Application of Optimization of Compressors used in Air conditioning units

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Application of Optimization Study of Compressors Used in Air Conditioning Systems

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ABSTRACT

Total life-cycle cost of a system refers to the costs incurred by the owner 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, and 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. Most compressors are designed with a focus on minimizing the initial cost. Minimal data are available on the selection of a compressor based on its design variables and the corresponding cost impact. A comprehensive link between the design requirements and their initial and operating cost implications is lacking in the industry. This project addresses this issue by analyzing the impact of design variables on the total life cycle cost of an automobile air conditioning (AC) unit when located in different climatic zones. Using an earlier, experimentally validated, analytical model, the isentropic efficiency and volumetric efficiency of a reciprocating compressor are varied to suit the environmental conditions of four climatically diverse cities (with respect to the temperature, average length of cooling season and humidity) i.e. Phoenix, AZ, Peoria, IL, Minneapolis, MN and Miami, FL. The model connects the efficiencies to primary design variables of a compressor, namely the polytrophic exponent, clearance ratio, geometry, etc. The lowest possible life-cycle cost and the corresponding compressor specifications are determined and reported. Using this in the design of AC units (residential/commercial, automobile) will result in the best compressor design for a given application. For colder climates (Peoria and Minneapolis) the optimum isentropic efficiency and volumetric efficiency of a compressor averaged over the cooling season at 51% and 69% respectively, whereas for hotter climates (Phoenix and Miami) the efficiencies averaged at 60% and 73% respectively.

1. INTRODUCTION

Total life cycle costs provide a realistic optimum which can be used to justify a particular compressor design. Other objective functions such as efficiency do not yield realistic optimums. Total life-cycle cost of a system is defined as the total cost incurred by the owner throughout the life of the system including planning, design, acquisition, support, operating costs and any other costs directly attributable to owning or using the asset. It involves two main components: i) Initial Cost (IC): cost to purchase the system, often defined by the total/maximum capacity and the quality of the system. ii) Operating Cost (OC): 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.

1.1 Assumptions

The following assumptions were made for the present study -

- The analytical model is largely restricted to a reciprocating type compressor
- It considers only a simple model of the condenser and evaporator
- Valve leakage is restricted to reed valves used in reciprocating compressors
- A steady state flow of refrigerant and air (changes in kinetic and potential energies through each component considered negligible)

2. ANALYTICAL MODEL

One of the first steps in the project was to create an analytical model of an AC unit that runs on a reciprocating type compressor. Taking minimum total cost of the system as the objective function, equations are derived 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. Considering the scope of this project, costs were assumed to depend only on the compressor characteristics. Two intermediate design variables were considered in this project: 1) isentropic efficiency 2) volumetric efficiency. These intermediate design variables are further linked to the primary design parameters 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 conservation of energy around the components of the AC unit, equations are derived linking the work of the compressor to the heat transfer rate at the condenser and the evaporator. The condenser and the evaporator heat exchangers are modeled using effectiveness NTU relations. Maximum heat transfer rate equations link the air-side and the refrigerant-side of the heat exchangers.

The operating cost of the AC unit was taken as a function of the compressor work input. The tradeoff here was that the operating cost of the AC unit decreases if the isentropic and volumetric efficiencies of the compressor are high. However, a higher efficiency compressor also incurs a larger initial cost. Thus, the compressor efficiencies must be optimized for a given application and working environment.

To validate the analytical model an experimental AC apparatus was used to produce data and compare to the results from the analytical model. The experimental apparatus has the capability to vary the compressor speed by varying the electric frequency of the drive. The compressor speed was varied up to 2000 rpm with a resolution of 1 rpm. The mass flow rate and temperature of the air over the condenser was controlled using dampers on the inlet of the condenser air duct and the return air duct. The damper position varied on a scale of 0 to 90 degrees. Velocity of air over different damper openings were recorded to arrive at the volumetric flow rate through condenser. The mass flow rate of air over the evaporator was controlled by controlling the fan speed. The fan speed was varied over 6 different speeds. By measuring the velocity of the air, the volumetric flow rate of air was determined.

For this project, the compressor speed, flow rate over condenser and evaporator are stratified 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 openings refer to 60deg, 45deg, and 90deg opening of the damper. For the evaporator, the respective fan speeds refer to 4, 5, and 6 on the scale. Permutating these three parameters yield 27 different combinations of settings. The AC unit was run for each of these settings till the AC unit achieved a steady state. Steady state was verified by taking temperature and pressure readings every five minutes. It was observed that after 15 minutes, the AC unit achieved a steady state. Once the steady state was reached, all the instrument readings were taken every five minutes for five repetitions. The data was averaged over the five readings to get the final set of data for that setting. This cycle was repeated for all the 27 sets of operating conditions.

2.1 Optimization

The experimental data was then used to validate the empirical parameters, such as the multipliers in the Nusselt correlations, used in the analytical model. Validation was 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 work in the model was computed using the equations for isentropic efficiency and volumetric efficiency and the primary design variables. The compressor power was compared for all the 27 different settings. The min/max feature in Engineering Equation Solver (EES) was used to minimize the RMS error as a function of the 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 due to refrigerant mass left in the clearance volume (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 determined from the validated model. The optimum primary design variables are then used in the analytical system model to compute the total cost of the AC unit.

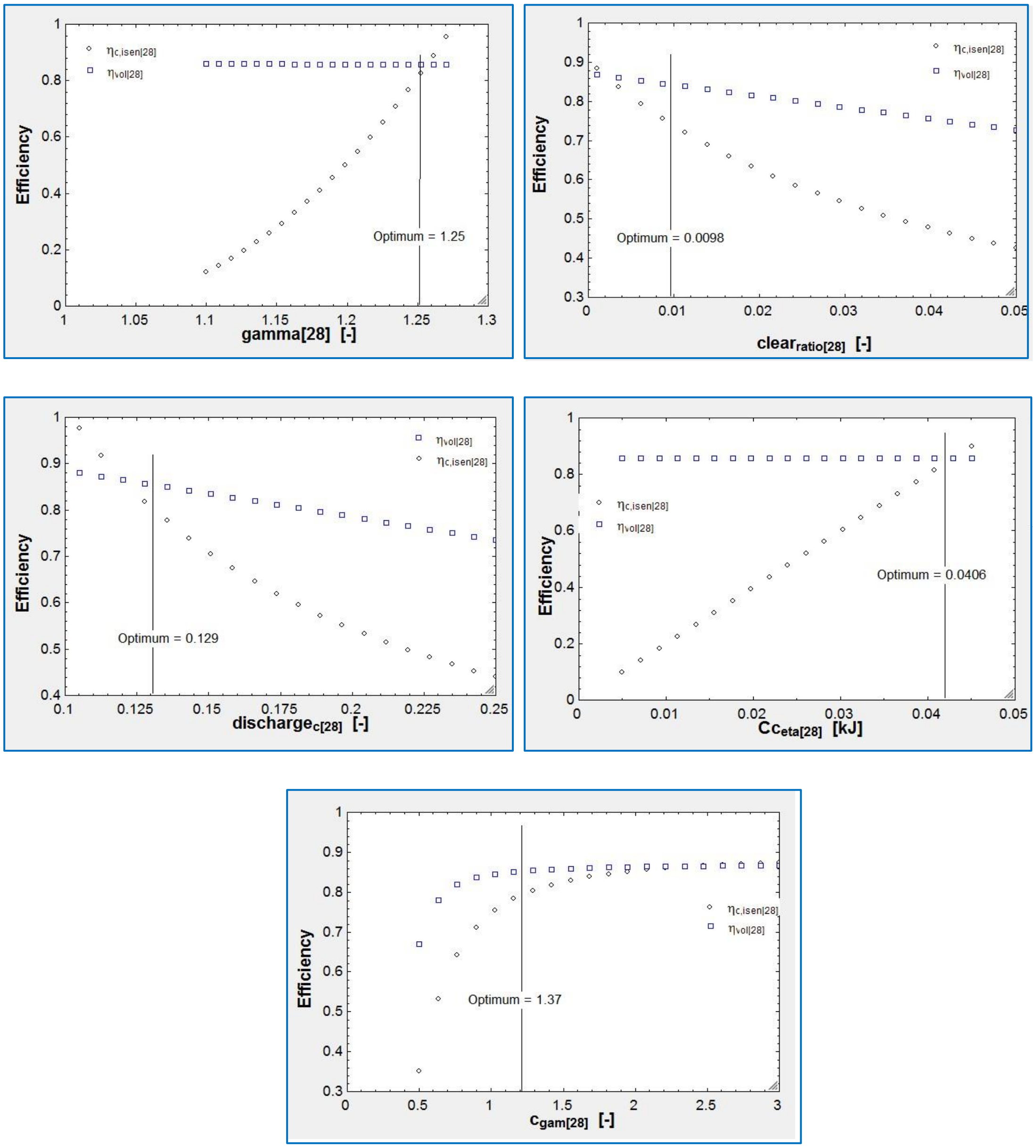

Figure 1: The variation of isentropic and volumetric efficiencies with the primary design variables of the system are as shown. The optimum values, based on minimum RMS error, for each design variable is identified on each plot.

2.2 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.

$$IC = \frac{CC_{comp} * W\_dot\_in}{\left(1 - eta_{c_{isen}}\right)^{1.37} * (1 - eta_{vol})^{1.57}} + CC_{rp} * r_p^{1.5}$$

Cost coefficients CC_{comp} and CC_{rp} are obtained using regression analysis.

Obtaining cost data was one of the challenging tasks of this project, as there is not much cost data available based on the primary design variables of compressors. Cost data was obtained from supplier websites based on the isentropic, volumetric efficiencies, and pressure ratio of the compressor. Using regression analysis, an equation was derived with empirical coefficients. Five compressors with different efficiencies and power requirements were considered for the regression analysis.

Table 1: Cost data based on isentropic and volumetric efficiencies of compressors used for regression analysis

Compressor#	Isentropic efficiency [%]	Volumetric efficiency [%]	Power rating [kW]	Cost [$]
1	68	70	0.7	240
2	71	68	0.7	253
3	70	65	0.6	175
4	70	70	1.1	430
5	75	72	1.3	720

Table 2: Cost data based on pressure ratios of compressors used for regression analysis

Compressor#	Pressure ratio [-]	Cost [$]
1	4	220
2	5.5	405
3	6.5	475
4	8	660
5	9.5	875

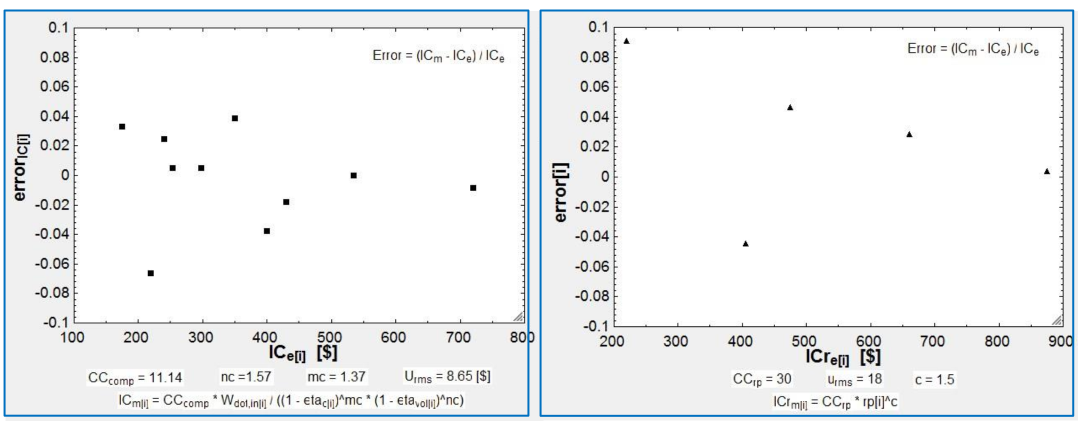

Figure 2- Results of Regression analysis carried out to determine the cost coefficients. The cost coefficients are optimized to obtain RMS error of less than 10%.

3. APPLICATION

3.1 Application: Automobile HVAC

Another paper by the same authors - Aralikatti and Zietlow (2018) describes an analytical model that links the primary design variables to the total life cycle costs. This validated model was used to compute the total cost of an automobile AC unit using inputs specific to automobile HVAC systems. The system inputs are suitably adopted for the average speed of vehicle, cooling loads, climatic conditions etc.

The cooling load for an automobile AC unit is divided into sensible and latent cooling loads. The heat gains considered in this project included heat gain from occupants, solar heat gain through the windows and the heat gain through the car envelope. The car envelope is assumed to be made up of thermal resistances from doors, windows, roof and the atmosphere inside and outside of the automobile. A schematic is shown in Figure 3. Thermal conductivity for a typical automobile door (sheet metal, insulation) was derived using ASHRAE Handbook of Fundamentals. The surface area of the door, roof, window glass and flooring was calculated for a typical sedan automobile. Fresh air requirements (ventilation) for two occupants were also included as part of the heat gain. Heat gains from infiltration air and fresh air (based on IAQ requirements) added to the cabin space are also included in this model.

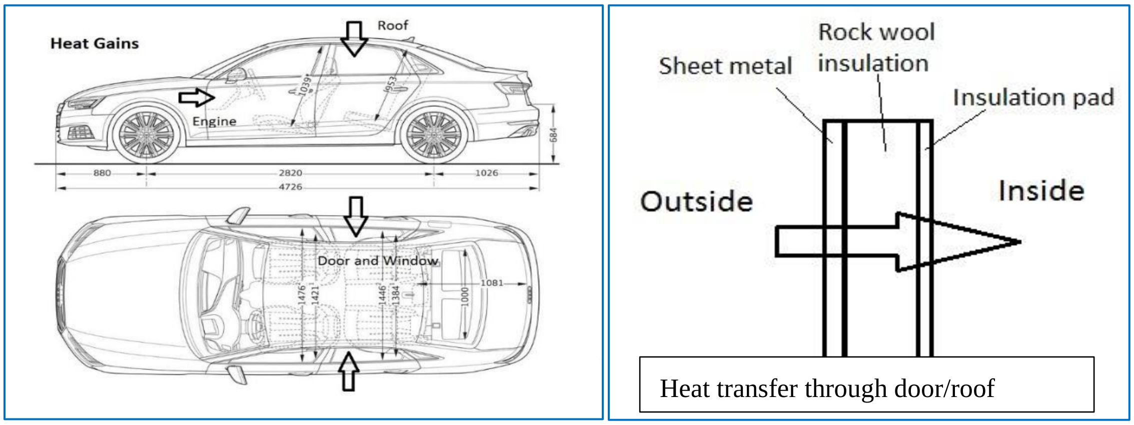

Figure 3: Schematic of thermal resistance network showing the heat gain between the environment and the system.

Heat gains from the engine, other miscellaneous items in the cabin space of the automobile, sensible and latent cooling required for infiltration of air and water vapor were all considered part of the cooling load in this model. The operating conditions of the automobile were determined for four different climatic conditions: 1) Phoenix, AZ for longer summer season and low humidity, 2) Peoria, IL for moderate summer season and moderate humidity, 3) Miami, FL for longer summer season and high humidity, and 4) Minneapolis, MN for shorter summer season and low humidity. The mean summer temperature, average duration of summer (number of cooling days), average humidity over the summer, the average solar heat gain factors were some of the factors considered to calculate the average rate of heat gain.

The performance of the AC unit was evaluated for each of these conditions and the optimum isentropic and volumetric efficiencies compressor were determined based on the lowest total cost of the AC unit.

4. RESULTS AND DISCUSSION

Optimum values of primary design variables for the AC unit are reported in Table 1.

Table 3: 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 were found in agreement to a paper titled 'Polytropic Exponents for Common Refrigerants' by Lenz (2002). The paper 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 varied between 0.99 and 1.29.

Next, the application of the analytical model to an automobile HVAC for the four different climatic conditions is presented.

4.1 Phoenix AZ

The longer summer season and lower humidity showed a higher total cost as the usage of the AC unit was higher. The higher operating time shifted the optimization more towards reducing the operating cost of the system. The optimum produces a compressor with higher isentropic (59.5%) and volumetric efficiencies (72.5%). The graphs show the variation of cost with isentropic and volumetric efficiencies. Also, a sensitivity analysis was done on the primary design variables. All the variables are normalized by calculating the percent change from the baseline case. Since the polytropic exponent produced a greater slope than the other variables then the total cost is most sensitive to this variable. The polytropic exponent accounts for the frictional losses in the system.

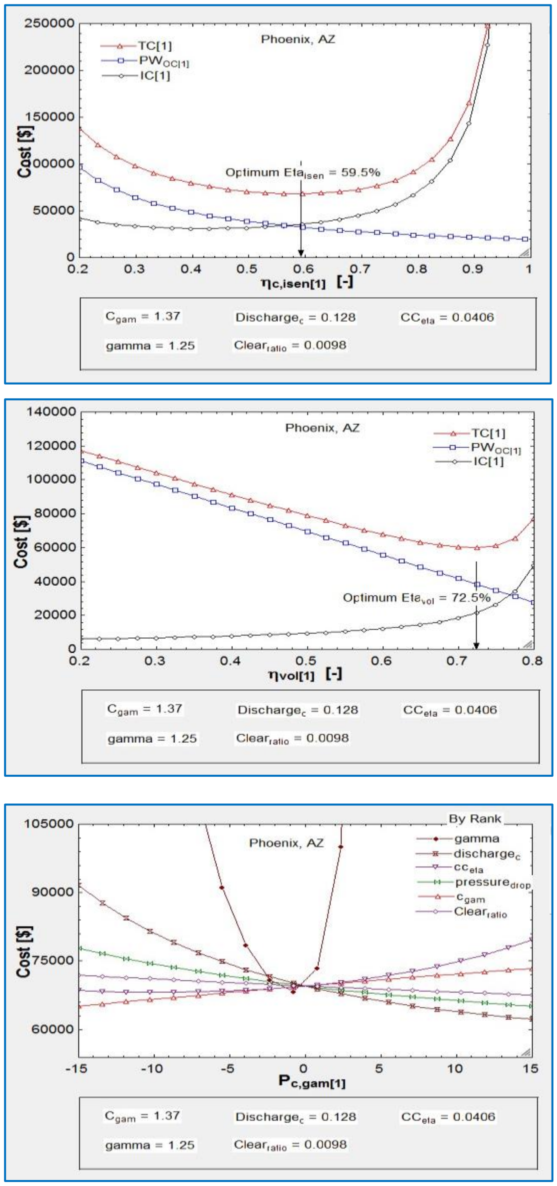

Figure 4: a) Graph shows the variation of initial (IC), operating (OC) and total life cycle cost (TC) with the isentropic efficiency of the compressor for Phoenix, AZ b) Graph shows the variation of initial (IC), operating (OC) and total life cycle costs (TC) with the volumetric efficiency of the compressor for Phoenix, AZ c) Graph shows the sensitivity of the Total costs to the primary design variables a compressor

4.2 Peoria IL

A moderate summer season and moderate humidity climate showed a significantly lower total cost as the usage of the AC unit was less. Thus, the optimization was shifted more towards reducing the initial cost of the system. The model recommends a compressor with lower isentropic (53%) and volumetric efficiencies (70%) relative to Phoenix, AZ. The sensitivity analysis shows the optimum was most sensitive to the polytropic exponent followed closely by isentropic efficiency coefficient and the discharge coefficient.

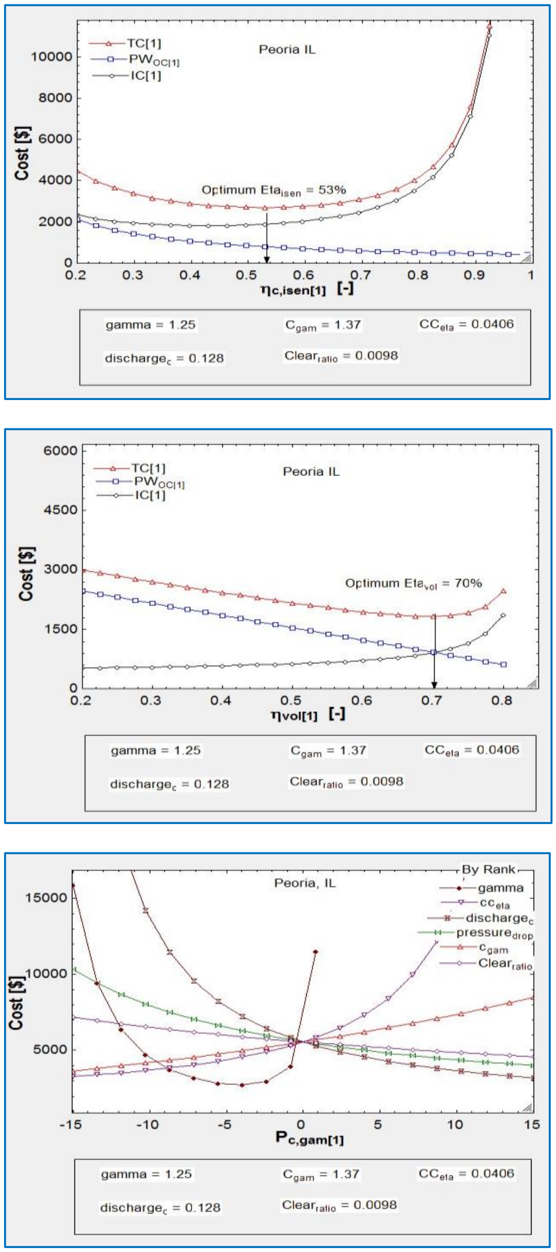

Figure 5: a) Graph shows the variation of initial (IC), operating (OC) and total life cycle cost (TC) with the isentropic efficiency of the compressor for Peoria, IL b) Graph shows the variation of initial (IC), operating (OC) and total life cycle costs (TC) with the volumetric efficiency of the compressor for Peoria, IL c) Graph shows the sensitivity of the Total costs to the primary design variables a compressor

4.3 Minneapolis, MN

Characterized by a short summer season and moderately humid climate generated a lower total cost. The usage of the AC unit was the least among other cities considered, thus the optimization was shifted more towards reducing the initial cost of the system. Thus, a less expensive compressor with a corresponding lower isentropic (49.5%) and volumetric efficiencies (67.5%) produced the minimum total cost. The graphs show the variation of cost with isentropic and volumetric efficiencies and also a sensitivity analysis of the total costs to the primary design parameters.

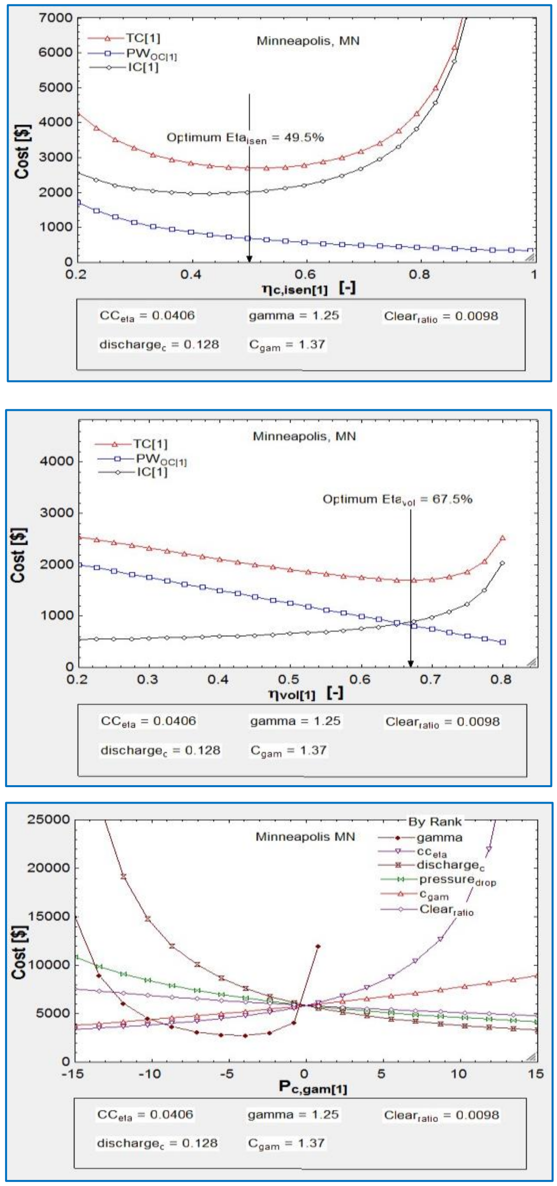

Figure 6: a) Graph shows the variation of initial (IC), operating (OC) and total life cycle cost (TC) with the isentropic efficiency of the compressor for Minneapolis, MN b) Graph shows the variation of initial (IC), operating (OC) and total life cycle costs (TC) with the volumetric efficiency of the compressor for Minneapolis, MN c) Graph shows the sensitivity of the Total costs to the primary design variables a compressor

4.4 Miami, FL

Characterized by a longer summer season and high humidity climate, the air conditioning system showed a significantly higher total cost as the usage of the AC unit was high. Thus, the optimization was shifted more towards reducing the operating cost of the system. Therefore, a more expensive compressor with higher isentropic (59.5%) and volumetric efficiencies (72.5%) could be justified.

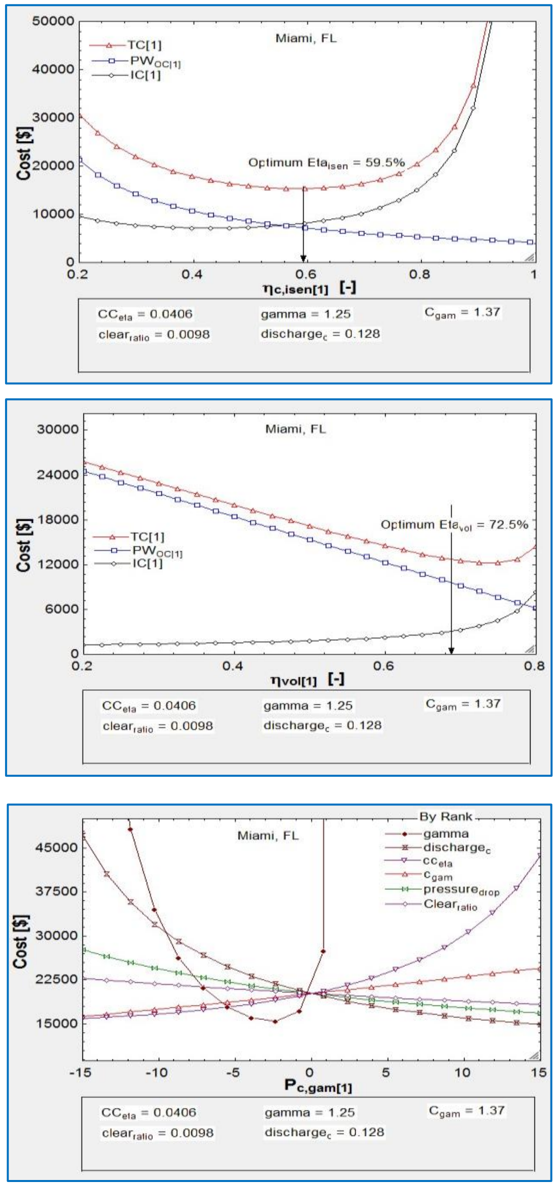

Figure 7: a) Graph shows the variation of initial (IC), operating (OC) and total life cycle cost (TC) with the isentropic efficiency of the compressor for Miami, FL b) Graph shows the variation of initial (IC), operating (OC) and total life cycle costs (TC) with the volumetric efficiency of the compressor for Miami, FL c) Graph shows the sensitivity of the Total costs to the primary design variables a compressor

5. CONCLUSION

The project successfully established the relation between the design variables of an air conditioning compressor and its total life cycle cost. The model details a logical progression of equations linking the primary design parameters to the objective function (minimizing total cost). The experimentally validated model was applied to an automobile AC to understand how the compressor optimization varied with operating conditions. The model is used to determine the best compressor for a given application which would yield the minimum total life cycle cost. For illustration, four different climate conditions, covering a wide range, were studied and the results discussed. A summary of the same is provided in the table below. The table also lists the latent and sensible cooling loads and the resulting optimum compressor efficiencies.

Table 4: Summary of model predictions for different climate conditions

City	Climate	Sensible load [kW]	Latent load [kW]	CDD [ºC-days]	Optimum values based on minimum total life cycle costs	
					Eta_isen	Eta_vol
Phoenix, AZ	High temp, low humidity	4.395	3.322	5207	59.5	72.5
Peoria, IL	Moderate temp, moderate humidity	2.568	0.287	1360	53	70
Minneapolis, MN	Low temp, moderate humidity	2.15	0.833	1036	49.5	67.5
Miami, FL	High temp, high humidity	3.763	1.345	4684	59.5	72.5

NOMENCLATURE

AC	*Air Conditioning*
AZ	*Arizona*
IL	*Illinois*
MN	*Minnesota*
FL	*Florida*
IC	*Initial Cost*
OC	*Operating Cost*
NTU	*Number of Transfer Units*
RPM	*Rotations Per Minute*
RMS	*Root Mean Square*
CC_{comp}	*Cost coefficient for power rating*
W_dot_in	*Power rating of the compressor (rate of work input)*
Eta_c_isen	*Isentropic efficiency*
Eta_vol	*Volumetric efficiency*
CC_{rp}	*Cost coefficient for pressure ratio*
r_p	*Pressure ratio*
EES	*Engineering Equation Solver*
CC_eta	*Coefficient of isentropic efficiency*
Gamma	*Polytropic exponent*
Clear_ratio	*Clearance ratio of the cylinder*
C_gam	*Coefficient of volumetric efficiency*
Discharge_c	*Discharge coefficient for the leakage flow rate through the valves*
HVAC	*Heating, Ventilating and Air-Conditioning*
ASHRAE	*American Society of Heating, Refrigerating and Air-conditioning Engineers*
TC	*Total life cycle Cost*

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