Ottimizzazione delle Caratteristiche Reologiche di Lubrificanti per l'Industria di Processo Finalizzati alla Riduzione delle Perdite di Potenza

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The aim of the paper is to optimize the tribological characteristics of lubricating oils that are used in the process industry during machining. In several cases, the machines employed are organized in several "stands" forming "lines" and are equipped by several spindles supported by journal bearings, fed by the same oil, but operating at different rotational speed. Often, the geometry and the size of the bearings and the loads acting on them are also different.
Typically, the spindles supported by the bearings rotate at increasing speeds from the feeding of the blank material to the outlet of the machined one. The power loss on the single spindle is different from the others, not only for the different rotational speed, but also because the oil with which the single bearing is fed has different temperature and, thus, different viscosity. At present, stand-ard mineral oils for typical use are employed.
Owing to the large power loss in these kinds of plants, an attractive idea for power saving is, therefore, to formulate a lubricating oil which, globally, along the entire line, has the best rheological characteristics (mainly in terms of vis-cosity) depending on the actual rotational speed of all spindles. In this way, the power dissipated during the machining process can be reduced while maintain-ing sufficient oil-film thicknesses in the various spindle journal bearings.
In this paper, the modelling of the line is presented, by using a TEHD model for the calculation of the power dissipated in each journal bearing and the lu-bricating oil characteristics are defined by means of a multivariate optimization on the parameters of viscosity, temperature and thickness of the oil film. Final-ly, the optimized dynamic viscosity curve is obtained and can be used for the formulation of an oil, not necessary of mineral origin, with suitable additives.

Keywords: *Power saving, Lubricants, Viscosity, Journal bearings.*

1. Introduction

Oil-film journal bearings in industrial field are still widely employed for their simplici-ty in high-load or high-speed applications. Typical applications are represented by machines with medium/large diameter shafts, operating at low Sommerfeld numbers and characterized by both low tangential speeds and high loads. Conversely other applications can be characterized by high speeds and low loads.
In the paper the case of a steel roll forming machine equipped with several bearings operating in different operating conditions has been considered.
The steel roll forming process is a multi-stage industrial process used to reduce the thickness of metal sheets or metal profile [1]. Typically, two counter-rotating rollers are used to generate the necessary force. The aim is to obtain the desired cross section shape. At each stage the cross-section surface decreases and so the length of the whole workpiece will change (see Fig. 1). The first stages have lower speeds, whereas the last ones have high rotational speeds. In this paper, the modelling of the line is pre-sented, by using a TEHD model for the calculation of the power dissipated in each journal bearing. Finally, the optimized dynamic viscosity curve that reduce the overall power loss, is obtained and can be used for the formulation of the oil.

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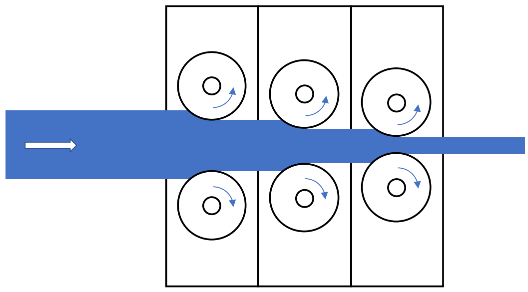

Fig. 1. Example of steel roll forming process.

2. Typical system description

The steel roll forming machine considered in the paper is composed by 10 equal stands with increasing rolling speed. Each stand is composed by 2 spindles in parallel configuration, rotating at the same speed as shown in Fig. 2.
Each spindle is supported by two oil-film plain journal bearings, namely the front and the rear ones. The front bearings of all the stands have the same geometry as well as the rear bearings but they have smaller dimensions with respect to the front ones.

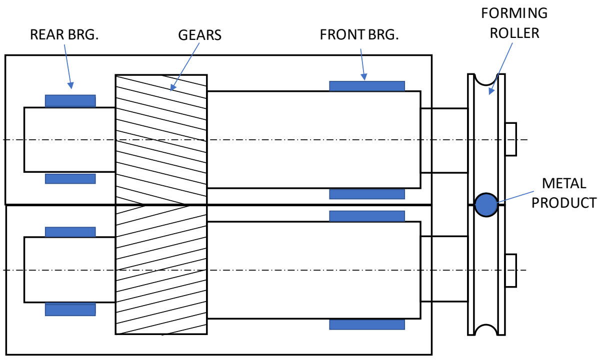

Fig. 2. Example of stand of steel roll forming machine.

For confidential reasons the rotational speed in each stand has been normalized with respect to the maximum rotational speed of the last stand and the estimated load on the bearings with respect to the maximum load obtained in the front bearing of the first stand. The normalized rotational speed and load are shown in Fig. 3. For instance, the speed in the last stand is about 7.5 times the speed in the first one and the load in the rear bearing of the last stand is about 26 times lower than the front bearing of the first stand.

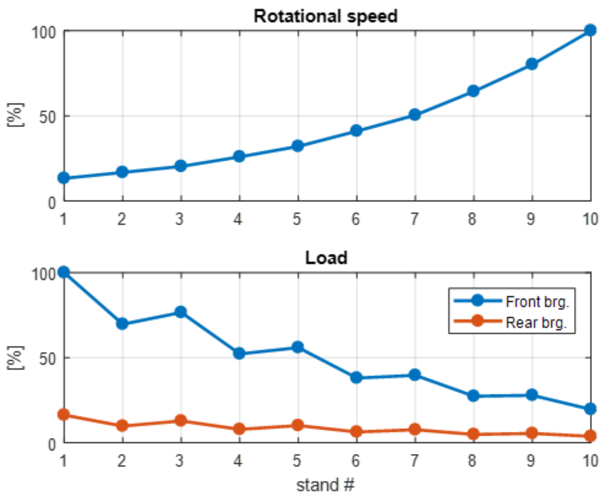

Fig. 1. Operating speed and load of the bearings.

3. Oil properties

The investigation for the reduction of power loss in the oil-film bearing is based on the simulation of the behaviour of all the bearings installed in the machine for different oil characteristics. The dependence of kinematic viscosity k_v on temperature T is given by Whalter's formula:

$$k_v(T) = \exp\left[\exp\left[A - B\ln(T)\right]\right] - 0.7 \qquad (1)$$

Constants A and B can be evaluated by the kinematic viscosity of the oil at 40°C ($kv_{40^\circ C}$) and 100°C ($kv_{100^\circ C}$) as:

$$B = \frac{\ln\left(\frac{\ln\left(kv_{100^\circ C} + 0.7\right)}{\ln\left(kv_{40^\circ C} + 0.7\right)}\right)}{\ln\left(\frac{T_{100^\circ C}}{T_{40^\circ C}}\right)} \qquad (2)$$
$$A = \ln\left(\ln\left(kv_{40^\circ C} + 0.7\right)\right) - B \cdot \ln\left(T_{40^\circ C}\right)$$

Mass density ρ and specific heat capacity c_p are given as follow:

$$\rho(T) = C - D \cdot T$$
$$c_p(T) = E + F \cdot T \qquad (3)$$

where C, D, E, F are all positive constants.
Therefore, the temperature behaviour of the oil is wholly defined by the two values of the kinematic viscosity at 40°C and 100°C that are $kv_{40^\circ C}$ and $kv_{100^\circ C}$. The reduction of power loss for the new oils will be evaluated in comparison to the actual oil used in the steel roll forming machine, named in the following as reference condition.

4. Bearing model

The TEHD model of the bearing includes the laws of hydrodynamic lubrication, the thermal effect due to shear stresses in the oil-film and the deformation of the bearing due to mechanical and thermal stresses.
The hydrodynamic model is based on the well-known Reynolds equation:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \frac{\partial}{\partial x}\left(\frac{\rho h U}{2}\right) - \rho V \qquad (4)$$

where x is the tangential direction, z the axial direction, h the oil-film thickness, p the pressure, μ the dynamic viscosity and U, V represent the velocity terms in the tangential and radial directions of the shaft.
Cavitation and turbulence effects are not negligible for plain journal bearings operating at high speed as the last stage in the steel roll forming machine. The cavitation problem has been solved on the basis of the Giacopini's algorithm [7], an extension of the Elrod algorithm [6], that uses the complementarity concept and also ensures the mass conservation. Turbulence has been considered by means of Constantinescu model [8].
Therefore, the following complementary problem must be solved:

$$\begin{cases} \dfrac{\partial}{\partial x}\left(\dfrac{h^3}{k_x\mu}\dfrac{\partial p}{\partial x}\right)+\dfrac{\partial}{\partial z}\left(\dfrac{h^3}{k_z\mu}\dfrac{\partial p}{\partial z}\right)= \\ =-\dfrac{\partial}{\partial x}\left(\dfrac{rhU}{2}\right)+\dfrac{\partial}{\partial x}\left(\dfrac{hU}{2}\right)+V(r-1) \\ r=\dfrac{\rho_0-\rho}{\rho_0}\geq 0 \\ p\geq 0 \\ p\cdot r=0 \end{cases} \quad (5)$$

In the active (non cavitated) region the fluid density is constant and equal to ρ_0. In the cavitated (non-active) region the density becomes lower due to the presence of vapor and gas bubbles. The pressure, instead, has a complementary behaviour. It assumes zero as value in the cavitated region, whereas it has a greater value in the other part of oil bearing. The solution of eq. (5) can be obtained through a linear complementarity problem (LCP) solver.

The turbulence has been considered using the Prandtl's mixing length hypothesis [9], by means of coefficients k_x and k_z:

$$\begin{aligned} k_x &= 12+0.53\cdot\left(k^2\cdot R_h\right)^{0.725} \\ k_z &= 12+0.296\cdot\left(k^2\cdot R_h\right)^{0.65} \\ k &= 0.125\cdot \mathrm{Re}_h^{0.07} \\ \mathrm{Re}_h &= Uh/\nu \end{aligned} \quad (6)$$

where Re_h is the global Reynolds number, R_h the local Reynolds number, h the oil-film thickness and ν the kinematic viscosity.

The distribution of the temperature in the bearing is obtained by means of a three-dimensional thermal model that include a portion of the shaft, the oil-film and the bearing. The energy equation for the oil-film is as follows:

$$\begin{aligned} &\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)= \\ &=k_{OIL}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right)+\mu\left[\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right] \end{aligned} \quad (7)$$

where c_p and k_{OIL} are the heat capacity of the thermal conductivity of the oil respectively. The use of a two-dimensional thermal model based on the assumption of adiabatic conditions at shaft and bearing surfaces and constant oil temperature in the oil-film thickness leads to the overestimation of the temperature in the oil-film especially in the case of bearings operating at high speeds where shear stresses can be very high. Therefore, a more accurate three-dimensional model must be adopted [5].

The temperature distributions in the bearing and shaft at the steady state are governed by the following equations:

$$\begin{aligned} -\nabla\left(k_{BRG}\nabla T\right)=0 \\ -\nabla\left(k_{SHAFT}\nabla T\right)=0 \end{aligned} \quad (8)$$

where k_{BRG} and k_{SHAFT} are the thermal conductivity of the bearing and shaft respectively.

Equation (7) and (8) have been solved by means of the finite element approach using a structured mesh for the oil-film and unstructured meshes for the bearing and shaft as shown in Fig. 4 for the lower front bearing in the first stand. In Fig. 4 the load acts in the upward direction.

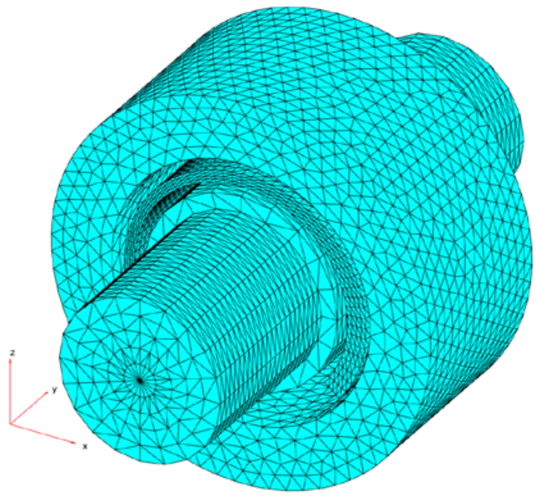

Fig. 2. Meshes used for the solution of the thermal problem.

Convective boundary conditions with convection coefficients $q=50$ W/m^2 are applied on all the non-active surfaces of the bearing and shaft.

An example of temperature distribution in the system is shown in Fig. 5 for the lower front bearing in the first stage of the machine in reference condition. In Fig. 5, a fixed temperature of 40°C is assumed as boundary condition for the inlet surface of the oil.

The effect of the deformation of the bearing because of the pressure distribution and bearing thermal expansion has been considered and evaluated by means of a finite element analysis. The resultant deformation of the bearing surface has been transformed in the change in oil-film thickness.

By considering an isotropic material, the deformation of the bearing (displacement **u**) due to thermal and mechanical stresses is governed by the elasticity equation:

$$-\nabla\left(C\otimes\nabla\mathbf{u}\right)=\frac{E}{1-2\nu}\alpha_t\nabla T \quad (9)$$

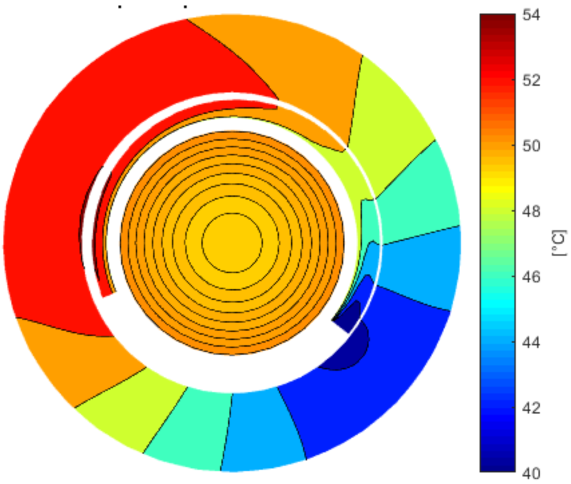

Fig. 3. Example of temperature distribution in the cross section (middle plane) of the bearing obtained with a three-dimensional thermal model.

where C is the tensor of mechanical properties, α_t the thermal expansion coefficient, E the Young’s modulus, and ν the Poisson’s ratio of the material.
Additional boundary conditions are applied on all surfaces of the pad to consider the traction stresses for the evaluation of the thermal deformation.
An example of bearing deformation is shown in Fig. 6 for the lower front bearing in the first stand of the machine in reference condition.
In conclusion, for a given static load, the following conditions must be satisfied:

- convergence of the pressure distribution in the oil-film;
- convergence of the temperature distribution in the system;
- convergence of bearing deformation;
- equilibrium of the forces on the shaft.

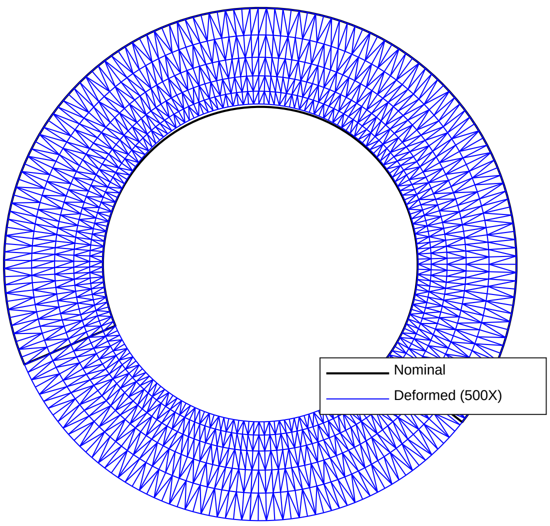

Fig. 4. Example of bearing deformation in the cross section (middle plane) of the bearing.

The results of the numerical simulations have been obtained by using the code developed by the authors and based on Matlab®. The optimization toolbox has been used for solving the equilibrium position of the system, the partial differential equation toolbox for solving the three-dimensional thermal model for the temperature distribution and the three-dimensional structural-mechanics model for the bearing deformation.

4.1. Results for the reference oil

The bearings of the machine operate in a wide range of conditions in terms of load and rotational speed. The bearings in the first stage run at low speed and high load whereas the bearing in the last stand operate at high speed and low load (see Fig. 3).
The maximum power loss for the reference oil is obtained in the last stands that runs at high speed as shown in Fig. 7 where all the values have been normalized with respect to the maximum value. Conversely the minimum oil-film thickness is obtained in the first stage of the machine as shown in Fig. 8.

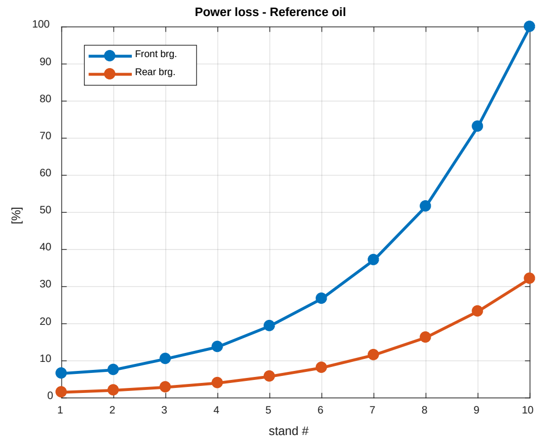

Fig. 5. Power losses in the bearings for the reference oil.

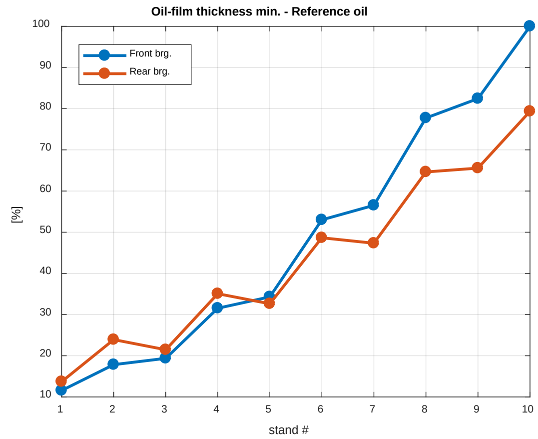

Fig. 6. Minimum oil-film thickness in the bearings for the reference oil.

The minimum oil-film thickness has been normalized with respect to the maximum value obtained in the last stand in Fig. 8.

Therefore, the design of the new oil will be a trade-off between the reduction of power loss that can be easily obtained by reducing the viscosity of the oil in the last stage and the limit of the minimum oil-film thickness reached in the first stages of the line.

4.2. New energy saving oil

The behaviour of all the 40 bearings in the steel roll forming machine has been simulated for different oil properties given by the two kinematic viscosities at 40°C and 100°C. The aim of the analysis is the reduction of the power loss. The percentage variation of the power loss with respect to the reference oil, that is the actual oil used in the system is shown in Fig. 9 where the black dot represents the reference condition. In Fig. 9 it is possible to note that the highest power loss reduction can be obtained by reducing the viscosity of the oil. However, the two viscosity parameters are not independent themselves. Real oils show a behaviour represented by a limited range of the viscosity index that relates the kinematic viscosities.

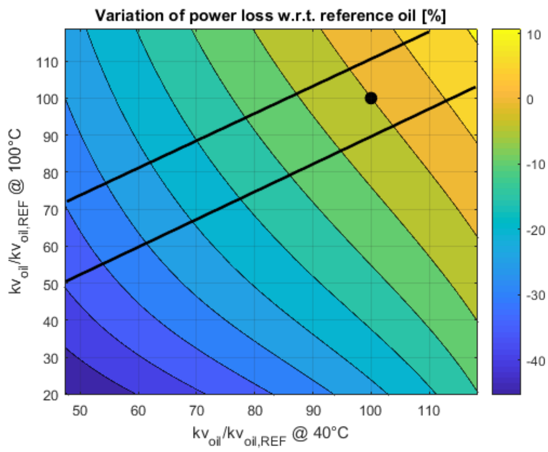

Fig. 7. Variation of power loss with respect to the reference oil.

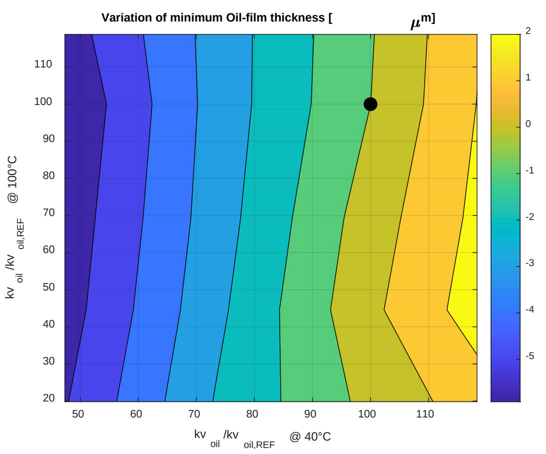

Fig. 8. Variation of minimum oil-film thickness with respect to the reference oil.

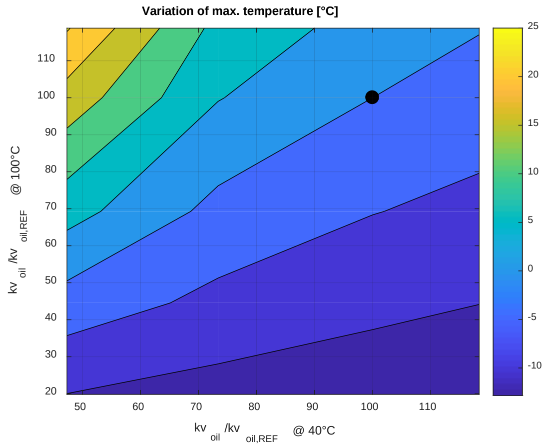

Fig. 9. Variation of maximum temperature with respect to the reference oil.

The two black lines in Fig. 9 represent the boundaries of the new energy saving oil. Conversely, the reduction of the oil viscosity leads to a reduction of the minimum oil-film thickness as shown in Fig. 10.
The value of the minimum oil-film thickness is critical for the bearings in the first stand that operate at high load and low speed. Furthermore, the reduction of the oil viscosity in the range given by the black lines in Fig. 9, in general, do not increase the maximum temperature in the bearings as shown in Fig. 11.

5. Conclusions

The reduction of the total power loss due to shear stresses in the oil of the bearings of a steel roll forming machine has been investigated in this paper.
The analysis has been performed by simulating the behavior of all the bearings by means of an accurate TEHD model. The reduction of the power loss can be obtained by the reduction of the oil viscosity. The following conclusions can be drawn:

- the maximum power loss is obtained in the bearings operating at high rotation speed (Fig. 7);
- the bearings operating at low rotational speed are critical because they show the minimum oil-film thickness (Fig. 8);
- the minimum oil-film thickness is mainly a function of the kinematic viscosity at 40°C (Fig. 10);
- the maximum oil temperature increases for an increase of the kinematic viscosity at 100°c and a reduction of the kinematic viscosity at 40°C (Fig. 11);
- the overall power loss of the machine can be reduced of 30% by using an oil with kinematic viscosities at 40°C and 100°c that are about half the values of the reference oil (Fig. 9).

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