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Modeling the Stiction Effect in Automatic Compressor Valves

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ABSTRACT

The so-called stiction effect (or stiction force) is one of the sources of thermodynamic losses in compressor suction and discharge systems. The losses associated with the stiction effect are caused by a deformation of the lubricating oil film between the valve and the seat. This generates a delay in the valve opening, since larger pressure differences between the cylinder and the discharge and/or suction chambers are needed to compensate for those effects. This paper advances a theoretical model for solving the dynamic behavior of a ring-shaped lubricant oil film between a discharge valve and the seat. The valve is allowed to move under the action of an external force due to the time-dependent pressure difference between the cylinder and the discharge chamber. The main contributions of the model are the consideration of a finite amount of oil between the valve and the seat and mathematical relationships for calculating the initial condition for the oil film thickness. The model results are compared with those obtained with the Khalifa and Liu (1998) method. At conditions typical of discharge systems of domestic refrigeration compressors, viscous effects are the dominant component in the oil stiction force under dynamic conditions.

1. INTRODUCTION

In order to maintain a pressure difference between the suction and discharge lines, domestic refrigeration compressors make use of automatic reed valves. Ideally, suction and discharge valves should open or close instantly whenever the instantaneous cylinder pressure equals those in the suction and discharge chambers, respectively. In real systems, however, a finite pressure difference is required to open an automatic valve due to inertial and viscous effects. One of such effects stems from the presence of a lubricant oil film between the valve and the seat. The so-called *stiction force* results from a combination of interfacial tension, capillarity effects and deformation (viscous flow) of the oil film. Stiction phenomena are a source of thermodynamic and volumetric losses in the compressor, as they cause an increase of the pressure difference required to open the valve and a delay in the valve opening.

Lorentzen (1955) was one of the first investigators to point out the influence of oil stiction phenomena on the dynamics of automatic reed valves. MacLaren and Kerr (1969) suggested that it was possible to obtain fitted coefficients to minimize discrepancies between experimental and numerical results for the dynamic behavior of compressor valves. In their experimental work, Giacomelli and Giorgetti (1974) concluded that the presence of oil between the valve and the seat causes a significant delay in the valve opening and, therefore, the amount of oil in that region should be kept at a minimum. Pringle (1976) observed that the phenomenon of cavitation in the oil film exerted a direct influence on the time required to open the valve and on its dynamics due to changes in the physical properties of the lubricant. Bauer (1990) pointed out that oil stiction phenomena become more important at higher compressor speeds and/or when the initial lubricant film thickness is larger, which is contrary to what has been observed in subsequent studies. According to Prasad and Panayil (1996), the volume of oil between the valve and the seat, the oil viscosity and the contact area between the valve and the seat are the most important parameters affecting oil stiction and, consequently, the valve dynamics at the time of its opening. Khalifa and Liu (1998) pointed out that the oil viscosity may not remain constant during valve opening due to heat transfer and refrigerant gas mass transfer to the lubricant oil. They concluded that the main contribution to the stiction force are the viscous effects associated with the deformation of the oil film between the valve and the seat.

In the present work, a calculation method is proposed for solving the dynamic behavior of a ring-shaped lubricant oil film between a discharge valve and the seat. The valve is subjected to an external force representing the time- (crank angle-) dependent pressure difference between the cylinder and the discharge chamber. As opposed to other models previously presented in the literature (Khalifa and Liu, 1998), the present method considers the existence of a finite amount of oil between the valve and the seat, and presents mathematical relationships for calculating the initial condition for the oil film thickness.

2. MODELING

2.1. Governing Equations

The valve opening process can be divided into two stages, before and after the break-up of the oil film. Before the film rupture, which is the main focus of this work, the dynamics of the valve is strongly influenced by the volume of lubricant between the valve and seat and by the stiffness of the valve. After the film break-up, the forces associated with the presence of the oil film, i.e., viscous, capillary action and surface tension forces, cease to exist and the dynamics of the valve is governed only by its mechanical properties (e.g., mass, stiffness and damping coefficients) and by the pressure difference between the compression and discharge chambers (in the case of a discharge valve).

The pressure variation in the lubricant film between the valve and seat shown in Fig. 1 can be calculated based on the Reynolds hydrodynamic lubrication equation (Booker, 1983). Thus,

$$\frac{\partial P}{\partial r} = \mu \frac{\partial^2 u_r(z)}{\partial z^2} \quad (1)$$

where the velocity profile $u_r(z)$ is given by,

$$u_r(z) = \frac{1}{2\mu} \frac{\partial P}{\partial r} z(z-x) \quad (2)$$

Integrating the continuity equation in differential form, one has,

$$\int_{R_{mi}}^r \int_0^x \frac{\partial}{\partial r} \left[\frac{\pi r}{\mu} \frac{\partial P}{\partial r} z(z-x) \right] dr dz + \int_{R_{mi}}^r \int_0^x \frac{\partial}{\partial z} [u_z] 2\pi r dr dz = 0 \quad (3)$$

As the seat is stationary, $u_z = 0$ at $z = 0$. By the same token, $u_z = dx/dt$ at $z = x$. Therefore, Eq. (3) can be written as,

$$\frac{\partial P}{\partial r} = \frac{6\mu}{r} \frac{dx}{x^3 dt} (r^2 - R_{mi}^2) + \frac{R_{mi}}{r} \frac{\partial P}{\partial r} \Big|_{R_{mi}} \quad (4)$$

where R_{mi} is the internal radius of the meniscus (see Fig. 1).

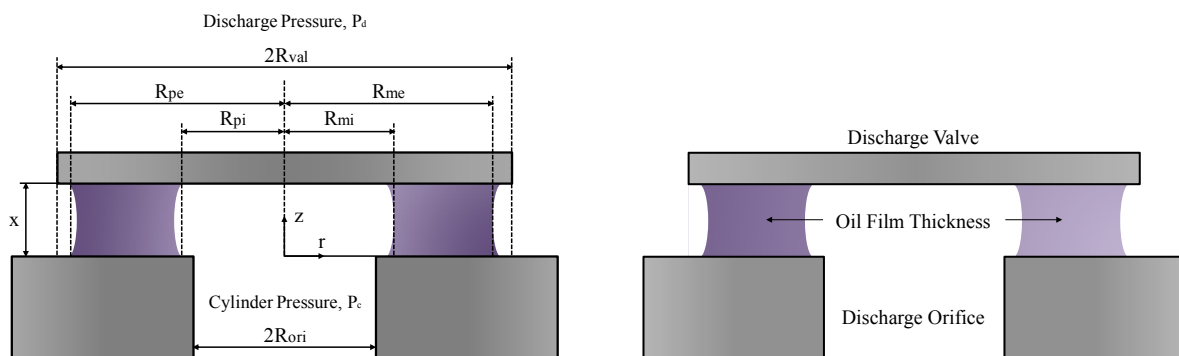


Figure 1: Geometry of the discharge valve model.

In dimensionless form, Eq. (4) is given by,

$$\frac{\partial P}{\partial \xi} = \frac{6\mu R_{mi}^2}{\xi x^3} \frac{dx}{dt} (\xi^2 - 1) + \frac{1}{\xi} \frac{\partial P}{\partial \xi} \Big|_1 \quad (5)$$

where $\xi = r/R_{mi}$ and $\lambda = R_{me}/R_{mi}$. Integration of Eq. (5) gives,

$$P(\xi) = \frac{3\mu R_{mi}^2}{x^3} \frac{dx}{dt} (\xi^2 - 2 \text{Ln}\xi) + \text{Ln}\xi \frac{\partial P}{\partial \xi} \Big|_1 + C \quad (6)$$

Application of the boundary conditions $P = P_{oc}$ at $\xi = 1$, and $P = P_{od}$ at $\xi = \lambda$ gives the values of the constants as follows,

$$C = P_{oc} - \frac{3\mu R_{mi}^2}{x^3} \frac{dx}{dt} \quad (7)$$

$$\frac{\partial P}{\partial \xi} \Big|_1 = \frac{P_{od} - P_{oc}}{\text{Ln}\lambda} - \frac{3\mu R_{mi}^2}{x^3 \text{Ln}\lambda} \frac{dx}{dt} (\lambda^2 - 2\text{Ln}\lambda - 1) \quad (8)$$

Therefore, Eq. (6) in its final form is given by,

$$P(\xi) = \frac{3\mu R_{mi}^2}{x^3} \frac{dx}{dt} \left(\xi^2 - 1 + \frac{\text{Ln}\xi}{\text{Ln}\lambda} (1 - \lambda^2) \right) + (P_{od} - P_{oc}) \frac{\text{Ln}\xi}{\text{Ln}\lambda} + P_{oc} \quad (9)$$

The pressures at the internal and external meniscii are calculated via the Young-Laplace equation using the gas pressure in the compression and discharge chambers. As the pressure difference between the two chambers at the onset of valve motion is relatively small, the gas-liquid interfacial tensions and the contact angles at the internal and external meniscii are assumed equal. Thus,

$$P_c - P_{oc} = \frac{2\gamma_{LG}^{int} \cos(\theta)}{x} \quad (10)$$

and,

$$P_d - P_{od} = \frac{2\gamma_{LG}^{ext} \cos(\theta)}{x} \quad (11)$$

Therefore, in terms of the compression and discharge chamber pressures, the pressure distribution in the liquid film is given by,

$$\Delta P = P(\xi) - P_d = \frac{3\mu R_{mi}^2}{x^3} \frac{dx}{dt} \left(\xi^2 - 1 + \frac{\text{Ln}\xi}{\text{Ln}\lambda} (1 - \lambda^2) \right) + (P_c - P_d) \left(1 - \frac{\text{Ln}\xi}{\text{Ln}\lambda} \right) - \frac{2\gamma_{LG} \cos(\theta)}{x} \quad (12)$$

The balance of forces on the valve in the z direction is given by,

$$F = \int_0^1 2\pi(P_c - P_d)R_{mi}^2 \xi d\xi + \int_1^\lambda 2\pi\Delta PR_{mi}^2 \xi d\xi - F_{\gamma LG} \quad (13)$$

where the first integral is the force due to the pressure difference between the cylinder and the discharge chamber in the region without an oil film, the second integral is the film stiction force, and the last term is due to the liquid-gas interfacial tension. Integration of Eq. (13) with Eq. (12) for the film region pressure distribution yields the following expression for the film stiction force,

$$F_{stic} = F_{visc} + F_{cap} + F_{\gamma LG} \quad (14)$$

where the first term is the force component due to viscous effects,

$$F_{visc} = \frac{3\pi\mu R_{mi}^4}{2x^3} \frac{dx}{dt} \left[\lambda^4 - 1 + \frac{2\lambda^2 - \lambda^4 - 1}{Ln\lambda} \right] \quad (15)$$

the second term is due to curvature of the meniscus (capillary force),

$$F_{cap} = \frac{2\pi\gamma_{LG} R_{mi}^2 \cos(\theta)}{x} (\lambda^2 - 1) \quad (16)$$

and the third term is the interfacial tension force,

$$F_{\gamma LG} = 2\pi R_{mi} \gamma_{LG} \sin(\theta) (\lambda - 1) \quad (17)$$

In the present study, valve motion is assumed normal to the valve seat and a single-degree-of-freedom model is used to compute the valve dynamics (Matos *et al.*, 2002).

$$m \frac{d^2x}{dt^2} = F_{ext} - F_{visc} - F_{cap} - F_{\gamma LG} - F_{spr} \quad (18)$$

where F_{ext} is the external force acting on the valve,

$$F_{ext} = \pi R_{mi}^2 (P_c - P_d) \left(\frac{\lambda^2 - 1}{2Ln\lambda} \right) \quad (19)$$

and F_{spr} is the valve spring force given by,

$$F_{spr} = kx \quad (20)$$

It is assumed that a finite oil volume is present between the valve and the seat and, for a constant contact angle, the oil volume is given by,

$$V_{ol} = \int_0^x \pi [f_{ext}(z)^2 - f_{int}(z)^2] dz \quad (21)$$

where the terms related to the external and internal meniscii are given by,

$$f_{ext}(z) = R_{val} + R_{ori} - R_{mi} + \frac{x}{2\cos(\theta)} - \sqrt{\left(\frac{x}{2\cos(\theta)}\right)^2 - \left(z - \frac{x}{2}\right)^2} \quad (22)$$

$$f_{int}(z) = R_{mi} - \frac{x}{2\cos(\theta)} + \sqrt{\left(\frac{x}{2\cos(\theta)}\right)^2 - \left(z - \frac{x}{2}\right)^2} \quad (23)$$

where R_{val} and R_{ori} are the valve and orifice radii, respectively.

The assumption of a finite oil volume between the valve and the seat is the main difference between the present model and that of Khalifa and Liu (1998), who also modeled the stiction effect using the Reynolds lubrication equation. In their work, the external and internal meniscii are assumed to be formed at the valve and orifice radii (R_{val} and R_{ori}), respectively (see Fig. 1), so the contact area between the film and the valve (the film pressure integration area) remains constant and equal to the maximum area throughout the film deformation process. The outcome of this assumption is threefold; firstly, there must be an continuous supply of oil to the gap between the valve and the seat during the valve opening process; secondly, a film rupture criterion cannot be applied to the

Khalifa and Liu (1998) model because no account is made of the change in cross section of the oil film; thirdly, the assumption of a constant contact area between the film and the valve may result in a superestimation of the oil stiction force. For completeness, the viscous, capillary and external forces according to the model of Khalifa and Liu (1998) are presented below,

$$F_{visc}^{kl} = \frac{3\pi\mu R_{ori}^4}{2x^3} \frac{dx}{dt} \left[\lambda_{kl}^4 - 1 + \frac{2\lambda_{kl}^2 - \lambda_{kl}^4 - 1}{Ln\lambda_{kl}} \right] \quad (24)$$

$$F_{cap}^{kl} = \frac{2\pi\gamma_{LG} R_{ori}^2 \cos(\theta)}{x} (\lambda_{kl}^2 - 1) \quad (25)$$

$$F_{ext}^{kl} = \pi R_{ori}^2 (P_c - P_d) \left(\frac{\lambda_{kl}^2 - 1}{2Ln\lambda_{kl}} \right) \quad (26)$$

where the dimensionless parameter λ_{kl} is defined as,

$$\lambda_{kl} = \frac{R_{val}}{R_{ori}} \quad (27)$$

2.2. Initial Condition for the Film Thickness

The knowledge of the initial oil film thickness is one of the main difficulties in the quantification of stiction effects and their influence on the dynamics of discharge valves. Here, it is assumed that (i) the initial condition for the oil film thickness occurs when the cylinder pressure equals that of the discharge chamber, and (ii) this condition is one of *static equilibrium*, i.e., there is no fluid motion in the oil film. Therefore, according to Eq. (18), the initial condition for the oil film thickness can be calculated from,

$$\frac{2\pi\gamma_{LG} R_{mi_0}^2 \cos(\theta)}{x_0} (\lambda_0^2 - 1) + 2\pi R_{mi_0} \gamma_{LG} \sin(\theta) (\lambda_0 - 1) = kx_0 \quad (28)$$

which can be solved together with the oil volume equation (Eq. 21) to determine the initial values of λ_0 , R_{mi_0} and x_0 .

3. RESULTS

The following parameters were used in the simulation: (i) valve spring coefficient (k): 270 N/m, (ii) valve density (ρ): 7860 kg/m³, (iii) valve radius (R_{val}): 4.25 mm, (iv) orifice radius (R_{ori}): 3.0 mm, (v) lubricant oil: POE ISO 10, (vi) gas: R-134a, (vii) contact angle (θ): 7.5°, (viii) interfacial tension (γ_{LG}): 12 mN/m. Film rupture was assumed to take place when the difference between the external and internal meniscii (R_{me} and R_{mi}) fell below 10 μ m. Film rupture was also used as the stop criterion in the present model.

A prescribed cylinder pressure variation typical of hermetic household refrigeration compressors was assumed (see Fig. 2). In this case, a crank angle of 0° corresponds to the instant at which the cylinder pressure is equal to the discharge chamber pressure, which was taken as the saturation pressure of R-134a at 54.4°C (~14.7 bar).

Figure 3 illustrates the behavior of the stiction force and of the film thickness (valve displacement) as a function of the oil volume for the static equilibrium case (Eq. 28). It should be noted that the oil volume is normalized by its maximum value, which corresponds to the ring-shaped volume between the valve and the seat, without the curvature associated with the external and internal meniscii. Thus,

$$V_{ol}^* = \pi x_0 (R_{val}^2 - R_{ori}^2) \quad (29)$$

where x_0 is the positive root of the following equation derived from the force balance at the static equilibrium condition,

$$kx_0^2 = 2\pi\gamma_{LG} \cos(\theta) (R_{val}^2 - R_{ori}^2) + 2\pi\gamma_{LG} \sin(\theta)(R_{val} - R_{ori}) x_0 \quad (30)$$

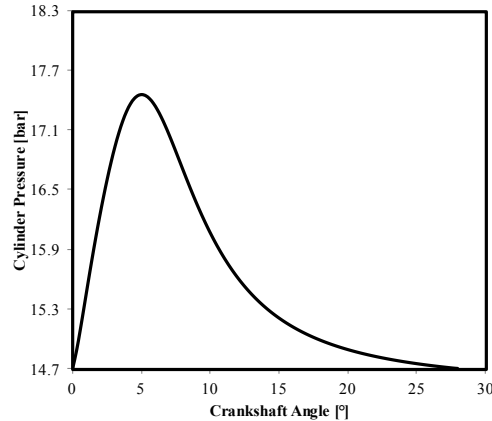


Figure 2: Cylinder pressure as a function of the crank angle.

As can be seen from Fig. 3, at the maximum volume, both the Khalifa and Liu (1998) and the present model give identical results for the stiction force and the film thickness at the static equilibrium condition. However, as the oil volume is decreased, significant discrepancies are observed between the models, which are, as mentioned previously, due to the fact that in the Khalifa and Liu (1998) model the space underneath the valve is completely filled with oil. Therefore, in their model, the stiction force at the static condition is uniquely dependent on the film thickness, which must decrease with the oil volume, thus giving rise to larger values of the stiction force (see Fig. 3.a). In the present model, as the oil volume is reduced, the area of the seat occupied by the liquid film (pressure integration area) is also reduced, which results in a smaller value of the stiction force.

As far as the film thickness is concerned (Fig. 3b), its increase with the oil volume stems from the increase in the stiction force that, at the static equilibrium condition, acts in the positive z direction against the valve spring force (whose tendency is to close the valve).

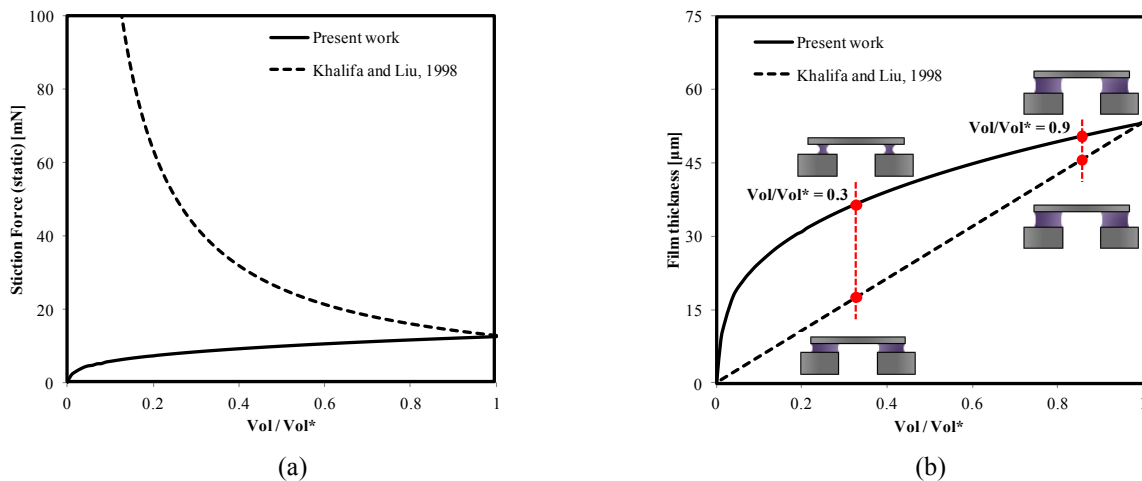


Figure 3: Variation of the (a) stiction force and (b) film thickness as a function of the oil volume for the static equilibrium condition.

Figure 4 shows the behavior of the discharge valve displacement and velocity as a function of the crank angle for a fixed oil content that is equal to the maximum possible value calculated according to Eq. (29). As can be seen, the valve opening process can be divided, before the rupture of the film, into two different parts. The *quasi-static region* is characterized by a resistance of the oil film to deformation (viscous effect) and by small values of valve

displacement. Conversely, in the *dynamic region*, the external force prevails over the oil stiction and valve spring forces, thus giving rise to larger values of valve displacement and velocity.

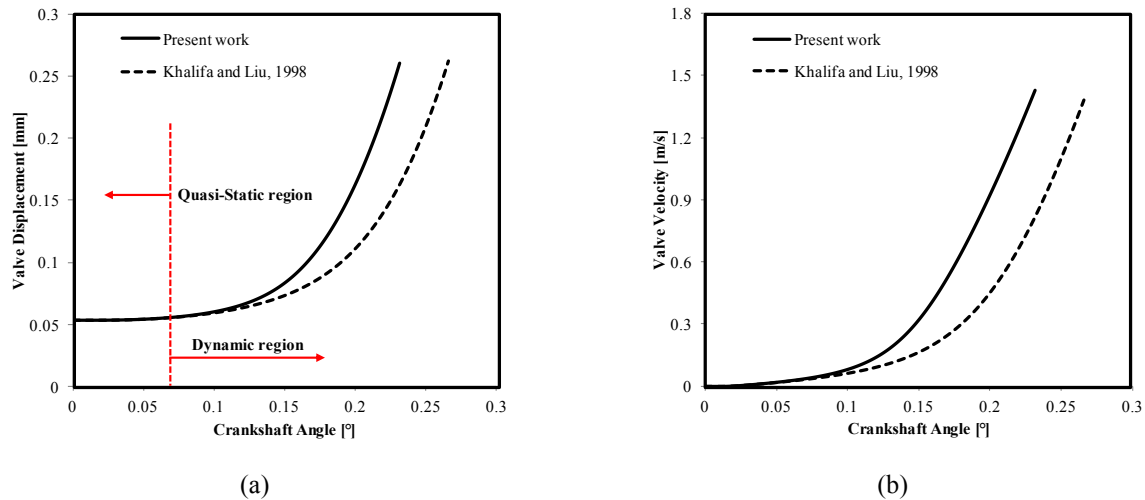


Figure 4: Discharge valve kinematic parameters: (a) Valve displacement; (b) Valve velocity.

It is important to mention that both the present and the Khalifa and Liu (1998) models yielded qualitatively similar results, bringing about the occurrence of the quasi-static and dynamic regions. While in the Khalifa and Liu (1998) approach a constant (maximum) integration area delimited by the valve and orifice radii is adopted, the present model uses a variable area defined by the points of contact between the oil and the valve and the oil and the seat (R_{mi} , R_{me} , R_{pi} and R_{pe}). This gives rise to values of stiction force and duration of the quasi-static regions that are smaller than those calculated using the Khalifa and Liu (1998) model.

Figure 5 presents the variation of the meniscus (capillary plus interfacial tension) and viscous components of the stiction force as a function of the crank angle. Again, the maximum oil volume has been used, which explains the convergence of the two models to the same value of the meniscus force as the initial condition (static equilibrium) is approached (see Fig. 5.a). In the dynamic region, the discrepancies between the two models become more significant as the crank angle (and the cylinder pressure) increases. This is because, once again, the pressure integration area in the film region decreases with an increase in the valve displacement for the present model, while it remains constant and equal to the projected area of the valve on the seat in the Khalifa and Liu (1998) model.

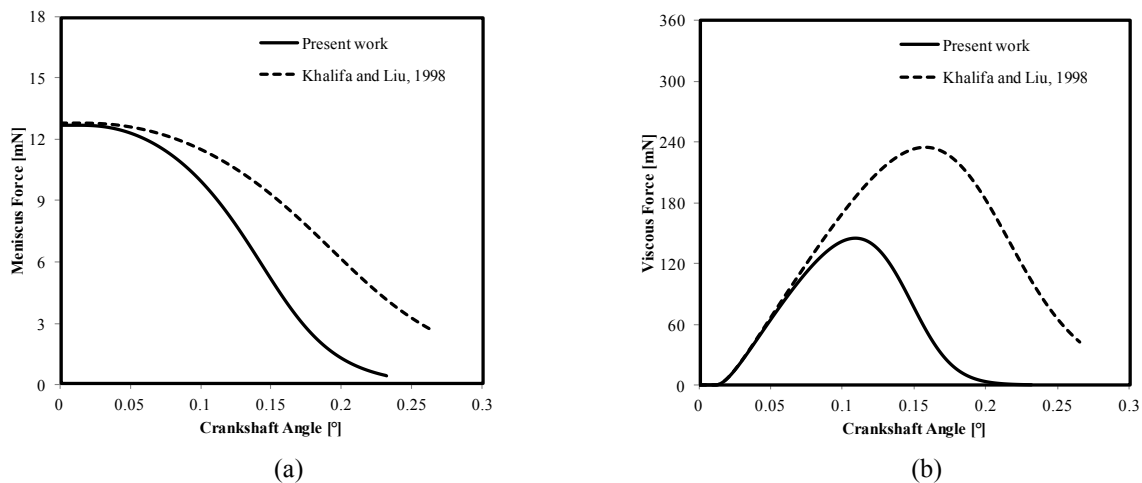


Figure 5: Oil film dynamic parameters: (a) Meniscus force; (b) Viscous force.

Figure 6 shows the behavior of the stiction force and of the resultant force (inertial force) as a function of the crank angle. At the initial condition (static equilibrium), a non-zero value of the stiction force points out the existence of a valve spring force, opposite in sign, associated with the initial oil film thickness. This scenario results in a larger cylinder pressure needed to overcome the forces that oppose the valve motion at the initial condition (i.e., discharge pressure, valve spring force, capillary and surface tension forces). As can be seen in Fig. 5(a), the viscous component, which is the dominant force during the dynamic region, is the largest obstacle to be surpassed by the external force, since the meniscus force is important, only, at the initial (static) condition.

The discharge valve operates according to the pressure difference between the cylinder and the discharge chamber. As mentioned above, ideally, it should open instantaneously as the cylinder pressure becomes equal to that in the discharge chamber. However, the valve opening is delayed as a result of the valve inertial and spring forces, and of the oil stiction force. Figure 6(b) presents the resultant force (i.e., the difference between the external force and the oil stiction force) calculated according to the present and Khalifa and Liu (1998) models. As can be seen, the largest value of the resultant force in the dynamic region for the present model implies that the valve opening process calculated with the Khalifa and Liu (1998) model takes longer and, consequently, larger cylinder pressures are needed to surpass the combined effect of viscous effects and valve stiffness in their case.

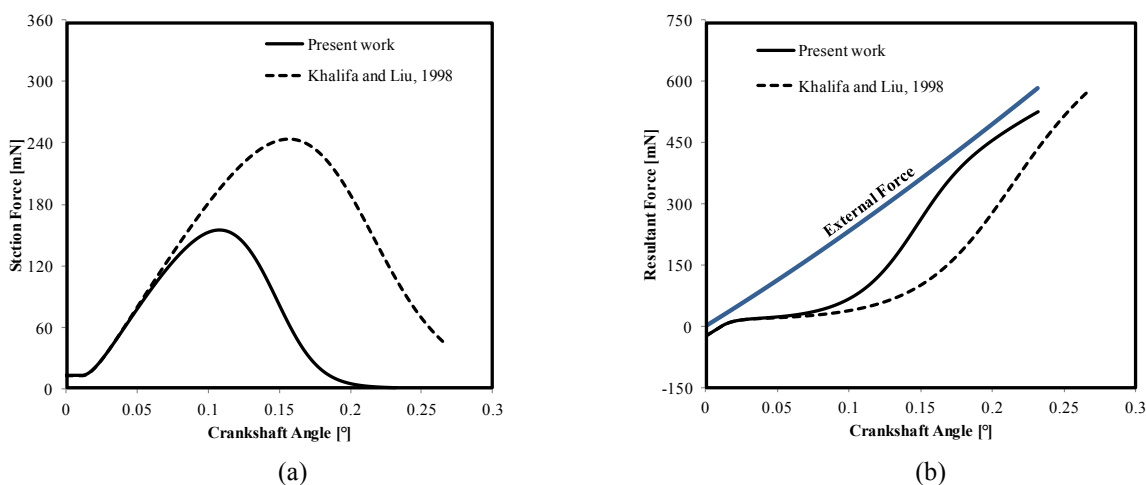


Figure 6: Behavior of the combined forces: (a) Stiction force; (b) Resultant force.

4. CONCLUSIONS

A new model that takes into account the volume of oil between the valve and the seat has been proposed to quantify the oil stiction phenomena in discharge valves. The model was also applied to determine the initial condition for the oil film thickness, when the pressure difference acting on the valve is nil (static equilibrium condition). For the conditions investigated here (typical of discharge valves of compressors for domestic refrigeration), the viscous component of the stiction force was found to be the dominant effect opposing the valve opening. This is in line with the conclusions of Khalifa and Liu (1998).

The stiction force calculated according to the model of Khalifa and Liu (1998) was systematically larger than those yielded by the present model. These differences were found to be related to the consideration of a finite volume of oil between the valve and the seat by the present method, which seems to be more physically consistent than the assumption of an infinite supply of oil to the valve-seat gap adopted by Khalifa and Liu (1998). Nevertheless, experimental work is needed in order to validate the present model.

NOMENCLATURE

F	Force	[N]	Subscripts:	
k	Spring stiffness	[N/m]	c	Cylinder
m	Effective valve mass	[kg]	cap	Capillary
P	Pressure	[Pa]	d	Discharge
R	Radius	[m]	ext	External
t	Time	[s]	int	Internal
V_{ol}	Volume	[m ³]	me	External meniscus
x	Film thickness	[m]	mi	Internal meniscus
z	Axial coordinate		oc	Oil/Cylinder
Greek Letters:			od	Oil/Discharge chamber
γ	Surface tension	[N/m]	ori	Orifice
θ	Contact angle	[rad]	pe	External point
λ	R_{mc}/R_{mi} ratio		pi	Internal point
μ	Dynamic viscosity	[Pa.s]	val	Valve
ξ	r/R_{mi} ratio		visc	Viscous

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