2012

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Alves, Marcus; Barbosa Jr., Jader; and Prata, Alvaro, "Analytical and CFD Modeling of the Fluid Flow in an Eccentric-Tube Centrifugal Oil Pump for Hermetic Compressors" (2012). *International Compressor Engineering Conference*. Paper 2045.
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Analytical and CFD Modeling of the Fluid Flow in an Eccentric-Tube Centrifugal Oil Pump for Hermetic Compressors

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ABSTRACT

A model is presented for calculating the volume flow rate in an eccentric-tube centrifugal oil pump for reciprocating compressors. The oil pump assembly consists of a pick-up tube with an inclined inlet that is mounted off-center with the crankshaft symmetry axis. The pick-up tube is connected to the shaft channel; a helical groove machined on the crankshaft that facilitates the supply of lubricant to the bearings. The analytical models for the pick-up tube and shaft channel were developed independently and were coupled via a numerical procedure to determine the steady-state volume flow rate in the assembly. The steady-state results were verified against a CFD model, which was also used to evaluate the oil flow in the pump assembly during a start-up transient. A parametric analysis was conducted to quantify the influence of the pump geometric parameters, i.e., the pick-up tube diameter and the shaft eccentricity.

1. INTRODUCTION

The purpose of the oil supply system in a hermetic reciprocating compressor is to provide, as swiftly as possible, a sufficient amount of lubricant oil to the journal bearings and other sliding surfaces, such as the piston-cylinder gap. Other desired features of the oil supply system are to function over a wide range of rotational speeds and to consume the least amount of energy.

The mathematical modeling approaches proposed up to the present date for calculating the fluid flow in oil supply systems can be divided into analytical (or semi-analytical) and those based on computational fluid dynamics (CFD). Kerpici *et al.* (2010) and Lückmann *et al.* (2009) reviewed the recent research efforts associated with the numerical simulation of the oil flow in pumping devices for rotary and reciprocating compressors. However, despite the recent advances in the processing speed of computers and numerical techniques, the large computing times and their associated costs are still an issue that prevents the use of sophisticated numerical analyses (i.e., transient and real geometry) in design and optimization of oil pumping systems in hermetic compressors.

Although hermetic compressor design does not generally start with the selection or design of the oil supply system, it becomes an important concern when the limiting operating conditions of a given bearing system are approached, for example, at low compressor rotation speeds. Therefore, simplified models that can be run at low computing costs are valuable in order to identify ranges of improved performance for a particular set of geometric parameters. So far, a considerable number of semi-analytical models of oil supply systems for hermetic compressors have been proposed in the literature (Cho *et al.*, 2002; Kim *et al.*, 2003; Kim, 2005; Alves *et al.*, 2011). It is obvious that the biggest advantage of semi-analytical models is the much shorter computing times than CFD-based models. However, the dependence on empirical correlations for a specific system component can be their main drawback, especially when a geometry optimization of the oil supply system is pursued.

The purpose of the present paper is to combine CFD modeling and analytical techniques to predict the oil flow rate in the eccentric-tube centrifugal oil pumping system shown schematically in Fig. 1. The lower part of the pump consists of an oil pick-up tube whose tip is immersed in the oil sump. The tube is mounted off-center with the crankshaft symmetry axis so that centrifugal forces originating from the spinning motion elevate the oil through the tube. The inlet of the pick-up tube is slanted towards the axis of rotation to facilitate the flow of oil into the system.

After leaving the pick-up tube, the oil enters the crankshaft, where it flows through a helical groove machined on its outer surface (the shaft channel) and feeds the shaft bearings. From the top of the shaft, the excess oil is expelled into the crankcase environment and falls back into the sump at the bottom of the crankcase. In the pick-up tube, the driving force for fluid flow is the centrifugal (body) forces that accelerate the oil in the slanted portion of the tube. Conversely, viscous shear contributes to the pumping of the oil in the crankshaft channel due to the relative motion between the rotating shaft and the bearings that are incorporated in the cylinder housing (not shown in Fig. 1). The fluid flow channel geometry in the eccentric-tube centrifugal oil-pumping system is shown in Fig. 2.

Figure 1: The eccentric tube centrifugal oil pumping system.

Figure 2: (a) Fluid flow domain and channels in the oil pumping system. (b) CFD mesh.

2. CFD MODELING

The fluid flow in the eccentric-tube centrifugal oil pumping system was solved numerically using the FLUENT package (Fluent, 2006). The following simplifying assumptions have been adopted: (i) there is no refrigerant dissolved in the oil; (ii) the flow is isothermal; (iii) the physical properties of the fluids (oil and refrigerant) are constant.

The computational mesh shown in Fig. 2 is comprised of approximately 400,000 tetrahedral volumes whose sizes have been distributed so that the grid refinement is increased close to the solid walls and interfaces between adjacent domains. The CFD model geometry was formulated so that the pressure in the upper reservoir is the same as that in the oil sump at all times. This is achieved via the pressure equalization boundary in the oil sump domain, which acts as a virtual connection between the two regions and eliminates the need of a larger computational grid, for example, encompassing the whole internal crankcase volume.

As in previous work (Lückmann *et al.*, 2009), the boundary conditions have been established so as to allow the existence of three distinct domains: oil sump (static), pick-up tube and shaft (moving), and upper reservoir (static). During the simulation, the oil that is elevated to the upper reservoir does not return to the sump, although the static pressure in the two static domains is maintained the same by the pressure equalization. The boundary conditions for the mass and momentum equations were specified as follows: (i) no-slip between fluid and solid wall in the sump, (ii) prescribed pressure at the base of the upper and at the top of the lower reservoirs, (iii) angular velocity of the walls of the pick-up tube-shaft domain set equal to the speed of the electrical motor (moving-mesh), (iv) all remaining walls and interfaces between domains are stationary. The initial condition is such that the oil is stationary prior to the start-up of the electric motor. The shaft has a constant speed from the time it is set into motion. The conditions of the numerical simulation are as follows: (i) shaft speed: 3000 rpm; (ii) length of the pick-up tube submerged in the oil sump: 15 mm; (iii) pressure: 1.2 bar; (iv) temperature: 75°C; (v) refrigerant: R-600a; (vi) lubricant oil: ISO VG 5; (vii) surface tension: 0.012 N m⁻¹; (viii) refrigerant density: 4.289 kg m⁻³; (ix) gas viscosity:

13.78×10^{-6} Pa s; (x) oil density: 922.02 kg m^{-3} ; (xi) oil viscosity: 3.64×10^{-3} Pa s. The geometric variables associated with the pick-up tube are as follows: $\beta = 72^\circ$, $D = 3 \text{ mm}$ and $e = 4.5 \text{ mm}$.

A 2nd-order upwind scheme has been utilized for discretizing the equations of motion together with a body force weighted scheme for the pressure interpolation. The PISO algorithm (Issa, 1986) was used for the pressure-velocity coupling for it handles well the small time steps that had to be employed in the present simulation. The convergence criteria for all velocities and mass were set at 10^{-4} RMS. The free-surface two-phase gas-liquid flow in the oil pumping system was solved using the volume of fluid (VOF) method. The interface reconstruction was resolved using the PLIC algorithm. The flow was modeled as turbulent and the default constants of the k - ϵ model provided by the commercial code have been used. A time step of $10 \mu\text{s}$ was used in the simulations.

Figure 3 shows the results for the contours of volumetric fraction for the oil phase in the fluid flow channel (pick-up tube and shaft channel) as a function of time. Since the acceleration of the motor is assumed infinite at $t = 0 \text{ s}$, there is already some oil in the slanted portion of the pick-up tube at 4 ms (see Fig. 3.a). It takes approximately 35 ms for the oil to reach the outlet of the pick-up tube, as demonstrated in Fig. 3.b. At 72 ms (see Fig. 3.c), there already is a significant amount of oil in the lower bearing. At a certain time between 80 and 100 ms, according to Fig. 3.d, the flow in the pick-up tube becomes established in the sense that the phase interface assumes the configuration that will be retained until the end of the simulation (when steady-state is observed). In this configuration, shown in detail in Fig. 4.a, a stratified flow pattern is initiated at the height of the degassing orifice and develops further downstream into the pick-up tube. At around 135 ms, according to Fig. 3.e, there is a significant amount of oil in the upper bearing and already a net flow into the upper reservoir. At approximately 200 ms, steady-state is already established, i.e., the time-average oil flow rate is uniform as a function of both space and time in the oil pumping system.

As seen in Fig. 4.a (steady-state), the liquid fills the lower portion of the tube and a stratified free-surface flow is established downstream of the degassing orifice (modeled as a pressure equalization boundary). This free-surface flow is similar in nature to that in an open channel since, due to the degassing orifice, the pressure in the upper part of the pick-up tube (downstream of the orifice) remains almost constant and there is no net gas flow into the shaft channel (see Fig. 3.f). Figure 4.b. shows the phase distribution in a cross-section of the upper part of the pick-up tube located at 10 mm below the inlet of the shaft channel. Apart from small local disturbances, the interface is virtually flat, giving rise to a stratified flow configuration.

In two-phase stratified flows in horizontal tubes, the shape of the interface is correlated by the Bond number, which relates gravitational and surface tension forces. According to Ng *et al.* (2001), if

$$Bo = \frac{(\rho_L - \rho_G)gD^2}{\sigma} > 250 \quad (1)$$

then the gas-liquid interface can be treated as flat. Here, by replacing the acceleration due to gravity by the centripetal acceleration, $\omega^2 e$, a modified Bond number of 440 was calculated. This corroborates the findings of the present CFD simulations that the interface is flat and, for the analytical model presented in Section 3, the gas-liquid interface in the pick-up tube will be treated as such.

Figure 5 presents the calculated pumped oil flow rates at the shaft outlet as a function of time. Again, approximately 200 ms are needed to reach steady-state. In the present simulation, the time required to reach steady-state was quite close to the “climbing-time” (Lückmann *et al.*, 2009), which was somewhere between 100 and 200 ms. The steady-state oil flow rate was calculated at approximately 62.7 ml min^{-1} . The dotted line corresponds to the steady-state pumped mass flow rate obtained with the semi-analytical model described in Section 3.

3. ANALYTICAL MODELING

The physical domain is divided into two regions, i.e., the pick-up tube and the shaft channel. The models for the two regions are developed independently and, to obtain the oil flow rate for a given set of operating conditions, they are solved together via an iterative procedure. The following assumptions have been adopted: (i) steady-state, (ii) constant physical properties, (iii) constant angular velocity, (iv) laminar flow.

Figure 3: Contours of volume oil fraction as a function of time (in ms): (a) 4, (b) 33, (c) 72, (d) 94, (e) 133, (f) 194.

The geometry of the pick-up tube model is shown in Fig. 6. The shape of the gas-liquid interface in the upper part is calculated assuming a rigid-body rotation of the fluid around the shaft symmetry axis (Fox *et al.*, 2008). Assuming that (i) there is no pressure change across the interface (flat interface assumption) and (ii) the momentum change in the flow direction is negligible, one has (in terms of dimensionless variables),

$$r^* dr^* = \frac{D}{2} \frac{g}{(\omega e)^2} dz^* \quad (2)$$

Integration from the point of onset of free-surface flow, at $z = H_B$, to a general position, z , gives the radial position of the free-surface in the vertical section of the pick-up tube as follows,

$$r_i^* = \left[\left(1 - \frac{D}{2e} \right)^2 + \frac{Dg}{(\omega e)^2} \left(z^* - \frac{2H_B}{D} \right) \right]^{\frac{1}{2}} \quad (3)$$

and the height of the liquid film in the pick-up tube is given by,

$$H_L^* = \frac{H_L}{e} = 1 + \frac{D}{2e} - r_i^* = \left[\left(1 - \frac{D}{2e} \right)^2 + \frac{Dg}{(\omega e)^2} \left(z^* - \frac{2H_B}{D} \right) \right]^{\frac{1}{2}} \quad (4)$$

Figure 4: (a) Phase distribution at steady-state. (b) Oil (red) and gas (yellow) in a cross-section of the tube at 10 mm below the shaft inlet.

Figure 5: CFD and simplified (semi-analytical) model predictions of the oil flow rate at the shaft outlet .

In Eqs. (3) and (4), the point of onset of free-surface flow, $z = H_B$, is *a priori* unknown. In order to calculate H_B , it was assumed, based on observations of the CFD results for steady-state, that at the inlet of the shaft channel the liquid film height was approximately one quarter of the tube diameter. Thus, the position along the vertical axis where the free-surface flow is initiated is given by,

$$H_B^* = \frac{2H_B}{D} = \frac{2H_S}{D} - \frac{\omega^2 e}{g} \left(\frac{1}{2} - \frac{3}{16} \frac{D}{e} \right) \quad (5)$$

where H_S is a geometric parameter, i.e., the height of the pick-up tube.

Due to the short length of the slanted section of the pick-up tube, frictionless flow was assumed between points 1 and 2 in Fig. 6. Thus, the linear momentum conservation equation in the slanted section is given by,

$$\frac{\partial p^*}{\partial x^*} = - \left(\frac{g}{\omega^2 e} \sin \beta - x^* \cos^2 \beta \right) \quad (6)$$

Assuming fully developed flow between points 2 and 3 (see Fig. 6), one has,

$$\frac{1}{r_1^*} \frac{\partial}{\partial r_1^*} \left(r_1^* \frac{\partial u^*}{\partial r_1^*} \right) = k \quad (7)$$

where k is dimensionless constant given by,

$$k = \frac{\bar{P}_w^2 \omega e}{4\pi^2 \nu_L} \left[\left(\frac{p_3^* - p_2^*}{l + l_c} \right) + \frac{1}{\rho_L (\omega e)^2} \left(\bar{\rho} g - \frac{dp}{dz} \Big|_{sc} \right) \right] \quad (8)$$

where $\bar{P}_w$ is the average liquid wetted perimeter in the region $2H_B/D \leq z^* \leq 2H_S/D$, calculated assuming a stratified two-phase flow regime (Alves, 2007). l is the length of the vertical section of the pick-up tube defined as,

$$l = H_S - e \tan \beta \quad (9)$$

l_c is the equivalent tube length associated with the minor loss due to the elbow connecting the slanted and the vertical segments. It was calculated from an interpolation of the data provided by Fox *et al.* (2008),

$$\frac{l_c}{D} = 2.09 \exp[0.038(90 - \beta)] \quad (10)$$

where β is in degrees. In Eq. (8), the first term inside the second pair of brackets is the gravitational pressure gradient associated with the two-phase region in the vertical section of the pick-up tube. The apparent density of the two-phase mixture given by,

$$\bar{\rho} = \rho_L (1 - \bar{\alpha}) + \rho_G \bar{\alpha} \quad (11)$$

where $\bar{\alpha}$ is the average gas fraction of the stratified two-phase flow in $2H_B/D \leq z^* \leq 2H_S/D$ (Alves, 2007).

The second term in the second pair of brackets on the right-hand side of Eq. (8) is the pressure gradient in the shaft channel, which is calculated via the pressure drop and fluid flow model for the shaft channel region to be discussed next. The pressure difference between points 2 and 3 is given by,

$$p_3^* - p_2^* = \frac{g}{(\omega e)^2} (h_o + e \tan \beta) - \frac{1}{2} \quad (12)$$

where h_o is the vertical distance between the submerged tip of the pick-up tube and the free interface of the oil sump, which constitutes a static pressure head. Thus, by incorporating the above relationships in Eq. (8), and integrating Eq. (7) with the following boundary conditions,

$$u^*(1) = 0; \quad \frac{\partial u^*}{\partial r_1^*}(0) = 0 \quad (13)$$

one can calculate the radial velocity profile in the vertical segment of the pick-up tube. Integration of this velocity profile in the channel cross-sectional area gives the volume flow rate in the pick-up tube as follows,

$$Q_{pt}^* = \frac{Q_{pt}}{\bar{P}_w^2 \omega e} = - \frac{\bar{P}_w^2 \omega e}{128\pi^3 \nu_L} \left[\left(\frac{p_3^* - p_2^*}{l + l_c} \right) + \frac{1}{\rho_L (\omega e)^2} \left(\bar{\rho} g - \frac{dp}{dz} \Big|_{sc} \right) \right] \quad (14)$$

The basic geometry of the shaft channel is shown in Fig. 7. It consists of a fixed outer barrel and a rotating shaft that incorporates the helix flights. Since an identical model was solved for the screw pump oil supply system (Alves *et al.*, 2011), only the final result will be presented here. The reader is referred to Alves *et al.* (2011) for further details of the model and its solution. Thus, the resulting expression for the shaft channel flow rate is given by,

$$Q_{sc}^* = \frac{Q_{sc}}{\Omega \tilde{H}^2 R_B} = -\frac{1}{2}(\tilde{\kappa} - 2)\tilde{\gamma} \cos \tilde{\beta} + \frac{2}{\tilde{\gamma}} \sum_{i=1}^{\infty} \frac{[1 - (-1)^i]}{\tilde{\alpha}_i^2} \left[\tilde{C}_1 (e^{\tilde{\alpha}_i} - 1) + \tilde{C}_2 (e^{-\tilde{\alpha}_i} - 1) + \frac{\tilde{\kappa}[1 - (-1)^i]}{\tilde{\alpha}_i^3} \right] \quad (15)$$

where,

$$\tilde{C}_1 = -\left[\frac{\tilde{\alpha}_i^2 \cos \tilde{\beta} - \tilde{\kappa}(e^{-\tilde{\alpha}_i} - 1)}{\tilde{\alpha}_i^3 (e^{-\tilde{\alpha}_i} - e^{\tilde{\alpha}_i})} \right] [1 - (-1)^i]; \quad \tilde{C}_2 = -\left[\frac{\tilde{\alpha}_i^2 \cos \tilde{\beta} - \tilde{\kappa}(e^{\tilde{\alpha}_i} - 1)}{\tilde{\alpha}_i^3 (e^{-\tilde{\alpha}_i} - e^{\tilde{\alpha}_i})} \right] [1 - (-1)^i] \quad (16)$$

Figure 6: Schematic description of the analytical model for the pick-up tube.

Figure 7: Basic geometry and system of coordinates adopted for the shaft channel.

The pumped mass flow rate is obtained via a coupled solution procedure for Eqs. (14) and (15) outlined as follows: (i) The flow rate in the pick-up tube is calculated with Eq. (14) assuming that the shaft channel on top of it is devoid of any oil; (ii) the pressure gradient in the shaft channel corresponding to this flow rate, i.e., with $Q_{sc} = Q_{pt}$, is determined via the bisection method using Eq. (15); (iii) this resulting pressure gradient in the shaft channel is incorporated in the new estimate of the flow rate in the screw pump (i.e., it is updated in Eq. (8)), which is compared to the flow rate previously calculated in (i). The procedure is repeated until convergence is obtained. The overall tolerance used was a normalized residuum of 10^{-5} calculated by the following expression,

$$RES = abs \left(\frac{Q_{pt,j-1} - Q_{pt,j}}{Q_{pt,j}} \right) \quad (17)$$

where j stands for the iteration number.

4. RESULTS

In order to verify the agreement between the semi-analytical and CFD models, Fig. 8 shows the steady-state pumped volume flow rate in the pick-up tube alone (without being connected to the shaft channel) as a function of the inclination angle of the slanted tube and of the tube diameter, for an eccentricity of 3 mm. The flow rate increases with the tube diameter, and reaches a maximum for an inclination angle of approximately 60° . The agreement between the analytical model (solid lines) and the CFD results (discrete points) is reasonable, though the point of maximum predicted by the CFD approach is not picked-up by the semi-analytical model, which also tends to under-

predict the CFD data for larger flow rates. Similar conclusions could be drawn for cases with an eccentricity of 4.5 mm. In this case, the maximum flow rate was observed for an inclination angle of 68°.

The pumped volume flow rate in the entire assembly (pick-up tube plus shaft channel) predicted by the semi-analytical and CFD models has been shown in Fig. 5 for the conditions outlined in Section 2. The analytical model over predicts the CFD steady-state volume flow rate by approximately 15%, which can be considered acceptable for engineering design calculations give the tremendous decrease in computing time associated with the analytical model.

Figure 9 illustrates the behavior of the pumped volume flow rate in the pick-up tube-shaft assembly as a function of the tube diameter, D , and shaft eccentricity, e . These results were generated for a fixed slanted tube inclination angle of 65°, which is representative of the values that gave the maximum flow rates in the CFD analysis. The shaft rotation speed was kept fixed at 3000 rpm. As can be seen, increasing both D and e contributes to an increase of the volume flow rate, since a larger diameter generates less friction and a larger eccentricity induces a higher centrifugal acceleration. Nevertheless, the influence of the shaft eccentricity on the volume flow rate seems to be more pronounced for the ranges of parameters investigated in this analysis. It should be mentioned, however, that increasing the shaft eccentricity also contributes to decreasing the contact area between the piston and the cylinder. While this contributes to reducing the viscous dissipation in the piston-cylinder gap (lower frictional losses), it may also produce more refrigerant leakage through the gap. Thus, an integrated analysis that takes into account the mass and energy balances in the cylinder and the oil supply system modeling is recommended in order to determine the most appropriate combination of pick-up tube diameter and shaft eccentricity.

Figure 8: Volume flow rate (pick-up tube only).

Figure 9: Volume flow rate (entire assembly).

Figure 10 shows the influence of the screw helix ratio of the shaft channel on the pumped volume flow rate as a function of the pick-up tube diameter, for a shaft eccentricity of 4.5 mm. The screw helix ratio is defined as,

$$\tau = \frac{\Lambda}{\tilde{R}_B} = 2\pi \tan \tilde{\beta} \quad (18)$$

The volume flow rate increases with the shaft channel screw pitch, since smaller values of screw helix ratio are associated with longer channels and hence a larger frictional pressure drop. Nevertheless, for a fixed pick-up tube diameter, the variation of the volume flow rate with the screw pitch is quite small and, for the conditions evaluated in the present study, the shaft channel screw pitch is not as important in determining the volume flow rate as the shaft eccentricity and the pick-up tube diameter.

The effect of the oil dynamic viscosity on the pumped volume flow rate is illustrated in Fig. 11, together with the influence of the pick-up tube submerged depth, h_o . The observed behavior is in line with results previously obtained

in the literature for oil supply systems that rely on centrifugal action for pumping the lubricant, such as reed-type pumps (Cho *et al.*, 2002; Kim and Lancey, 2003; Kim, 2005; Lückmann *et al.*, 2009). As can be seen, the flow rate decreases with increasing viscosity, as the centrifugal force is independent from the fluid viscosity and the friction losses are directly proportional to it. A higher flow rate can be obtained with a more viscous fluid if a larger value of h_o is used (i.e., a larger volume of oil in the sump), which increases the hydrostatic head acting on the pump.

Figure 10: Oil flow rate (entire assembly) as a function of the shaft channel screw helix ratio.

Figure 11: Oil flow rate (entire assembly) as a function of viscosity and of the pick-up tube submerged depth.

5. CONCLUSIONS

Numerical (CFD) and semi-analytical techniques were used in the simulation of an eccentric tube centrifugal oil pumping system for hermetic compressors. While CFD is capable of predicting the transient behavior of the oil pump during the electric motor start-up, the semi-analytical model can only predict the steady-state pumped mass flow rate. The agreement between the two methods in the prediction of the steady-state flow rate is within 15%.

The pumped volume flow rate was seen to increase with the pick-up tube diameter and shaft eccentricity, as larger diameters generate less friction and larger eccentricities induce a higher centrifugal acceleration. The effect of the shaft eccentricity was more pronounced for the conditions investigated here. The influence of the shaft channel screw pitch was also investigated, but it was concluded that it is not as important in determining the volume flow rate as the shaft eccentricity or the pick-up tube diameter.

NOMENCLATURE

Roman					
D	diameter	(m)	r_l	ancillary radial coordinate	(m)
e	eccentricity	(m)	$r_1^* = 2\pi r_l / \bar{P}_w$	dimensionless radial coordinate	(-)
g	acceleration due to gravity	(m/s ²)	R_B	barrel radius (shaft)	(m)
p	pressure	(N/m ²)	u	velocity	(m/s)
$p^* = p / \rho_L \omega^2 e^2$	dimensionless pressure	(-)	$u^* = u / \omega e$	dimensionless velocity	(-)
Q	flow rate	(m ³ /s)	x	axial coordinate	(m)
r	radial coordinate	(m)	$x^* = x / e$	dimensionless axial coordinate	(-)
$r^* = r / e$	dimensionless radial coordinate	(-)	z	axial coordinate	(m)
			$z^* = 2z / D$	dimensionless axial coordinate	(-)

Greek			ρ	density	(kg/m ³)
$\tilde{\alpha}_i$	eigenvalues	(–)	σ	surface tension	(N/m)
β	slanted tube angle	(rad)	τ	screw helix ratio (shaft)	(–)
$\tilde{\beta}$	screw helix angle (shaft)	(rad)	Subscripts		
$\tilde{\gamma}$	screw aspect ratio (shaft)	(–)	G	gas	
Λ	screw pitch (shaft)	(m)	L	liquid	
$\tilde{\kappa}$	screw curvature ratio (shaft)	(–)	pt	pick-up tube	
Ω	angular velocity	(rad/s)	sc	shaft channel	
ν	kinematic viscosity	(m ² /s)			

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ACKNOWLEDGEMENTS

The material presented in this paper is a result of a long-standing technical-scientific partnership between UFSC and Embraco. Financial support from FINEP and CNPq through grant No. 573581/2008-8 (National Institute of Science and Technology in Cooling and Thermophysics) is duly acknowledged.