PREDICTION OF BEARING LOADS DURING THE STARTUP OF RECIPROCATING COMPRESSORS

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Abstract. *Most simulation methodologies for compressors available in the literature consider a steady state operating condition. However, several phenomena that affect the compressor energy efficiency, noise level and reliability are associated with transient effects that occur the startup and shutdown processes. This paper presents a numerical analysis of the compressor startup, with special attention to the bearing loads that take place in the first compression cycles. Numerical results are compared to experimental data and good agreement is verified, demonstrating that the model can be used for the analysis of compressor startup.*

Keywords*: reciprocating compressor, compressor startup, bearing system.*

1. INTRODUCTION

In refrigeration systems the cabinet temperature is usually used as control parameter to switch on and switch off the compressor so as to guarantee the required temperature in each specific compartment. During such operation transients, the compressor is subject to critical conditions, characterized by higher loads on the mechanical kit. For instance, in the first cycles of the compressor startup valves and mufflers operate in a critical condition, since the equalized system pressure leads to a high mass flow rate of refrigerant. Additionally, the suction valve is kept open longer and, depending on its design, may present oscillating motions that significantly increase the noise level as the compressor speed is increased. Another important aspect is that during the startup the fully hydrodynamic lubrication in the bearings may not occur, with a mixed or even boundary lubrication regime resulting instead. Such lubrication conditions can originate metallic contact between shaft and bearings, increasing mechanical losses and causing wear.

The vast majority of studies dealing with compressor transients are focused on the system performance. For this reason, a very simple modeling is adopted for compressor, with no attention given to mufflers, valves and bearings (Koury et al., 2001; Jabardo et al., 2002; Browne and Bansal, 2002). For instance, in such approaches the compressor mass flow rate is usually obtained from a curve fitting of experimental data obtained for different compressor speeds. On the other hand, a number of interesting studies on compressor body vibrations during startup and shut-off are available in the literature. Gerhold and Hamilton (1974) presented a mathematical model for a single cylinder refrigeration compressor, developed to predict the displacement of the compressor suspension mounting points and its frame center of gravity during the shutdown operation. The pressure inside the cylinder was evaluated for a perfect gas, with the compression and expansion processes following a polytropic line defined from experimental data.

Dufour et al. (1995) studied numerically and experimentally the transient dynamic behavior of a single cylinder reciprocating compressor during the startup, steady-state and shut-off regimes, with the aim of evaluating the compressor displacement. The pressure force acting on the piston during startup was obtained by a curve fitting of the experimental data. Porkhial et al. (2002) considered an experimental analysis of the transient behavior of a small reciprocating compressor adopted in household refrigerators. Based on the experimental data, an empirical model was developed to predict the transient performance of the compressor under different conditions. The authors verified a greater energy consumption in the transient as compared to the steady-state regime.

Kim et al. (2004) analyzed the dynamic characteristics of a reciprocating compressor with flexible suspension springs. Combined wire torque data were used until a certain speed and after that the torque input was switched to the primary wire torque. Experimental data for pressure inside the cylinder for the steady-state condition was prescribed to evaluate the force acting on the piston surface. Chen (2006) presented a method to estimate the motor startup speed profile for different prescribed load torque-speed relationships. The author pointed out the importance of supplying the proper motor speed-torque curve in order to obtain accurate results.

Currently, there is a great demand for compressors with higher energy efficiency and lower noise levels, preferably at no further cost. Refrigeration compressors belong to a very competitive market in which the time required to design and launch new products is very short. This paper reports the development of a numerical model, experimentally validated, to simulate the compressor startup and shutdown and aimed at assisting their design. As a first stage, the study considers the startup of a reciprocating compressor for a thermal equilibrium condition, which occurs under a cyclic operation condition.

2. MATHEMATICAL MODEL

The methodology adopted to simulate the compressor accounts for piston displacement as a function of crankshaft angle, the thermodynamic process inside the cylinder, mass flow rate through the valves, valve dynamics, gas pulsation inside the mufflers and refrigerant thermodynamic properties (Ussyk, 1984). Several parameters are calculated during the compressor cycle, such as instantaneous pressure throughout the compressor, mass flow rate, valve dynamics, energy and mass losses, refrigerating capacity, etc.

The compressor modeling was carried out by mathematically describing each one of the compressor components. The control volumes considered are: compression chamber, suction and discharge mufflers, internal environment, compressor housing, electric motor and bearings. A steady state condition is assumed for all temperatures with the exception of the gas inside the cylinder. Moreover, the compressor is simulated decoupled from the refrigeration system and, therefore, the variation in the pressures in the suction and discharge lines during startup had to be derived from experimental data.

2.1. Compressor chamber, valves and muflers

An integral formulation of the energy conservation equation is used to model the thermodynamic process for the gas inside the cylinder. The internal temperatures of the compressor have to be supplied as inputs for the simulation program. This is accomplished by an interface with a second simulation code, which evaluates the temperature for eight control volumes through energy balances and using some of the compressor simulation program outputs. Mass flow rates in the energy equation are associated with leakage through the clearance between the piston and cylinder and to the flow through the valves. The former is evaluated by assuming the presence of pure refrigerant in the clearance, whereas the latter is estimated through a procedure explained in the next section.

Reed valves used in hermetic compressors are called automatic because they open and close depending on the pressure difference between the cylinder and the suction/discharge chamber, established by the piston motion. Such valves are usually made of steel and their dynamics can be expressed in a simplified manner through a one-degree-of-freedom model. The effective force area definition is used to evaluate the force over the valve reed during the motion. On the other hand, the mass flow rate through the valve is evaluated from data for effective flow area. For a certain pressure drop, the effective flow area expresses the ratio between the actual mass flow rate and the theoretical value given by an isentropic flow condition.

Gas pulsation in mufflers was simulated via a one-dimensional formulation for the flow dynamics, represented by conservation equations for mass, momentum and energy. Wall effects on the flow were taken into account through estimates of friction force and heat transfer, obtained from standard correlations developed for steady flow through pipes. The pressure drop at geometric singularities was determined from classical correlations in terms of the area ratio. The system of differential equations for fluid flow in mufflers is discretized through a finite volume methodology and solved by a numerical procedure (Deschamps et al., 2002). Convective transport at the control volume faces was approximated with a first order upwind interpolation scheme. A fully implicit time discretization scheme was applied to unsteady terms in the equations. The system of algebraic equations was solved with the Tridiagonal Matrix Algorithm (TDMA), in a segregated approach. The coupling between pressure and velocity was handled through the SIMPLEC algorithm extended to flows of arbitrary Mach number.

The control volume balance equations for the compressor components are simultaneously and iteratively solved since they depend on all compressor energy fluxes. The transient equations associated with the compressor simulation code are solved via a fourth-order Runge-Kutta method. Thermodynamic properties for the refrigerant were evaluated through a program link to the REFPROP database.

2.2. Bearings

A typical bearing system of a reciprocating compressor adopted for refrigeration purposes is composed of the piston-cylinder assembly bearing, piston pin-small eye bearing, crankpin-big eye bearing, thrust bearing and crankshaft bearings, as identified in Fig. 1a. The compressor considered in this paper had three radial bearings on the crankshaft, represented by the main bearing, secondary bearing and the crankpin-big eye bearing. Fig. 1b presents a free-body diagram indicating all forces that act on the crankshaft. The resulting moment of each force is evaluated around the action line of force F_{mb} in the main bearing.

Assuming a rigid mechanism, with no deformation in the connecting rod, piston or crankshaft, and that the crankshaft inertia is sufficiently small, the equilibrium equations for the force and moment can be represented as follows:

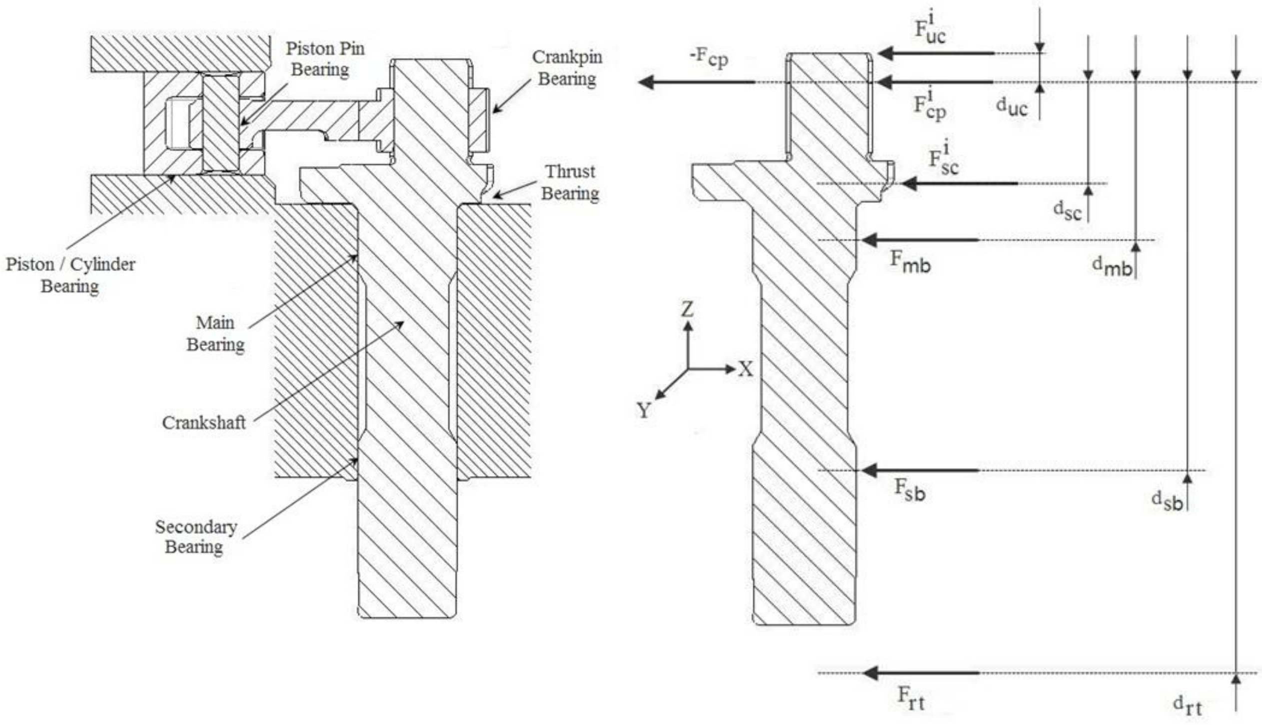
$$\sum F_x = F_{mb,x} + F_{sb,x} + F_{rt,x} + F^i_{sc,x} + F^i_{uc,x} + F^i_{cp,x} - F_{cp,x} = 0 \tag{1}$$

$$\sum F_y = F_{mb,y} + F_{sb,y} + F_{rt,y} + F^i_{sc,y} + F^i_{uc,y} + F^i_{cp,y} - F_{cp,y} = 0 \tag{2}$$

$$\begin{aligned}\sum M_{mb,y} = &F_{sb,x}(d_{mb} - d_{sb}) + F_{rt,x}(d_{mb} - d_{rt}) + F^i_{sc,x}(d_{mb} - d_{sc}) + \\ &F^i_{uc,x}(d_{mb} + d_{uc}) + (F^i_{cp,x} - F_{cp,x})d_{mb} = 0\end{aligned} \tag{3}$$

$$\begin{aligned}\sum M_{mb,x} = &F_{sb,y}(d_{mb} - d_{sb}) + F_{rt,y}(d_{mb} - d_{rt}) + F^i_{sc,y}(d_{mb} - d_{sc}) + \\ &F^i_{uc,y}(d_{mb} + d_{uc}) + (F^i_{cp,y} - F_{cp,y})d_{mb} = 0\end{aligned} \tag{4}$$

The distance to the line of action of a force, *d*, and the directions *x* and *y* used as subscripts for the force components are indicated in Fig. 1b. The solution of the above system of equations allows the determination of loads F_{mb} and F_{sb}, on the main and secondary crankshaft bearings. It should be noted, that the magnitude and direction of each force vary according to the shaft angular position.



(a) (b)

Figure 1: Typical bearing system of a reciprocating compressor (a) and crankshaft free-body diagram (b).

The inertia forces of the upper counterweight, F^i_{uc}, shaft counterweight, F^i_{sc}, and crankpin, F^i_{cp}, can be obtained from the following general equation:

$$F^i_n = m_n \omega^2 r_n \tag{5}$$

where m_n and r_n are, respectively, the mass and the rotation radius of the n-th component of the bearing system and ω is the compressor shaft angular velocity.

The force of the crankshaft on the connecting rod can be obtained from a balance of forces on the crankpin, which involves the force on the connecting rod itself, F_{cr}. From the free body diagram for the piston shown in Fig. 2, the sum of forces in the *x* direction can be expressed as

$$F_{cr} = -(F_{gc} + F_p^i + F_{cb})/\cos\alpha \tag{6}$$

where F_{gc}, F_p^i and F_{cb} are the force associated with the gas compression, piston inertia and cylinder bearing, respectively. The connecting rod angle, α, can be defined as:

$$\alpha = arc\,sin[(e\,sen\theta - d_m)/L_{cr}] \tag{11}$$

As Fig. 2 shows, e and L_{cr} are the crank eccentricity and the connecting rod length, respectively. On the other hand, d_m represents a misalignment between crankshaft and cylinder, whereas θ is the crankshaft angle. The piston inertia force, F_p^i, is given by the product of its acceleration and mass, whereas the force due to the gas compression, F_{gc}, is obtained from the pressure difference between the cylinder and the compressor shell multiplied by the piston cross sectional area. Finally, F_{cb} is the friction force between the piston and the cylinder.

A balance of forces for the crankpin bearing is required to evaluate the force acting on the eccentric rod, F_{cp}. Considering the directions x and y, the components of the force acting on the connecting rod, F_{cr}, can be obtained by the following expressions:

$$F_{cp,x} = F_{cr}\cos\alpha - F_{cp,x}^i \tag{7}$$

$$F_{cp,y} = F_{cr}\sin\alpha - F_{cp,y}^i \tag{8}$$

The load on the bearings is of paramount importance in terms of the compressor efficiency, because it affects directly the power losses by friction. In addition, such loads also affect the lubrication condition that prevails in each bearing. An inappropriate lubrication condition can generate premature wear and also an increase of the voltage required for the compressor startup.

If the bearing speed is very low there will be no pressure build up in the lubricant film and the loading will be supported by metallic contact; a condition known as boundary lubrication. As the speed is increased, the hydrodynamic pressure in the lubricant film is increased and a mixed lubrication takes place, with the loading being supplied by both the hydrodynamic pressure and the contact pressure between the two surfaces. At high speeds the hydrodynamic pressure reaches such a level that both surfaces are completely separated by a lubricant film.

The Stribeck curve, illustrated in Fig. 3, gives the friction factor as a function of the Stribeck number and plays an important role in identifying boundary, mixed and hydrodynamic lubrication regimes. The boundary regime is characterized by metallic contact in the bearing, whereas the mixed regime corresponds to a transition region in which a thin, unstable, oil film is formed. Finally, in the fully hydrodynamic regime a stable oil film prevails and guarantees an adequate lubrication for the bearing. During the compressor startup, the boundary and mixed lubrication regimes occur due to a low speed condition in the bearings. Hence, since the hydrodynamic condition is only achieved after some cycles, it is crucial to understand such phenomenon in order to design highly efficient, reliable compressors.

Manke (1991) developed a methodology to simulate the dynamics of finite radial bearings through the solution of the Reynolds equation, considering reciprocating compressors under a cyclic steady-state operating condition. However, the model considers a fully hydrodynamic condition and, therefore, cannot account for occurrences of contact between the shaft and bearings, a condition that may occur during the startup and shutdown transients.

Nogueira et al. (2002) has presented an extensive study on friction factors in different lubrication regimes (hydrodynamic, boundary and mixed), using an experimental setup to measure the rotation and load for the shaft. The authors proposed a correlation for the friction factor based on a parameter referred to as solid contact. They verified that for high quality surface finishing the transition between the hydrodynamic and the boundary lubrication regimes is abrupt, while for significant roughness the transition is smooth.

In the present study, the methodology developed by Manke (1991) was adopted to simulate the crankshaft bearings. In the case of mixed lubrication during the early moments of the compressor startup, the linear relationship for the friction factor proposed by Nogueira et al. (2002) was employed. In order to account for the two lubrication regimes, the friction factor needed to evaluate the mechanical loss in the radial bearings was written as a function of the Stribeck number. Accordingly, the mechanical loss in the bearing, $\dot{E}_n$, is given by:

$$\dot{E}_n = \left(f^{+} \frac{R_n}{c_n} \right) F_n R_n \omega \tag{9}$$

where F_n is the load on the bearing.

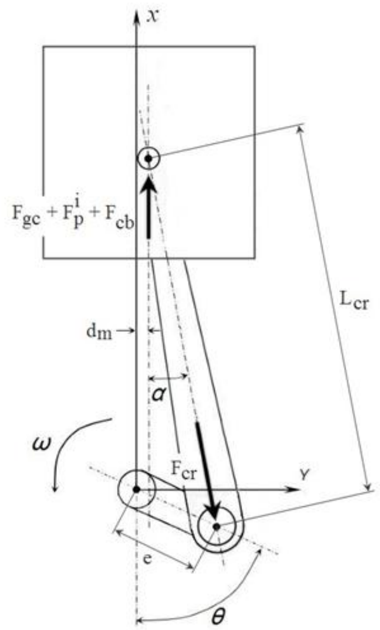


Figure 2: Free body diagram for the piston.

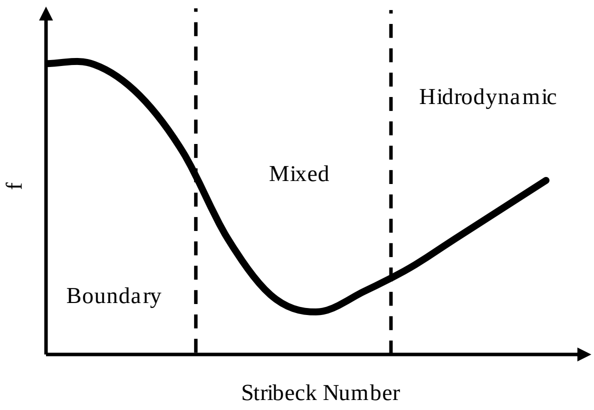


Figure 3: Friction factor as a function of the Stribeck number; adapted from Czichos, 1978.

The modified friction factor, f^+, appearing in Eq. (9) is defined as follows:

$$f^+ = f\,\frac{c_n}{R_n} \tag{10}$$

where c_n is the bearing radial clearance, R_n is the bearing radius and f is the friction factor.

The thrust bearing and the piston-cylinder assembly are modeled as axial bearings fully filled with oil. When the Stribeck number indicates a mixed lubrication condition, the corresponding friction factors for both axial bearings are multiplied by a correction factor. Finally, the Stribeck number is defined as follows:

$$S_n = \frac{\mu_o \omega (2R_n W_n)}{F_n} \tag{11}$$

where μ_o is the oil viscosity and W_n is the bearing width.

2.3. Rotor-crankshaft assembly

The evaluation of the mechanical kit acceleration during the compressor startup requires that the total resistive torque acting on the crankshaft mechanism are known. Such torque is caused by bearing friction and the force required to compress the gas inside the cylinder. The torque supplied by the motor can be described by its performance curve, considering also the action of an auxiliary coil. Therefore, the instantaneous angular shaft velocity is given by $\omega = \omega^0 + \dot{\omega}\Delta t$, where the angular velocity $\dot{\omega}$ is given by:

$$\dot{\omega} = \frac{Tq_{motor} - Tq_{resist}}{I_t} \tag{13}$$

where ω^0 is the angular velocity at previous time, $\dot{\omega}$ is the angular acceleration of rotor and crankshaft, Δt is the time discretization, I_t is the rotor and crankshaft inertia. Finally, Tq_{motor} and Tq_{resist} are the electrical motor and total resistive torque, respectively.

3. RESULTS

The methodology developed to simulate the reciprocating compressor under a transient operation condition was verified with reference to experimental data. The Fig. 4a shows numerical and experimental results for the piston position, x_p, over time. It is possible to verify a good agreement between experimental and numerical prediction. The

Fig. 4b shows numerical and experimental results for the variation in pressure inside the compression chamber, p_c, as time function. As can be seen, there is a good agreement between the two sets of results, supporting the validation of the methodology here presented. An interesting detail that can also be observed in this figure is the associated low speed of the compressor during the first two cycles. After 6 cycles the compressor has almost reached its full velocity of a steady cyclic operation condition, but suction and discharge pressures are far from the stabilized values.

Figure 5 depicts the main forces acting on the crankshaft mechanism during the initial compressor cycles. It is possible to notice that the loads on the main bearing, secondary bearing and crankpin bearing increase with time. This is a consequence of pressure rise in the discharge line as the compressor speed climbs (Fig. 5b) due to the motor torque.

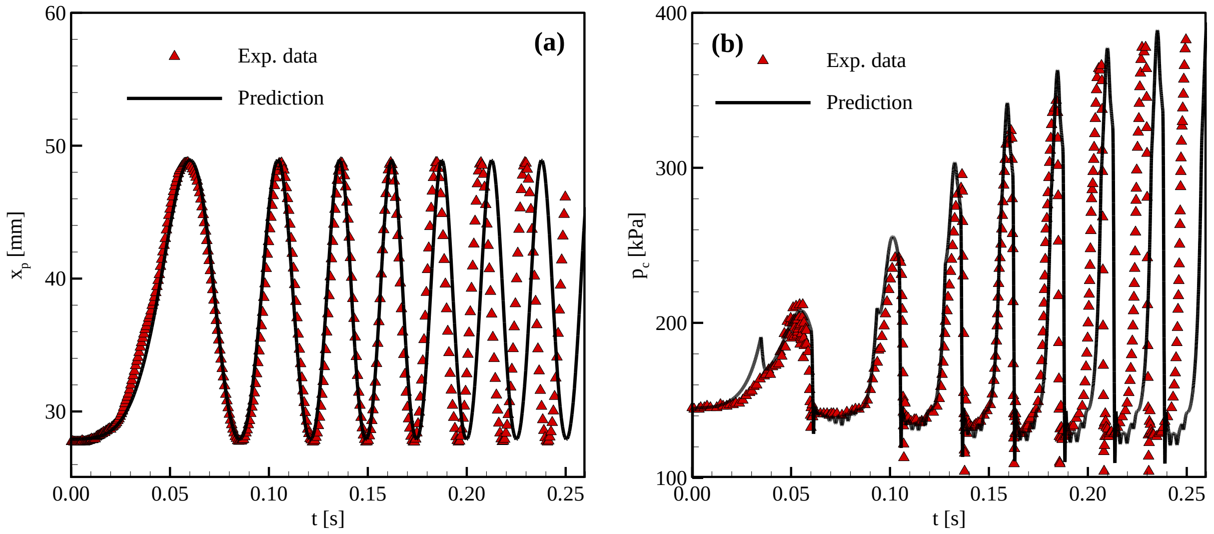


Figure 4: Numerical and experimental results for piston position (a) and pressure in the compression chamber (b).

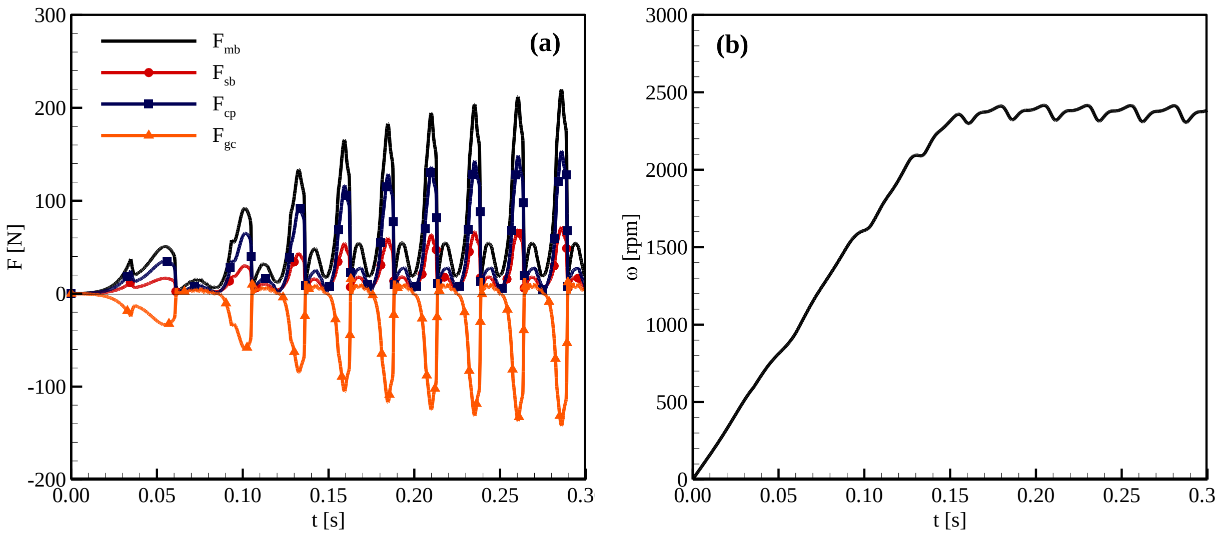


Figure 5: Numerical results for the main forces over mechanism (a) and crankshaft speed (b).

The mechanical loss for a particular bearing, $\dot{E}_n$, is a function of both the load and the compressor speed. The greatest values of $\dot{E}_n$are found in the first cycle (Fig. 6a) due to the highest friction factor observed there. As can be seen from Fig. 6b, the Stribeck number for the initial cycles remains low for a longer period of time. In fact, the compressor low speed in the first cycle gives rise to the boundary lubrication regime. For instance, Fig. 6a shows that the mechanical loss in the first cycle is far greater than losses predicted for the remaining cycles, which is a clear indication of boundary lubrication in the first cycle. Although Fig. 6b indicates low values of Stribeck number during the subsequent cycles, the compressor speed is high enough to avoid such a critical lubrication condition, returning mechanical losses considerable lower than that in the first cycle.

Figure 7 shows load components, Fx and Fy, acting on the main bearing, secondary bearing and crankpin bearing for four cycles, specified by the time instant in which the cycle starts, t_i, and the period Δt_c it takes to be completed: i) t_i = 0.14936s e Δt_c = 25.5ms; ii) t_i = 0.70871s e Δt_c = 25.5ms; iii) t_i = 1.30606s e Δt_c = 20.75ms e iv) t_i = 2.34401s e

Δt_c = 20.7ms. By examining Fig. 7, one can observe that loads in the x direction are greater than those in the y direction, with the main bearing withstanding the highest loads. It is interesting to note that a fully cyclic condition has already been reached at cycle iii), since the associated load diagram is virtually equal to the one of cycle iv).

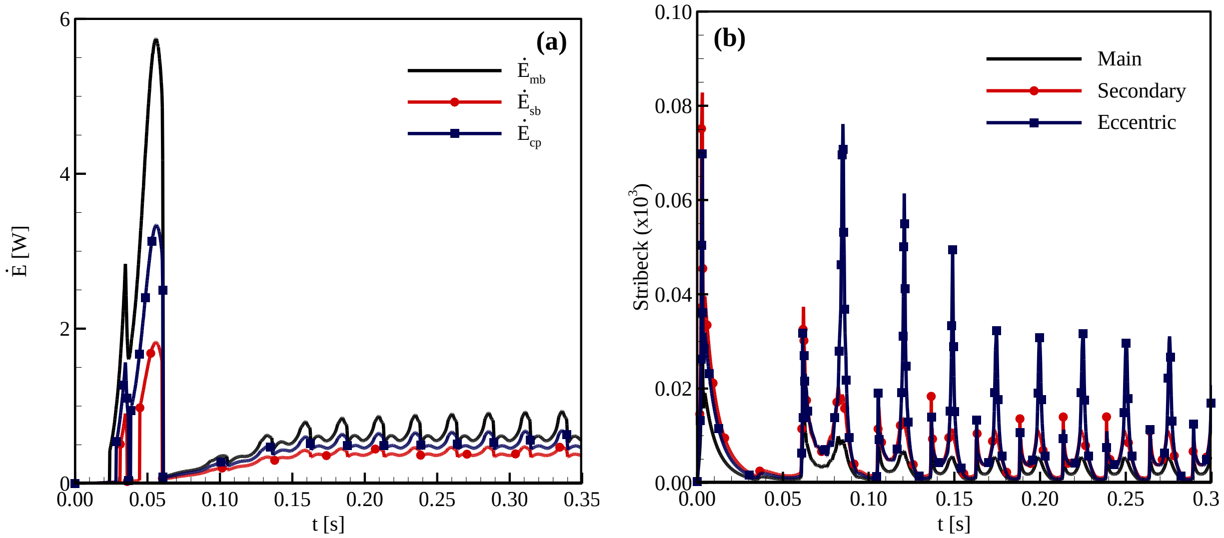


Figure 6: Mechanical loss (a) and the Stribeck number for main, secondary and eccentric bearings.

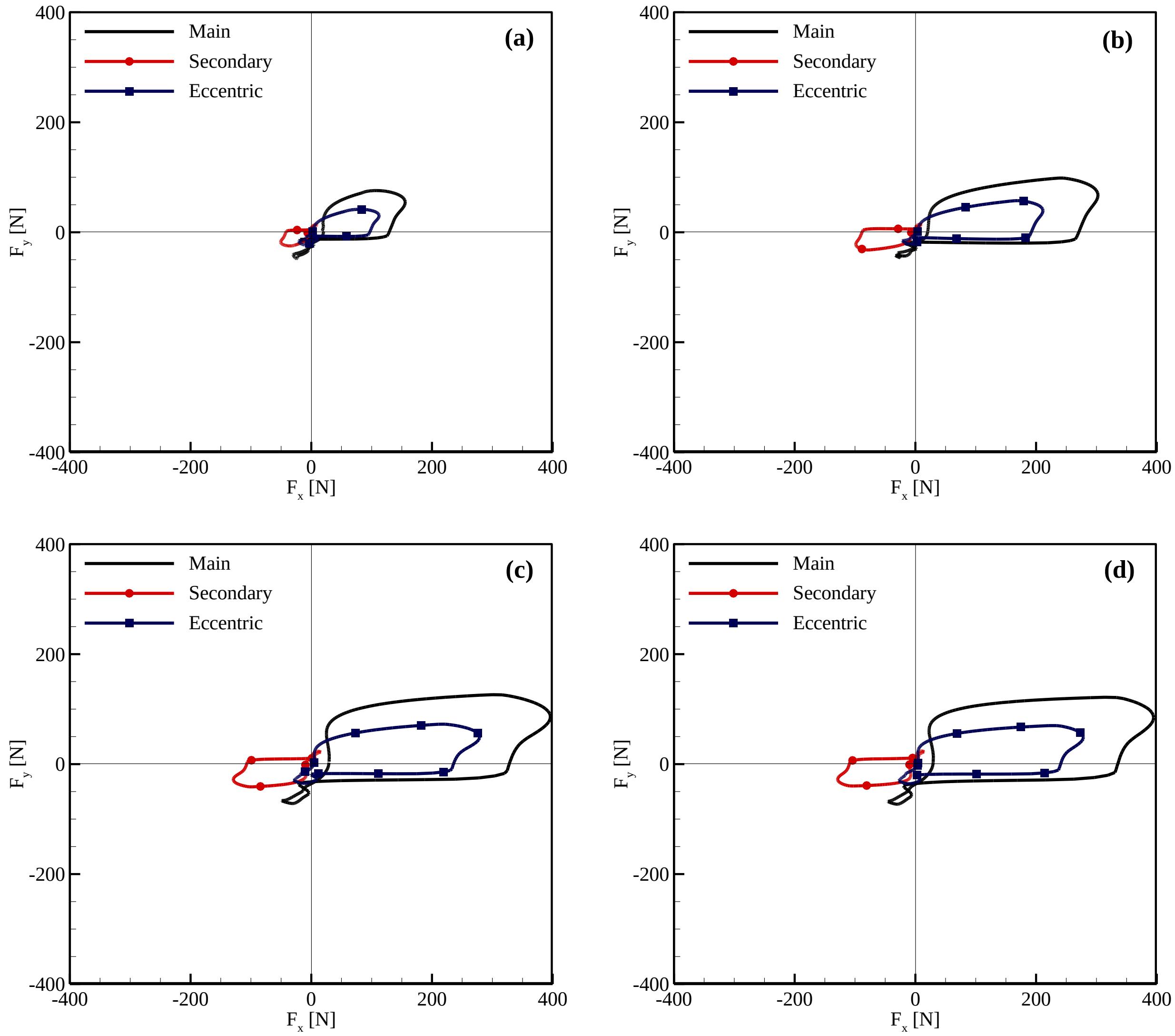


Figure 7: Load components for different speeds of crankshaft.

4. CONCLUSIONS

Transients associated with the compressor startup affect the mechanical kit load, the performance of valves and mufflers, and the lubrication regime of bearings. Most studies in the literature dealing with such transients consider the compressor coupled to a refrigeration system, with a detailed account of the system performance, but usually describing the compressor through curve fittings of experimental data. This paper presented a numerical methodology, experimentally validated, to simulate transients in compressors and taking into account the performance of all the compressor components. Numerical predictions were compared to experimental data and a good agreement was verified, demonstrating that the model can be used as a valuable tool for the analysis of compressors.

5. ACKNOWLEDGEMENTS

This study forms part of a joint technical-scientific program between the Federal University of Santa Catarina and EMBRACO, partially funded by CNPq (Brazilian Research Council) through Grant No. 573581/2008-8 (National Institute of Science and Technology in Refrigeration and Thermophysics). Support from FINEP (Federal Agency of Research and Projects Financing) is also acknowledged.

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