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Development And Validation Of Integrated Design Framework For Compressor System Model

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ABSTRACT

The performance of a refrigerator is largely guided by the efficiency of its drive unit, namely, the compressor and understanding the compressor behavior requires a detailed study of its dynamics, flow-thermals, electrical and controls aspects. Having a simulation model which captures these physics reasonably well is a critical part of the design and performance prediction of a compressor. The current paper describes a systematic approach of making a system level simulation framework by first developing individual models and then integrating them into a single framework to capture the multi-physics interaction of the different sub-components. This framework, developed in MATLAB/SIMULINK contains five modules, namely, Dynamics, Thermal, Motor, Controls and Post-processor. Piston Dynamics is modeled as a spring-mass system with adjustable static equilibrium and head-crash prevention algorithm. The thermodynamics model essentially captures the valve physics. The valves in a reciprocating compressor contribute to pressure losses (pressure profile deviation from ideal suction and discharge processes, valve dynamics, leakages, pressure pulsations) and thermal losses (refrigerant back-flow caused by incorrect valve timing). The starting point of simulating these details is considering the gas dynamics and coupling it to the valve motion. The prediction of this model is validated against test data of a baseline compressor. As part of the integrated design framework, a permanent magnet linear motor is simulated as a resistance-inductance network with a series, velocity dependent voltage. To control the desired operating conditions (capacity, stroke, clearance etc.) upon the integrated system model, a set of controllers were designed to control the motor.

1. INTRODUCTION

Successful compressor development requires careful conceptual study, and further design and prototyping from that study. To have an efficient design, the conceptual study must include analytical model(s) of not just several individual components but also of overall system level design. This approach is indicated in Figure 1

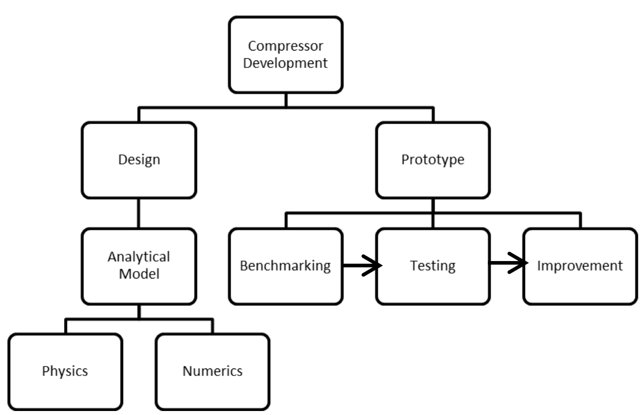


Figure 1 Compressor Development

This paper describes several critical component modules in the electrodynamically oscillating reciprocating compressor system and system level model built in MATLAB/SIMULINK which integrates all these modules. The flow chart of the system level model is as shown in Figure 2.

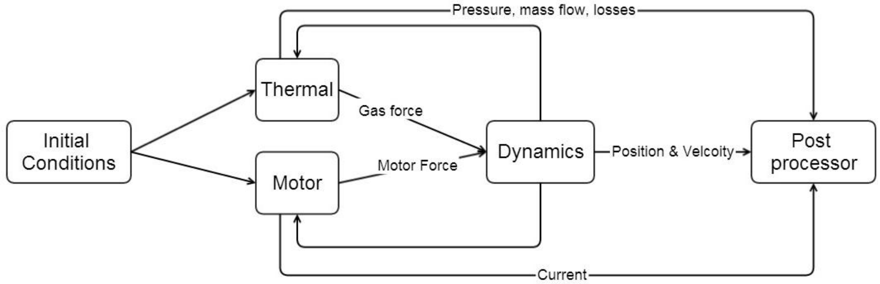


Figure 2: Simulation Block Diagram

2. Simulation Model

2.1 Dynamics

The dynamics module is the simplest of the module where basic spring mass damper equation is solved for

$$M\ddot{x} + C\dot{x} + Kx = F_{gas} + F_{motor} \tag{1}$$

The input for this module is gas force and motor force which can be easily converted to acceleration. This acceleration is used to estimate position and velocity using kinematics equations

$$v_{t+\Delta t} = v_t + a_t \Delta t \tag{2}$$

$$x_{t+\Delta t} = x_t + v_t \Delta t + \frac{1}{2} a_t \Delta t^2 \tag{3}$$

2.2 Thermal

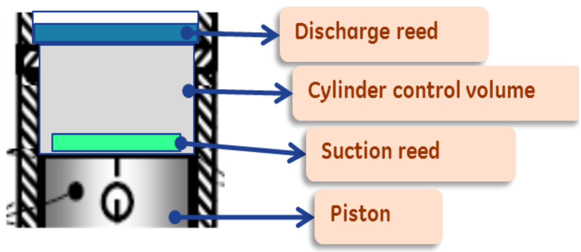
Thermal model is the most critical module in this integrated simulation model. The output of the dynamic module is feedback into the thermal module for pressure, gas force, mass flow, and valve losses calculation.

Compression process: A schematic for a typical compressor is show in Figure 3. The efficiency of a compressor can be estimated by considering the gas dynamics (pressure and temperature evolution) within the cylinder control volume. Compression and expansion are a closed control volume process with no mass efflux/ influx from the control volume. The starting point is considering the gas dynamics to be a polytropic process as shown in Equation (4).

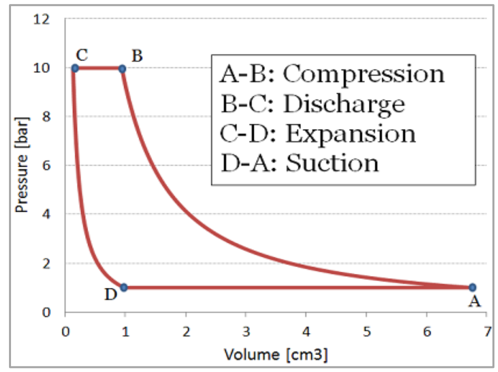
$$P_t = P_{t-1} \left\{ \frac{m_{t-1}}{\rho_0 V} \right\}^n \tag{4}$$

Suction and discharge mass flow is calculated using the Equation (5). Details of this equations with derivation is available in [Zhou, 2001]

$$m_t = m_{t-1} \pm \int C_d A_0 \sqrt{2\rho \Delta P}\, dt \tag{5}$$



3a. Compressor control volume



3b. Ideal cycle P-V map

Valves: The valves in a reciprocating refrigerator compressor contribute to losses such as pressure difference (from ideal discharge and suction process), leakages, and pressure pulsations. It is thus imperative to model the valve physics towards prediction of compressor performance parameters**.** Discharge and suction involve mass exchange and the control volume state depends on the dynamics of the discharge valve that sits on the cylinder bore on one side and the suction valve that is mounted on the piston. Instead of assuming perfect valve, the valve flutter and corresponding losses are also simulated using a 1-D model which models valve as spring-damper system as presented in the Equation (6) below [Huang 2002].

$$\mathrm{m_d \ddot{x}_d + c_d \dot{x}_d + k_d x_d = (P_{cyl} - P_d) A_{dv} + F_{prestress}} \tag{6}$$

Valves losses: There is a possibility of mass exchange due to leakages around the valve circumference when they are closed, due to surface irregularities, assembly issues etc. These can be modeled as orifice flows with a discharge coefficient. These coefficients however would need to be empirically determined on a case to case basis. This form of mass exchange loss (suction and discharge loss) can be computed as given in Equation (7) and Equation (8)

$$m_{svLeak} = Cd_{svLeak} A_{svLeak} \left(\sqrt{2\rho_{cyl} (p_{\mathrm{cyl}} - p_s)} \right) \tag{7}$$

$$m_{dvLeak} = Cd_{dvLeak} A_{dvLeak} \left(\sqrt{2\rho_{cyl} (p_{cyl} - p_d)} \right) \tag{8}$$

Oil Stiction: The oil formulation can be addressed by a time varying equivalent damping of the valve [Khalifa, 1998]. The damping constant is a function of the time varying oil contact factor X. Figure 4 shows the presence of an oil film between the valve reed (depicted as a disc) and the valve seat. This damping constant is predicted by Equation (9) Due to the higher temperatures, stiction on discharge side is neglected.

$$C_{stiction}(t) = \frac{3\pi\mu r_{sv}^4}{2h^3} . \left(1 - X_0(t)^4 + \frac{1 - 2X_0(t)^2 + X_0(t)^4}{log(X_0(t))} \right) \tag{9}$$

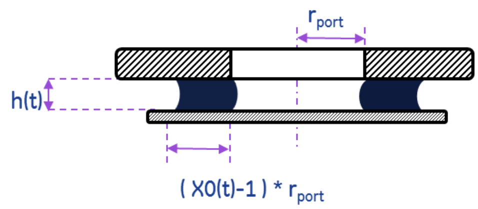


Figure 3: Oil film between the valve seat and reed causes stiction

Blow-by: If there is a mass exchange due to blow-by, this is considered as a case of Couette flow and given by: Equation (10). Detailed explanation and derivation of this is presented in [Mantri2014]

$$m_{bb} = \Delta t\, \rho_{cyl}\, (\pi\, D_{piston} c) \left(\frac{v}{2} - \frac{c^2}{12\mu} \frac{\Delta P_{bb}}{L_{bb}} \right) \tag{10}$$

2.3 Motors

Physical Structure of the Motor
The motor used in the compressor is a linear permanent magnet motor. The basic structure of the motor may be represented by the two-dimensional cross section view below:

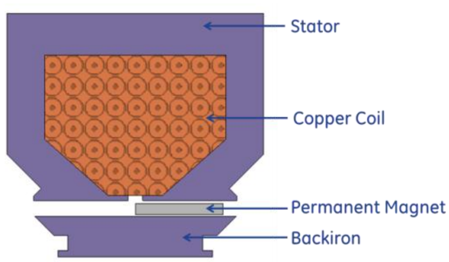


Figure 4 Layout of Motor and stator

The back-iron and the stator are held stationary to the compressor assembly. The stator holds the coil with several turns. The permanent magnet is held with an air gap between the stator and the back-iron. The magnet is assembled to rigidly move with the piston. When the coil is excited with sinusoidal excitation the generated alternating magnetic field due to the coil interacts with the magnetic field of the permanent magnet to generate force on the permanent magnet, due to which the piston reciprocates.

Equivalent Circuit of the Motor
Due to the permanent magnet oscillating in the air gap between the coil, voltage or back EMF is generated in the coil. The magnitude of the back EMF is proportional to the air gap flux density and the number of turns and the velocity of the magnet. The frequency of the back EMF is equal to the frequency of the piston oscillation. This behavior is similar to the behavior of the DC motor, the equivalent circuit of which is well established. The equivalent circuit has three components, the resistor, the inductance and the back EMF. This can be represented by the circuit diagram below:

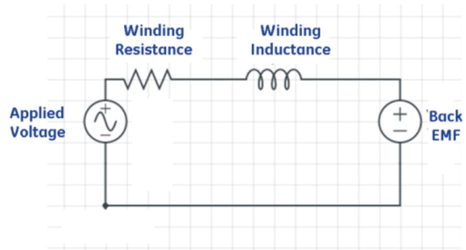


Figure 5 Layout of RLC circuit of motor

The equation for the motor can be written as:

$$\alpha \, {}^{dx}/_{dt} = \left[V - iR - L \, {}^{di}/_{dt}\right] \tag{11}$$

Where α is the motor force constant expressed in Force per ampere, *R* is the resistance of the motor winding *L* is the motor inductance, *V* is applied terminal voltage, *i* is the resulting current and the displacement of the piston is *x*.

The force output of the motor is proportional to the current and is given by $F=\alpha*i$. As can be seen the applied voltage needs to overcome the back EMF due to the velocity by forcing a desired amount of current through the motor. Depending on the current drawn, the stroke of the compressor can be controlled. The frequency of the applied voltage will also dictate the frequency of the piston motion.

This equivalent circuit implementation is used in the control model of the stroke and operating frequency of compressor as explained in the section below.

2.4 Controls

Following three different kinds of control are designed and integrated in this simulation frame work

Voltage Control (Mass Flow Control): Mass Flow in a compressor needs to be regulated based on the load. Mass Flow Control is a simple PI Control which generates the output voltage based on the Mass Flow error signal.

Current Control (Clearance Control): The EER and capacity are proportional to the stroke of the linear compressor. To obtain maximum EER it is required to drive the stroke of the compressor to close to full stroke without going over the full stroke. The control force of a linear compressor is adjusted based on reference clearance stroke by the Clearance Control. The current (proportional to motor force) applied to the motor is varied based on the reference clearance calculated from the minimum value of the stroke from gas model. A digital controller is designed using pole zero cancellation method which outputs the required current based on the clearance error.

Frequency Control (Resonant Frequency Tracking): When the operating electrical frequency is equal to the mechanical resonant frequency then maximum efficiency can be obtained. This happens when motor current is in phase with piston velocity.
The phase between velocity and current signal is obtained by detecting their zero crossings and then it is driven to zero by using a proportional controller. The output of the proportional controller is a variation of frequency. New inverter frequency is obtained by adding the previous frequency to variation of frequency.

2.5 Model Summary

Table 1 Summary of physics and mathematics for several components of the model

Process	Physics	Mathematics
Piston motion	Kinematics, Newton's Laws	Equation 1-3
Gas compression	Polytropic	Equation 4
Oil Stiction	Valve damping	Equation 9
Blow-by leakage	Couette flow	Equation 10
Valve Motion	1 D spring mass system	Equation 7
Controller	PI Control, Phase difference at resonance	Laplace transform
Motor	RLC Circuit	Equation 11

2.6 Post-Processor

In the post-processor model, following equations are used for prediction of key performance parameters, such as cooling capacity, EER and power input. All the losses, such as friction loss, motor loss, valve losses and blow-by are also computed. A pie chart presenting the loss pareto is presented for better understanding of loss distribution.

$$Cooling\ capacity = \left[\int \left(\mathrm{m_t} - \left(m_{svLeak} + m_{dvLeak} + m_{bb} \right) \right)\ dt \right] f \Delta h \qquad (12)$$

$$P_{in} = \int F_{motor} v\ dt\ + iron\ loss + cu.loss + inv.loss \qquad (13)$$

$$EER = \frac{cooling\ capacity}{P_{in}} \qquad (14)$$

3. SIMULATION RESULTS AND VALIDATION

The model is simulated with parameters for a certain typical household refrigerator compressor.

Component level validation: Since the thermal module is the most critical part of the model, detailed care has been taken to validate the model.
As shown in the figure 6, a reasonable level of accuracy is obtained including the pressure pulsation for the gas module. The shift in time required is attributed to delay caused due to simulating the model only as a component.

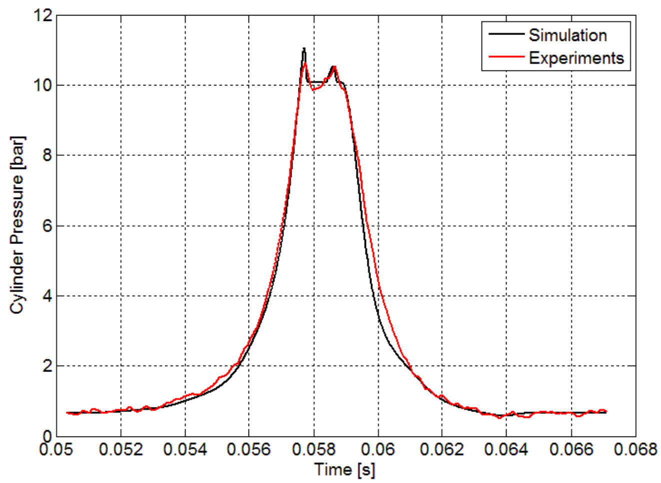


Figure 6 Gas module Validation

System level validation: The system level validation of the entire integrated model matches within 3 % of the test data. Discrepancy in the current profile is attributed to idealistic sinusoidal current wave assumption for simulation and pulse modulator in the motor. The pressure profile captures the first peak from test data but subsequent peaks are not captured because of 1-D model used. However, the effect on compressor overall performance and EER is less than 1 %.

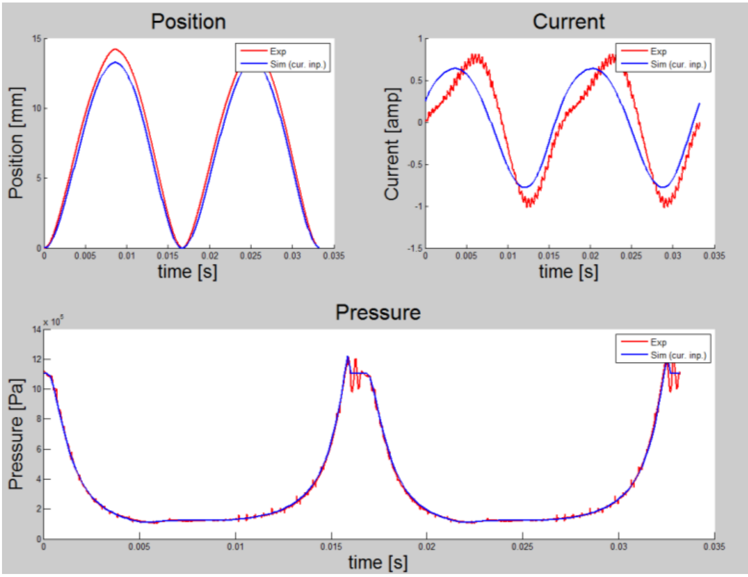


Figure 7 System Level Validation

Finally, a loss pareto chart can be used for studying losses due to different components of the compressor. The stiction and blow-by losses for this analysis were negligible and removed from this analysis.

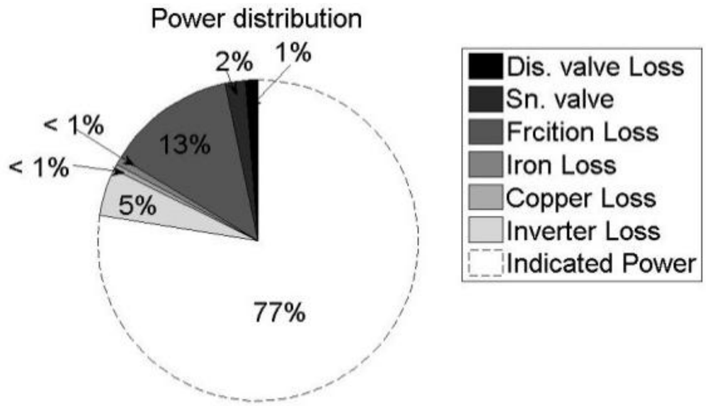


Figure 8 Loss Pareto

4. CONCLUSIONS

A novel and innovative technique for modeling the system level dynamics of compressor is developed and simulated in MATLAB/SIMULINK environment. Each module is carefully modelled with reasonable assumptions to reach an optimum solution within acceptable limits on accuracy and speed. This model is validated both at component level and system level and used successfully to optimize parameters and prediction of compressor performance.

NOMENCLATURE

A	$Orifice\ area$
C	$Damping\ constant$
C_d	$Discharge\ coefficent$
D_{piston}	$Piston\ Diameter$
F_{gas}	$Gas\ force$
F_{motor}	$Motor\ force$
M	$Piston\ Mass$
K	$Spring\ Constant$
L_{bb}	$Blowby\ engagement\ length$
L_e	$Inductance$
R_e	$Resistance$
P_{in}	$Total\ Power\ Input$
PR	$Pressure\ ratio$
S	$Piston\ stroke$
V	$Voltage$
V_{swept}	$Swept\ Volume$
X_{cl}	$Piston$ - Cylinder $head\ clearance$
X_0	$Radial\ extent\ of\ the\ oil\ wetting\ the\ port\ normalized\ by\ the\ port\ radius$
a	$Piston\ acceleration$
c	$Local\ radial\ clearance$
f	$frequency$
h	$Oil\ film\ height$
m	$Mass\ flow$
n	$Polytropic\ constant$
p_{cyl}	$Cylinder\ pressure$
r_{sv}	$Suction\ port\ radius$
v	$Velocity$
x	$Displacement$

Abbreviations

EER	$Energy\ Efficiency\ Ratio$

Greek

α	$Motor\ constant$
Δh	$Cooling\ Enthalpy$
ΔP_{bb}	$Pressure\ drop\ due\ to\ blow\ by$
μ	$Oil\ Viscocity$
ρ_{cyl}	$Refrigerant\ Density$

Subscripts

d	$Discharge$
$dvLeak$	$Discharge\ valve\ leak$
s	$Suction$
$svLeak$	$Suction\ valve\ leak$
t	$current\ time\ step$
$t+1$	$next\ time\ step$
bb	$Blowby$

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