EXERGY ANALYSIS OF SCROLL COMPRESSORS WORKING WITH R22, R407C, AND R417A AS REFRIGERANT FOR HVAC SYSTEM

by

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The rise in crisis of power enthralls the world economically and the options for conventional and non-conventional energy resources have been searched out. No system exists in this world with 100% efficiency due to several irreversibility's. If the output obtained from the system is maximum for a given input, maximum amount of energy can be saved globally. To understand the thermodynamic losses occurring in the system and to predict the available energy that can be tapped from the system, exergy plays a major role. Experimental study on exergy in a system can pave the way to understand the complete behavior of the system exergually. Conceptually exergy studies are based on simulation, to provide a new dimension to the concept of exergy experimental validation have been promoted. The analogy of exergy analysis of three refrigerants working in scroll compressors and their exergual features are explained in this paper. The refrigerants R22, R417A, and R407C and their thermo dynamical behavior, irreversibility were experimented in an air conditioning system with three scroll compressors, interaction between the system and the refrigerant in terms of pressure drop and heat transfer, friction has been implemented for the calculation of exergy. The entire system performance on the basis of refrigerant is validated in each part of the air conditioning system. The resultant coefficient of performanace of R407C is 2.41% less than R22 in a R22 designed scroll compressor with minimal exergy losses. The second law efficiency of 50 to 55% obtained in R22 has fewer rules over R407C and R417A which has 48 to 52%. The diminutive deviation of results encourages R417A refrigerant to be used as a substitute for R22. Thus the exergual prediction of performance of refrigerant and second law efficiency can identify the use of eco-friendly refrigerant in scroll compressor.

Key words: *exergy analysis, scroll compressor, alternate refrigerant, HVAC system, eco-friendly refrigerant*

Introduction

Air conditioners are the major consumers of power. In heating, ventilation and air conditioning (HVAC) system the compressor consumes 85% of the total power input and the remaining 25% is utilized by the other components. Due to the systems irreversibility, 15% of the energy supplied to the compressor is thermodynamically trapped and converted in to unavailable energy. Hence in the current scenario the study of compressor is imperative. In compressor the exergual losses are due to pressure drop and heat transfer characteristics in a system. The technical innovations and the energy crises force the researchers to intensify the system with minimum exergual losses, which results in the development of scroll compressor. The previous studies in scroll compressor focused thermodynamic analysis and optimization. This includes the study of exergy that can improve the technology to avoid exergy losses. Exergy analysis can be applied to the optimization and performance evaluation of energy systems. A thermodynamic model for a variable speed scroll compressor with refrigerant injection was developed and the deviations of the predicted from the measured value were within 10% for approximately 90% of experimental data [1]. This model included the energy balance in the low-pressure shell compressor, suction gas heating, motor efficiency, and volumetric efficiency considering gas leakages as a function of compressor frequency. Except using the general design optimization model includes multi-variable, direct search, inequality constraint shows that the maximum deviation of cooling capacity, coefficient of performance (COP) are below 2.53 and 1.69% [2].The compressor components are analyzed in terms of nine different elements. Using simulation program based on the overall compressor model, a parametric study of the scroll compressor was performed.

R22 has been widely accepted as the leading refrigerant for higher COP, due to the environmental issue this refrigerant has to be phased out. The identification of direct substitute for this refrigerant becomes an issue. R407C heat pump having a large temperature difference shows improvement in compression process for a steady operating condition [3]. Thermo-economic analysis on second law based thermodynamics applied to evaporative cooling system coupled to an adsorption dehumidifier, with exergetic manufacturing cost (EMC) application to different conditions for optimization of operating cost and predicted the effectiveness of a direct evaporative cooling as 90% and indirect evaporative cooling is between 70 and 80% [4]. The hermetic scroll compressor with variable speed scroll for different working condition the significant energy saving on average equal to about 20% by adopting a scroll compressor speed control algorithm [5].

The mathematical modeling of scroll compressor with steady-state energy balance equation using lumped capacitance method and the temperature of compressor elements to get power consumption and efficiency of compressor and it is explained that the radial and flank leakage gap size, the compressor efficiency decreases from 49.20 to 38.51%, and 47.88 to 43.09% [6]. The exergy analysis of vapor compression refrigeration system for investigating the effects of the evaporating and condensing temperatures on pressure loss, it was found that the evaporating and condensing temperatures have strong effects on exergy loss that affects second law efficiency and COP, the total exergy loss decreases with decreasing temperature difference between evaporator and refrigerated space is also verified [7]. The heat pump system with economizer coupled scroll compressor can extend effectively its operating ranges and provide a technological method to enable the heat pump to run steadily in severe weather conditions, the essential parameter that affects crucially the performance of heat pump system using experimental data of the heat pump prototype the relative intermediate pressure is also reported to be in-between 1.1 and 1.3, respectively [8]. Thermodynamics and heat transfer modeling of scroll pump, its functioning and comparison of the performance of scroll pump with other pumps are given in [9]. The exergy analysis of heat pump system with economizer coupled with scroll compressor, a descriptive approach to exergual effects in heat pump system are given in [10]. The basic heat transfer mechanism with the concept of two phase heat transfer and exergy losses in terms the exergo-economical prediction for heat exchanging materials, the imperfections in materials to avoid irreversibility is considered [11]. Two-dimensional model has been developed to study the fluid flow and heat transfer in the working chamber of scroll compressor, the unsteady continuity, momentum, and energy equations for the gas flow in scroll compressor were formulated, the curvilinear, moving and deforming scroll chamber were formulated, the results shows that available empirical correlations with lumped parameter approach are inadequate in predicting the heat transfer within the scroll compressor chamber, the operating speed of 3000 rpm and maximum scroll angle of 400° is considered in [12]. The experimental study of convective heat transfer in the compressor operating, at variable speeds, using 13 thermocouples installed inside the compressor and temperature measured at the discharge port, has been performed to verify simulation result with relevant heat transfer coefficient. The whole consecutive compression process in the scroll compressor is simulated in detail by solving equations of mass and energy balance for the refrigerant [13].

Methodology

System description

The system consists of three scroll compressors with individual oil separators. The oil separator is connected to the condenser which is fixed with a receiver for storing the liquid refrigerant. The system ends with the evaporator, called as insulated room. There are four valves (valve 1, valve 2, valve 3, valve 4) placed before the compressor and three valves after the compressor (valve A, valve B, valve C) for the operation of refrigerants individually in each compressor. Sensors were placed at each inlet and exit of compressor and condenser. The sensors are connected to a 12 channel display to monitor the temperature. The values obtained from the sensors are connected to a data logger through the display. The pressure and temperature gauges were placed at each section to obtain the reading simultaneously. The setup containing compressor, condenser, and expansion valve are placed outside the room to avoid the heat and noise liberation from the instruments. The compressor used is of 1.5 TR capacity with radial compliance; the details about the compressor are encrypted in tab. 1.The condenser and evaporator used is

Table 1. Specifications of the scroll compressor used in the experimental plant

No.	Compressor types	Refrig- erant	Input power	Lubricant used	Motor outside diameter	Capacity [W]
1.	Compressor 1	R22	220 V, three phase	Mineral oil	140 mm	5250 (1.5TR)
2.	Compressor 2	R417A	220 V, three phase	Mineral oil	140 mm	5250 (1.5TR)
3.	Compressor 3	R407C	220 V, three phase	POE (polyolester)	140 mm	5250 (1.5TR)

designed to operate under optimum conditions for refrigerants R22, R417A, and R407C.

Experimental procedure

The refrigerants were charged individually with mineral oil in each compressor, the preliminary leak test was conducted and leakages were completely controlled since the exergy efficiency lies on less entropy losses. The schematic of the experimental plant layout is depicted in fig. 1. Initially the system is operated with R22 with mineral oil. The system is allowed to run for 45 minute to 1 hour, during which valve 1 is opened and valve 2, valve 3, and valve 4 are closed. So the refrigerant flows through compressor 1. The time taken to attain 25 °C in the room

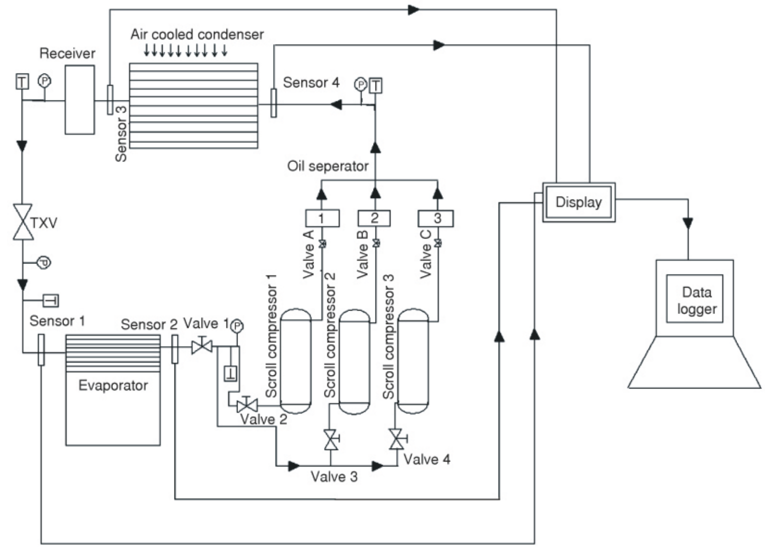

Figure 1. Experimental plant lay-out

were calculated for compressor 1. Similarly by varying the evaporator temperature as 2, 4, 6, 8, and 10 °C. The corresponding pressure, and temperature readings are recorded in the data logger. After the operation of a complete cycle with R22 valve A with valve 2, valve 3, valve 4 are closed. The refrigerant R22 is continuously pumped into the compressor without discharge, till the pressure gauge shows negative pressure and the valve 2 is completely closed. The compressor 2 containing refrigerant R417A is switched on and the valve 1 and valve 3 is opened along with valve B. A set of readings were taken after 1 hour of the operation of R417A at 25 °C. Now the valve B is closed, it is confirmed from the negative pressure that all the refrigerants were pumped into the compressor and valve 3 is closed.

Finally the system with R407C is operated by opening the valve C and valve 4 and the compressor 3 is switched on. Similar procedure is repeated for R407C with various evaporator temperatures of 2, 4, 6, 8, and 10 °C. All the refrigerants were pumped into scroll compressor by closing valve C and valve 4. During the operation of single compressor the valves pertained to other compressors are closed so that no leak of refrigerant from one cycle to the other is entertained.

Experimental analysis

The analysis carried out for a specific operating condition by maintaining the evaporator coil temperature as 10 °C the room maintained at 25 °C with condenser temperature of 45 °C.

Heat transfer loss between compressor wall and the surroundings is:

$$\text{Loss} = Q_\mathrm{w} \left(1 - \frac{T_0}{T_\mathrm{w}}\right) \tag{1}$$

where Q_w is the heat transfer through the wall, T_W – the wall temperature, and T_0 – the ambient temperature.

Due to mixing of re-expansion gases, the fresh charge which leads to the exergy loss:

$$\text{Loss} \quad T_0 S_{\text{gen}} \tag{2}$$

The irreversibility occurring in a system is mainly due to wire drawing, different modes of losses in a scroll compressor is due to mixing, friction, conduction, suction wire drawing, and discharge wire drawing.

Exergy loss due to mixing

The exergy destruction occurs inside the compressor. This loss is due to mixing of re-expansion of the compressed refrigerant vapor and fresh charge at the time of expansion:

$$I_{\text{mix}} \quad T_0 \;\; m_{\text{i}} \;{}_{T_{\text{i}}}^{T_0} \frac{Cp\text{d}\theta}{\theta} \quad \frac{Q_{\text{mix}}}{T} \tag{3}$$

$$Q_{\text{mix}} \quad m_{\text{i}}\,(h_{\text{i}} \quad h) \tag{4}$$

The mixing takes place inside the compressor, the loss in exergy mainly occurs in-between the scroll mating parts. By varying the scroll angle according to the arbitrary motion the amount of loss in exergy can be predicted by using the formula (3) where T_0 is the absolute temperature of the environment, T_{i} – the absolute temperature just downstream of an adiabatic throttling process, θ – the angle, and h – the enthalpy.

Exergy loss due to friction

The exergy destruction rate associated with a rate of friction work that is dissipated as a heat transfer to cylinder wall:

$$I_{\text{f}} \quad T_0 S_{\text{gen}} \tag{5}$$

where

$$S_{\text{gen}} \quad \frac{m\Delta P}{\rho T_{\text{in}}} \tag{6}$$

Exergy loss due to conduction

This exergy loss occurs by the conduction from the shell to the atmosphere:

$$I_{\text{cond}} \quad Q_{\text{w}} \;\; 1 \quad \frac{T_0}{T_{\text{w}}} \tag{7}$$

Exergy loss due to suction wire drawing

The fluid flowing into the system, the exergy destruction rate due to adiabatic throttling is given by:

$$I_{\text{suc}} \quad T_0 m(S_{\text{i}} \quad S_{\text{i}}^{1}) \tag{8}$$

Exergy loss due to discharge wire drawing

$$I_{\text{dis}} \quad T_0 m(S_{\text{i}}^{1} \quad S_{\text{i}}^{11}) \tag{9}$$

Exergy loss due to convection

$$I_{\mathrm{con}} \quad h_2 \quad h_3 \tag{10}$$

Work wasted in compressor

$$W_{\mathrm{r}} \quad I_{\mathrm{con}} \quad T_0(S_3 \quad S_2) \tag{11}$$

Exergy loss in evaporator

The entropy generation in evaporator is due to pressure loss:

$$\Delta P_{\mathrm{evap}} \quad S_1 \quad S_4 \quad 2 \; \frac{h_1 \quad h_4}{T_4 \quad T_1} \tag{12}$$

Exergy loss in condenser

The entropy generation due to pressure loss inside the condenser is given by:

$$\Delta P_{\mathrm{cond}} \quad S_3 \quad S_2 \quad 2 \; \frac{h_2 \quad h_3}{T_2 \quad T_3} \quad C_{\mathrm{pv}} \ln \frac{T_2}{T_2^1} \tag{13}$$

Total exergy loss in a system is the summation of all exergy losses in various auxiliaries of the system:

$$I_{\mathrm{total}} \quad I_{\mathrm{mix}} \quad \Delta P_{\mathrm{cond}} \quad \Delta P_{\mathrm{evap}} \quad I_{\mathrm{con}} \quad I_{\mathrm{dis}} \quad I_{\mathrm{suc}} \quad I_{\mathrm{cond}} \tag{14}$$

Second law efficiency

$$\eta_{\mathrm{II}} \quad \frac{W_{\mathrm{rev}}}{W_{\mathrm{ac}}} \tag{15}$$

where $W_{\mathrm{ac}} = W_{\mathrm{rev}} + W_{\mathrm{r}}$

$$W_{\mathrm{rev}} \quad Q_{\mathrm{evap}} \; \frac{T_0}{T_{\mathrm{evap}}} \quad 1 \tag{16}$$

Work rejected in compressor

$$W_{\mathrm{rej}} \quad T_0(S_2 \quad S_1) \tag{17}$$

Work rejected in expansion valve

$$W_{\mathrm{rej}} \quad T_0(S_4 \quad S_3) \tag{18}$$

Work rejected in evaporator

$$W_{\mathrm{rej}} \quad T_0(S_1 \quad S_4) \quad T_0 \; \frac{Q_{\mathrm{b}}}{T_{\mathrm{space}}} \tag{19}$$

$$Q_{\mathrm{b}} \quad T_{\mathrm{c}}(S_1 \quad S_4) \tag{20}$$

Results and discussion

The evaporator pressure loss associated with the condensing temperature, shows the influence of condensing temperature on evaporator pressure loss (fig. 2). Even though the pressure loss inside the evaporator is very small, the loss is accounted since the concept of exergy depends on pressure loss and heat transfer. It is observed that the pressure loss for refrigerant R22 is 0.01% less compared to R417A and 0.04% less compared to R407C. As the temperature of the refrigerant increases to the increase in length of the evaporator the pressure drop crop up due to friction and heat transfer. The density decreases in each component of mixture refrigerant is due to heat and friction inside the tube which influences more pressure loss for eco-friendly refrigerants than conventional refrigerant.

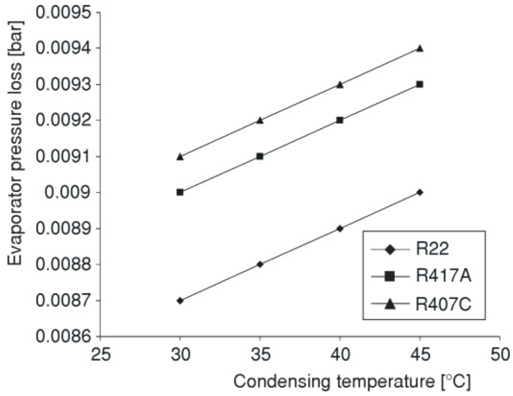

Figure 2. Influence of condensing temperature on evaporator pressure loss

The work rejection is frequent with any air-conditioning cycle operating at different temperatures. The increase in evaporator temperature persuades the work rejection capability of the system which is encrypted in fig. 3. It is observed that as the evaporator temperature increases the mass flow rate through the system is less, whereas if the evaporator temperature is low the mass flow rate increases in the system which leads to more work rejection. For the mixture refrigerants the entropy losses is more due to the property of poor heat holding capacity.

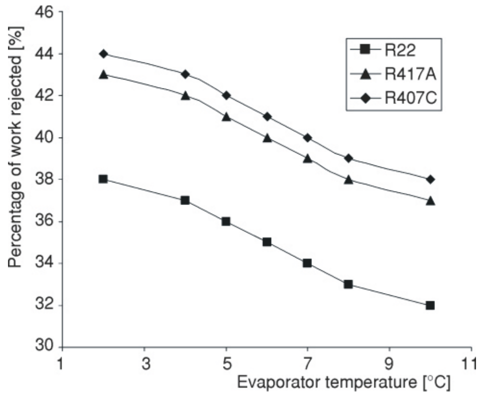

Figure 3. Percentage of work rejected for the increase in evaporator temperature

The influence of evaporator temperature on condenser pressure is shown in fig. 4. The pressure loss occurring in condenser is relatively high compared to that in evaporator. This is due to the high pressure and temperature of the refrigerant at the compressor discharge, *i. e.* the condenser inlet. Depending on the evaporator temperature the mass flow rate in the condenser is increased or decreased, so the pressure drop inside the condenser is determined by the change in mass flow rate. Due to high vapour pressure, the pressure drop for mixed refrigerants is high.

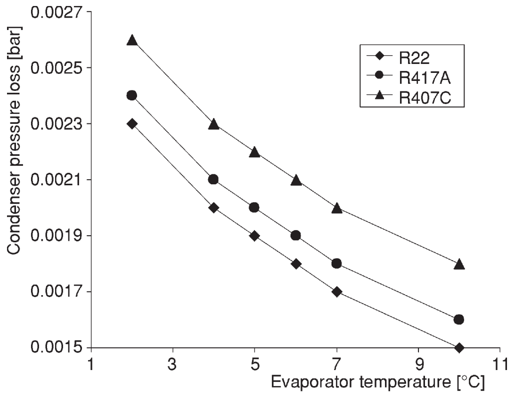

Figure 4. The influence of condenser pressure loss for the increase in evaporator temperature

The increase in COP determines the performance of the system. The effect of condenser temperature has the variation in COP which is shown in fig. 5. It is observed that the increase in condenser temperature decreases the COP for constant evaporating temperature of 10 °C.The heat loss for R417A and R407C is larger than R22 which results in lower coefficient of performance for R417A and R407C.

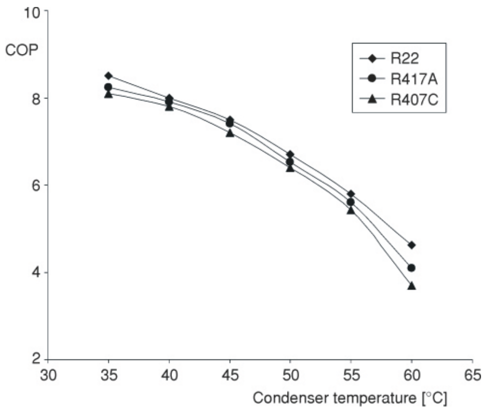

Figure 5. The variation of COP for increase in condenser temperature

The compressor contributes the major work requirement for the air conditioning system, the losses or leakage (radial leakage, axial leakage) of refrigerant inside the compressor causes the loss in exergy of the system which decreases the energy efficiency ratio of the system. It is observed that as the compressor discharge temperature increases the exergy loss increase which is shown in fig. 6. As the temperature increases inside the compressor the refrigerants R417A and R407C follows the same trend as R22.

The variation in COP to the increase in evaporator temperature has the better influence than the temperature variation in condenser, there is a drift in increase of COP for the evaporator temperature of 4 and 7 °C as shown in fig. 7, and this shows that if the cycle is operated at 4 and 7 °C better COP can be obtained.

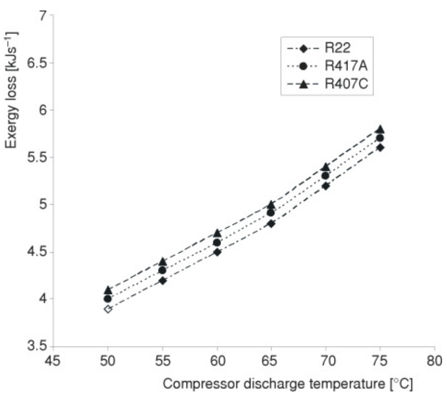

Figure 6. The influence of compressor discharge temperature on exergy loss

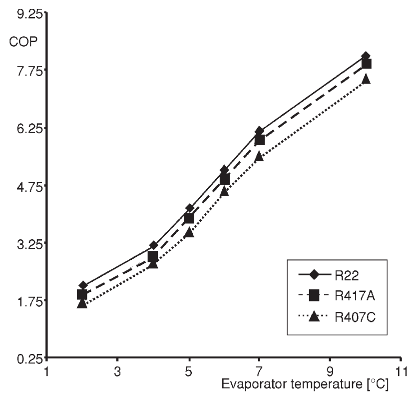

Figure 7. The variation of COP for increase in evaporator temperature

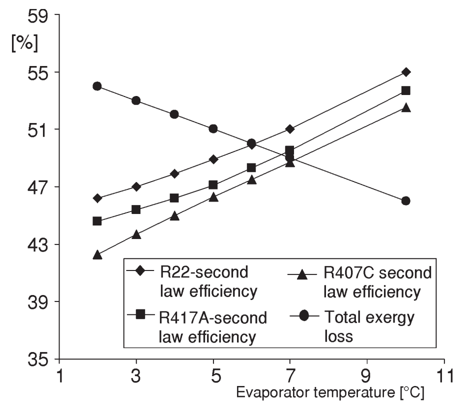

Figure 8. Influence of evaporator temperature on the percentage of exergy loss

The second law efficiency determines the performance of the system as show in fig. 8. It is observed that the percentage of second law efficiency is improved to 55% for R22, 54% for R417A, and 52% of R407C for the cycle operating at the maximum evaporating temperature of 10 °C. To have better second law efficiency the system has to be operated at higher operating temperature. The total exergy loss occurs at 50% of the second law efficiency for R22, 48% of R417A, and 46% of R407C. This shows that the second law efficiency of R22 is more compared to eco-friendly refrigerants. When the mixture refrigerant gets heated up due to compression it dissipates more heat energy due to the decrease in density and viscosity of difluoromethane (R32), pentafluoroethane (R125), and R134a which is a mixture in R407C and R125. R134a and R600 mixture in R417A makes the refrigerant to lose more heat in the form of exergy that results in higher exergy loss which is responsible for lower second law efficiency in eco-friendly refrigerants.

Exergy destruction starts from the compressor suction temperature, depending upon the compressor suction temperature and pressure drop in the suction tube for refrigerants R22, R417A, and R407C (fig. 9). If the suction temperature is above 14 °C the exergy loss is enhanced, so as to minimize the exergy loss, the compressor has to be operated within the suction temperature of 14 °C. Practically it is impossible to operate the system with a suction temperature of 14 °C in an air-conditioning application. The exergy loss in the compressor suction has to be scarified.

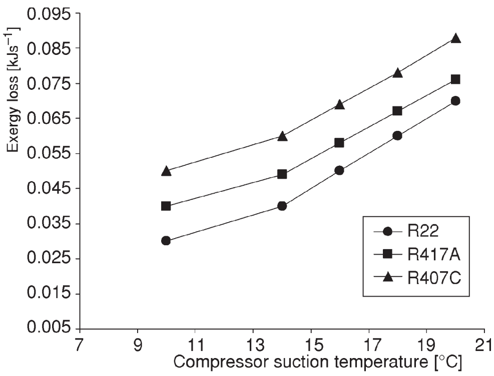

Figure 9. The variation in exergy loss for the increase in compressor suction temperature

Conclusions

The refrigerant based exergy analysis reveals the suitability of refrigerant for scroll compressor. Compressors with individual refrigerants and mineral oil can perform better thereby accurate prediction of exergy can lead to accurate results.

The total exergy losses for R22 is found to be 1% less than R417A and 1.5% less than R407C.To conserve the exergy loss by using eco-friendly refrigerant R417A can be used as an alternative than R407C.

To obtain good second law efficiency and minimum exergy the system has to be operated with 4 °C evaporator temperature, 35 to 40 °C condenser temperature, within 65 °C of compressor discharge temperature, and 14 °C of compressor suction temperature.

For the mixture refrigerant to have good performance and less exergy loss the air conditioning cycle has to operate under constant temperatures (with fixed evaporator and condenser temperature).

Nomenclature

C_p – specific heat of constant pressure, [kJkg^{-1}K^{-1}]
I – identifier for inlet flow stream
I_{con} – convection loss
I_{cond} – conduction loss
m – mass flow rate, [kgs^{-1}]
m_i – mass flow inlet, [kgs^{-1}]
ΔP – pressure difference, [bar]
Q_b – heat rejected from evaporator, [kW]
S_{gen} – entropy generation, [kJkg^{-1}K^{-1}]
S_i – specific entropy before throttling, [kJkg^{-1}K^{-1}]
S_i^1 – specific entropy after throttling, [kJkg^{-1}K^{-1}]
S_i^{11} – specific entropy at the discharge, [kJkg^{-1}K^{-1}]
S_1 – entropy at the compressor inlet, [kJkg^{-1}K^{-1}]
S_2 – entropy above saturation point, [kJkg^{-1}K^{-1}]
S_2^1 – entropy in saturation point, [kJkg^{-1}K^{-1}]
S_3 – entropy at the exit of condenser, [kJkg^{-1}K^{-1}]
S_4 – entropy at the inlet of evaporator, [kJkg^{-1}K^{-1}]
T – apsolute temperature, [K]
T_c – condenser temperature, [K]
T_w – wall temperature, [K]
T_0 – absolute temperature of the environment, [K]
T_1 – temperature at the compressor inlet, [K]
T_2 – temperature above saturation point, [K]
T_2^1 – temperature in saturation point, [K]

T_3 – temperature at the exit of condenser, [K]
T_4 – temperature at the inlet of evaporator, [K]
TR – tonage of refrigeration
W_{rej} – amount of work rejected, [kJs^{-1}]

Greek letters

θ – angle of rotation (deg.)
ρ – density, [kgm^{-3}]

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