

2014

Friction Model Development for a reciprocating compressor

Parag Mantri

GE, India, parag.mantri@ge.com

Bhavesh Kachhia

GE, India, Bhavesh.Kachhia@ge.com

Bhaskar Tamma

GE, India, Bhaskar.Tamma@ge.com

Aditya Bhakta

GE, India, aditya.bhakta@ge.com

Follow this and additional works at: <https://docs.lib.purdue.edu/icec>

Mantri, Parag; Kachhia, Bhavesh; Tamma, Bhaskar; and Bhakta, Aditya, "Friction Model Development for a reciprocating compressor" (2014). *International Compressor Engineering Conference*. Paper 2263.
<https://docs.lib.purdue.edu/icec/2263>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

Parametric study of friction model for a reciprocating compressor

Parag MANTRI*, Bhaskar TAMMA, Bhavesh KACHHIA, Aditya BHAKTA

General Electric Global Research Center,
Bangalore, Karnataka, India
Parag.Mantri@ge.com

* Corresponding Author

ABSTRACT

Friction loss has a significant impact on the performance of a reciprocating compressor. Piston-cylinder friction is a major contributor compared to the other contributors like thrust bearing, piston pin and crank. In the present work, the piston-cylinder interaction inside a small hermetic compressor is modeled using the Reynolds equation which is solved using finite difference method. The model provides key compressor design parameters such as minimum oil film thickness, oil pressure distribution between piston-cylinder, normal forces and friction power loss. Using the above formulation, parametric study has been carried out to infer the impact of different operating and design considerations.

Keywords: compressor, friction, valve design, minimum oil film thickness, pressure distribution, parametric study

1. INTRODUCTION

Reciprocating compressors technology is mature but still has high friction losses. Measuring friction loss and friction forces is extremely complicated yet highly important for design of compressor. Estimation of friction force and associated losses in overall compressor performance can be critical in design of a compressor and its performance prediction. Though the desired motion of the piston is purely axial, imbalance of forces cannot be avoided completely and causes eccentricity in piston motion. This eccentricity results in both unwanted friction and hydrodynamic forces. While published literature is abundant with study of friction and lubrication in piston in reciprocating compressor, this paper brings novelty in solution approach and presenting a way for using the results for optimization design parameters through Design of experiments (DoE).

2. MODELING

The motion of piston in both axial and radial direction is shown in Figure 1 below. The eccentricities at top and bottom of the piston are caused by its motion in radial direction. The following section discusses the formulation of dynamic forces acting on the piston and connecting rod which dictate the piston motion.

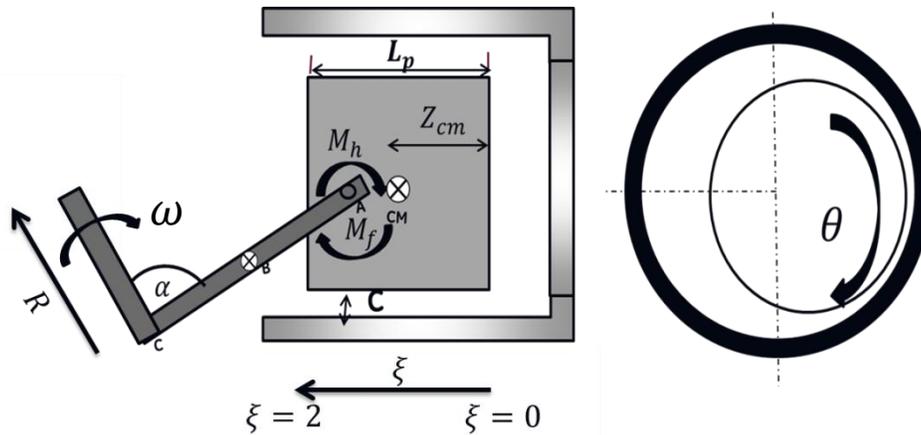


Figure 1 Piston Cylinder Schematic (adapted from [Prata2000])

2.1 Equations of motion

Dynamic force balance equations on piston with connecting rod are developed in published literatures [Kellaci 2010, Prata 2000, Rigola 2009] and can be represented as following

$$\begin{bmatrix} m_p \left(1 - \frac{Z_{cm}}{L_p}\right) & \frac{Z_{cm}}{L_p} \\ \frac{I_p}{L_p} & -\frac{I_p}{L_p} \end{bmatrix} \begin{bmatrix} \ddot{\xi}_t \\ \ddot{\xi}_b \end{bmatrix} = \begin{bmatrix} F_h + R_x \\ M_h + M_f \end{bmatrix} \quad (1)$$

Here the radial component of reaction force, R_x at connecting rod on piston end, is given by the equations below [Prata2001]

$$R_x = \frac{1}{L_{AC}} \left[(R_z L_{Ac} + m_b \ddot{z}_{cr} L_{BC}) \tan \alpha - m_b \ddot{x}_{cr} L_{BC} - \frac{I_b \ddot{\alpha}}{\cos \alpha} \right] \quad (2)$$

Hydrodynamic (F_h, M_h) and viscous (F_f, M_f) forces and moments used in the above equations are [Prata1998]

$$F_h = - \int_0^{L_p} \int_0^{2\pi} P(\theta, z) R \cos \theta d\theta dz \quad (3)$$

$$M_h = - \int_0^L \int_0^{2\pi} [pP(\theta, z) R \cos \theta] (z_p - z) d\theta dz \quad (4)$$

$$F_f = - \int_0^L \int_0^{2\pi} \left(\frac{h}{2} \frac{\partial P}{\partial z} + \mu \frac{V_p}{h} \right) R d\theta dz \quad (5)$$

$$M_f = - \int_0^L \int_0^{2\pi} \left(\frac{h}{2} \frac{\partial P}{\partial z} + \mu \frac{V_p}{h} \right) R \cos \theta R d\theta dz \quad (6)$$

For calculating oil pressure, P , as a function of polar coordinates θ , ξ and film thickness, h as function of eccentricities, following representation of Reynolds equations is used [Prata2000]

$$\frac{\partial}{\partial \theta} \left(h^3 \frac{\partial P}{\partial \theta} \right) + \frac{\partial}{\partial \xi} \left(h^3 \frac{\partial P}{\partial \xi} \right) = -12\mu R^2 \left(\frac{V_p}{2R} \frac{\partial h}{\partial \xi} - \frac{\partial h}{\partial t} \right) \quad (7)$$

$$h = c \left\{ 1 - \left[\varepsilon_t - (\varepsilon_t - \varepsilon_b) \xi \frac{R}{L_p} \right] \cos \theta \right\} \quad (8)$$

Using above equations parameters such as minimum oil film thickness, pressure distribution, viscous and shear forces, are calculated. The solution methodology is presented in the next section.

2.2 Solution Algorithm

This paper presents a combination of using MATLAB's ® inbuilt ODE solvers® for solving ordinary differential equations and self-defined finite difference scheme for solving PDEs from Reynolds equations. Initial values of all the eccentricities and their first time derivatives is assumed zero. Taking the partial derivatives of the film thickness equation with respect to θ and ξ

$$\frac{\partial h}{\partial \xi} = c \cos \theta \left\{ (\varepsilon_t - \varepsilon_b) \xi \frac{R}{L_p} \cos \theta \right\} \quad (9)$$

The derivatives of pressure terms can be expressed using central difference scheme as follows

$$\frac{\partial^2 P}{\partial \theta^2} = \frac{P_{i,j+1} - 2P_{i,j} + P_{i,j-1}}{\Delta \phi^2} \quad (10)$$

$$\frac{\partial^2 P}{\partial \xi^2} = \frac{P_{i+1,j} - 2P_{i,j} + P_{i-1,j}}{\Delta \xi^2} \quad (11)$$

Where i is the discretization coordinate along θ and j is the discretization coordinate along ξ .

Expanding Equations (7) and (8) and substituting Equations (10) and (11) in it, we obtain expression of instantaneous pressure as presented in the equation below. Mathematical expression for each term is present in the appendix at the end of the paper.

$$P_{i,j} = \frac{C_1 P_{i+j} + C_2 P_{i-1,j} + C_3 P_{i,j+1} + C_4 P_{i,j-1} + G}{D} \quad (12)$$

3. RESULTS AND DISCUSSION

To establish a baseline case, parameters from a typical household refrigerator compressor [Prata2000] are used as follows

Table 1 Parameters used for simulation

Parameter	Symbol	Value
Crank Radius	R	10.5 mm
Piston-cylinder Radial Clearance	c	3μ
Crank angular velocity	ω	3000 rpm
Length of Piston skirt	L_p	$2R$
Length of connecting rod	L_{AC}	$3.473R$
Mass of Piston	m_p	34.1 g
Lubricating oil viscosity	μ	3 cP

Typical results obtained with such analysis are shown in the Figure 2 below. The pressure distribution at each crank angle is plotted as a function of the piston radial, (θ) and axial (ξ) coordinates. It can be seen that the oil pressure is higher towards the top of piston (*i. e.* $\xi = 0$) and equal to the cylinder pressure as it is the boundary condition for the

piston-cylinder clearance at the top while it is lower at the piston, ($\xi = 2$) skirt bottom equaling the boundary condition there.

Minimum oil film thickness is plotted for last 2 cycles to establish solution convergence. The minimum film thickness ranges from 0.9 to 1.3 μm and it is lowest at TDC and BDC where the piston velocity goes to zero leading to low hydrodynamic force as can be seen in Figure 2. Since the piston and cylinder surface finish is mirror finish (i.e. $R_a < 0.1$ microns), boundary lubrications doesn't occur during the high speed regime of piston.

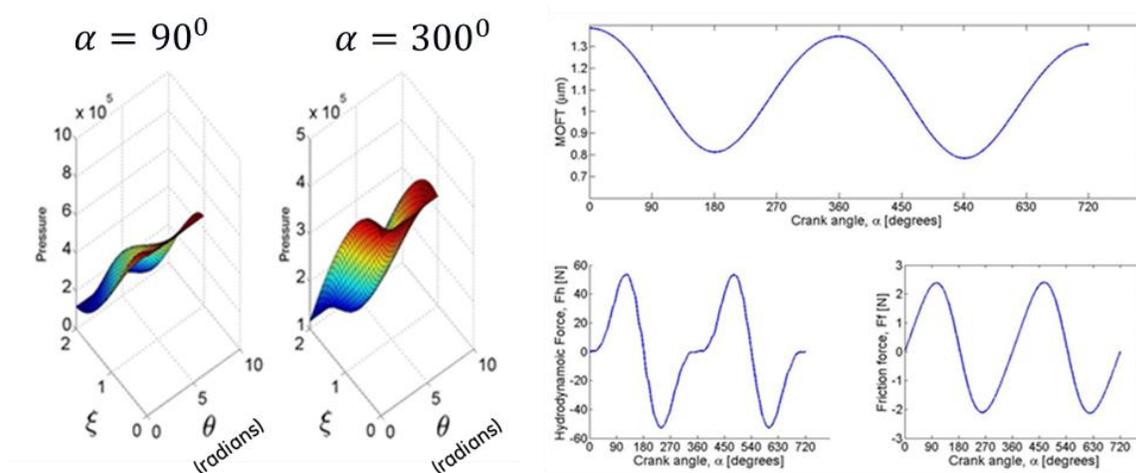


Figure 2 Pressure distribution, film thickness and force distribution

3.1 Blow-by

Another important parameter to influence the friction losses and parameters affecting it is the blow-by. The radial clearance between the piston and the cylinder offers a passage for the refrigerant to escape due to the pressure difference across this channel. A schematic of the scenario is seen in Figure 3. When the piston moves towards the TDC the refrigerant is compressed and the pressure increases introducing a pressure difference between the compression chamber and the openings to the shell environment. This constitutes the blow-by loss within the compressor since the work done in compressing the refrigerant escaping through the radial clearance will not be useful for cooling. The amount of refrigerant leakage depends on the nominal radial clearance. The engagement length for blow-by changes as the piston reciprocates about its mean position. There could be different engagement paths depending on the oil flooding of openings. An accurate representation of the blow-by model will require incorporating these aspects.

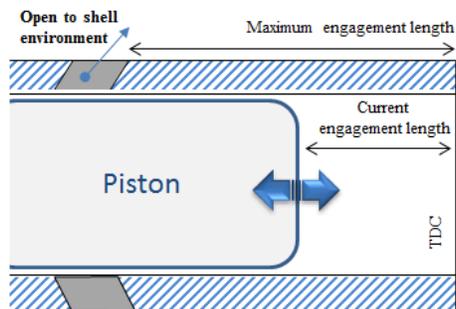


Figure 3: Blow-by through the piston cylinder radial clearance

The flow through the annular region between piston-cylinder can be approximated by the Couette flow model and the effect of the piston profile can be considered by updating the radial clearance locally. The mass exchange due to blow-by, m_{bb} , in time, Δt is given by

$$m_{bb} = \Delta t \rho_{cyl} (\pi D_p c) \left(\frac{v_p}{2} - \frac{c^2 \Delta p}{12\mu L_{bb}} \right) \quad (13)$$

The comparison between flow rate estimates using the Couette flow model and experiments is shown in Figure 4a, where measurements are done for the piston being held at different distances from TDC. At increased distances from the TDC, the Couette model over-predicts the flow rate possibly because at such distances the engagement length is not large enough for the flow to be developed. Nevertheless, the model was found to be accurate in evaluating the effect of different nominal radial clearances and piston profiles on blow-by. The effect of radial clearance can be seen in Figure 4b from the model predictions for the piston at 8 mm distance from the cylinder head. The blow-by rate increases as the radial clearance increases and is a factor to consider when designing the piston-cylinder clearance for friction and blowby.

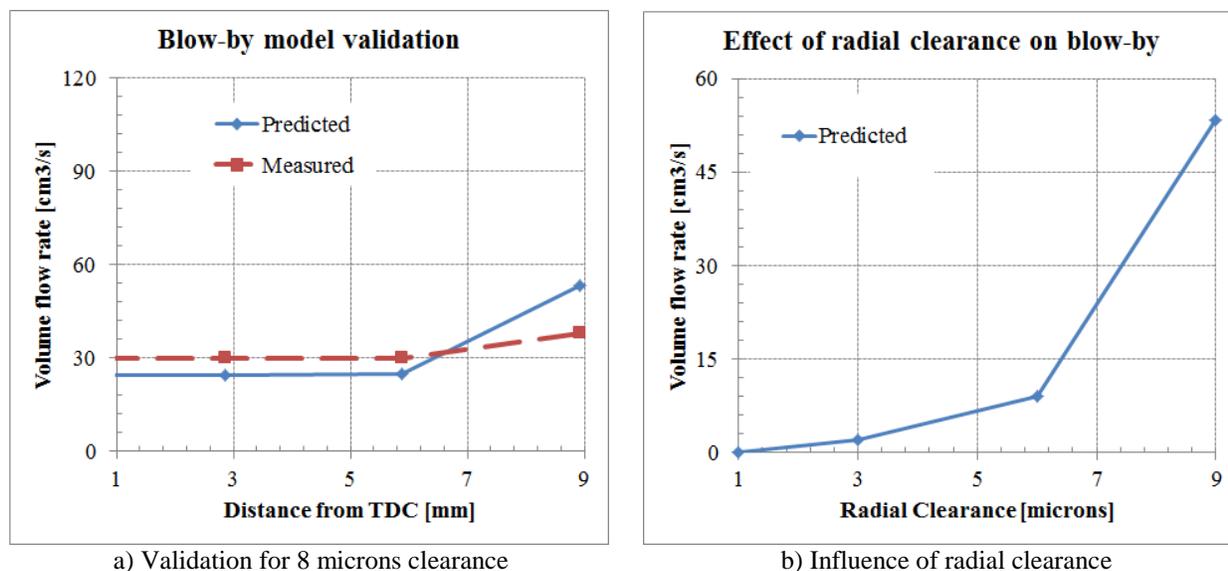


Figure 4: Blow-by model

4. Design of Experiments (DoE)

A parametric study has been carried out to investigate and understand the effect of pump angular speed, ω , clearance, c , and oil viscosity, μ . The range of parameters for this study is listed in Table 2 below.

Table 2 DoE Parameters

Parameter	Symbol	Units	Min.	Max.
Crank angular velocity	ω	<i>rpm</i>	500	3500
Piston Cylinder Clearance	c	<i>microns</i>	1	5
Lube Oil Viscosity	μ	<i>cP</i>	2.5	7.5

Two parameters, MOFT (minimum oil film thickness) and total friction power consumption is used as design/output parameters. Results of this study are shown in the figure 5 below. We see that with increasing pump speed, ω and increasing viscosity, μ the minimum oil film thickness, h_{min} and friction power, P_{in} increases. While small clearance is good for keeping oil leakage to minimum, it also results in increased friction and hence friction power, P_{in} . Therefore the designer would need to consider and evaluate the trade-off between these two factors.

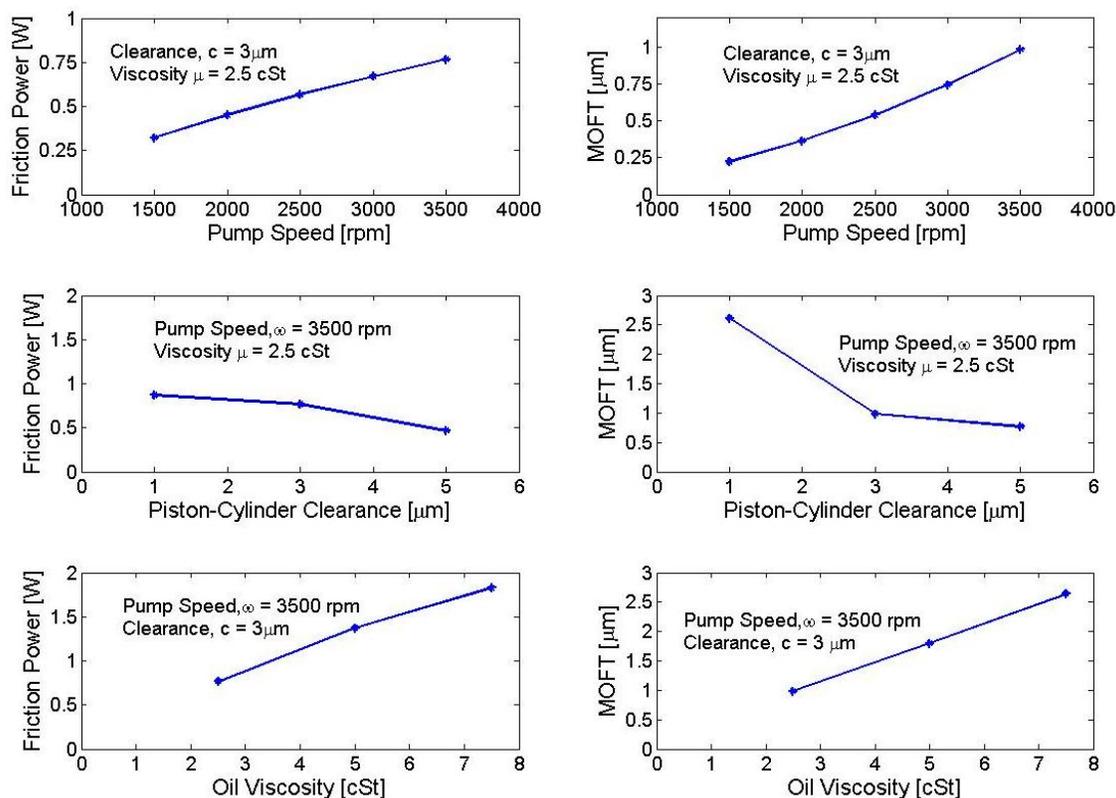


Figure 5 Parameters sensitivity analysis

As can be seen from the above plots in Figure 5, the following observations can be made

- Running at low rpm results in low friction power loss. Hence running at variable speed with majority of time at low speed is efficient when compared to running at constant speed of 3500 rpm.
- During start-ups, the oil viscosity is high due to low oil temperatures and hence needs higher starting torque (or current)
- Using lower viscosity fluids (oil or gas) reduces the friction power linearly
- With increase in piston-cylinder radial clearance the friction reduces (Figure 5) and blow by increases (refer Figure 4). Hence a trade-off of these two losses needs to be considered in the design.

4. CONCLUSIONS

Formulation for solving pressure distribution, film thickness, forces, and power requirement for reciprocating compressor is adapted from literature and solved in novel manner with discretization of PDEs in finite difference equations and then solving ODEs with inbuilt MATABL solver. The results from in-house tests are used to validate the model and perform DoE on certain key parameters to understand the impact of these parameters on MOFT and power. It is seen that radial clearance needs to be optimized as compromise between power consumption, oil leakage and blow-by loss.

NOMENCLATURE

D_p	Piston Diameter
F_f	Viscous friction force
F_h	Hydrodynamic force
F_{rx}	Radial force

I_B	Connecting rod moment of inertia
I_p	Piston moment of inertia about wrist pin
L_{BC}	Connecting rod length from its CM to crankpin
L_{AB}	Connecting rod length from its CM to wrist pin
L_{bb}	Piston engagement length
L_p	Piston Length
M_h	Hydrodynamic moment
M_f	Viscous moment
P_{in}	Friction Power
R	Piston radius
R_x, R_z	Horizontal & Radial component of connecting rod forces
c	Piston cylinder radial clearance
e_t, e_b	Top & bottom eccentricity
h	Oil film thickness
m_b	Connecting rod mass
m_{bb}	Mass outflow due to blowby
m_p	Piston Mass
p_{cyl}	Cylinder pressure
t	time coordinate
v_p	Piston velocity
$\ddot{x}_{cr}, \ddot{z}_{cr}$	Horizontal & axial component of connecting rod acceleration
z	Axial coordinate along piston length
Z_{cm}	Axial distance from piston top to CM
Z_p	Axial distance from piston top to wrist pin

Greek

α	Crank angle
ξ	Dimensionless axial coordinate $\left(\frac{L_p}{R}\right)$
Δp	Pressure drop across piston engagement length
Δt	Time step
$\varepsilon_t, \varepsilon_b$	Dimensionless top & bottom eccentricity respectively $\left(\frac{e_t}{c}, \frac{e_b}{c}\right)$
θ	Angular coordinate along piston circumference
ρ_{cyl}	Refrigerant density
μ	Viscosity

Abbreviations

CM	Center of Mass
MOFT	Minimum Oil Film Thickness
TDC	Top Dead Center
BDC	Bottom Dead Center

Superscripts

.	First derivative w.r.t. time
..	Second derivative w.r.t. time

Subscripts

<i>min</i>	<i>minimum</i>
<i>max</i>	<i>maximum</i>
<i>cyl</i>	<i>Cylinder</i>

APPENDIX

Variable	Mathematical Equivalent	Variable	Mathematical Equivalent
C_1	$\left(\frac{1}{\Delta\xi^2} + \frac{3H_1}{2h\Delta\xi}\right)$	H_2	$\frac{\partial h}{\partial \phi}$
C_2	$\left(\frac{1}{\Delta\xi^2} - \frac{3H_1}{2h\Delta\xi}\right)$	H_3	$\frac{\partial h}{\partial t}$
C_3	$\left(\frac{1}{\Delta\theta^2} + \frac{3H_2}{2h\Delta\theta}\right)$	D	$\left(\frac{2}{\Delta\xi^2} + \frac{2}{\Delta\theta^2}\right)$
C_4	$\left(\frac{1}{\Delta\theta^2} + \frac{3H_2}{2h\Delta\theta}\right)$	G	$\frac{12\mu R^2}{h^3} \left(\frac{V_p}{2R} H_1 - H_3\right)$
H_1	$\frac{\partial h}{\partial x}$		

REFERENCES

Kellaci, A., Mazouzi, R., Kheldij, B., Bounif, A., 2010, The Effect of Lubricant rheology on piston skirt/cylinder Contact for an Internal Combustion Engine. *Mechanicka*. Vol. 81, no. 1, page 30-36

Prata, A. T, Fernandes, J. R. S., Fagotti, F., 2000, Dynamic Analysis of Piston Secondary Motion for Small Reciprocating Compressors, *Journal of Tribology*, vol 122, p. 752-760

Prata, A. T, Fernandes, J. R. S., Fagotti, F., 1998, Numerical Simulation of the Piston Secondary Motion in a Reciprocating Compressor, *International Compressor Engineering Conference*. Paper 1234.

Prata, A. T, Fernandes, J. R. S., Fagotti, F., 2001, Piston Lubrication in Reciprocating Compressors, *Engenharia Termica*. n. 1, p. 56-63

Rigola, J., Perez-Segarra, C, D., Olivia A., 2009, Numerical Simulation of Piston Leakage Over Hermetic Reciprocating Compressors Behavior, 2009, *Institution of Mechanical Engineers - International Conference on Compressors and their Systems*, P 375- 384

ACKNOWLEDGEMENT

The authors would like to thank General Electric for supporting this research and providing required permission to publish this work.