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Optimization of Compressors used in Air Conditioning Systems

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ABSTRACT

Strong market pressures cause most compressors in air conditioning (AC) units to be designed by minimizing the initial cost. Since most manufacturers don't own the air conditioning system, the operating cost of the system over the life of the component, is often neglected. The objective of this project is to create an analytical model that simulates an actual AC unit and then use it to arrive at the optimized design variables based on the lowest total life-cycle cost. Performance of a compressor is affected by several factors including compressor speed, suction and discharge pressures, and component geometry and valve efficiencies. The effect of these factors on the compressor performance in terms of volumetric and isentropic efficiencies is modeled for a reciprocating compressor. Experimental data is collected from an AC unit to verify and validate the analytical model at different operating conditions. Energy balances between the refrigerant and air side of the heat exchangers showed an average error less than 10% which confirms the accuracy of the instrumentation. A parameter optimization is conducted on the empirical coefficients used in the analytical model by minimizing the RMS error between the model and experimental data. This validated model can be used to optimize the compressor to obtain the lowest total life-cycle cost for different working environments.

1. INTRODUCTION

Much data is available on the working of compressors used in air conditioning (AC) units and the design variables that affect their isentropic and volumetric efficiencies. Predictions from theoretical models and supporting experimental evidences have shown thermodynamic, mechanical, and fluid characteristics that play a part in determining the compressor efficiencies. However, not much data is available on the cost impact of these design variables and design of optimized AC units from a total life cycle cost point of view. Most HVAC system designs are biased towards a lower initial cost, the design variables of the system are sub optimized in a way then leads to a higher operating cost. Accretion of the operating costs over the life span of the system often yields higher operating cost compared to the optimum system. This project makes an effort towards solving this problem by connecting the design variables to both initial and operating costs of the AC unit.

The total life cycle cost of a system is the cost incurred by the end user over the entire life of the system. The two main components of this are: 1) Initial Cost: cost to purchase the system, often defined by the total/maximum capacity and quality of the system. 2) Operating Cost: the cost of operation over the entire life of the system, defined by the daily usage and the efficiencies of the system and its components. The goal of this project is to optimize the performance of the compressors used in AC units based on the primary design variable that impact the compressor efficiencies. The objective function of the optimization is to minimize the total life cycle cost of the AC unit.

2. LITERATURE SURVEY

2.1 Air Conditioning Unit

An air conditioning unit provides cooling and humidity control for a building. A refrigerant is used as a working fluid to extract and transfer the heat from the controlled space (room air) to the environment (atmosphere).

The air conditioning cycle consists of four essential steps:

- The refrigerant starts its cycle in a gaseous state. The compressor compresses the refrigerant up to high pressure and temperature.
- The gaseous refrigerant then enters a heat exchanger (condenser), where it is cooled and condenses into liquid.
- An expansion valve drops the refrigerant pressure. After the pressure reaches the saturation pressure, the temperature drops along with the pressure.
- The refrigerant mixture passes through another heat exchanger (evaporator), where it gains heat from its surrounding medium and evaporates to a gaseous state again. The refrigerant repeats the cycle again by passing through the compressor and so on.

In the process, heat is absorbed from indoors and transferred outdoors, resulting in cooling of the inside air.

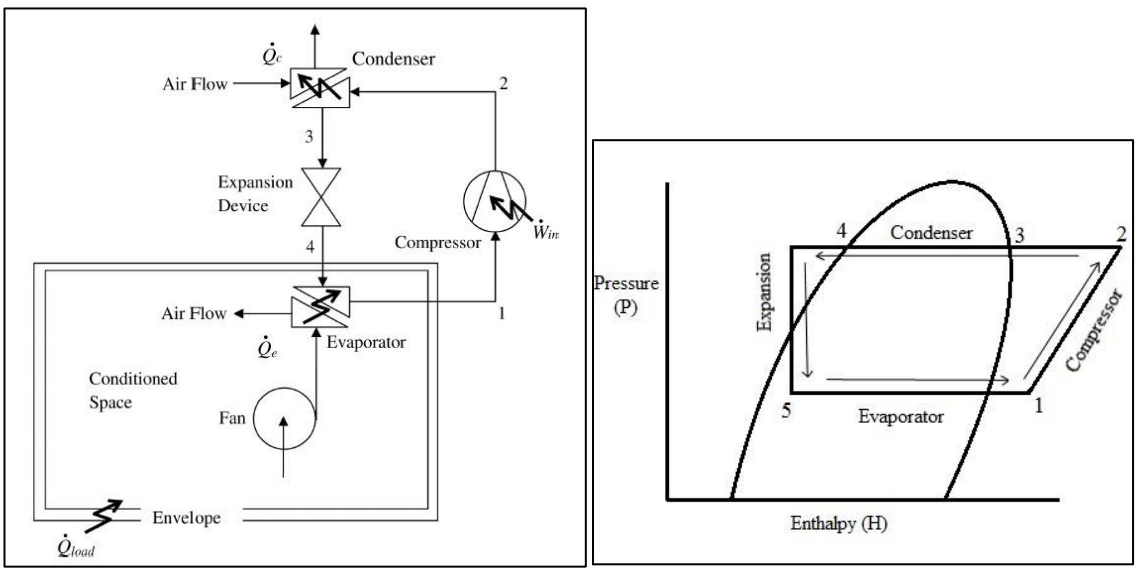

Figure 1: a) Schematic diagram of air conditioning cycle with all its components. Adapted from *Optimization of Cooling Systems* (p.4) by D.C. Zietlow, 2016 New York, NY: *Momentum Press Thermal Science and Energy Engineering Collection.* Copyright (2016) by Momentum Press. b) Property plot of pressure-enthalpy variations through different state points in the air conditioning cycle.

2.2 Compressors

A reciprocating compressor is a positive displacement machine, in which the density and pressure of the gas entering the cylinder is increased through compression. Compressors have a major effect on both the initial and operating costs. Performance of a compressor is defined by the isentropic efficiency ($\eta_{isen} = \dot{W}_s/\dot{W}_a$) and the volumetric efficiency ($\eta_{vol} = \dot{V}_a/\dot{V}_s$) along with its power input.

Where, $\dot{V}$a = Actual volumetric flow rate of the refrigerant
$\dot{V}$s = Ideal volumetric flow rate of the refrigerant
$\dot{W}$s = Ideal power input to the compressor
$\dot{W}$a = Actual power input to the compressor

In an actual compressor, factors like valve opening/closing time, surface finish, fluid flow disturbances, clearance volume, leakage through valves, and piston blow-by play a role in determining the η_{isen} and η_{vol}.

Perez-Segara, Rigola, Soria and Oliva (2005) in their paper, discuss the efficiencies of reciprocating compressors from a thermodynamic point of view. Equations relating basic compressor characteristics like compression ratio, clearance ratio, and polytropic exponent to the isentropic and volumetric efficiencies are discussed. Losses in the form of re-expansion of compressed gases in the clearance volume, pressure drops in the suction and discharge lines, inefficiencies in opening/closing of valves are accounted for in the efficiency equations. The below equations for isentropic and volumetric efficiencies in this project are adopted from this paper.

$$\eta_{vol} = \frac{Vol\ flow\ rate}{Vol\ flow\ rate\ @\ zero\ clearance\ ratio}$$

$$\eta_{vol} = \frac{[V_{cl}\left[1 - c\left(p_r^{\frac{1}{C_{gam}}\gamma} - 1\right)\right] - leak\_dot]f_n}{V_{cl}\ f_n}$$

$$\eta_{isen} = CC_{\eta}.\gamma.\frac{[p_r^{\frac{\gamma-1}{\gamma}} - 1]}{\frac{P_2V'_2 - P_1V_1}{1-\gamma}}$$

Swept volume, V_{cl} = $\frac{bore\ dia^2}{4}$ x stroke x no.of cylinders
f_n = Compressor speed
p_r = Pressure ratio
c = Clearance ratio
γ = Polytropic exponent
refrigerant leak rate, leak_dot = m_{leak} / rho

m_{leak} = $discharge_c$ x $Area_{leak}$ x $\sqrt{(2.\ \rho_{leak}.\ pressure_{drop})}$ x g_c
$Area_{leak}$ = 3.14 x $port_{dia}$ x gap
g_c = gravitational constant from Newtons II law

Gap = $\frac{(flatness+parallelism+surface\_roughness)}{3}$

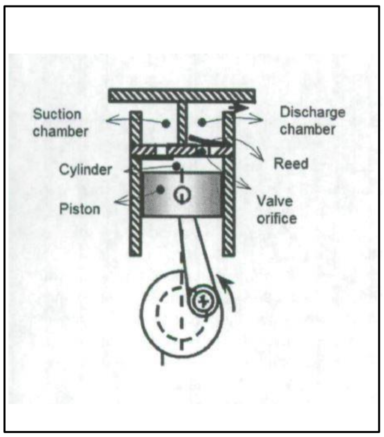

Figure 2: Shows the schematic and working of Reed valve in a reciprocating compressor. Adapted from "Influence of piston on effective areas of reed-type valves of small reciprocating compressors", by E.L.Pereira, 2011, *HVAC&R Research*, 17(2): p.218-230, Copyright (2011) American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc.

Hiller and Glicksman (1976) in their paper on detailed modeling and computer simulation of reciprocating refrigeration compressors, discuss the factors affecting the efficiencies of a reciprocating compressors: Real gas properties, valve dynamics, pressure variation at suction and discharge manifolds, frictional losses due to pistons, piston rings, etc. For the sake of simplicity, these factors have been accounted for, cumulatively, as the polytropic exponent in this analytical model.

Pereira and Deschamps (2011) in their paper discuss about the reed valves used in small reciprocating compressors. Reed valves are actuated by the pressure difference between the cylinder and the suction/discharge chamber which is established by the piston motion. The reed valves open and close the ports on the piston wall. The sealing between the valve and valve seating area depends on the machining parameters of the valve and the piston wall. Parallelism, flatness and the surface roughness are the important machining parameters which are related to the refrigerant leakage. The refrigerant leakage affects the volumetric and hence the isentropic efficiency of the compressor. This has been accounted for, in this project to calculate the actual volumetric flow of the refrigerant after subtracting the leakage flow rate from the swept volume of the cylinder.

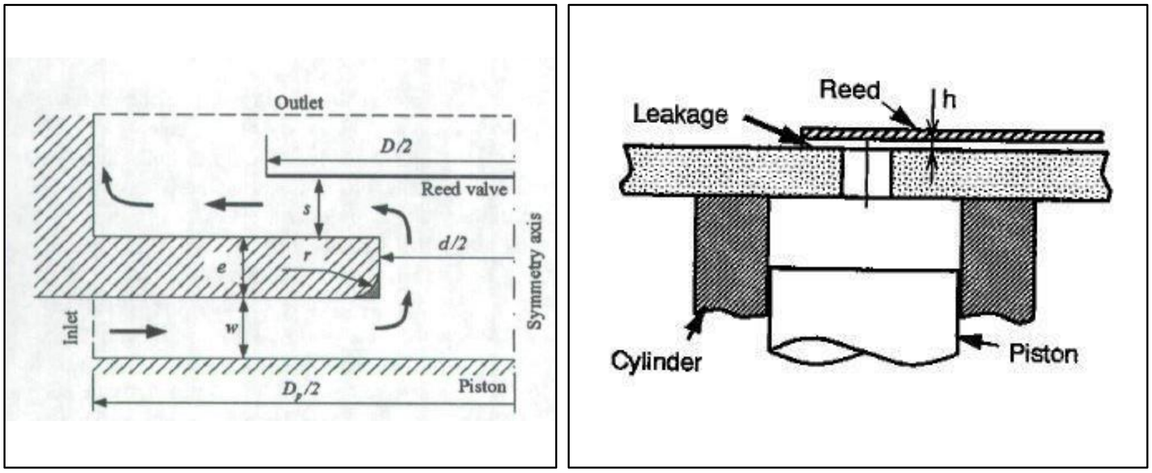

Figure 3: Schematic of Reed valve showing the geometry and primary design variables critical to the efficiency of compressors. Adapted from "Influence of piston on effective areas of reed-type valves of small reciprocating compressors", by E.L.Pereira, 2011, *HVAC&R Research*, 17(2): p.218-230, Copyright (2011) American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc.

Paper by Fujiwara and Kazama (1998) discuss the relation between the valve geometry and the leakage flow rate. Equations relate the radial clearance, port diameter, clearance volume, pressure drop to the leakage flow rate of refrigerant. The equation defines an empirical coefficient called the discharge coefficient which is based on the flow properties of the refrigerant and is used as a primary design variable in this project.

Mass flow rate of leakage fluid, $q_m = C.A.\sqrt{2.\,\rho.\,(dP)}\ .\ g\_c$

Where, C = discharge coefficient based on the fluid and its thermodynamic state ρ = density of fluid dP = pressure difference between the cylinder and the discharge chamber

A = Leakage area based on the gap between valve and valve seating

$A = \pi\, d\, h$

g_c = gravitational constant from Newtons II law

Where d is the discharge port diameter h = clearance height between reed valve and valve seat (defined by the machining parameters: parallelism, flatness and surface roughness).

2.3 Cost Data

Cost coefficients are the empirical coefficients required to calculate the initial cost of the compressor based on the power rating, efficiencies and the pressure ratio. Obtaining cost data was one of the challenging tasks of this project, as there is not much cost data available based on the design variables of compressors. Cost data was obtained from supplier websites based on the isentropic, volumetric efficiencies, and pressure ratio of the compressor. Five compressors with different efficiencies and power requirement are considered for the regression analysis. To obtain accurate data for cost coefficients, the rms error of the regression analysis was kept below 10%.

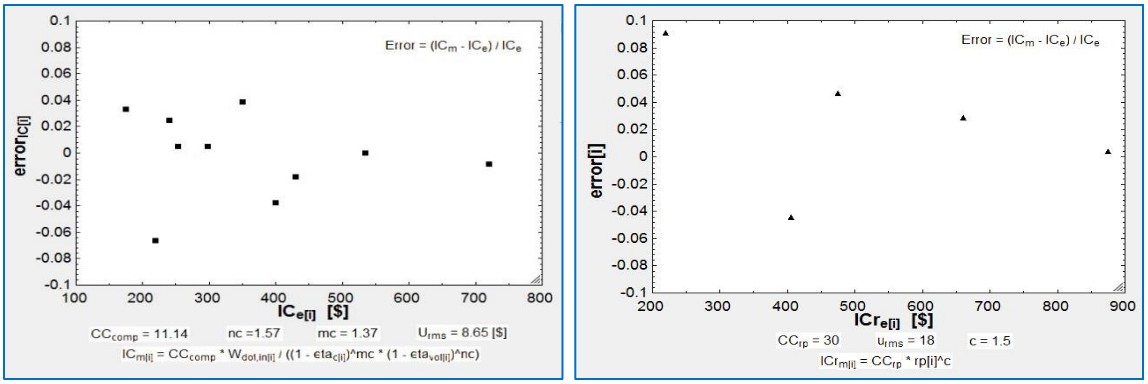

Figure 4: Shows results of regression analysis carried out to determine the cost coefficients. The cost coefficients are optimized to obtain an rms error of less than 10%.

3. ANALYTICAL MODEL

The analytical model is developed by using the minimum total cost of the system as the objective function for the optimization. Equations are developed to link the design variables of the compressor to the initial and operating costs of the system. Initial cost is defined as a function of the work input, isentropic efficiency and the volumetric efficiency of the compressor. Two intermediate design variables are considered in this project: 1) Isentropic efficiency 2) Volumetric efficiency. These intermediate design variables are further linked to the primary design variables of a compressor like the pressure ratio, valve geometry (surface roughness, parallelism and flatness of the valve and valve seating area), clearance ratio, and the frictional losses between cylinder wall and the piston. Using equations for conservation of energy around the components of the AC unit, equations are derived linking the power of the compressor to the heat transfer rates in the condenser and the evaporator. The condenser and the evaporator heat exchangers are modeled using effectiveness-NTU relations..
The operating cost of the AC unit is taken as a function of the compressor power input. The tradeoff here is that the operating cost of the AC unit decreases if the isentropic and volumetric efficiencies of the compressor increase. However, a higher efficiency compressor also incurs a higher initial cost. Thus, the compressor efficiencies have to be optimized for a given application and working environment. The analytical model shows the relation between the primary design variables, the efficiencies and the costs of the system. The apparatus used for collecting experimental data had the capability to vary the compressor speed by varying the electric frequency of the drive. The compressor speed is varied up to 4000 rpm with a resolution of 1 rpm. The mass flow rate and the temperature of air through the condenser are controlled using dampers on the inlet of the condenser air duct and the return air duct. The damper position varies on a scale of 0 to 90deg. Velocity of air for different damper openings is recorded to arrive at the volumetric flow rate through condenser. The mass flow rate of air through the evaporator is varied by controlling the fan speed. The fan speed is varied over 6 different speeds. Velocity of air in both instances is measured using an anemometer. Using the velocity of air, the volumetric flow rate of air is determined.

For this project, the compressor speed, flow rate over condenser and evaporator are divided into 3 levels: low, medium, and high. For the compressor speed, the respective levels are 1000, 1500, and 2000 rpm. For the condenser, the respective damper settings are 60˚, 45˚, and 90˚. For the evaporator, the respective fan speeds corresponded to 4, 5, and 6 on the scale. Permutating these three parameters yielded 27 different combinations of settings. The AC unit is run for each of these settings till the AC unit achieved a steady state. Steady state is verified by taking multiple temperature and pressure readings at regular intervals until a steady state condition is achieved. It is observed that after about 15 minutes, the AC unit achieved a steady state. Once the steady state is reached, all the instrument readings are taken every five minutes over 25 minutes. The data is averaged over the five readings to get the final set of data for that setting. This cycle is repeated for all the 27 settings.

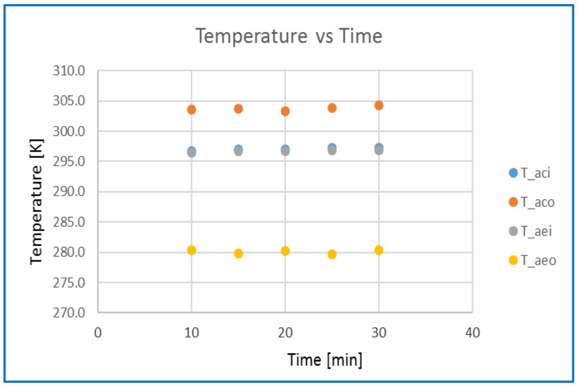

Figure 5: Shows the variation of temperature readings in steady state. The sensors have an accuracy of +/- 1°C. T_aci is the temperature of the air at the condenser inlet, T_aco is the temperature of air at the condenser outlet, T_aei is the temperature of air at the evaporator inlet and T_aeo is the temperature of air at the evaporator outlet.

4. RESULTS AND DISCUSSION

4.1 Energy Balance

The accuracy of the instrumentation for the AC unit is verified by conducting an energy balance on the AC unit. An EES code is written to compute the energy balance between the refrigerant-side and the air-side of the condenser and evaporator. The heat lost by the refrigerant in the condenser should be equal to the heat gained by the air flowing over the condenser tubes. Similarly, the heat gained by the refrigerant in the evaporator should be equal to the heat lost by the air flowing over the evaporator tubes.

$$\% \text{ error condenser} = \frac{Q_{dot_{cr}} - Q_{dot_{ac}}}{Q_{dot_{ac}}} \text{x } 100$$

$$\% \text{ error evaporator} = \frac{Q_{dot_{ea}} - Q_{dot_{er}}}{Q_{dot_{ea}}} \text{x } 100$$

Q_dot_cr = Rate of heat lost by refrigerant in the condenser [kW]
Q_dot_ac = Rate of heat gained by air in the condenser [kW]
Q_dot_er = Rate of heat gained by refrigerant in the evaporator [kW]
Q_dot_ea = Rate of heat lost by air in the evaporator [kW]

To accurately calculate the energy balance, the enthalpy of the water condensed at the evaporator must also be included in the heat transfer rate. Most air conditioning processes include a dehumidifying process, thus, the volumetric flow rate of condensate from the evaporator is measured along with its temperature to calculate its heat transfer rate.
The energy lost in the form of heat from the compressor to the environment must also be determined. This heat energy is generated from the friction between the piston and the cylinder walls as well as the friction between the refrigerant molecules being compressed in the cylinder. During the compression cycle, some of the heat energy is transferred to the cylinder walls which in turn is transferred to the outside environment. This is measured using the heat transfer equation Q = U*A*Delta_T. Where U is the overall heat transfer coefficient, calculated for the compressor material and taking into account the circulation of air over the compressor due to the belt drive, A is the surface area of compressor exposed to the outside environment and DELTA_T is the temperature difference between the compressor and the outside environment. The temperature of the compressor surface is measured using a thermal image camera. The surface area of the compressor exposed to the environment is calculated using its geometry and specification sheet.
The results of the energy balance are compared using percentage error between the air and refrigerant. This is found to be less than 10% error, which is acceptable.

The graph below shows the variation of error in kW over the 27 different operating conditions.

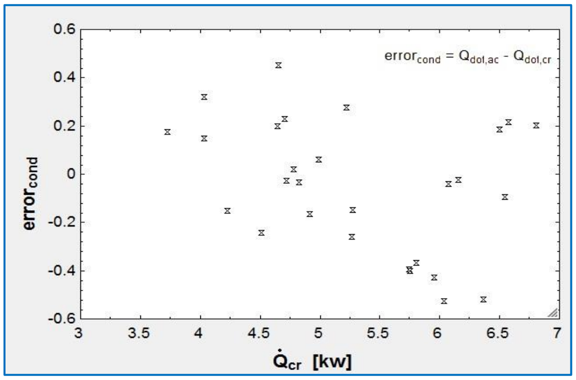

Figure 6: Shows a plot of heat transfer energy balance @ air side and refrigerant side of a condenser over the 27 experimental readings. The error percentage is < 10%

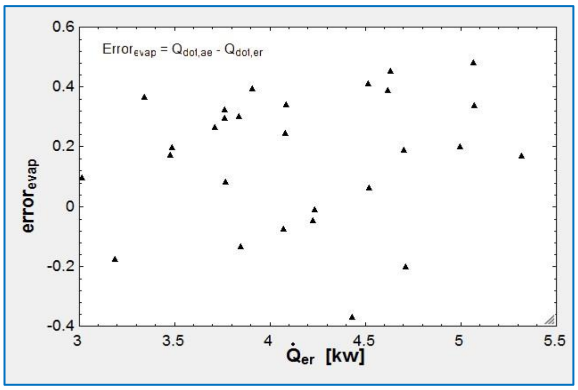

Figure 7: Shows a plot of heat transfer energy balance in kW @ air side and refrigerant side of an evaporator over the 27 experimental readings. The error percentage is < 10%

4.2 Optimization

The experimental data is then used to validate the empirical coefficients used in the analytical model. Validation is based on the root mean square (RMS) error computed between the compressor power calculated from the model and the actual compressor power input. The compressor power in the model is computed using the equations for isentropic efficiency and volumetric efficiency and the primary design variables. Cost equations are not included in this EES code and only the compressor power is compared for all the 27 different settings. The min/max feature in the EES software is used to minimize the RMS error as a function of the following primary design variables: coefficient of isentropic efficiency (CC_eta), polytropic exponent of the compression cycle (gamma), clearance ratio of the cylinder (clear_ratio), coefficient of volumetric efficiency (C_gam), the pressure drop caused by the refrigerant flowing through inlets and outlets of the compressor (pressure_drop) and the discharge coefficient for the leakage flow rate through the valves (discharge_c). The optimum values for these primary design variables are obtained using data from all the 27 settings. The optimum primary design variables are then used in the analytical model to compute the total cost of the AC unit for one set of operating conditions. Using the above data, the total cost of an AC unit is calculated.

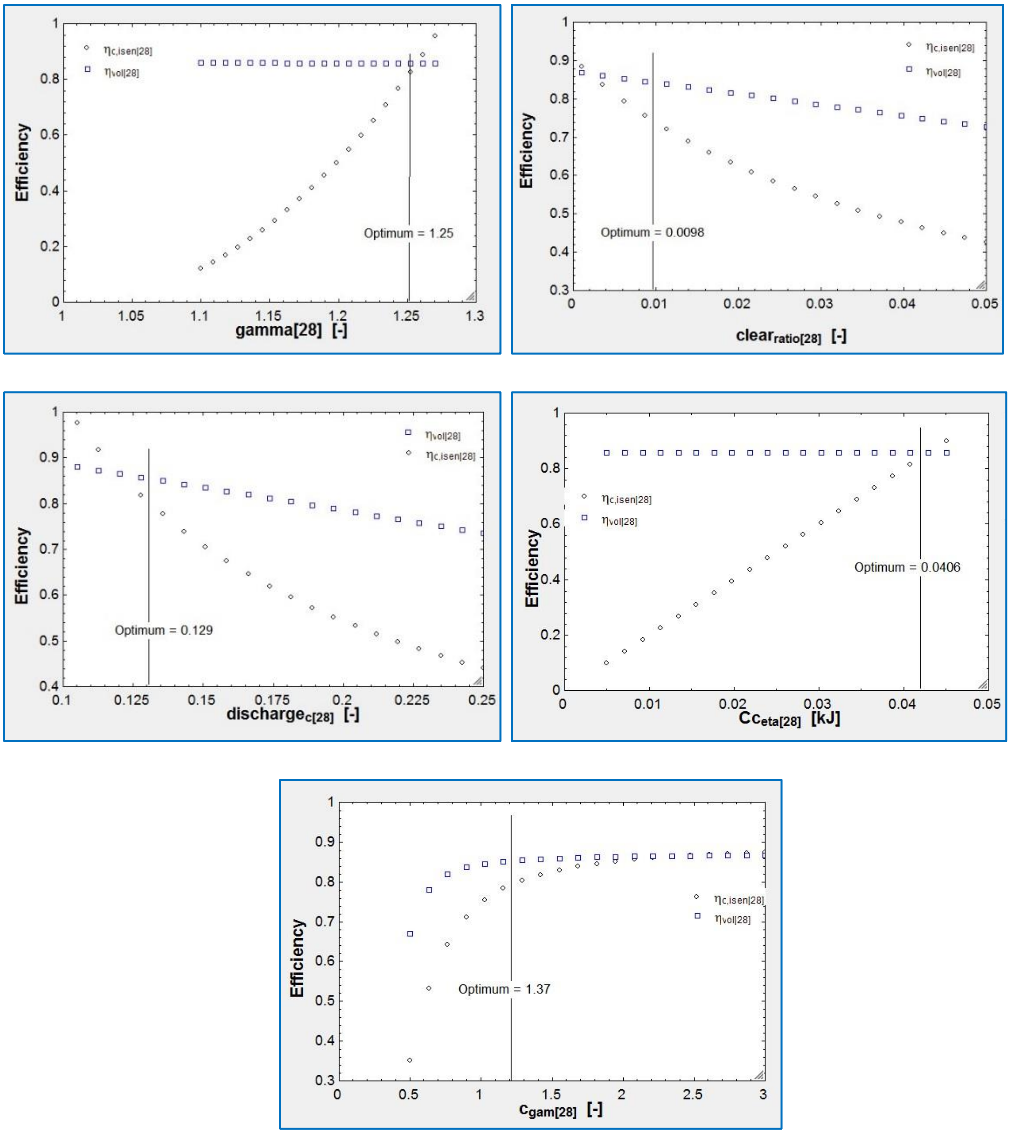

Figure 9: Above plots show the variation of isentropic $\eta_{c,isen(28)}$ and volumetric efficiencies $\eta_{vol(28)}$ with the primary design variables of the system. These parameters were optimized to achieve a low RMS error.

Optimum values of primary design variables for the AC unit are found to be

Table 1: Optimization results of primary design variables

Primary design variables	Description	Value
Gamma	Polytropic Exponent	1.25
clear_ratio	Clearance ratio	0.0098
discharge_c	Discharge coefficient of refrigerant	0.1289
Cc_eta	Isentropic efficiency coefficient	0.0406
c_gam	Volumetric efficiency coefficient	1.37

The polytropic exponent obtained from this project is found to be in agreement with Lenz (2002) which calculates polytropic exponents based on isentropic conditions and constant specific heat work by using numerical solution of pressure and volume changes, suction and discharge pressures and the efficiency of the compressor. The polytropic exponents vary between 0.99 and 1.29.

5. CONCLUSION

The project successfully establishes a relation between the design variables of an air conditioning compressor and the total life cycle cost of an AC unit. The model details a logical progression of equations linking the primary design variables to the objective function (minimizing total cost). The validated model can be applied to an automobile AC unit or any other AC application to understand how the compressor characteristics vary with operating conditions. This model can be used to determine the best compressor for a given application which would yield the minimum total life cycle cost.

This model also shows a relation between the compressor efficiencies and the valve gap. The valve gap influences leakage flow rate of refrigerant. The greater the gap, the higher is the leakage, resulting in lower volumetric efficiency and hence lower isentropic efficiency as shown in Figure 10. The valve gap is a function of the valve machining parameters like the parallelism, flatness and surface roughness of the valve and the valve seating area. The dependency of efficiencies on the poly-tropic exponent is shown earlier in Figure 9. Thus correlating, it can be shown that the poly-tropic exponent and the valve gap are inversely proportional (in the γ range $1 \sim 1.4$).

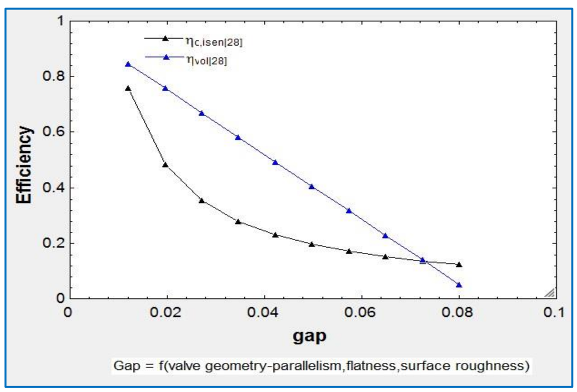

Figure 10: Plot shows variation of isentropic and volumetric efficiencies with gap between valve and valve seat.

Thus, the model can be used to determine the best compressor for a given application based on its operating conditions. It also provides an insight into the compressor design variables and how they affect the total life cycle costs.

REFERENCES

Assaf, Day, Winchester, Wolffe (2010). Final Report- Design of a Rapid-Cycling Compressor controller.

Ferreira, R., Deschamps, C., & Prata, A. (1989). Pressure distribution along valve reeds of hermetic compressors. *Experimental Thermal and Fluid Science*,*2*(2), 201-207.

Fujiwara, M. and Kazama, T. (1998) Performance Improvement of a Reciprocating Air Microcompressor. *International Compressor Engineering Conference.* Paper 1328.

Hiller, C. C. and Glicksman, L. R. (1976). Detailed Modeling and Computer Simulation of Reciprocating Refrigeration Compressors. *International Compressor Engineering Conference.* Paper 162.

Lenz, J. R. (2002). Polytropic Exponents for Common Refrigerants. *International Compressor Engineering Conference.* Paper 1528.

Niazi, A., Dai, J. S., Balabani, S., &Seneviratne, L. (2006). Product Cost Estimation: Technique Classification and Methodology Review. *Journal of Manufacturing Science and Engineering*,*128*(2), 563.

Pereira, E., & Deschamps, C. (2011). Influence of piston on effective areas of reed-type valves of small reciprocating compressors. *HVAC&R Research*,*17*(2), 218-230.

Pérez-Segarra, C., Rigola, J., Sòria, M., & Oliva, A. (2005). Detailed thermodynamic characterization of hermetic reciprocating compressors. *International Journal of Refrigeration,28*(4), 579-593. doi:10.1016/j.ijrefrig.2004.09.014
Prasad, B. G., Shiva Shiva (1998), Heat Transfer in Reciprocating Compressors - A Review. *International Compressor Engineering Conference.* Paper 1349.
Schreiner, Joao E.; Deschamps, Cesar J., and Barbosa, Jader R.(2010). Theoretical Analysis of the Volumetric Efficiency Reduction in Reciprocating Compressors due to In-Cylinder Thermodynamics. International Compressor Engineering Conference. Paper 2004.
Wall, G. (1988). On the Optimization of Refrigeration Machinery. *International Refrigeration and Air Conditioning Conference.* Paper 51.
Zietlow, D.C. (2016). Optimization of Cooling Systems. *Momentum Press Thermal Science and Energy Engineering Collection.*
Handbook of ASHRAE - Fundamentals. *American Society of Heating Refrigerating and Air Conditioning Engineers.*