THE PREDICTION OF BEARING LUBRICANT TEMPERATURES AND COOLING REQUIREMENTS FOR A CENTRIFUGAL PUMP

by

Charles A. Cappellino

Analysis Engineer

and

James C. Osborne

Supervisor, Product Development

Goulds Pumps, Incorporated

Engineered Products Division

Seneca Falls, New York

Charles A. Cappellino is an Analysis Engineer at the Engineered Products Division of Goulds Pumps, Inc. He is primarily responsible for the development of mechanical and hydraulic analysis techniques for centrifugal pumps. This includes activity in the areas of rotordynamics, thermal analysis, pump performance prediction and impeller flow analysis.

Mr. Cappellino received a B.S. degree in Mechanical Engineering from Clarkson College of Technology and is a member of ASME.

James C. Osborne is Supervisor, Product Development at the Engineered Products Division of Goulds Pumps, Inc. His primary responsibilities include the design and development of centrifugal pumps for the petroleum, paper and chemical industries.

Mr. Osborne received a B.S. degree in Mechanical Engineering from Rensselaer Polytechnic Institute. He is a member of ASME and is a registered professional engineer in the State of New York.

ABSTRACT

A thermal analysis of a centrifugal pump bearing frame using fundamental thermodynamic and heat transfer principles is presented. This analysis enables prediction of bearing lubricant temperatures for a wide range of operating and environmental conditions.

An experimental test program run in conjunction with the analytical effort is described. This discussion includes details on the design of a high temperature test facility, the instrumentation and the test procedure. Test measurements are compared to analytical predictions revealing good agreement for several different geometries and operating conditions.

Typical results are presented graphically and their practical usefulness explained. The use of a computer program based on this work is also described. This provides a means of quickly establishing the minimum cooling requirements based on the actual pumpage and ambient conditions of a field installation.

INTRODUCTION

Antifriction bearing failures are a primary cause of centrifugal pump downtime. Excessively high bearing lubricant temperature is one cause of such failures. Bearing cooling is therefore often used to avoid problems. How lubricant temperatures and the effect of cooling options can be analytically determined is shown, providing the user with a tool for evaluating various options for reducing pump downtime due to improper lubrication. Lubricant temperature limits, types of cooling and an analytical method to predict lubricant temperatures, as well as a corroborating test program and some typical results, are presented.

Hydrocarbon oils are the most common bearing lubricants. Unfortunately, most oils break down and lose their lubricating properties rapidly at elevated temperatures (Figure 1). The commonly used limit for such oils is 180°F, but 200°F is not an unreasonable limit, provided a good maintenance program is followed. High temperature lubricants, however, can be used to extend the operating range to the limit of the bearing. These lubricants can be synthetic oils or hydrocarbon oils with high temperature additives, provided they have satisfactory properties at the operating temperature. This may necessitate more viscous oil for high temperature operation than is normally used elsewhere. Bearings are normally limited to about 240°F and test data shows that they run no more than 10°F hotter than the oil sump where lubricant temperature is measured. High temperature lubricants, therefore, are limited to 230°F to protect the bearings. Special bearings can be made for higher temperatures, but they are expensive and add complexity to the equipment, so they are seldom used in pumps.

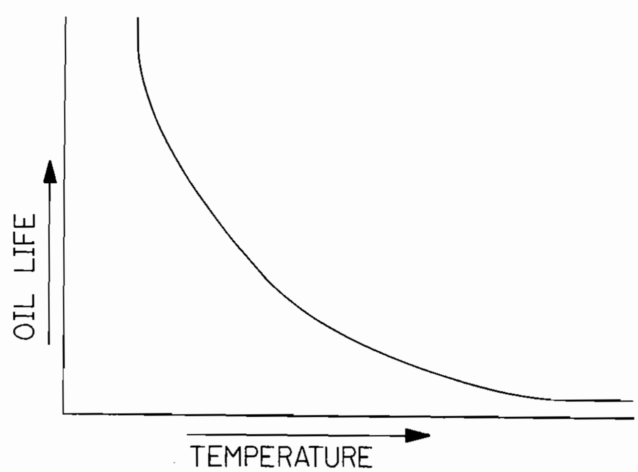

Figure 1. Typical Hydrocarbon Oil Life.

When the lubricant temperature exceeds safe limits, it can be controlled by cooling the bearing frame from an external source. This can be accomplished with either a liquid run through heat exchangers or air blown over the bearing frame. The most common cooling method is a jacket cast in the bearing frame and cooled with water. Some older jackets, however, were made to encircle the bearings (Figure 2), thus cooling only the outer race which then contracts and preloads the bearing. The preload generates more heat and the situation can quickly result in a seriously overloaded bearing and early failure. Such designs are not common today and should be avoided.

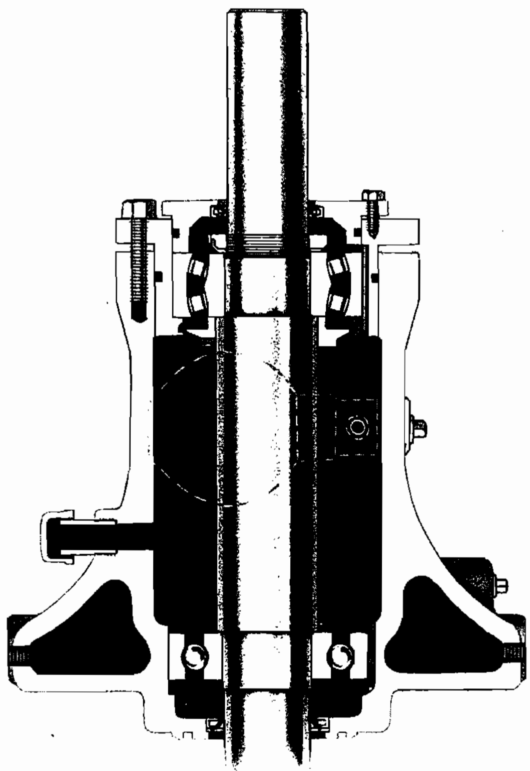

Figure 2. Poorly Designed Bearing Frame Cooling Jacket.

A well designed cast bearing frame cooling jacket, located on the bottom of the bearing frame (Figure 3) so only the oil is cooled, is a good functional design and is suitable for most services. When very high temperatures are involved, however, the high rate of heat transfer needed to maintain the proper lubricant temperature is better handled by small heat exchangers similar to that shown in Figure 4. This type of cooler is normally 10 to 20 percent more efficient than a cast-in jacket and is easy to clean or replace so peak efficiency can be maintained. Besides being efficient, this method is ideal because

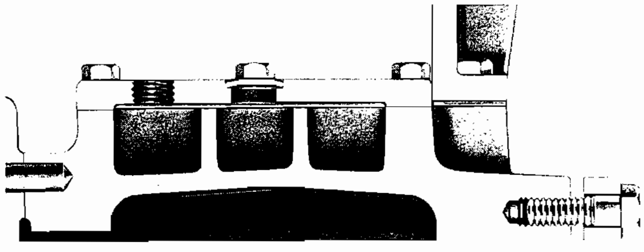

Figure 3. Well Designed Bearing Frame Cooling Jacket.

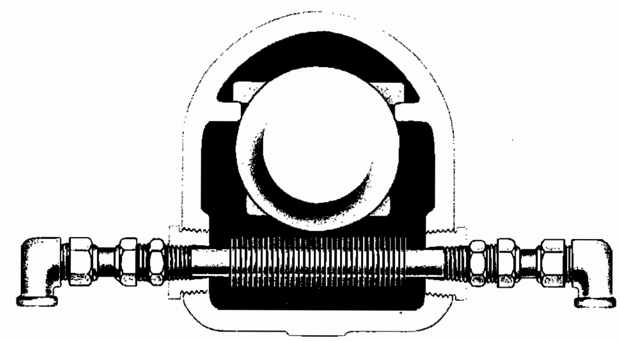

Figure 4. High Efficiency, High Temperature Bearing Frame Cooler.

it cools only the oil, which in turn bathes the entire bearing to withdraw heat evenly, thus cooling the bearing without inducing other problems.

Stuffingbox cooling jackets (Figure 5) may also be used on high temperature services. Their primary purpose is to control the seal environment, but they also significantly reduce heat flow to the bearings. The effect of stuffingbox cooling on the bearing lubricant temperature, therefore, must be considered in determining the most economical cooling package for a particular installation.

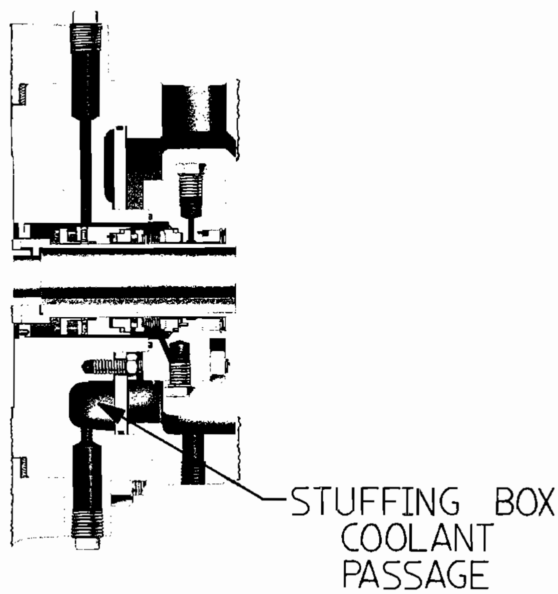

Figure 5. Stuffingbox Cooling Jacket.

Bearing frame cooling can also be provided by mounting fans on the pump shaft to air cool the frame by forced convection. This method of cooling is quite effective and can be used to limit lubricant temperatures satisfactorily on all but the hottest services.

Any or all of these cooling options can be used depending on need, user preference, environmental regulations, etc. They all have their own advantages and disadvantages, but the one thing they have in common is that they all add cost. The initial cost of buying a cooler usually isn't prohibitive, but the cooler is only a small part of the total system needed to use it. A cooling system generally can include:

- Heat exchangers and jackets supplied with the pump.
- Piping to supply cooling water.

• Water treatment facility, cooling tower, etc., as needed to meet water quality standards for discharge or reuse.

• Wells, pumps, city water as applicable to supply water.

• Trucking water to remote arid sites.

The total cost to buy, install and maintain such a system can obviously be significant. Decisions to use cooling on a pump, therefore, should be carefully considered and cooling should only be used when bearing lubricant temperatures would exceed safe limits without it [1].

Unfortunately, determining exactly when and how much cooling is needed to limit the lubricant temperature is difficult. Many factors, including pumpage temperature, ambient conditions, speed, lubrication, pump design and cooling, influence the final bearing lubricant temperature. All of these factors can vary widely between installations, making lubricant temperature prediction a complex problem. Due to the complexity of the problem, detailed studies are not routinely done on an application by application basis, so most cooling recommendations are based on a very limited amount of test data or simply past practice, and most of the important variables are ignored for simplicity. This results in workable but conservative guidelines, which specify more cooling than is needed. Field history shows that this has generally been a reliable approach to the problem, but it is not the most economical.

Adherence to these guidelines has generally provided reliable service, but if cooling requirements could be accurately predicted, wasteful cooling could be curtailed without jeopardizing equipment reliability. This would cut users' costs by eliminating or down sizing the entire cooling water distribution systems.

To achieve this capability, an analytical thermal model was developed, and an experimental test program was performed to confirm the analytical work. A line of end suction, heavy duty, single stage pumps (Figure 6), designed to meet American Petroleum Institute Standard 610 (API-610) [2] requirements, was used as the subject of this analysis and testing program. These pumps were chosen because they are often used in high temperature services and in hot climates, so cooling is commonly used and the potential benefits, therefore, are high.

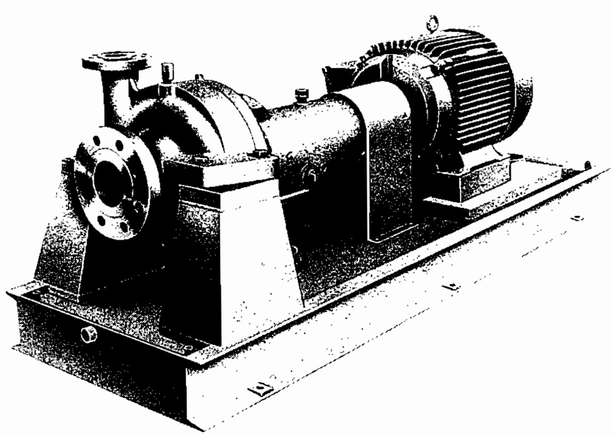

Figure 6. Subject of This Study: High Temperature Process Pump.

ANALYTICAL MODEL

A significant amount of analysis work was performed with the intention of meeting three objectives. First, the analysis would provide a better understanding of the mechanisms and variables affecting bearing lubricant temperatures and cooling requirements. Second, it would identify important measurements and instrumentation requirements for an extensive test program that would follow. Finally, it would provide a means of predicting bearing lubricant temperatures and cooling requirements for any standard operating condition of the end suction API-610 product line.

The Energy Balance Equation

A heat energy balance was used as the foundation for this analysis. For a steady state bearing lubricant temperature to be reached, the heat energy entering the lubricant must equal the heat energy leaving the lubricant. This balance and its components are described in equation (1) and shown pictorially in Figure 7.

Rate of Heat Generated by the Bearings	+	Rate of Heat Conducted from the Pumpage	=	Rate of Heat Convected & Radiated to the Environment	+	Rate of Heat Removed by the Oil Cooler	+	Rate of Heat Removed by Stuffingbox Cooling	(1)

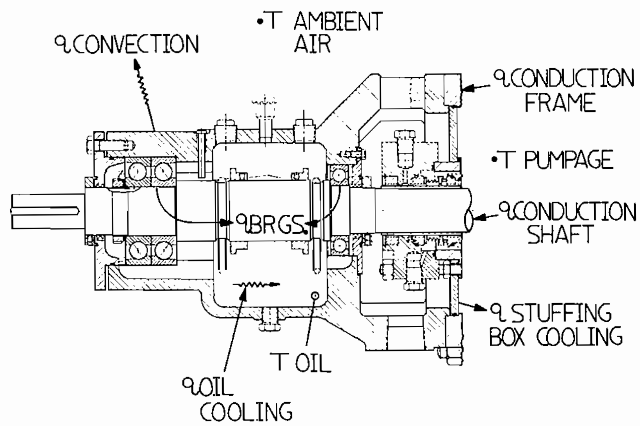

Figure 7. Heat Transfer Paths.

The solution of equation (1) requires the use of fundamental heat transfer principles and a means of calculating bearing heat generation. Each component of equation (1) is subsequently described and all except the bearing heat generation appear in the following forms:

$$q = U\,A\,\Delta T \qquad (2)$$

or

$$q = h\,A\,\Delta T \qquad (3)$$

where

- q = rate of heat transfer, Btu/hr.
- U = overall heat transfer coefficient, Btu/hr ft^2°F.
- h = convection heat transfer coefficient, Btu/hr ft^2°F.
- A = area basis of overall heat transfer, ft^2.
- ΔT = temperature difference, °F.

Values of U and h can be determined from readily available heat transfer literature and techniques [3,4,5,6,7,8].

Bearing Heat Generation

An inevitable source of heat generation in rotating equipment is the bearings. The rate of heat generation is a function of many variables including bearing geometry, speed, load, lubricant qualities and lubrication method. In this analysis, equation (4) was used to estimate the rate of heat generation by the bearings [9,10].

$$q_{brgs.} = 0.485\ N\ [0.083\ f_1\ P_\beta\ d_m + 1.183 \times 10^{-6}\ f_0\ (\nu N)^{2/3}\ d_m^3] \quad (4)$$

This equation can be divided into load dependent heat generation (equation (5)) and viscous churning or speed dependent heat generation (equation (6)).

$$q_{load} = 0.485\ N\ [0.083\ f_1\ P_\beta\ d_m] \quad (5)$$

$$q_{viscous} = 0.485\ N\ [1.183 \times 10^{-6}\ f_0\ (\nu N)^{2/3}\ d_m^3] \quad (6)$$

where

$q_{brgs.}$ = rate of bearing heat generation, Btu/hr.
q_{load} = load dependent bearing heat generation, Btu/hr.
$q_{viscous}$ = viscous churning or speed dependent bearing heat generation, Btu/hr.
N = rotational speed, rpm.
f_1 = load torque factor.
P_β = applied bearing load, lbs.
d_m = bearing mean diameter, in.
f_0 = viscous torque factor.
ν = lubricant kinematic viscosity, centistokes.

The ability to divide bearing heat generation into its components, along with the capability of evaluating the effects of various parameters (bearing load, size, speed, etc.), provides the designer with a valuable analytical tool. For instance, the sizing of antifriction bearings is often a balance between bearing load capacity and heat generation (larger bearings carry more load, but generate more heat). An optimum balance could be determined through the use of this analysis.

Effect of Pumpage Temperatures

As stated previously, the line of pumps which is the subject of this analysis serves the petroleum industry and often handles high temperature pumpage. A pumpage temperature higher than the lubricating oil temperature will result in heat conduction through the shaft and bearing frame, which can become a significant factor in determining the lubricating oil temperature. The rate of heat conduction through the shaft and bearing frame is calculated using equations (7) and (8), respectively.

$$q_{shaft} = U_s\ A_s\ [T_{pumpage} - T_{oil}] \quad (7)$$

$$q_{frame} = U_f\ A_f\ [T_{pumpage} - T_{oil}] \quad (8)$$

where

q_{shaft} = rate of heat conduction through shaft from pumpage, Btu/hr.
q_{frame} = rate of heat conduction through frame from pumpage, Btu/hr.
U_s, U_f = overall heat transfer coefficients of shaft and frame, respectively, Btu/hr ft^2°F.
A_s, A_f = area basis of overall heat transfer for shaft and frame, respectively, ft^2.
$T_{pumpage}$ = temperature of pumpage, °F.
T_{oil} = temperature of lubricating oil, °F.

In some cases, the pumpage temperature is lower than the lubricating oil temperature. This situation leads to heat loss through the shaft and bearing frame to the pumpage. Equations (7) and (8) can also be used to determine this heat loss.

Other Sources of Heat

Other sources of heat can be easily incorporated into the analysis if desired. Examples include a bearing frame heating element used in extremely cold climates or the amount of heat absorbed as a result of the desert sun striking the bearing frame surface.

Convective Heat Loss

A major component of heat loss is the amount of energy convected to the surrounding air. This heat loss can be determined by

$$q_{air} = h_{air}\ A_{sf}\ [T_{oil} - T_{air}] \quad (9)$$

where

q_{air} = rate of heat convected and radiated to the surrounding environment, Btu/hr.
h_{air} = average convection heat transfer coefficient, Btu/hr ft^2°F.
A_{sf} = surface area of bearing frame dissipating heat, ft^2.
T_{oil} = lubricating oil temperature, °F.
T_{air} = surrounding air temperature, °F.

A value of h_{air} can be calculated for still air or for air blown over the bearing frame by fans. This capability provides a quick and easy method of evaluating various environmental conditions, as well as the effect of optional cooling fans.

Bearing Frame Cooling

As discussed previously, industrial pumps offer some type of bearing frame or lubricating oil cooling as an option to maintain proper oil temperatures. The oil cooling element involved in this analysis is shown in Figure 4. Heat is removed from the oil by convection to the finned tube, then transferred to the cooling water and carried away. The rate of heat transfer is determined by

$$q_{oil\ cooler} = U_c\ A_c\ [T_{oil} - T_c] \quad (10)$$

where

$q_{oil\ cooler}$ = rate of heat removal by the oil cooler, Btu/hr.
U_c = air cooler overall heat transfer coefficient, which is based on cooler geometry, conduction through the tube wall and convective heat transfer coefficients, Btu/hr ft^2°F.
A_c = total oil cooler heat transfer area, ft^2.
T_{oil} = lubricating oil temperature, °F.
T_c = average of inlet and outlet coolant temperatures, °F.

The coolant outlet temperature can be easily calculated knowing the coolant inlet temperature, flowrate and specific heat. This information can be useful when the coolant is passed along to other pumps or when regulations must be met for coolants discharged into the environment.

Stuffingbox Cooling

Earlier discussion indicated stuffingbox cooling can significantly reduce the flow of heat to the lubricating oil. This is due to the proximity of the stuffingbox cooling chamber to the shaft and to the point of attachment of the bearing frame (Figure 5). As the coolant passes through the cooling chamber, a large amount of heat is absorbed from the surrounding surfaces. A large amount of heat is thus blocked from entering the bearing frame. The rate of heat transfer can be determined by

$$q_{sb} = h_{sb} A_{sb} [T_{sb} - T_c] \quad (11)$$

$$q_{sbc} = C\, q_{sb} \quad (12)$$

where

q_{sb} = the entire amount of heat absorbed by the stuffing-box coolant, Btu/hr.
h_{sb} = average convection heat transfer coefficient, Btu/hr ft^2°F.
A_{sb} = area basis of overall heat transfer.
T_{sb} = average temperature of the surface area inside the coolant passageway, °F.
T_c = average of inlet and outlet coolant temperatures, °F.
q_{sbc} = amount of heat blocked from entering the lubricating oil through the bearing frame arms and shaft due to the presence of stuffingbox cooling, Btu/hr.
C = empirically derived coefficient.

Analysis Summary

All components of the energy balance equation (1) are described in equations (2) through (12). By substitution, equation (1) becomes

$$q_{brgs} + U_s A_s [T_{pumpage} - T_{oil}] + U_f A_f [T_{pumpage} - T_{oil}]$$
$$= h_{air} A_{sf} [T_{oil} - T_{air}] - U_c A_c [T_{oil} - T_c]$$
$$- C\, h_{sb} A_{sb} [T_{sb} - T_c] \quad (13)$$

This equation allows prediction of bearing lubricant temperatures and cooling requirements for any standard operating condition. It also provides an excellent means of evaluating and understanding the thermal effects of design changes and new cooling options.

EXPERIMENTAL TEST PROGRAM

An experimental test program was established with the intent of providing a wide range of data to support the analytical effort. The test program was divided into two categories: conventional tank testing with water temperatures ranging from 60°F to 150°F, and high temperature testing using a specially designed test loop. The high temperature testing involved pumpage temperatures in the 400°F to 550°F range and will be the focus of the experimental test program discussion.

High Temperature Test Facility Design

The design objectives for the test facility were:

1. The ability to test various sizes of the end suction API-610 process pump product line.
2. Develop and maintain high pumpage temperatures.
3. Develop and maintain a range of air temperatures.
4. Not require a large expenditure of capital or construction time.

To meet these objectives, a simple system consisting of a pump, motor, piping, expansion tank and valve was developed as shown in Figure 8. This system does not incorporate any type of heating element to raise the pumpage temperature. The horsepower consumed by the pump is transferred directly to the fluid and subsequently converted to heat energy by valved pressure breakdown and piping losses. Since the piping system is a closed loop, the pumpage temperature can be raised to a desired level by simply varying the pump horsepower through changes in impeller diameter and flowrate. The test loop was enclosed by a small shed, which made it possible to increase or decrease the air temperature surrounding the pump through the use of exhaust fans and openings to the ouside. This provided enough flexibility to meet the range of air temperatures required for the test program.

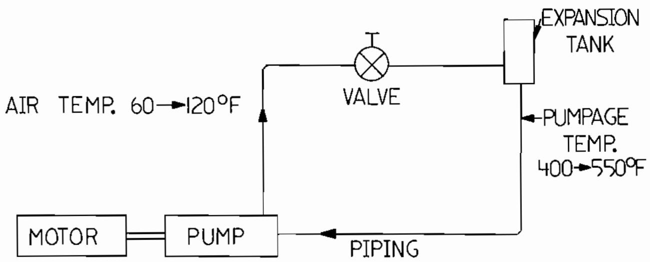

Figure 8. Schematic Diagram of High Temperature Test Facility.

Instrumentation

The pumps installed in the high temperature test loop were highly instrumented with thermocouples. These were placed at points of specific interest throughout the pump (Figure 9), so that the analytical model could be verified. Temperature readings of each thermocouple were recorded throughout the entire test program.

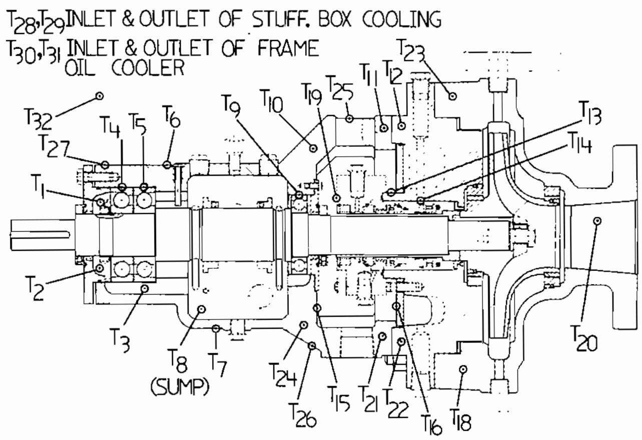

Figure 9. Thermocouple Locations for High Temperature Test Program.

Range of Test Program

A full range of the end suction API-610 process pump product line (5 hp to 700 hp) was involved in the testing program with concentration on the smaller pump sizes due to size and availability considerations. Pumpage temperatures ranged from 60°F to 550°F, surrounding air temperatures from 60°F to 120°F, and speeds from 1200 to 3600 rpm. Cooling arrangements included frame oil cooling, stuffingbox cooling, and various cooling fan arrangements. These cooling arrangements were tested separately and in combination, thus providing a wide range of data at many different operating conditions.

PRESENTATION AND DISCUSSION OF RESULTS

Comparison of Oil Temperature Prediction to Test Data

A representative sample of pump test conditions are compared with analytical predictions in Table 1. The sample contains four different bearing frame sizes, two speeds, a wide range of ambient and pumpage temperatures, and three different cooling conditions (none, frame oil cooling and stuffingbox

Table 1. Representative Sample of Pump Test Conditions and Absolute Error of Predictions.

Pump Size	Speed (rpm)	Ambient Temp. (°F)	Pumpage Temp. (°F)	Cooling Type	Coolant Inlet Temp. (°F)	Coolant Flow Rate (gpm)	Absolute Error of Predicted Oil Temp. (°F)	Absolute Error of Predicted Coolant Outlet Temp. (°F)
1×2-11	3550	85	136	None	--	---	+5	--
1½×3-13	3550	73	152	None	--	---	+8	--
2×4-9	1750	68	114	None	--	---	+1	--
2×4-11	3550	65	403	None	--	---	−1	--
4×6-16	3550	77	77	None	--	---	0	--
8×10-21	1780	80	159	None	--	---	+2	--
1½×3-11	3550	120	502	Frame	67	1	+8	−1
1½×3-11	3550	92	446	Frame	71	.3	+8	0
2×4-11	3550	60	503	Frame	47	2.2	+1	0
2×4-11	3550	86	411	Frame	53	.5	−3	−1
1½×3-11	3550	101	382	Stuffingbox	73	2.5	−2	−5
2×4-11	3550	89	397	Stuffingbox	54	2.7	−4	−6

cooling) with various coolant temperatures and flowrates. The last two columns in Table 1 list the absolute prediction error (°F) for the oil temperature and the coolant outlet temperature.

An examination of these columns reveals good agreement regardless of size, speed, ambient temperature, pumpage temperature and cooling arrangement. Although this sampling represents less than 20 percent of all tested conditions, the results listed are very indicative of the general results. Oil temperature predictions generally fell within 10°F of actual values with most agreeing within 5°F. This agreement is entirely adequate to make accurate cooling requirement recommendations and supports the use of the analytical technique for design purposes.

Examination of a Typical Set of Test Data

Knowing the temperature distribution throughout a pump is very beneficial to understanding the flow of heat from one area to another. This allows fine tuning of assumptions and actual calculation of many pertinent heat transfer quantities.

One of the more interesting findings resulting from the test data analysis is the determination of exactly how stuffingbox cooling tends to control bearing lubricant temperatures. The pump shaft appears to be the obvious path for heat conduction from the pumpage into the lubricating oil. The use of equations (7) and (8) to predict the amount of heat flow through the shaft and bearing frame indicates that more heat passes through the bearing frame than through the shaft. Thus, a method of reducing the amount of heat flow through the bearing frame could significantly reduce lubricating oil temperatures. An examination of Figures 10 and 11 show the substantial effect of stuffingbox cooling on the bearing frame temperatures. This particular case shows that for a condition without stuffingbox cooling (Figure 10), thermocouples T_{11} and T_{12} indicated 308°F and 338°F, respectively. These high temperatures and the 30°F difference indicate a significant flow of heat into the lubricating oil. The temperature distribution resulting from the use of stuffingbox cooling with the same ambient and pumpage temperatures is shown in Figure 11. Thermocouples T_{11} and T_{12} indicated 184°F and 189°F, respectively. A significant decrease in the temperature difference and the heat flow is evident. Thus, Figures 10 and 11 show how stuffingbox cooling can effectively prevent process heat from entering the bearing frame.

Graphical Presentation of Results

A graphical display of the results of a study such as this is usually beneficial. It can provide a good picture of trends and a simple means to utilize the results for practical purposes. Unfortunately, the analysis presented here is too complex to be completely described graphically. It is possible, however, to plot results for some common cases to show the effect of certain variables and to demonstrate how this analytical model can be used to reduce cooling costs. Two common cases which can be particularly useful for practical guidance in most actual installations are shown in Figures 12 and 13. The data shown on Figure 12 applies to a typical non-cooled unit. Effects of bearing frame

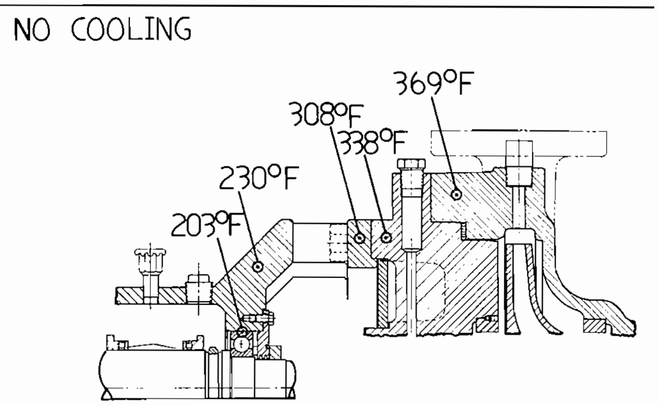

Figure 10. Temperature Distribution without Stuffingbox Cooling.

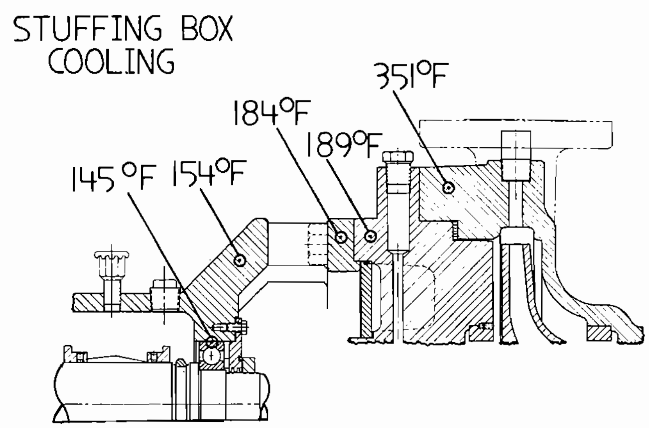

Figure 11. Temperature Distribution with Stuffingbox Cooling.

cooling on the same unit are shown in Figure 13. It must be noted that the trends shown on these curves are typical of what was found throughout the study, but the actual results apply only to one specific case for the group of pumps using one particular bearing frame. These results are not applicable to other models or sizes.

Lubricant temperature as a function of ambient and pumpage temperatures for a pump running at 3550 rpm without any cooling is shown in Figure 12. The same data for the same pump run at 1780 rpm are presented in Figure 14. These plots provide a quick check on the necessity of cooling for a particular installation, and demonstrate the flexibility such data provides.

As an example, API-610 require that ring oil lubricated pumps be supplied with sufficient cooling to maintain the oil temperature below 180°F when the ambient temperature is 110°F. An examination of Figure 12 shows that some type of cooling would be needed with pumpages above 275°F, if API-610 limits were used. However, in specific instances, API-610 limits could be unnecessarily conservative, and other approaches may be worth considering. This can be easily done using a graph such as Figure 12. For example:

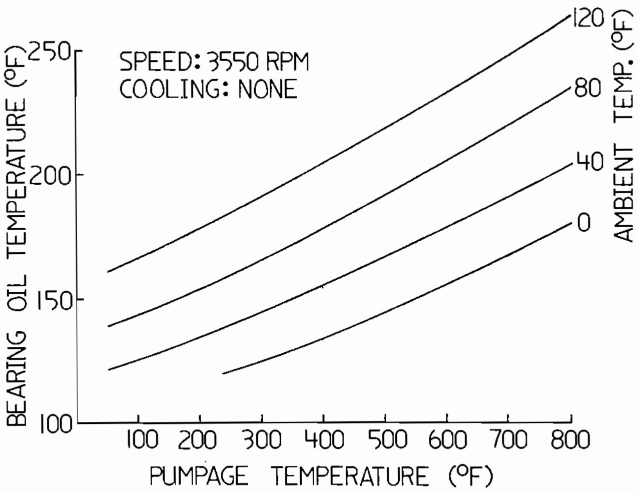

Figure 12. Oil Temperature Versus Ambient and Pumpage Temperature at 3550 RPM.

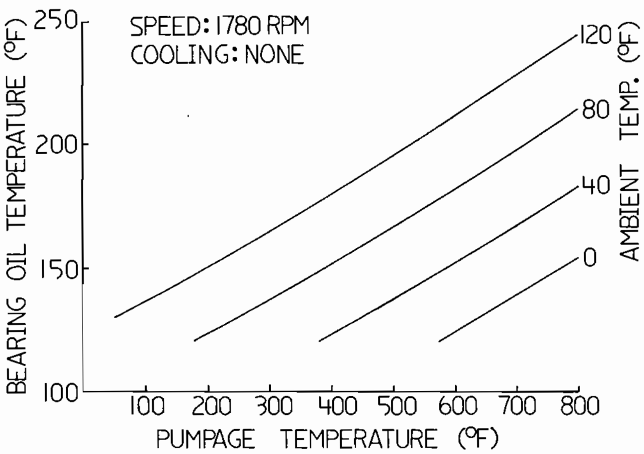

Figure 13. Oil Temperature Versus Pumpage Temperature and Cooling Flowrate.

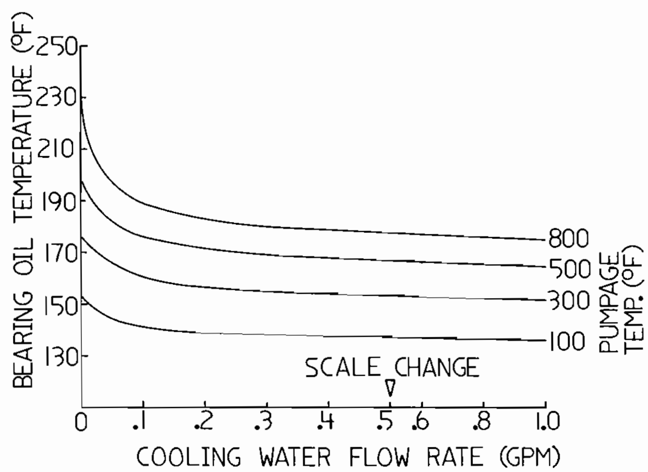

Figure 14. Oil Temperature Versus Ambient and Pumpage Temperature at 1780 RPM.

1. Pumpages up to 500°F could be handled without exceeding a lubricant temperature of 180°F if the ambient temperature didn't exceed 60°F.
2. Even with 110°F ambient temperature, pumpages of almost 650°F can be handled if lubricants are used that can withstand the 230°F operating temperature.
3. If the unit can be run at 1780 rpm, API limits can be met with pumpages up to 450°F as shown in Figure 14. With this data, actual conditions and various options can be evaluated to determine if cooling is needed.

Bearing lubricant temperature is a function of pumpage temperature and coolant flowrate as is shown in Figure 13. Once it is decided that cooling is needed, this plot can be used to determine how much is needed. In this case, a very small flow of cooling water has a significant effect on the bearing lubricant temperature as depicted in Figure 13. This is directly applicable only to a specific unit with assumed ambient and coolant temperatures, but the basic results were consistent throughout this study. In terms of reducing oil temperatures, the major benefits of cooling are obtained with a coolant flow of ¼ gpm, and flows greater than 1 gpm produce only marginal benefits. Greater coolant flowrates would be of interest only in cases where the coolant discharge temperature is a matter of concern, as would be the case when the coolant is piped in series with other equipment or is being discharged into a lake or stream. In such cases, temperature rise through the cooler should be calculated to preclude subsequent operating or legal problems.

Computer Program

Typical results are shown in Figures 12, 13 and 14 and the usefulness of such data is demonstrated. Similar plots can be made for other pump sizes to provide a portfolio of data which can be used to establish a satisfactory cooling package for many common installations. The number of graphs needed to handle all variations, however, would be unwieldy and some problems which require iterative solutions would be quite cumbersome to handle graphically. Unusual or complex problems, therefore, are best handled by calculation. Fortunately, the entire analytical model can be put on a hand held programmable calculator so such calculations can be made quickly and easily. With this tool, any combination of conditions, design options and cooling options can be analyzed to determine lubricant temperature and coolant temperature. This is particularly useful for cases such as the combined use of bearing and stuffingbox cooling, which is beyond the reach of simpler methods.

CONCLUSION

An analytical method has been developed which enables the prediction of lubricating oil temperatures and cooling requirements for a line of API-610 end suction pumps. Comparing analytically predicted lubricant temperatures to test measurements shows good agreement for an extensive range of pump sizes, speeds, ambient temperatures, pumpage temperatures and cooling arrangements. Thus, this analytical method possesses capabilities which make it a useful tool in the process of pump design and application.

This analysis procedure provides two major benefits. First, it gives the designer the ability to evaluate the thermal effects of design changes and new cooling options. This allows design refinement, while limiting the extent of expensive development testing. Secondly, graphs and a computer program based on the analysis provide an accurate means of determining whether costly cooling options are required for field installations. If cooling is required, it provides a means of evaluating and selecting the least costly of several available options. Thus, the manufacturer can make recommendations with confidence and the user can avoid unnecessary capital expenditures and maintenance costs for superfluous cooling equipment.

This analysis is currently being used on a regular basis for the line of API-610 end suction pumps involved in this study. A similar program has been undertaken for a line of chemical process pumps built to meet ANSI B73.1 requirements [11]. Results to date indicate that the general principles and procedures can be easily and accurately applied to other model lines to obtain similar capabilities.

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