APPLICATION OF FINNED HOUSINGS IN HORIZONTAL GEARBOXES WITH SPLASH LUBRICATION

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Abstract. *Part of the power transmitted by a set of gears is transformed in heat. The maximum mechanical power which can be applied to a gearbox keeping its oil temperature under the maximum allowable temperature is one of the most important parameters when dimensioning a gear box unit. In some cases, the heat transferred by the gearbox housing is not enough to keep this temperature under the desired limit. Thus, it is necessary to use an additional cooling system like fins (around the housing), fans, cooling coil, heat exchangers and others. This paper presents a thermal analysis of gearbox finned housings in order to obtain the optimal fin size. The gearbox is experimentally analyzed to determine the influence of fins on the heat transfer. Thermocouples are mounted on the housing to obtain the temperature in several points. Furthermore, a finite difference procedure is developed to predict the temperature at the fins. Numerical results for the fin efficiency, optimized fins length and the optimal gap between the fins are computed. The experimental and numerical results for the gearboxes systems analyzed in the present study is very important to help technicians and engineers to project and manufacture a more effective gearbox housing.*

Keywords*: Heat Transfer, Gearbox Finned Housing, Finite Difference Method, Thermal Measurements.*

1. INTRODUCTION

The development of new materials brings evolution to gearboxes designs, mainly in its weight-power ratio. With the gearbox area reduction, the maximum mechanical power which can be applied to a gearbox keeping its oil temperature under the maximum allowable temperature became lower.

The oil temperature from a gearbox during its work has to be stabilized in an equal or inferior value to a considered acceptable limit. This limit can be defined by the application or be defined by the customer. In general, it is the maximum temperature that internal components as bearings and seals reach without breaking down or without big changes in oil characteristics happening (Flender GmbH, 1985) and a certain value to guarantee safety for the people around of the equipment.

However, in some cases, the thermal dissipation by the superficial area of the housing is not enough. Thus, it is necessary to use an additional cooling system like fans, cooling coil, heat exchangers and others. A general alternative applied for increasing the thermal exchange with the environment is the use of fins on the housing.

The main focus of this work is the use of vertical two-dimensional channels formed by parallel plates (or fins) around the housing in gearboxes and give information on using them in different power and environmental conditions using technical and scientific information to determine the dimensions and proper installation of the fins in the gearboxes housings.

The fins length, gap between each other and the formula to get the new capacity of mechanical power with the use of finned housings is presented in this work.

This study is of high priority for the company Flender Brasil Ltda. when considering a criterion for the evaluation of the fins application in its gearboxes. Even though the finned housings are frequently used by the company to increase the thermal exchange, there is no study about fins sizes and their position in the housings to reach the optimized design.

2. COOLING THEORY

2.1. Determination of the maximum mechanical power by the thermal dissipation

The oil gearbox has to operate in a temperature below its limits, and for this reason it is necessary to know the maximum mechanical power in order to analyze the wear and other internal mechanical losses. These mechanical losses become heat and have to be dissipated by the gearbox housing. The heat can be calculated by the energy balance between the mechanical losses and the thermal power to be dissipated by the housing.

The mechanical losses, P_V, is obtained on the basis of equation 2.1:

$$P_V = P_1(1-\eta) \tag{2.1}$$

where: P_V = mechanical losses [W]
P_1 = mechanical power [W]
η = power factor

The mechanical power losses become heat and this is the energy $\dot{Q}$ to be dissipated by the gearbox housing and defined by the equation 2.2:

$$\dot{Q} = U.A.\Delta T \tag{2.2}$$

where: U = medium global convective factor [W/m²·K]
A = gearbox housing superficial área [m²]
ΔT = temperature difference between oil and environment [K]

The thermal equilibrium is given by the equations 2.3 to 2.5:

$$P_V = \dot{Q} \tag{2.3}$$

$$P_1.(1-\eta) = U.A.\Delta T \tag{2.4}$$

$$P_1 = \frac{U.A.\Delta T}{(1-\eta)} \tag{2.5}$$

The maximum mechanical power depends on the difference of temperature between the oil and external air, the convective global average coefficient of heat transference, the gearbox housing superficial area and its power factor. However, it is not possible to increase these variables indiscriminately. They are limited by the environment, mechanical factors or by physical space.

2.2. Increase of gearbox housing cooling

Part of the mechanical power is transformed in heat by the wear from the rotative parts and from the contact between oil and gears. This heat has to be dissipated by the gearbox housing to maintain the oil temperature inside its acceptable limits. If the oil temperature exceeds these limits its viscosity can become very low or the oil can even deteriorate and lubrication fails can happen, with mechanical damages to the machine. Other internal components can also be affected because of operating in high temperatures. Rolling bearings and seals could have their lifespan reduced due to lack of lubricant film, caused by the low oil viscosity or by the material that does not handle high variations of temperature. In special cases, the customer can also define the maximum housing temperature in values that will not bring the risk of burning or damage to other machines.

Methods for increasing the thermal exchange of the gearbox with the environment are presented below.

Radial or axial fans in the axles of the high rotation gearboxes are normally used to improve the thermal exchange between the housing and the environment, because it increases the air circulation around the machine, with the increase of the average convection coefficient. Coils can also be used with the same purpose. Ducts are installed in the gearbox interior where a fluid (generally water) flows, a temperature lower than the oil enters and heat is removed from the gearbox. In critical cases, heat exchangers can be used to control the oil temperature. In this case, hot and cold fluid can move in the same direction, opposite direction or in a draining crossed in a construction of pipes inside a shell.

Sometimes it is not possible to use active cooling in the machine or mechanical systems and the thermal exchange can only happen by the natural convection with the environment. It can happen because of the following reasons: physical space limit, impossibility to install fans next to the gearbox (as fine grains industries or dust), non-availability of water (or other fluids) for the coils or thermal exchangers, or because of the customer's project. In natural convection the cooling happens due to fluid motion caused by the density variations, resulting from temperature distributions in the case of heat transfer, which happen because of local gravitational and centrifugal forces (Burmeister, 1993). Another alternative, very used and efficient, if well projected, relates to the fins application on the gearboxes housings. The fins are a frequently encountered configuration in natural convection cooling in air (Bar-Cohen and Rohsenow, 1984). Small plates are welded in one or to the two parts of gearboxes housings (upper and lower housing) vertically or horizontally. The position of these plates (fins) depends on the direction of air flow on the gearbox. They are added to the housing to increase its superficial area to improve the thermal exchange with the environment. The figure 2.1 shows a typical gearbox with its upper and lower housings and the fins in the lower housing part.

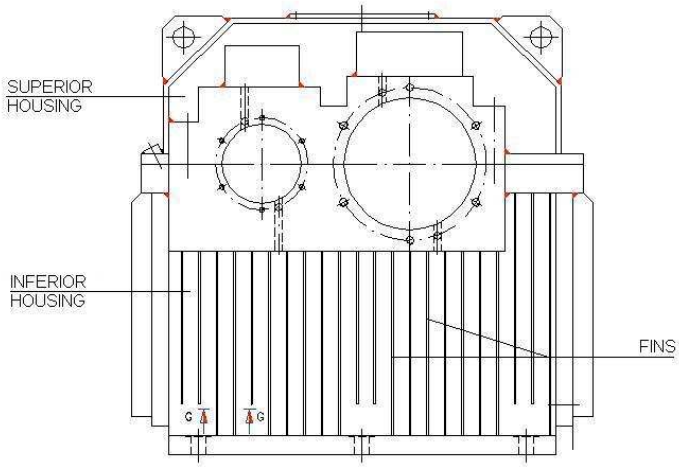

Figure 2.1 – Externally finned gearbox housing with vertical rectangular fins.

3. MATHEMATICAL MODEL

3.1. Initial considerations

Some analysis and studies of adjacent subjects as temperature calculation and oil thermal distribution inside the gearbox, effect of the splash oil to the gearbox housing due the axles and gears rotation, and others had not been considered for going beyond the main objective of this paper.

The oil temperature is calculated using the gearbox efficiency, a global convective coefficient, the gearbox superficial area and the outside temperature. A part of the mechanical power transmitted by the engine is lost by the wear between the rotative parts and the movement of these parts inside the oil. The other part from mechanical power is used to give power for the machine connected to the gearbox. This lost energy is transferred to the oil as heat and has to be dissipated by the gearbox housing. The oil temperature is used to define the installation and dimension of the fins in the gearbox housing.

The temperature drop due to housing and fin thickness are not considered. The Biot number (Incropera and de Witt, 1998) relates the heat transfer resistance inside and at the surface of a body. Using average values of housing and fins dimensions is possible to reckon the Biot number magnitude in some housing regions and make important considerations. Very low Biot numbers (<<1) are observed in the housing thickness and in the fin section, not allowing to make any account of the temperature along of housing thickness. Then, the fin and the internal housing surface temperature are the same. In a new consideration the oil temperature from inside housing is considered equal to the internal surface temperature, as showed in Figure 3.1. However, along the fins (*x* axis) the Biot number is considerable (>1) and the fin temperature distribution is determined by a numerical method that is presented in this paper.

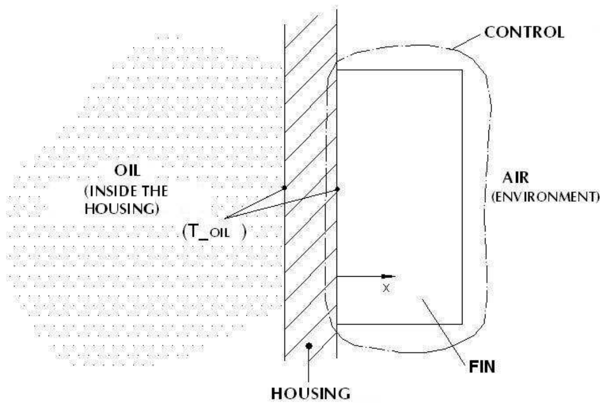

Figure 3.1 – Analyzed control volume in one fin.

The splash lubrication is considered in the gearbox analysis. This lubrication type happens with part of the shafts and gears dip into troughs of oil, splashing the oil onto the other parts of the gears and bearings. The oil level must be in the half of the highest bearing in the gearbox. Practically all the housing lower part is in contact with the oil and the upper part is in contact with air and splash oil. As in this upper part the oil is not in constant contact with the housing as the lower part, the coefficient of heat transference between the oil and the inferior housing is much higher than the lower part. In this paper it is studied only the lower externally finned housing, where the temperature is higher than the upper part.

3.2. Numerical Treatment

The basic idea of any numerical method consists on transforming a continuous domain into a discrete problem with a finite number of nodal points (Figure 3.2) and the solution is gotten by an algebraic equations system. A frequently used and readily applicable method in numerical solution is the method of finite-differences approximation of the partial derivates. The method is considered approximate in that the derivative at a given point is represented by a derivative taken over a finite interval across the point (Özişik, 2002).

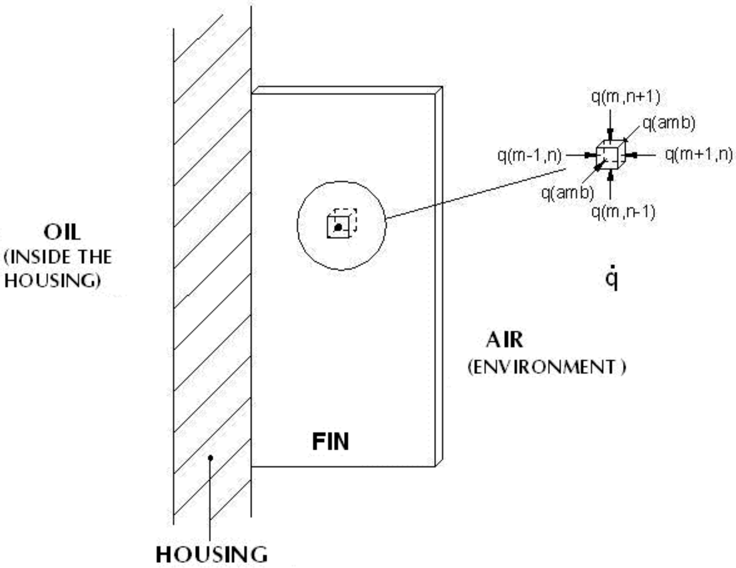

Figure 3.2 – Notation for the mesh used in the finite differences approach.

The numerical determination of the temperature distributions demands an equation for the energy conservation written in the appropriate form for each one of the nodal points with unknown temperature. The resultant set of equations must simultaneously be calculated for the temperature in each node.

For any interior node in a bidimensional system without internal heat generation and with uniform thermal conductivity, the accurate form of the energy conservation requirement is represented by equation 3.1. However as the system it is characterized by a nodal net is necessary to work with an approached form or in finite differences of this equation.

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}=0 \tag{3.1}$$

With energy balance around the nodal region it is possible to determine the equations in finite differences. As the real direction of the thermal flow is unknown, it is convenient to formulate the energy balance considering that all the thermal flows are directed into the node. The directions are determined correctly if the heat transference equations are coherent with this consideration. As a condition of stationary state, the sum of all heat flux entering in the node will be equal to the zero.

This energy balance can be considered as a control volume around the interior node (*m,n*) showed in Figure 3.5. For these conditions, as it is adopted in this paper, the energy exchange is influenced by the heat conduction between the node (*m,n*) and its four adjacent nodes.

To analyze the conduction part in the system it is considered that the conduction heat transfer happen only along the bands that are guided in directions *x* or *y*. Then, simplified forms of Fourier law can be used (Burmeister, 1993). As an example, the rate of heat conduction from the node (*m-1, n*) to the node (*m,n*) can be expressed as in Equation 3.2 below:

$$q_{(m-1,n)\to(m,n)}=k(\Delta y.1)\frac{T_{m-1,n}-T_{m,n}}{\Delta x} \tag{3.2}$$

The heat transfer area is the *(Δy.1)* and $(T_{m-1,n} - T_{m,n}) / \Delta y$ is the finite differences approach of the temperature gradient in the border between the two nodes. The remaining rates of conduction can be written as following (Equations 3.3 to 3.5):

$$q_{(m+1,n)\to(m,n)}=k(\Delta y.1)\frac{T_{m+1,n}-T_{m,n}}{\Delta x} \tag{3.3}$$

$$q_{(m,n+1)\to(m,n)} = k(\Delta x.1)\frac{T_{m,n+1}-T_{m,n}}{\Delta y} \quad (3.4)$$

$$q_{(m,n-1)\to(m,n)} = k(\Delta x.1)\frac{T_{m,n-1}-T_{m,n}}{\Delta y} \quad (3.5)$$

Solving the energy balance equations, the finite difference equation for an interior node results are as expressed in Equation 3.6:

$$T_{m,n+1} + T_{m,n-1} + T_{m+1,n} + T_{m-1,n} - 4T_{m,n} = 0 \quad (3.6)$$

For each nodal point with unknown temperature it is necessary an equation in finite differences. However, in this case there are not only interior nodes, but external points to the conditions of heat transfer by convection. For these points the finite differences equation can also, in an analogous form, be obtained by the energy balance method.

The fins are divided in a nodal mesh and each one of its unknown temperature nodes generates a finite differences equation. The set of these equations is solved by a numerical method presented in the next section.

3.3. Maximum gearbox power by the thermal dissipation

The heat transformed by part of total mechanical power must be dissipated by the housing total area (fins superficial area and the housing non-finned region) to reach a thermal balance in the gearbox and for the oil temperature to maintain its acceptable limits.

The finned housing external area, A_{total}, is defined by Equation 3.7:

$$\mathrm{A_{total}} = \mathrm{A_{wo\ fins}} - \mathrm{N}\cdot\mathrm{A_{fs}} + \mathrm{N}\cdot\mathrm{A_{l}} \quad (3.7)$$

where:

$A_{wo\ fins}$ = Gearbox original area without fins	[m²]
A_{fs} = Fin front section area	[m²]
A_l = Fin lateral area	[m²]
N = Number of fins	[m²]

Then, the maximum mechanical power can be rewritten as Equation 3.8:

$$P_1 = \frac{U.A_{total}.\Delta T}{(1-\eta)} \quad (3.8)$$

This equation reveals the maximum mechanical power to be transmitted by the gearbox where the heat generated by the mechanical losses is balanced with the maximum thermal dissipation of finned housing. To determine this maximum mechanical power, the maximum oil temperature, T_{oil} (normally 95°C), the ambient temperature, T_{amb}, and the new gearbox total area with the fins have to be known.

3.4. Optimized gap between the fins

To define the optimized gap between the fins, $S_{optimized}$, some correlations are used to maximize the heat transference in a series of isothermal plates. The existence of an optimal value for the series results of the fact that, even so the coefficient of heat exchange in each fin is lower with the reduction of the gap, S, the number of wings that can be placed in a specific volume increase. Then, $S_{optimized}$ maximizes the heat transference in the series supplying a maximum value of average U and A_{total}. For fins with uniform thermal flow, the total volumetric tax of heat transference increases simply with the reduction of S. However, is necessary to keep the surface temperature below the established limits and is not possible to reduce S to extremely small values. Thus, $S_{optimized}$ can be defined as the value of S that supplies the maximum volumetric heat transfer for unit of difference of temperatures (in gearbox, the difference between the surface temperature and the ambient temperature).

The optimized gap between the fins, $S_{optimized}$, it is a function of the parameter P (Equation 3.9) and is defined by Equation 3.10:

$$P = \frac{c_p.\rho^2.g.\beta.\Delta T}{\mu.k.L} \quad (3.9)$$

where: c_p = specific heat capacity of air [J/kg·K]
ρ = air density [kg/m³]
g = gravity acceleration [m/s²]
β = thermal expansion coefficient of air [1/K]
μ = kinematics viscosity of air [N·s/m²]
k = thermal conductivity of air [W/m·K]
L = average length [m]
ΔT = difference between the surface temperature and the ambient temperature[K]

$$S_{optimized} = \frac{2.7}{P^{0.25}} \tag{3.10}$$

4. NUMERICAL ANALYSIS

For numerical analysis equations, the Figure 4.1 shows a fin with its nodes and the nodes' equations from Equation 4.1 to 4.9:

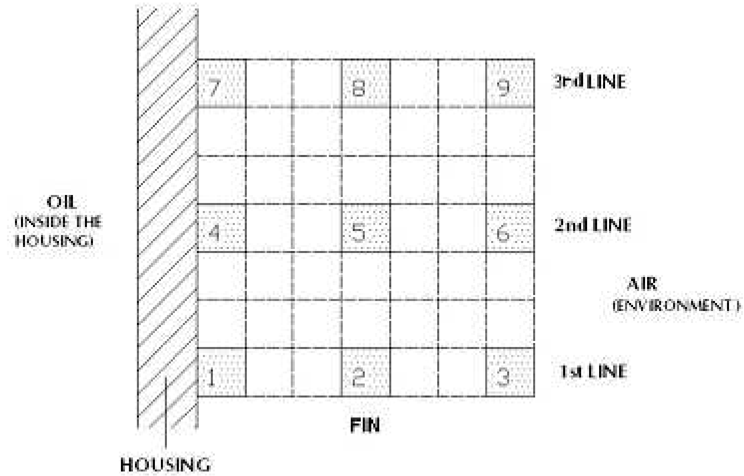

Figure 4.1 – Fin detail with division by nodes.

First Line:

$$\text{Node (1)} - (2.T_{forward} + T_{upper}) - 2\left(\frac{h.dw}{k} + 2\right)T_{node} = \frac{-2.h.dw}{k}.T_{\infty} - T_{base} \tag{4.1}$$

$$\text{Node (2)} - T_{upper} + T_{forward} + T_{backward} - 4.T_{node} = -T_{base} \tag{4.2}$$

$$\text{Node (3)} - (2.T_{backward} + T_{upper}) - 2\left(\frac{h.dw}{k} + 2\right)T_{node} = \frac{-2.h.dw}{k}.T_{\infty} - T_{base} \tag{4.3}$$

Center lines (from the second line until the penultimate line):

$$\text{Node (4)} - (2.T_{forward} + T_{upper} + T_{lower}) - 2\left(\frac{h.dw}{k} + 2\right)T_{node} = \frac{-2.h.dw}{k}.T_{\infty} \tag{4.4}$$

$$\text{Node (5)} - T_{upper} + T_{lower} + T_{forward} + T_{backward} - 4.T_{node} = 0 \tag{4.5}$$

$$\text{Node (6)} - (2.T_{backward} + T_{upper} + T_{lower}) - 2\left(\frac{h.dw}{k} + 2\right)T_{node} = \frac{-2.h.dw}{k}.T_{\infty} \tag{4.6}$$

Last line:

$$\text{Node (7)} - (.T_{forward} + T_{lower}) - 2\left(\frac{h.dw}{k} + 1\right)T_{node} = \frac{-2.h.dw}{k}.T_{\infty} \tag{4.7}$$

$$\text{Node (8)} - 2.T_{lower} + T_{forward} + T_{backward} - 2\left(\frac{h.dw}{k} + 2\right)T_{node} = -2\left(\frac{h.dw}{k} + 1\right)T_{\infty} \tag{4.8}$$

$$\text{Node (9)} - (T_{backward} + T_{lower}) - 2\left(\frac{h.dw}{k} + 1\right)T_{node} = \frac{-2.h.dw}{k}.T_{\infty} \tag{4.9}$$

In this numerical modeling, the finite differences equations set is transformed in a coefficients matrix and independent terms and the result of these matrices is used in computational algorhytm, following the equations below:

$$[A]x[T]=[C] \tag{4.10}$$

$$[T]=[C]x[A]^{-1} \tag{4.11}$$

where: $[T]$ = Temperature matrix in each node
$[C]$ = Independent terms matrix
$[A]^{-1}$ = Coefficients T inverse matrix

5. RESULTS

5.1. Experimental Results

For the mathematical model analysis and the used methodology, some measurements in a Flender gearbox had been carried out. The model used is SDN-250 with finned housing.

The temperatures measurements were taken in Flender Brasil Ltda during the equipment test without output load using the instructions catalogue (Flender Brasil, 1994). During approximately 2 hours the equipment was functioning until the temperature reached a stability point. A bimetallic thermometer was used to take the temperature because of its good precision converting a temperature change into mechanical displacement (Chids, 2001). The thermometer has the precision of 0.5°C. Some points in the fin had been defined and measurements were taken keeping the thermometer to a constant distance of 15 cm from the measurement point.

5.2 Theoretical Results

The experimental values and numerical values were compared. The input data in the numerical method (Table 5.1) had been approached to the maximum of the real conditions of the environment and the fins real dimensions used in the analysis. This comparison has as an objective of validating the adopted model.

Table 5.1 – Input dada for the numerical model

Model	SDN-250 with finned housing
T_{amb}	27º C
T_{oil}	45º C
Housing material	SAE 1020 (k = 51,9 W/m·ºC)
Convective coefficient (*U*)	17.5 W/m²·ºC (Flender Brasil, 1994)
Fin thickness (*E*)	0.0032 m
Height (*H*)	0.0381 m
Average Length (*L*)	0.255 m
Original Superficial area	1.87 m²
Power Factor	96.6 %

Figure 5.1 shows the measurement points from the most internal point of the fin (1) to the most external point (9) in the horizontal direction.

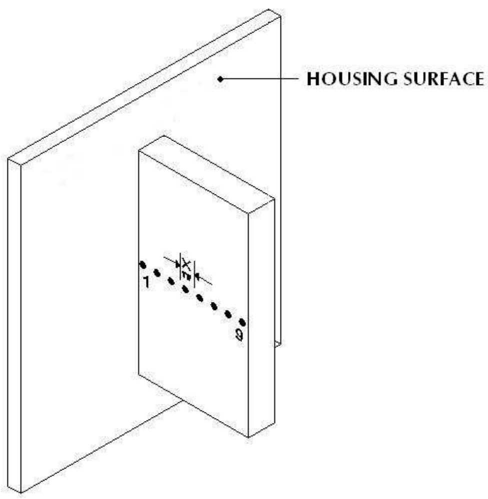

Figure 5.1 – Fin measurement points.

The Table 5.2 shows the experimental values compared to the theoretical results from the mathematical model.

Table 5.2 – Comparative values experimentally obtained and by the mathematical model

Point	Experimental Values [ºC]	Numerical Values [ºC]
1	45,5	45,0
2	44,5	44,5
3	44,0	44,1
4	43,5	43,7
5	43,0	43,4
6	42,5	43,0
7	42,5	42,9
8	42,5	42,7
9	42,0	42,6

In Figure 5.2 is observed a good agreement between the experimental and theoretical results in the two cases:

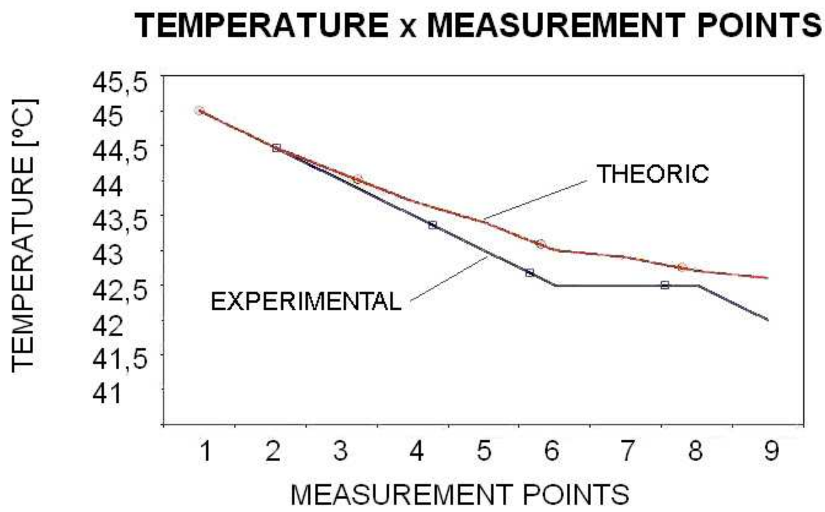

Figure 5.2 – Comparative graph between experimental and theoretical values

The Figure 5.3 shows clearly the temperature reduction in the fin as measurement is taken farther from the gearbox housing it is. These thermal results are very important to estimate the thermal gradient and to define the fin length.

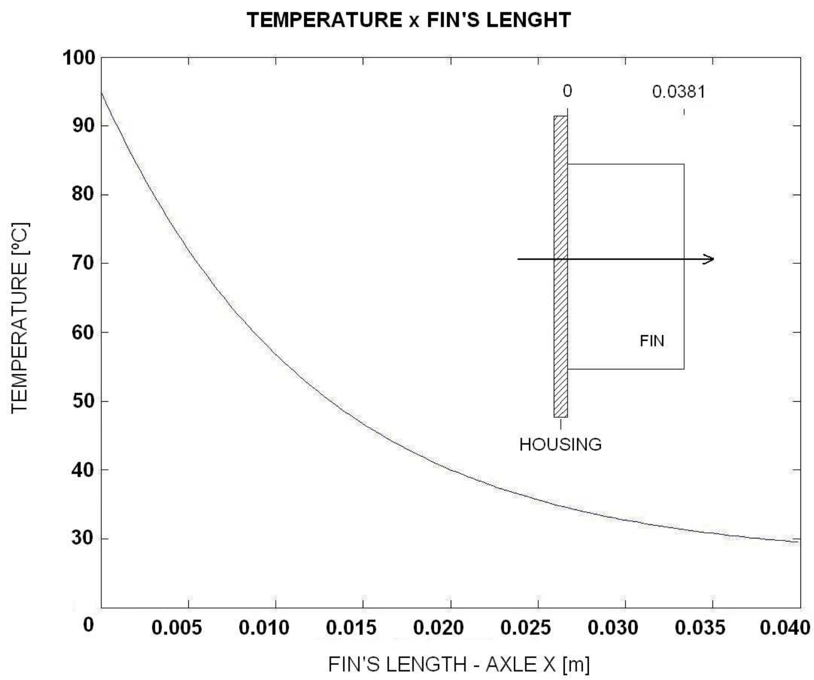

Figure 5.3 – Temperature variation in function of the fin length

In a new analysis, in the vertical direction (along the fin height axis), the thermal behavior obtained from the mathematical model is showed in Figure 5.4:

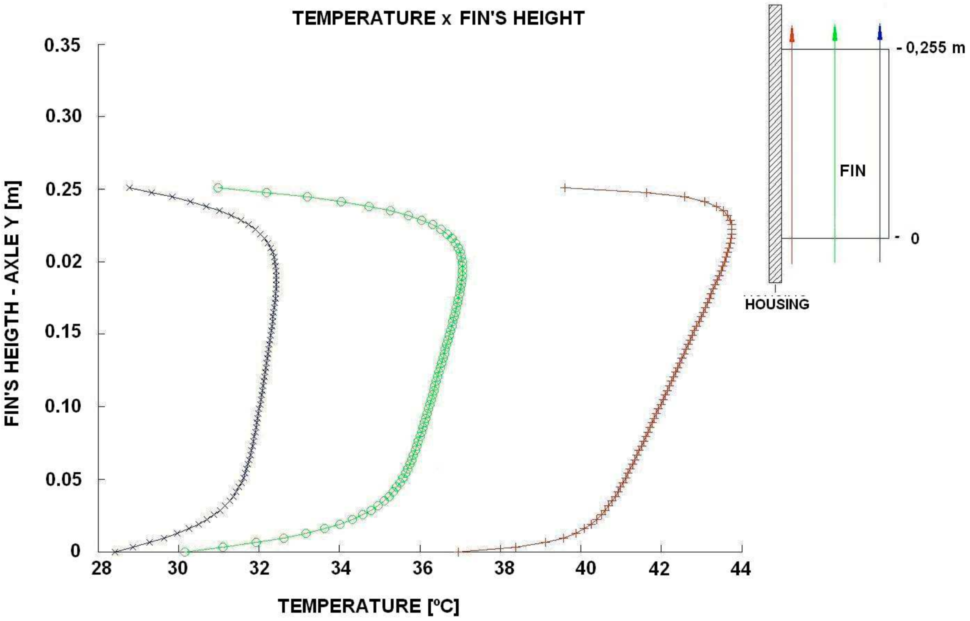

Figure 5.4 – Variation of the temperature in function of the fin height

The air thermo-physical properties to calculate the parameter P to obtain the optimized gap between the fins (Equation 3.9) are gotten from tables presented by Incropera and Witt (1998), and shown below:

$$c_p = 1.0103\ kJ/kg.K;\ \rho = 1.3737\ Kg/m^3;\ \mu = 1.8264\,\mathrm{x}10^{-5}\ N.s/m^2;\ k = 0.0623\ W/m.K\ .$$

The gravity acceleration, g, and average height, L (taken as the average height of the inferior housing) are:

$$g = 9.8\ m/s^2;\ L = 0.255\ m$$

The thermal expansion coefficient for ideal gas, T^{-1} is:

$$\beta = \left(\frac{300+368}{2}\right)^{-1} = 0.0030\ K^{-1}$$

The film temperature used is an average between base fin temperature and the ambient temperature. Therefore the value of P coefficient is:

$$P = 8.443x10^{7}\ \ \mathrm{m}^{4}$$

Finally, the optimized gap calculated for the fins, given by Equation 3.10, is:

$$S_{otimo} = \frac{2.7}{P^{0.25}} = 0.0282\ m$$

The experimentally used values and calculated gaps are showed in Table 5.3:

Table 5.3 – Gaps values between the fins

Used gap between the fins [mm]	Optimized gap calculated [mm]
30.0	**28.2**

The value adopted for the gap by Flender Brasil Ltda. in this case is very next to the obtained theoretical value using the mathematical model.

Using the numerical method for the calculation of the maximum allowable mechanical power are considered the same values of ambient and oil temperature adopted in the Flender selection catalogues (Flender Brasil, 1989) that are 20ºC and 95ºC respectively.

From Equation 3.7 is obtained the total area:

$$A_{total} = 1.87 - 84*(0.255*0.0032) + 2*(0.0381+0.0032)*0.255*84*0.8956] = 3.368 \ m^2$$

Therefore, the maximum allowable mechanical power is (Equation 3.8):

$$P_1 = \frac{17.5*3.4*(95-20)}{(1-0.969)} = 141 \ kW$$

The informed value in the Flender catalogue (without fins) and the obtained in the theoretical model, considering the use of fins, are presented in Table 5.4:

Table 5.4 – Increase of maximum allowable mechanical power with fins

Maximum allowable mechanical power (without fins) - PG1 [kW] (Flender Brasil, 1989)	Maximum allowable mechanical power (with fins) - $PG1_{fins}$ [kW]	Increase
110	**141**	**28 %**

The increase of the fins application in the housing inferior part in comparison with the same housing without fins is 28%.

6. CONCLUSIONS

The main objective of this paper is to demonstrate the effectiveness and establish a criterion for the use of fins in industrial gearboxes with varied power and environment conditions and to present a numerical result with a satisfactory degree of precision compared with the experimental data.

Analyzing the obtained results, it is possible to conclude that the adopted gap is very good as compared to the optimized gap calculated, making possible a maximum volumetric heat dissipation, as suggested in the numerical treatment part.

Another important analysis is related to the fin height. The reduction of its height from 1½ "(38,1 mm) to 1" (25,4 mm) is possible, allowing material economy without fin efficiency loss.

The difference of temperature between the fin thickness extremities is very small and can be neglected. Therefore, thinner fins could be used. This thickness must be sufficient for the manufacture process to be viable, must be stiff enough not to deform and must not present risks of cuts or injuries for operators. To optimize the fins designs and structures, some bibliographies are indicated (Bejan *et al.*, 1996; Bejan, 2000).

The maximum mechanical power which can be applied to a gearbox in order to keep its oil temperature under the maximum allowable temperature is one of the most important parameters while dimensioning a gear box unit. Varying each input parameter in the numerical method is possible to obtain the maximum mechanical power and determine the fins configuration in the gearbox housing.

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8. RESPONSIBILITY NOTICE