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TEMPERATURE INFLUENCE ON BEARING SCUFFING FAILURE

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Abstract
Scuffing is a complex phenomenon of severe adhesive wear generated under particular combinations including contact pressure, lubrication, speed and friction. Scuffing involves the sudden collapse of the lubricant film and is generally regarded as resulting from thermal phenomena. Under particular combinations including contact pressure, lubrication, speed and friction, a critical temperature is reached in the vicinity of the contact. At this temperature, the breakdown film with local welding or adhesion of the contacting surfaces can appear. The paper presents a temperature distribution model for ball bearings contacts related to the scuffing risk. The own experimental and theoretical results are in good agreement with the literature.

Keywords*: Angular contact ball bearing, Scuffing, Temperature, Heat generation.*

1. INTRODUCTION

There is little agreement in the literature on what scuffing is, what causes it, or what appearance is. This has resulted in significant confusion because various researchers have point out very different physical processes into the general heading of scuffing.

Scuffing involves the sudden collapse of the lubricant film and is generally regarded as resulting from thermal phenomena.

Under particular combinations including contact pressure, lubrication, speed and friction, a critical temperature is reached in the vicinity of the contact. At this temperature, the breakdown film with local welding or adhesion of the contacting surfaces can appear. As result of local adhesion, the friction increases and causes a still higher temperature, and then the process can become catastrophic.

Scuffing can take place either at low sliding velocity and high normal pressure, by softening and melting a thick surface layer of the contacting bodies, or at high sliding velocities and low normal pressure, by a process of abrasion and adhesion [1].

Nowadays, the skidding and/or scuffing become an important limiting factor for high-speed rolling bearings used in aircraft turbine engines.

However, details of the circumstances and the way in which this type of distress occur are still the subject of debate, so that, various mechanisms have been proposed over the past few decades to characterize the scuffing failure.

Among the multiple models, none is sufficiently effective and reliable, covering a wide range of operating conditions, surface topography, material and lubricant composition.

Block (1958) gave one of the earliest and certainly the most well known theory in scuffing. He postulated that scuffing occurs when the total surface temperature in the contact zone of two rubbing surfaces reaches a critical value.

This one can be determined only by the combination of lubricant and rubbing surfaces.

Jacobson (1990) has suggested a completely new interpretation of scuffing. In his model, when the contact's fluid reaches its limiting shear stress, lateral flow of the oil film occurs. This then relieves the pressure on the flattened asperities in the contact, so they can pop up, to promote scuffing.

Lee and Cheng (1995) have been developed a new theory in prediction of the threshold of the scuffing. They considered the contacts operating in a wide range of lubrication regimes, including both the boundary and the partial EHL regimes.

The critical temperature-pressure model is based on the thermal adsorbtion/desorbtion behavior of the interacting surfaces.

According to this new theory, failures occur when the contact temperature exceeds a certain critical value, which is a function of the lubricant pressure generated by the hydrodynamic action in the EHL contact.

In all these models, the critical role played by the temperature is evident.

In order to predict scuffing failure in ball bearings the authors developed a temperature distribution model that leads to a critical scuffing value.

2. TEMPERATURE DISTRIBUTION MODEL

The total contact temperature on the ball race contact ellipse is a sum of both the bulk and flash temperatures.

The bulk temperature is easily measured, but the flash temperature must be calculated.

2.1 Assumptions

The calculation of flash temperature is based on the theory of a moving heat source over a semi-infinite solid, formulated by Jaeger.

As the Peclet's number is greater than 10 for the ball races contacts [2] the heat flow in the direction perpendicular to the movement may be neglected.

From point of view of scuffing failure, the sliding speeds on the contact ellipse (especially on the inner one) in the tested ball bearing (7206) are very important and similar to the rolling speed.

The difference between the ball and inner race temperatures is significant.

That justifies an unequal heat partition, Λ, on each two contacting surfaces, by: $\Lambda = \sqrt{U_{ball} / U_{race}}$.

For a point on a surface contact area P(χ), the temperatures distribution was calculated using an iterative procedure on discretized ellipse as

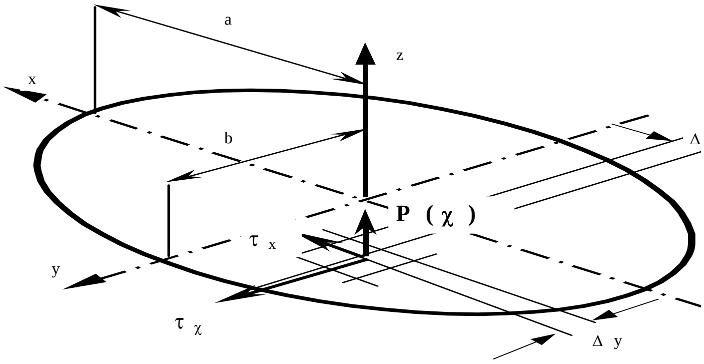

Figure 1: *Contact ellipse discretization*

2.2 Flash and surface temperature computation

The network on discretized ellipse uses the steps: $\Delta x = a/10$ and $\Delta y = b/10$

To evaluate flash temperatures, T_f, on rolling direction, we considered:

$$T_{f\,ball} = \sqrt{\frac{1}{\pi\rho_s c_s u_{ball} k_s}} \cdot \int_{y_0}^{y_{i+1}} \left[\frac{k_f}{h_c} \cdot \left(T_{c\,race(\chi)} - T_{c\,ball(\chi)} \right) + \Lambda \cdot q_{ball(\chi)} \right] \frac{d\chi}{\sqrt{y_{i+1} - \chi}} \quad (1)$$

$$T_{f\,race} = \sqrt{\frac{1}{\pi\rho_s c_s u_{race} k_s}} \cdot \int_{y_0}^{y_{i+1}} \left[\frac{k_f}{h_c} \cdot \left(T_{cball(\chi)} - T_{crace(\chi)} \right) + (1-\Lambda) \cdot q_{race(\chi)} \right] \frac{d\chi}{\sqrt{y_{i+1} - \chi}} \quad (2)$$

These relations reflect the cumulative character of energy dissipated on ellipse contact

To evaluate surface temperatures, T_c , we add the flash temperature as above computed to the average temperature T_m , measured by an infrared thermometer.

We obtained:

$$T_{cball}(y) = T_{m_{ball}} + \sqrt{\frac{1}{\pi\rho_s c_s u_{ball} k_s}} \cdot \int_{-b}^{y} \left[\frac{k_f}{h_c} \cdot \left(T_{crace(\chi)} - T_{cball(\chi)} \right) + \Lambda \cdot q_{ball(\chi)} \right] \frac{d\chi}{\sqrt{y - \chi}} \quad (3)$$

$$T_{crace}(y) = T_{m_{race}} + \sqrt{\frac{1}{\pi \rho_s c_s u_{race} k_s}} \cdot \int_{-b}^{y} \left[\frac{k_f}{h_c} \cdot \left(T_{cball(\chi)} - T_{crace(\chi)} \right) + (1-\Lambda) \cdot q_{race_{(\chi)}} \right] \frac{d\chi}{\sqrt{y-\chi}} \qquad (4)$$

where:

u_{ball} , u_{race} - ball and inner race tangential speed, respectively;

k_s, k_f - thermal conductivity of solid (ball and inner race and lubricant, respectively;

h_c - central film thickness;

$T_{c_{ball}(\chi)}, T_{c_{race}(\chi)}$ - ball and race surface temperatures in the point of coordinates $(x_j, y_i - \Delta y)$;

$q_{ball(\chi)}$, $q_{race(\chi)}$ - total heat generated by viscous friction q_f and on the contact asperities q_a, when there is no complete separation through lubricant in the point of coordinates $(x_j, y_i - \Delta y)$.

The temperature on every point on ball-race contact takes account of three essential components:

(1) $(k_f / h_c) \cdot \left[T_{c_{race(ball)}} - T_{c_{ball(race)}} \right]$ is the heat generated by the thermal transfer through lubricant, between ball and race;

(2) $q_f\ (\chi) = \tau(\chi) \cdot V(\chi)$ is the energy dissipated as the result of film viscous friction;

(3) $q_a(\chi) = FA \cdot V(\chi)$ is the energy dissipated as result of direct contact on asperities when there is no complete separation through lubricant.

There are: $\tau(\chi)$ - the lubricant shear stress; $V(\chi)$ - the ball-race sliding speed; FA - the asperities friction force. They are computed as described in [3]

The amount of *(1)*, *(2)*, *(3)* components in the temperatures computation is strongly related to the bearing lubrication regime and the tangential and sliding speeds of contact surfaces.

2.3 Results

In normal operating conditions for the 7206C ball bearing studied (with negligible sliding on ball-inner race), the figures 2 and 3 present flash and surface temperature distributions, T_f and T_c, respectively.

They are obtained across the rolling direction on ball-inner race contact ellipse, according to the computation model above described.

The lubricant is ELF 154 NS, with determined properties and rheological characteristics.

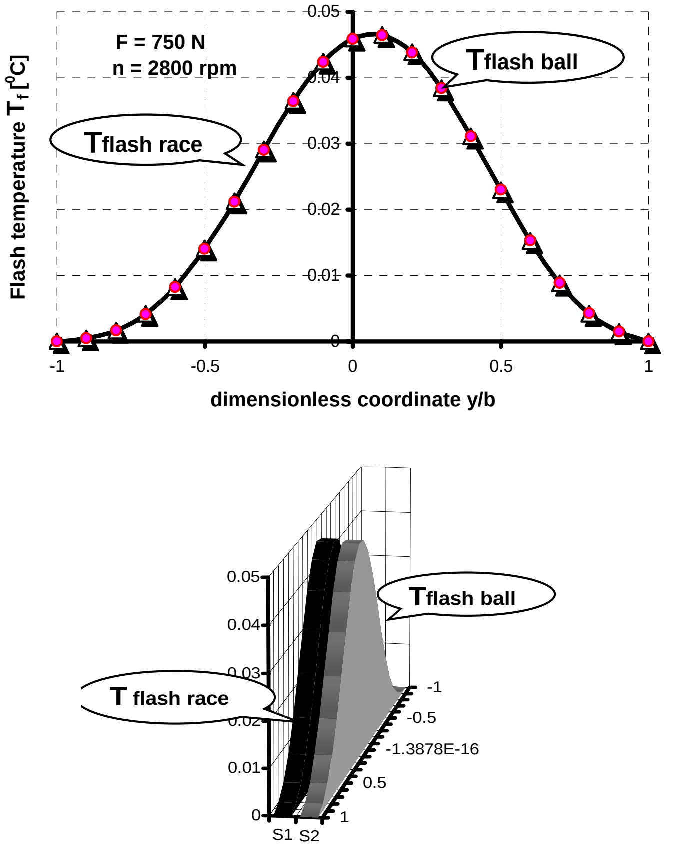

Figure 2: *Flash temperature distribution*

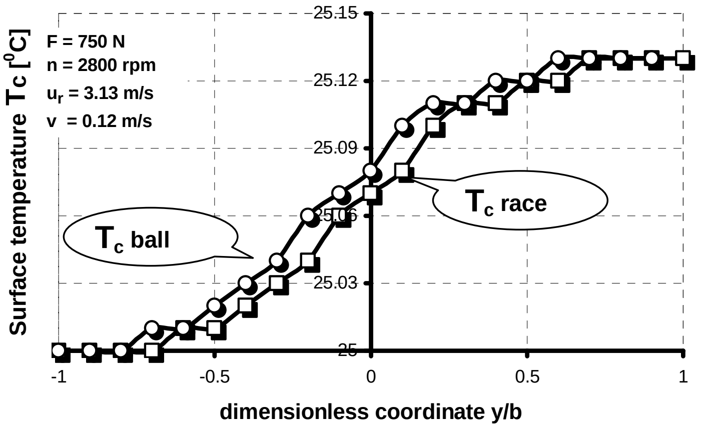

Figure 3: *Surface temperature distribution (T_m = 25 ^{0}C)*

Observation

In the prescribed operating conditions as above mentioned in figures 2 and 3, the sliding speeds are very small (v = 0.12 m/s) compared to rolling speed (u_r = 3.13 m/s). This is the reason why the flash temperatures T_f are almost the same on ball and inner race (figure 2) and the surface temperature T_c has no significant rise (figure 3).

Another example (figures 4 and 5) presents the temperature distributions, T_f and T_c, respectively, but for different operating conditions. The contact geometry of the specimens is the same as in the ball-inner ring contact from 7206 ball bearing (the ellipticity factor k = 8.964).

The simulation criteria reefer to material, geometrical characteristics and the lubrication regime of the practical system, which is 7206 angular ball-bearing.

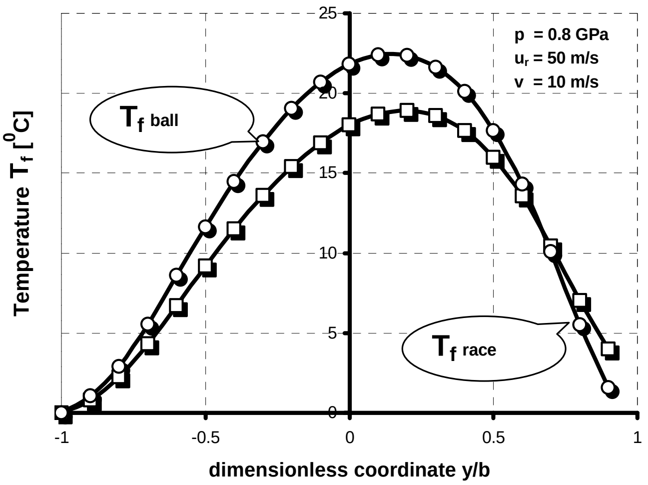

Figure 4: *Flash temperature distribution on specimens*

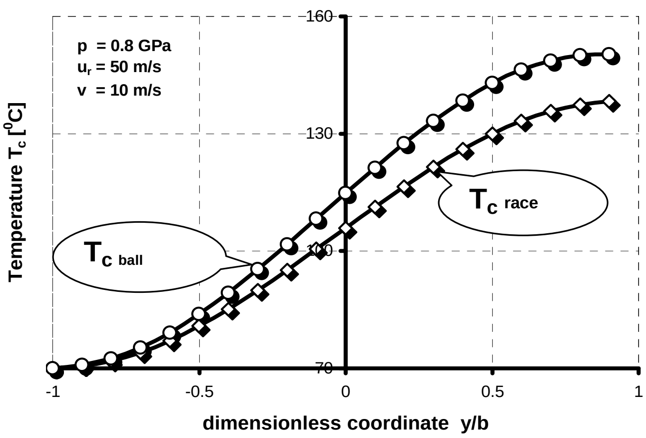

Figure 5: *Surface temperature distribution on specimens (T_m = 70 ^{0}C)*

Observation

The contact pressure is p = 0.8 GPa (the same as in precedent example described in figures 2 and 3). The rolling and sliding speeds are considerably greater (v = 10 m/s, u_r = 50m/s) than those from the precedent example.

It can be noticed the important increase of contact temperatures values. There is the result of the gross sliding measured on the contact.

Taking account into these temperatures high values we must think to scuffing risk for the contact subjected to sliding.

2.4 Validation

The temperature distribution model is in good agreement with the literature (Zhu-Cheng, 1989; Nélias, 1994; Spikes-Enthoven-Cann, 1 996).

Figure 6 illustrates the comparison with Zhu and Cheng results, using the same bearing geometry and the same working conditions. The Zhu and Cheng theoretical model was valid only for a complete elastohydrodynamic lubrication regime. It can be remark a good agreement between the two models.

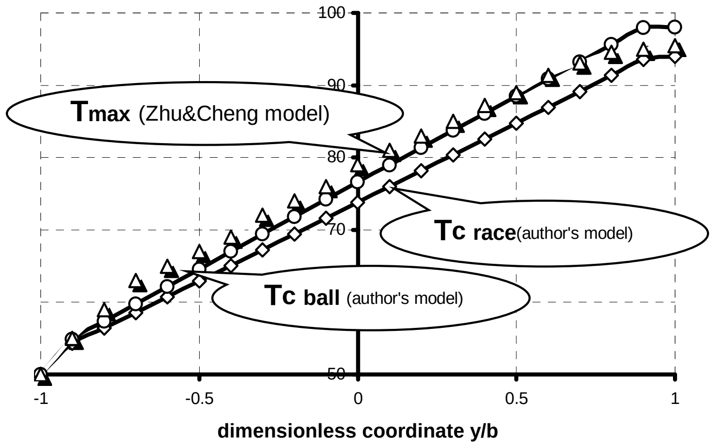

Figure 6: *Surface temperature distribution T_c (F = 750 N; n = 5500 rpm; T_0 = 50 ^{0}C; V = 2 m/s)*

3. SCUFFING TEMPERATURE

The scuffing failure is the result of a permanent competition between speed and pressure in the contact and also supposes the settlement of a complex function depending to mechanical, physical and chemical effects.

The scuffing approaches analysis and our original experiments on bearing's contacts have detached the idea that, any scuffing mechanism considered, there is an energetically unbalance in the rolling contact. This unbalance generates disruptions in lubrication conditions and the scuffing risk appears.

So, we consider the most adequate model to estimate the scuffing limits an energetically one.

Energy dissipation quantified by $\mu{\cdot}p{\cdot}V^{0.8}$ criterion (resulting of our theoretical and experimental study) is corroborated with scuffing temperature.

Our distribution model under a scuffing particular combination including contact pressure, lubrication, speed and friction estimates this critical temperature. Also, a Rayngerr MX4 infrared thermometer has measured average bearing temperature when the scuffing is initiated.

Figure 7 presents the surface temperature, computed by the relations (1-4) for a slide to roll ratio ξ = 83% on the inner race-ball contact ellipse. The average measured temperature before the scuffing occurrence was about 162 ^{0}C, according to those calculated and presented in figure 3.

The surface temperatures values (T = $153.7\ ^0C$) corroborated with energy dissipated q on ellipse contact **(**$\mu\cdot\ p\ \cdot V^{0.8} = 1.552\ \cdot 10^8$ W/m^{-2} **)** calculated as described in [3] offers a covering view of the scuffing failure risk in the ball bearing.

Figure 8 presents the surface temperatures for ball $T_{c\ ball}$ and race $T_{c\ race}$., for a slide to roll ratio ξ = 76,5% on the inner race-ball contact ellipse. It can be remark that the computed scuffing temperature exceeds 150 ^{0}C (for ball 156.4 ^{0}C and for race 152.2 ^{0}C).

These scuffing critical values obtained by the temperature distribution model as above presented are in good agreement with temperature experimental values *(≈160* ^0C). measured by an infrared thermometer (Rayngerr MX4).

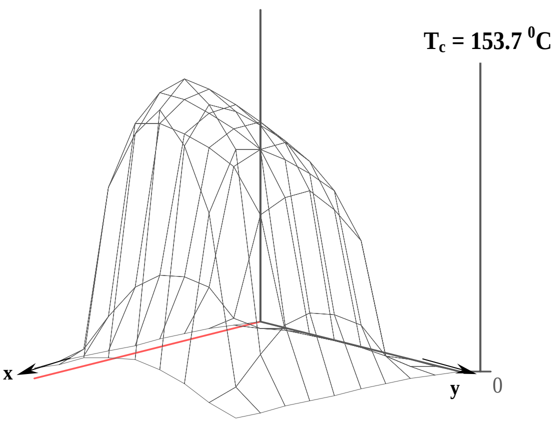

Figure 7: *Surface temperature distribution (F = 750 N; n = 2800 rpm)*

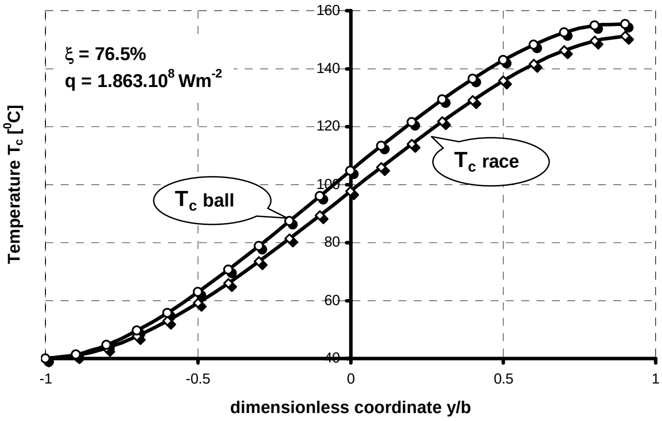

Figure 8: *Surface temperatures* $T_{c\ ball}$ *and* $T_{c\ race}$ *with scuffing risk*

4. CONCLUSIONS

The scuffing results indicate that there is a complex interdependence between mechanical, physical and chemical effects in the rolling contacts This is a fact to support our theoretical modelisation which relies on an energetically equilibrium existent in the bearing for all range of operating conditions.

The temperature distribution model is a wide covering one because it includes the thermal transfer between ball and races and also the heat generated both by the viscous friction in lubricant film and the boundary fiction on the asperity contact.

However the scuffing critical temperature isn't a sufficiently effective and reliable criterion to evaluate this type of distress.

Corroboration with the energy dissipation criterion offers a better evaluation of the bearing performance related to scuffing.

4. REFERENCES

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