Proceedings of FEDSM2005
2005 ASME Fluids Engineering Division Summer Meeting and Exhibition
June 19-23, 2005, Houston, TX, USA

FEDSM2005-77382

HIGH TEMPERATURE OVERHUNG PUMPS: COOLING OPTIMIZATION

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ABSTRACT

A typical industrial application of high temperature pumps involves handling of fluids up to 400 °C. This is critical for pump bearing housing, where thermal dissipation is not effective due to geometric configuration. Therefore, without any external cooling system, bearings and lubricating oil temperatures can exceed allowable values prescribed by both API 610 Reference Standard [1] and bearing manufacturer [2]. Particularly, for a overhung pump, when pumped fluid temperature is above 200 °C, external cooling system is necessary and water is usually used for this purpose. Consequently, water availability must be taken into account when considering pump's location, which is particularly difficult in desert areas. From these considerations was the idea to enhance the heat transfer of the pump support, in order to avoid any need of cooling water. The problem has been dealt with numerical analysis and experimental tests. First, we have considered the original support in the most critical situation, the stand-by condition, where no forced convection (fan) is effective. From the results pertaining to currently used support, we have got the hints to improve heat transfer by a full redesign. Finally an experimental validation has been set up. The measures gained allow us to validate hypothesis taken into consideration in the numerical simulation.

KEYWORDS

Heat transfer, natural convection, convection coefficient, thermal analysis.

INTRODUCTION

For a pump in stand-by condition heat transfer [3] occurs by conduction within the body of the pump and by natural convection in the surrounding air.

In heat conduction the variables that affect the process are the conductivity of the material and the temperature gradient.

In heat convection the physic of the problem is more complex, resulting in many parameters involved in the process. Therefore, we give a brief explanation of the mechanisms behind natural convection.

In convection conditions the fluid immediately adjacent to the body forms a thin slowed-down region called boundary layer. Heat is conducted into this layer, which sweeps it away.

The total heat transfer rate depends on the knowledge of convection coefficients. Although these coefficients may be obtained by solving the boundary layer equations, it is only for simple flow situations that such solutions are readily effected. The first step of this paper will be the calculation of these coefficients using empirical correlations that have been developed for common engineering geometries. The particular form of these equations is obtained by correlating measured convection heat transfer results in terms of appropriate dimensionless groups. The aforementioned coefficients will represent the boundary conditions for the numerical thermal analysis.

NOMENCLATURE

A area [m^2]
D diameter [m]
g gravitational acceleration [m/s^2]
h heat transfer coefficient [W/m^2K]
k thermal conductivity [W/m.K]
L characteristic length [m]
P perimeter [m]
Q total heat transfer rate [W]
T_f fluid temperature [K]
T_s surface temperature [K]

T_∞ free stream temperature [K]

Dimensionless parameters

Gr Grashof number
Nu Nusselt number
Pr Prandtl number
Ra Rayleigh number

Greek letters

α thermal diffusivity [m^2/s]
β coefficient of thermal expansion [K^{-1}]
μ dynamic viscosity [Kg/m.s]
ν kinematic viscosity [m^2/s]
ρ mass density [Kg/m^3]

Subscripts

L the length on which a parameter is based
D the diameter on which a parameter is based

CONVECTIVE HEAT TRANSFER

Consider the flow of a fluid at temperature T_∞ over a surface of area A at uniform temperature T_s. The total heat transfer is:

$$Q=h.(T_s- T_\infty).A \quad (1)$$

The most critical variable is the so called convection coefficient, h. The determination of h is a complicated task because it depends on the boundary layer that develops on the surface. Accordingly, h is a function of many variables such as geometry, roughness, properties of the fluid. From heat transfer theory we know that:

$$h_L=fn(k, T_s- T_\infty, L, v, \alpha, g, \beta) \quad (2)$$

Where L is a length that must be specified for a given problem. In natural convection, when fluid buoys up from a hot body or down from a cold one, h depends on the temperature difference $T_s- T_\infty=\Delta T$, typically as $\Delta T^{1/4}$ or $\Delta T^{1/3}$.

In forced convection, in which a fluid is forced past a body, h is independent of ΔT.

There are eight variables in W, m, s and °C. From the dimensional analysis we know that we have to look for 8-4= 4 pi-groups. For h and a characteristics length L the groups may be chosen as:

$$Nu_L=h.L/k \quad (3)$$
$$Pr= \nu/\alpha \quad (4)$$
$$\Pi_3=L^3/ \nu^2.g \quad (5)$$
$$\Pi_4= \beta. (T_s- T_\infty) \quad (6)$$

Nu_L is the average Nusselt number and is inversely proportional to the thickness of the thermal boundary layer. The Nusselt number is to the thermal boundary layer what the friction coefficient is to the velocity boundary layer. From knowledge of Nu_L the average heat transfer coefficient may be found.

Π_3 characterizes the importance of buoyant forces relative to viscous forces. Π_4 characterizes the thermal expansion of the fluid. For an ideal gas:

$$\beta =1/ T_\infty \quad (7)$$

In heat transfer theory it is usually used the product of Π_3 and Π4, the so called Grashof number:

$$\Pi_3.\ \Pi_4=Gr_L=g.\ \beta.\Delta T.L^3/ \nu^2 \quad (8)$$

The Grashof number plays the same role in free convection that the Reynolds number plays in forced convection.

In most situations we correlate data with the following functional equation:

$$Nu_L=fn(Gr_L, Pr) \quad (9)$$

In the dimensionless analysis is usually used the product of Gr and Pr. This is called the Rayleigh number, Ra_L.

$$Ra_L=Gr_LPr= g.\ \beta.\Delta T.L^3/ (\alpha .\nu) \quad (10)$$

Rayleigh number is normally used to highlight the transition from laminar to turbulent flow. For example, for a vertical plate the critical Rayleigh number is nearly 10^9.

Accordingly the correlations of natural convection give:

$$Nu_L=fn(Ra_L, Pr) \quad (11)$$

Generally the correlations are on the form:

$$Nu_L=C.Ra_L^n \quad (12)$$

Where:
C is a constant depending on geometry
n=1/4 for laminar flow n=1/3 for turbulent flow

It follows that for turbulent flow h is independent of L. Note that all properties are evaluated at the film temperature, $T_f=(T_s+T_\infty)/2$.

For simple geometry these equations have been obtained using an integral method. For example, in 1930, Squire and Eckert found the equation for a vertical plate:

$$Nu_L=0.678.Ra_L^{1/4}.[Pr/(0.952+Pr)]^{1/4} \quad (13)$$

Such results can be used in the case of more complex geometry. For example, in the case of the bearing support we can divide the external surface into many domains in order to apply correlations found for simpler geometry. In particular we used the results relative to vertical surfaces, horizontal surfaces (upward and downward), horizontal cylinders. A special approach is dedicated to estimate heat transfer for an array of vertical fins.

As pointed out before, many of the variables involved in convective heat transfer depend on the surface temperature T_s which is unknown in our application. This implies an iterative process. To evaluate the temperature field in our support, a first analysis run has been performed, assuming an uniform heat transfer coefficient of 10 W/m^2K. Afterwards the new coefficients have been calculated by means of the formulae shown below. The process continued until the convergence was found.

Vertical surface:

The following equation, from Churchill and Chu [4], is used to evaluate the heat transfer along the lateral surfaces of the bearing support.

$$Nu_L=0.68+0.67Ra_L^{1/4}.[1+(0.492/Pr)^{9/16})]^{-4/9} \quad (14)$$

It is valid for $Ra_L<10^9$ (laminar flow)

In our application:

$g=9.81\ m/s^2$
$\beta=1/T_\infty=1/316=0.00316\ K^{-1}$
$T_s=80\ °C=353\ K$
$T_\infty=43\ °C=316\ K$
$T_f=61\ °C=334\ K$
$L=0.2\ m$
$k=28x10^{-3}\ W/m.K$
$\alpha=26.2x10^{-6}\ m^2/s$
$\nu=18.4x10^{-6}\ m^2/s$
$Pr=0.702$

$Ra_L= g.\ \beta.\Delta T.L^3/(\alpha\ .\nu)=19.03x10^6$

The flow on the lateral surface is then laminar. The average Nusselt number can now be evaluated:

$Nu_L=0.68+0.67Ra_L^{1/4}.[1+(0.492/Pr)^{9/16})]^{-4/9}=34.6$

Hence:

$h=Nu_L.k/L=4.8\ W/m^2\ K$

Horizontal surface:(upward)

In this case the boundary layer becomes increasingly unstable as Ra is increased.

Raithby and Hollands [5] suggest:

$$Nu_{L*}=0.56.Ra_{L*}^{1/4}/[1+(0.49/Pr)^{9/16}]^{4/9} \quad (15)$$

Where $L^*=A/P$

Recommended for $1<Ra_{L*}<10^7$

Applying this correlation to the upper surface of the support, we have that:

$g=9.81\ m/s^2$
$\beta=1/T_\infty=1/316=0.00316\ K^{-1}$
$T_s=83\ °C=356\ K$
$T_\infty=43\ °C=316\ K$
$T_f=63\ °C=336\ K$
$A=0.00972\ m^2$
$P=0.404\ m$
$L^*=0.024\ m$
$k=28.8x10^{-3}\ W/m.K$
$\alpha=28.2x10^{-6}\ m^2/s$
$\nu=19.5x10^{-6}\ m^2/s$
$Pr=0.707$

$Ra_L= g.\ \beta.\Delta T.L^{*3}/(\alpha\ .\nu)=3.12x10^4$

$Nu_{L*}=5.7\ W/m^2\ K$

Horizontal surface:(downward)

For the lower side of hot plates the flow is generally stable. Fujii and Imura [6]evaluated a correlation for plates with slight inclinations, applicable also to horizontal plates:

$$Nu_L=0.58.Ra_L^{1/5} \quad (16)$$

This is valid for $10^9<Ra_L<10^{11}$

In our example:

$g=9.81\ m/s^2$
$\beta=1/T_\infty=1/316=0.00316\ K^{-1}$
$T_s=80\ °C=353\ K$
$T_\infty=43\ °C=316\ K$
$T_f=61\ °C=334\ K$
$L=0.18\ m$
$k=28x10^{-3}\ W/m.K$
$\alpha=26.2x10^{-6}\ m^2/s$
$\nu=18.4x10^{-6}\ m^2/s$
$Pr=0.702$

$Ra_L= g.\ \beta.\Delta T.L^3/(\alpha\ .\nu)=1.39x10^7$

$Nu_L=0.58Ra_L^{1/5}=15.6$

We obtain:

$h=Nu_L.k/L=2.4\ W/m^2\ K$

Horizontal cylinders:

This approximation can be applied to simulate heat transfer along the region of the shaft exposed to air. Churchill and Chu [7] provide this correlation:

$$Nu_D=0.36+0.518Ra_D^{1/4}/[1+(0.559/Pr)^{9/16}]^{-4/9} \quad (17)$$

This equation is valid in the range $10^{-6}<Ra_D<10^9$

In our application we have the following parameters:

$g=9.81\ m/s^2$
$\beta=1/T_\infty=1/316=0.00316\ K^{-1}$
$T_s=150\ °C=423\ K$
$T_\infty=43\ °C=316\ K$
$T_f=96\ °C=369\ K$

$D = 0.04$ m
$k = 31\times10^{-3}$ W/(mK)
$\alpha = 33.3\times10^{-6}$ m^2/s
$\nu = 23.5\times10^{-6}$ m^2/s
$Pr = 0.706$

$$Ra_D = g.\,\beta.\Delta T.D^3/\,(\alpha\,.\nu) = 2.71\times10^5$$

$$Nu_D = 0.36 + 0.518 Ra_D^{1/4}/[1+(0.559/Pr)^{9/16}]^{-4/9} = 9.3$$

$$h_D = Nu_D.k/D = 7.2 \text{ W/m}^2\text{ K}$$

<u>Finned deflector:</u>

To increase heat transfer from the pump shaft, a finned deflector has been designed, which is to be fitted on the shaft.

The geometry of this dissipator consists of many round plates closely spaced (Fig. 1):

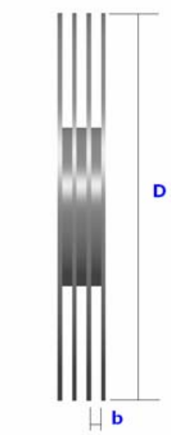


Fig. 1

If the plates are spaced far enough apart, the natural convection boundary layers that develop along the surface of the plates will not interfere with each other. In this case, the heat transfer coefficients would be the same as for a single flat plate. However, if the heated plates are moved closer together, the boundary layers on adjacent plates will merge. As a consequence the natural heat transfer coefficients will fall below the single plate value. Once the length of the deflector has been set, if the number of fins increases, the total heat transfer area also increases while the value of h decreases. Therefore, there will be an optimum value for the spacing between plates.

To model heat transfer in these conditions, we can use the results obtained for rectangular fins. Elenbaas [8] established that the optimum spacing between the plates in air could be obtained by setting the channel Rayleigh number (based upon plate spacing b and channel aspect ratio b/D) equal to 50:

$$Ra_{b\ b/D} = (b/D).\ g.\ \beta.\Delta T.b^3/\,(\alpha\,.\nu) = 50 \qquad (18)$$

$$b_{opt} = 2.659.[(D.\alpha.\nu)/\,(g.\beta.\Delta T)]^{1/4} \qquad (19)$$

In our deflector the variables are as follow:

$g = 9.81$ m/s^2
$\beta = 1/\,T_\infty = 1/316 = 0.00316$ K^{-1}
$T_s = 105°C = 378$ K
$T_\infty = 43$ °C $= 316$ K
$T_f = 63$ °C $= 347$ K
$D = 0.135$ m
$k = 29.8\times10^{-3}$ W/mK
$\alpha = 29.3\times10^{-6}$ m^2/s
$\nu = 20.7\times10^{-6}$ m^2/s

We obtain:

$$b_{opt} = 0.0068 \text{ m} = 6.8 \text{ mm}$$

We approximated this value to 7 mm. We can now evaluate the heat transfer coefficient using the Elenbaas correlation:

$$Nu_b = 0.0417.Ra.[1-e^{-(35/Ra)}]^{3/4} = 1.25 \qquad (20)$$

$$h_b = Nu_b.k/b = 5.3 \text{ W/m}^2\text{ K}$$

We do not report the calculation of the heat transfer coefficients for other parts of the support, such as arms, flanges, etc. They have been evaluated by means of the before mentioned formulae.
All the coefficients that we have evaluated so far constitute the boundary conditions for the thermal analysis.

NUMERICAL ANALYSIS

Figure 2 shows the pump taken into account in this paper. The critical elements for the thermal transmission are the contact surface between the support and the cover, the shape and the length of the support arms. So, we focused our attention on these elements.

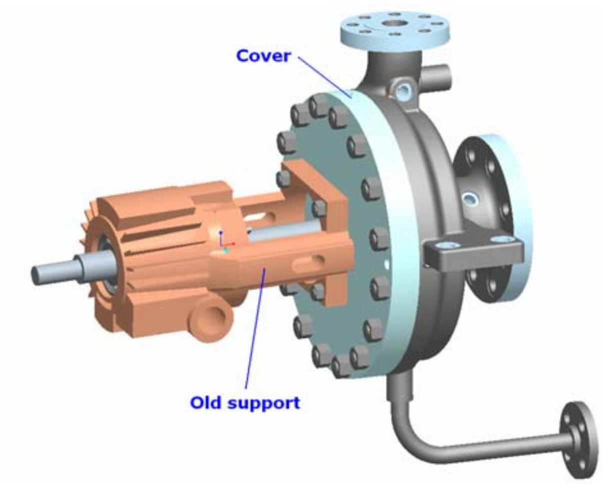


Fig. 2

The original support (Fig. 3) has two arms, set at the centerline of the bearing; moreover, the flange is fully in contact with the pump cover.

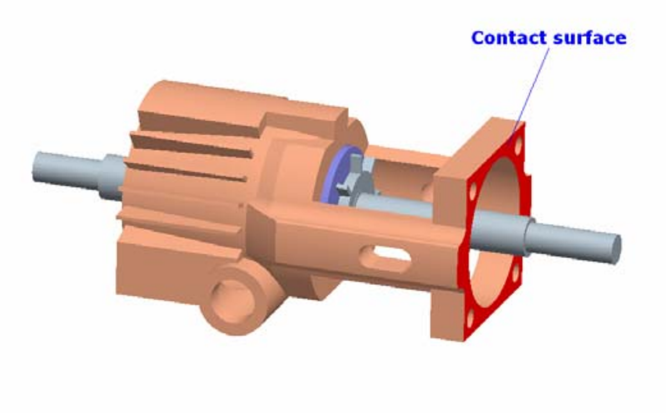


Fig. 3

To reduce the heat flux from the pump casing we decreased the surface in direct contact with the casing. Next, the support arms have been properly lengthened to take advantage of thermal gradient and at the same time they have been moved upward.

The last modification, concerned the bearing housing: we introduced a jacket to drastically reduce the thermal flux from the arms. Figure 4 shows the new support.

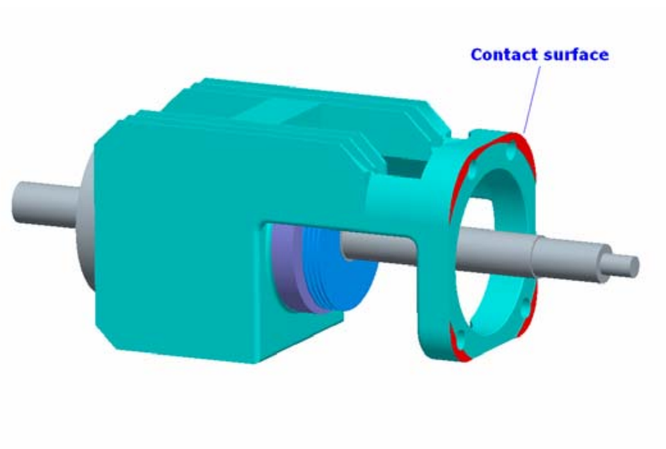


Fig. 4

The thermal analysis has been performed by means of the finite element method [9]. We have evaluated the temperature distribution, in steady state conditions, when a prescribed temperature is applied to the frontal surface of the support. The dissipation taken into account is the natural convection, simulating the worst working condition. Hereby we list the characteristics of the numerical model.

Materials

The following table shows the materials taken into account. Appropriate thermal properties have been considered [10-12].

Element	Common name	ASME	UNS
Support	Carbon Steel	SA-216 WCB	J03002
Shaft	INOX steel	SA-479 410	S41000
Bearings	Cr steel	-	-
Covers	Carbon Steel	SA-516 Gr 70	K02700
Finned deflector	Aluminum	SB-221	A93003

Geometry

We modelled the support, the shaft, bearings, covers and the finned deflector. To simplify the model, we neglected all the geometric features that do not affect the thermal behaviour (rounds, chamfers, etc.)

Results

Let's consider the results corresponding to a pumped fluid temperature of 380 °C. The ambient temperature is set to 43 °C (as prescribed by API 610 reference standard).

Figure 5 shows the temperature contour for the whole support.

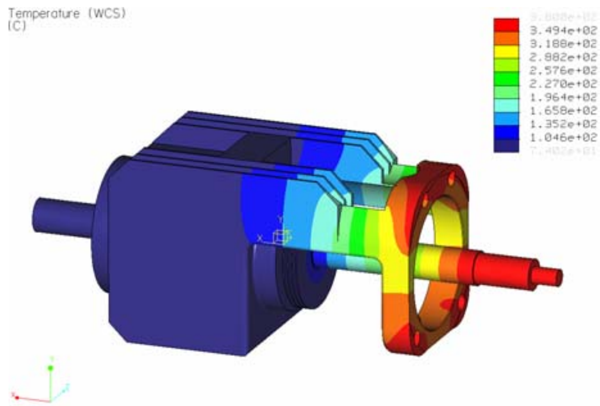


Fig. 5

In figure 6 it is possible to see a longitudinal section, where it is clearly visible the bearing housing configuration (legend levels have been changed to emphasize temperature distribution in bearing housing).

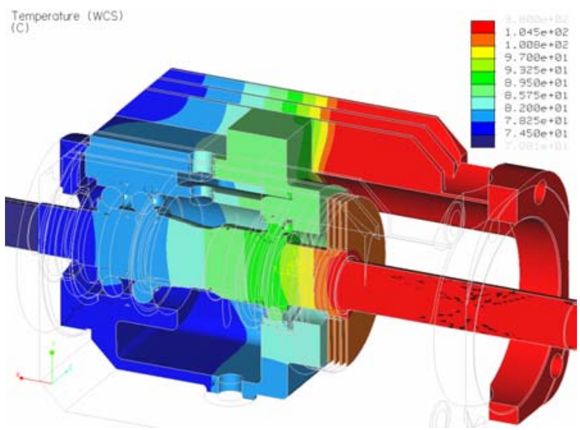


Fig. 6

Maximum temperature in the oil reservoir is about 82 °C. This is the maximum value permitted by API 610 reference standard (par. 5.10.2.4). Figure 7 shows the temperature distribution of the radial bearing: the maximum temperature, 93 °C, is lower than the recommended value (110 °C). The difference between outer and internal races is nearly 10 °C.

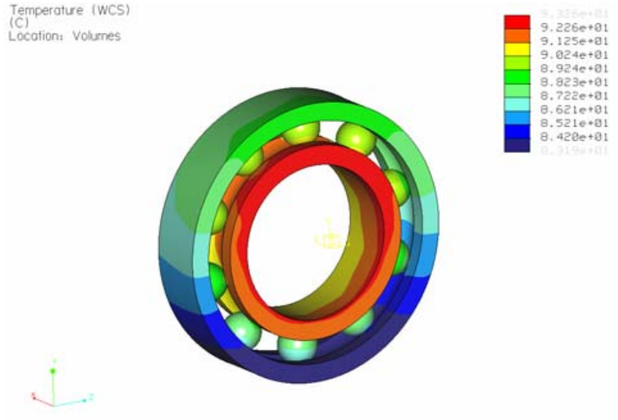


Fig. 7

The table summarizes the numerical results:

Tmax oil [°C]		Tmax bearing [°C]	
Old support	New support	Old support	New Support
97	82	121	93

EXPERIMENTAL TESTS

To validate the results found with numerical analysis, we set up an experimental campaign. The support has been heated to a temperature of 380 °C (fig. 8).

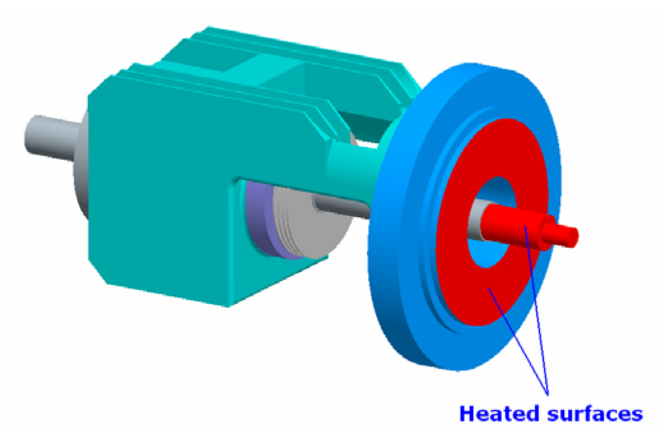


Fig. 8

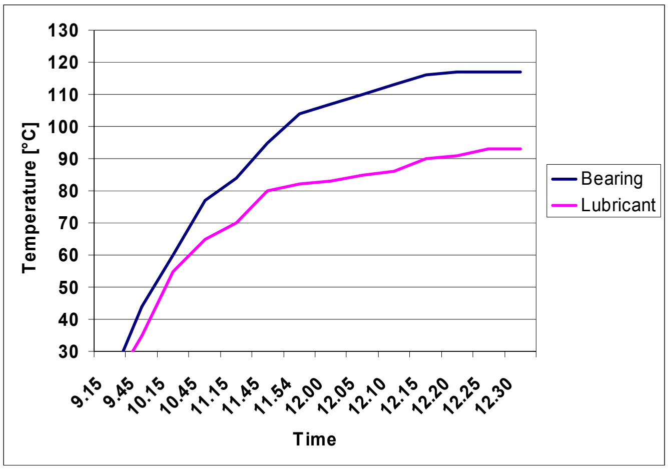
We kept this condition for a time long enough to have steady conditions in all the support. We monitored the temperature across the arms, along the shaft and also of the line bearing and its lubricant.

After three hours we reached a steady state conditions: the bearing temperature was nearly 90 °C, the oil in the reservoir was about 76 °C. We plot the results corresponding to a boundary condition of a temperature fluid of 380 °C and an ambient temperature of 30 °C.

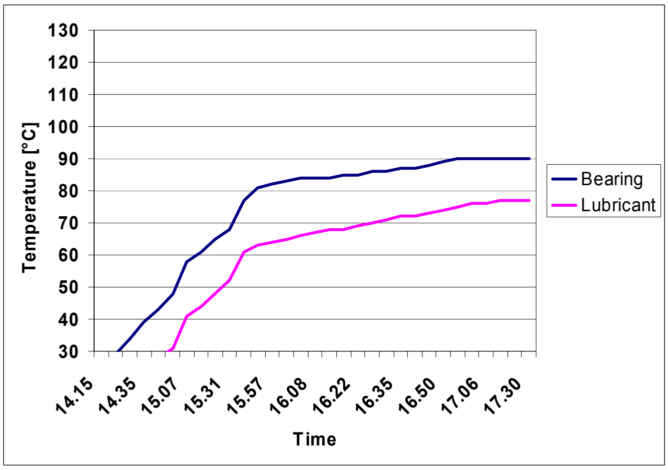
We report the experimental results for both the old and the new support.

Operative conditions: T_{fluid}= 380 °C T_{amb}= 30 °C

Old Support:



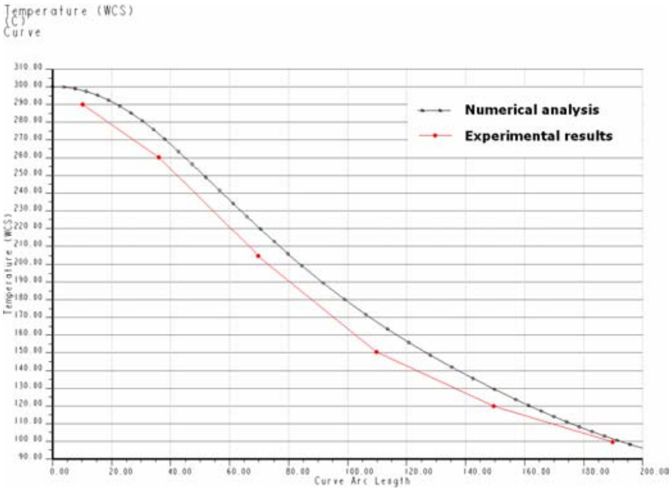
New support:



The table summarizes the experimental results:

Tmax oil [°C]		Tmax bearing [°C]	
Old support	New support	Old support	New Support
93	76	117	90

To compare the experimental results with the numerical one, we run the analysis with ambient temperature of 30 °C. The graph in the next page shows the temperature along the arms, both for the numerical and the experimental results.



From this graph we can see that the temperature evaluated from the numerical analysis is higher than the temperature measured on the prototype (the maximum difference is about 10 %).

The following table reports the comparison for the oil/bearing temperatures: the numerical model overestimates by a 1-2 % the temperatures measured.

New support			
Tmax oil [°C]		Tmax bearing [°C]	
Numerical analysis	Experimental results	Numerical analysis	Experimental results
78	76	91	90

SUMMARY AND CONCLUSIONS

This paper has dealt with the re-design of a bearing support of an overhung pump, with the main purpose of increasing the thermal performances.

We showed how with a design aimed to optimize the heat transfer, it is possible to increase considerably the thermal dissipation. For the support examined in this paper the maximum pumped fluid temperature raised from 200 °C to 380 °C. The temperature range obtained for the support considered in this paper can be granted to all the overhung pump line.

The design stage took remarkable advantage from the simulation tools, that allowed us to minimize the number of prototypes realized in the pre-production period.

Experimental tests, carried on in our test room, confirmed the quality of the numerical model, showing a good match between the temperature evaluated by the software and the temperature measured on the prototype.

ACKNOWLEDGMENTS

I would like to thank all the people in Pompe Gabbioneta's organization that gave me their support in doing this work.

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