertical Inline Centrifugal Pumps for Chemical Process," ASME Standard B73.2M-1991 (1991).

3. Cappellino, C. A. and Osborne, J. C., "The Prediction of Bearing Lubricant Temperature and Cooling Requirements for a Centrifugal Pump", *Proceedings of the Second International Pump Symposium*, The Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1985).

4. Shigley, J.E., *Mechanical Engineering Design*, Third Edition, New York, New York: McGraw Hill Book Co., Inc. (1977).

5. Krieth, F., *Principle of Heat Transfer*, Third Edition, New York, New York: Intext Educational Publishers (1973).

6. SFK Industries, Inc., SKF General Catalog, pp. 46-50 (1991).