Thermodynamic performance evaluation of HFC refrigerants for the chiller system simulated by hot gas bypass cycle

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ABSTRACT

The competitive of the building air-conditioning equipment pushes manufactures to develop and improve the performance and efficiency of their products. This process is cost and time consuming for testing the whole units. In this study, for the first time a modified proposed test block with hot gas bypass cycle (HGBP), based on the experimental compressor map, has been used as a model to investigate the theoretical performance of the chiller system. The model assumes one dimensional steady state conditions and uses an experimental compressor map to predict the thermodynamic performance of the system. A computational model, using the software program EES (EES-V8.4), in a single stage of the chiller system with HGBP cycle based on the first and second law analyses are presented for the prediction of the effects of the pressure ratio, degree of superheating and sub-cooling, condenser temperature, outdoor air temperature and humidity ratio on the exergy destruction, exergetic efficiency, gas power and coefficient of performance (COP) of the chiller system. The performance of the implemented model with various hydrofluorocarbon (HFC) refrigerants such as R134a, R152a and R423a are compared. The results, firstly, indicates that the modified model with applying the energy and exergy analysis works for all kind of working refrigerants including HFC refrigerants. Secondly, the performance of the compression refrigeration system based on the experimental compressor map data to choose the right compressor can help engineers to design a high quality unit before going to design the system practically. The results indicate that R423a has a higher coefficient of performance as compared to the rest of the refrigerants. The developed simulation model helped to establish the strong dependence between system performances and the compressor map.

1. INTRODUCTION

The significant improvements in refrigeration system require technological developments. As a result of these developments, applications and fields of refrigeration have become widespread. The refrigeration systems are used in a wide range of applications from domestic to industrial applications. Vapour compression refrigeration system are one of the best and efficient method for air-conditioning purposes among the various methods of cooling and heating systems [A. Arora et al. 2008, S. S. Seyitoglu 2005].

There are two advantages of vapour compression refrigeration cycles. First, a phase change process from liquid to vapour requires a large amount of thermal energy, and therefore removed a specified heat from the air-conditioned space [S. Devotta et. Al 2015, F. Samuel et.al. 2000]. Second the continuous extraction of heat as a result of isothermal nature of vaporization without reaching the temperature of the working fluid to the temperature of the environment [ E. Halimic, et. Al 2003].

In the last few decades, it is observed that
some of refrigerants cause ozone depletion layer and global warming. Chlorofluorocarbons (CFCs), due to their favorable characteristics such as low freezing point, non-flammability, non-toxicity and chemically stable behavior with other materials, have been used over the last few decades. Unfortunately, it was discovered that the chlorine destroys the earth’s ozone layer and causes serious health problems [B. O. Bolaji. 2010].

The production and marketing of substances that have high ozone depletion layer are regulated by the Montreal Protocol signed by the international community in 1987. The use of CFCs prohibited completely in 2010. The Kyoto protocol (UN- FCC, 2011) supported the plans to control the sustainable development and minimize the effect of global warming including the restriction of HFCFs. However, the problem of the global warming is still important issues with some of new refrigerants including HFCs.

The first law of thermodynamics, which is based on the energy balance, is a key point for analyzation of different systems and different refrigerants. However, it does not provide any information regarding how and what the details of the process are. Based on that, it can only discuss and compare the energy transfer and the efficiency of the system and this analyzation are not enough to show how much of this energy can be lost due to irreversibility or interaction between different parts of the refrigeration system [Yumrutas, et.al. 2002]. For these reasons, the analyzation process requires using the exergy balance, which provide the entire irreversibility of each part of the system and it gives a more realistic view of the process.

The competitive of the building airconditioning equipment pushes manufactures to develop and improve the performance and efficiency of their products. This process is costly and time consuming for evaluation of the performance of the equipment. The equipment and configuration of a typical hot gas by-pass test block cycle is described thoroughly in existing literature, primarily in work by [R. Yumrutas et al. At 2002].

J. McGovern. 1984]. Thermodynamic models of the individual components in the test block cycle exist, are well-established, and are used frequently in the thermal and fluid sciences. A simplified schematic of the test block layout is shown in figure 1.

![Figure 1: Test block diagram simulated by HGBP Cycle [9]](image)

The definitions of detailed information regarding the theoretical model development and solution methodology for the test are presented by [J. McGovern. 1984]. In this work, the contribution was to expand the model as an entire tool not only to test the performance of the compressor but to evaluate the energy and exergy analyses of the chiller system simulated by hot gas bypass cycle. The model does not include the evaporator and it compensates with the circulation of the water between the condenser and the cooling tower. In addition to that, the important modifications have been done to the analyzation of the condenser and cooling tower equations to improve the capability of using the software for the different case of the application at different boundary conditions such as outside...
temperature and humidity ratio of the atmosphere. The modifications in this work to the previous model, which was used by Gessler [J. McGovern. 1984 ], are:

- The compressor, condenser, cooling tower and throttling valves are assumed to be non-adiabatic and irreversible instead of adiabatic and reversible. For the cooling tower, different equations have been used from the previous work and which are more efficient and accurate than the previous equations by [9].

- The kinetic energy term is included for each part of the cycle.

- The exergy analysis is also used with the energy analysis which is only used in the previous work.

- Three different HFC refrigerants have been used instead of using only one refrigerant.

This paper ignores the pressure losses through the pipelines and the mixer and potential energies.

In this paper, thermodynamic analysis is applied to the vapor compression refrigeration cycle simulated by HGBP cycle. However, to the best knowledge of the author the exergy analysis is not used for this type of theoretical analysis of the chiller system simulated by HGBP cycle based on the compressor experimental map data. The expressions for the exergy losses (lost works) for the individual processes that make up the cycle as well as the coefficient of performance (COP) and second law efficiency for the entire cycle are obtained. Effects of condensing, degree of subcooling and degree of superheating temperatures on the exergy losses, second law efficiency and COP are investigated. Equation solver software with built-in thermodynamic functions is used to carry out the numerical calculations. Various hydrofluorocarbon (HFC) refrigerants such as R134a, R152a and R423a are compared.

2. Thermodynamic Methodology

The vapour compression refrigeration simulated by hot gas by pass cycle is analyzed using the first law and second law of thermodynamics. The rate forms of the energy and entropy balance equations [K. Wark.2002 ] are used throughout the model and engineering assumptions can be applied to each part of the model. The conservation of mass balance to the control volume is applied on the process and yields

\[
\left(\frac{dm}{dt}\right)_{cv} = \sum m_{in} - \sum m_{ot}
\]

This equation is applied to each part of the cycle, with the appropriate number of inputs and outputs.

2.1. The Steady Flow Energy Balance

This method is applied to each device within the whole system. The following mathematical expression, which is based on the first law of thermodynamics, is used to analyze the control volume of the system.

\[
\left(\frac{dE}{dt}\right)_{cv} = \left(\dot{Q} - \dot{W} + \sum_{in} \dot{m}(h_{in} + \frac{C^2}{2} + gz) - \sum_{ot} \dot{m}(h_{ot} + \frac{C^2}{2} + gz)\right)
\]

where \(dE\) denotes for energy transfer of the system (J), \(t\) stands for time (s), \(\dot{m}\) is the mass flowrate of the refrigerant (kg/s) and \(h\) is the specific enthalpy (J/kg). \(C^2 / 2\) and \(gz\) are the specific kinetic energy and the potential energy of the system respectively. \(\dot{Q}\) and \(\dot{W}\) are accounted for the heat rate and work rate flows of the control volume with its surroundings.

The subscripts \(in\) and \(ot\) are to denote the flow inlet and outlet of the system states respectively. According to the first law of thermodynamics, for energy conversion devices, the steady flow energy equation becomes:
Generally, most of researchers neglect the effect of kinetic and potential energies within the system, however, in this work the kinetic energy change has been included but it also neglects the potential energy. Therefore the equation 3 becomes:

\[
\frac{d\Omega}{dt} = \sum \left(1 - \frac{T_0}{T_j}\right) \dot{Q}_j - \dot{W} - p_0 \frac{dV_{cv}}{dt} + \sum \dot{m}\Omega - \sum \dot{m}\Omega - D\dot{S}_{irr}
\]

where \( \frac{d\Omega}{dt} \) is the time rate of change of exergy. The term \( (1 - T_0 / T_j)\dot{Q}_j \) and \( \dot{W} - p_0 \frac{dV_{cv}}{dt} \) represents the amount of exergy by heat transfer and work transfer respectively. The two terms of \( \dot{m}\Omega \) represents the amount of exergy transfer where the mass flowrate enters and exits the control volume. The steady state exergy rate balance becomes:

\[
D\dot{S}_{irr} = \sum \left(1 - \frac{T_0}{T_j}\right) \dot{Q}_j - \dot{W} + \sum \dot{m}\Omega - \sum \dot{m}\Omega
\]

Exergy destructions, using exergy balance, are evaluated in each component for the chiller system simulated by HGBP cycle and specified as follows.

- **Compressor**

The exergy destruction, \( T_0D\dot{S}_{irr} \) for non-adiabatic compressor can be calculated by applying the energy and entropy balances, respectively as shown below. The kinetic energy is included and the stagnation enthalpy, \( h_0 \), has been used instead of using static enthalpy. However, the potential energy is neglected.

\[
\dot{m}T_0(s_{0,in} - s_{0,in}) - \dot{m}(h_{0,in} + h_{0,in}) - T_0D\dot{S}_{irr} + \dot{W} = 0
\]

The value of the term, \( T_0D\dot{S}_{irr} \), is represented the exergy losses for the compressor.

- **Condenser**
The energy and exergy balances for refrigerant within the condenser are as follows

\[ \dot{m}_0 (s_{0,ot} - s_{0,in}) - \dot{m} (h_{0,ot,s} + h_{0,in}) - T_o \Delta S_{\text{irr}} = 0 \]  
(9)

- Cooling tower

The performance of a cooling tower is assessed by its effectiveness. In the literature, several definitions are used for effectiveness, but they are specific to certain boundary conditions and configuration. An equation used by [9], which was based on the work of [R. P. Mandi, 2005], was not robust and not applicable to a large class of heat and mass transfer exchanger devices.

In this work, an energy-based effectiveness and modified heat capacity ratio [P. Narayan, et.al. 2010] based on the total enthalpy rate change are used and it is accurate in all ranges of humidity and temperature levels. At the beginning the exergy equation, which is written in the following, developed by [T. Muangnoi et. Al 2007] used to evaluate the exergy flow of the fluid.

\[ X_{\text{air}} = \dot{m}_{\text{air}} \frac{c_p \omega_{\text{air}}}{R_v T_o} \left( \left( \frac{T_{\omega,\text{air}} - T_o}{T_{\omega,\text{air}} - T_i} \right) \ln \frac{T_{\omega,\text{air}} / T_o}{1 + 1.608 \omega_{\text{air}} / \omega_{\omega,\text{air}}} \right) \]  
(10)

However, the exergy destruction \( T_o \Delta S_{\text{irr}} \) of the cooling tower is calculated as follows.

\[ T_o \Delta S_{\text{irr}} = \dot{m}_{\text{water,in}} \left( h_{\text{water,in}} - h_{\text{water,out}} \right) + \dot{m}_{\text{water,in}} \left( s_{\text{water,out}} - s_{\text{water,in}} \right) \]  
(11)

Both of the above equations are used in the evaluation of the exergy analysis and estimation of their exergy destruction values are approximately the same. The exergy destruction of the air in the cooling tower is based on the height of the cooling tower with the known value of the thermodynamic properties at each level.

- Throttling valve

In the vapour compression refrigeration simulated by HGBP, two throttle refrigeration valves have been used one for the normal cycle which is placed after the condensing process and the other one is used to bypass the most amount of steam and back it to the mixer before send it to the compressor. The exergy loss within the throttle valve is as follows.

\[ 0 = D \dot{S}_{\text{irr}} - \dot{m}_i \left( s_{\text{out}} - s_{\text{in}} \right) \]  
(12)

- Mixer, flow split and orifice

For these parts only the energy analysis have been used and based on the equations used by [R. E. Sonntag, C.et. al 2009]. In the future work, the exergy analysis for these components will be used.

3- Results and Discussion

The experimental compressor map [K. Wark. 2002] has been used to characterize the compressor for the chiller system simulated by hot gas by pass cycle (HGBP). The steady state one dimensional analysis has been applied to the compressor unit. The modified test block simulated by HGBP cycle used to analyze the thermodynamic performance of the system. The engineering equation solver program (EES) used to carry out the energy and exergy balances of the system. Figure 2 shows the experimental compressor map between head and flow coefficients which are represents as \( \theta \) and \( \Omega \) respectively. It means that the theoretical investigation to evaluate the performance of the proposed method for the chiller system based on the experimental compressor map that has been used as an indicator to evaluate the rest of the properties of the whole system.
First of all, the thermodynamic performance of the system with different refrigerants such as R134a, R152a and R423a has been analyzed based on the characteristic of the compressor. The coefficient of performance, the work transfer to the gas within the compressor and the exergetic efficiency are presented as a function of the different pressure ratio of the compressor. Figure 3 presents the relationship between the coefficient of performance and the pressure ratio. It is obvious that the Refrigerant, R423a, has a higher coefficient of performance as compared to the R134a and R152a at different pressure ratio of the compressor. The reason is clear and because of the refrigerant R432a requires a less power input than the rest of the refrigerants to circulate the gas through the cycle.

![Figure 3: Coefficient of performance vs pressure ratio of the compressor](image)

The work transfer to the gas as a function of the compressor pressure ratio plotted in figure 4. The figure explains that the work transfer to the refrigerant R134a is higher than that both of refrigerants R423a and R152a respectively. This is due to a high value of exergy destruction occurred within the compressor for the gas as compared to the R152a and R423a.

![Figure 4: The work transfer to the gas vs pressure ratio of the compressor](image)

The variation in COP as a function of condensing temperature is shown in figure 5. It is observed that the COP of the refrigerants decreases as the condensing temperature increases. This is due to decrease in refrigerating capacity as the condenser temperature increases. The R423 has a higher COP than that with R134a and R152a. The COP of R134a is less than that of R423a and higher than that of R152a at lower value of condensing temperature and approximately approaches to each other at the higher value of condensing temperatures. It is obvious that the refrigerant R432a requires a less energy rate input in the form of the power to circulate the gas through the compressor. As a result of that, the coefficient of performance of the refrigerant R134a is less than that of the R432a and higher that of R152a at lower value of condensing temperature.
The effect of pressure ratio on the exergetic efficiency is plotted in figure 6. The refrigerant R152a has the highest value of exergetic efficiency as compared to R134a and R423a. However, this gas has a less value of energy efficiency.

The effect of relative humidity and dead state temperature has a similar effect on the coefficient of performance which are plotted in figure 7 and 8. With increasing dead state temperature and relative humidity the coefficient of performance decreases as well. However, in both cases the R423a has the highest value of coefficient of performance.

From figures 11, 12, 13, and 14, it may be inferred that maximum exergy destruction occurred in compressor and is followed by condenser, cooling tower and expansion valve for R134a and R152a. However, for the R423a the maximum exergy destruction occurred in compressor and is followed by cooling tower,
condenser and expansion valve. It is obvious that R423a has the lowest exergy destruction for the compressor when it is compared to R134a and R152a. The exergy destruction of the first valve which is located between the compressor and the mixer for the refrigerant R423 is high at low dead state temperature and is low at high dead state temperature when you compared to the other refrigerants such as R134a and R152a respectively as shown in figure 14.

Figure 10: Effect of degree of superheating on exergetic efficiency

Figure 11: Effect of dead state temperature on compressor exergy destruction

Figure 12: Effect of dead state temperature on condenser exergy destruction

Figure 13: Effect of dead state temperature on cooling tower exergy destruction

Figure 14: Effect of dead state temperature on extension valve exergy destruction

Conclusion

• The modified model simulated by hot gas by pass cycle, based on the experimental compressor map, has been used to evaluate theoretical investigation of the thermodynamic performance of the system. The engineering equation solver program (EES) used to carry out the energy and exergy balances of the system. The use of experimental compressor map data can help the designer to accurately select the compressor and investigate the thermodynamic performance of the vapour compression refrigeration cycle.

• The condenser temperature has a strong effect on the exergy losses in the compressor, cooling tower, condenser, and on the second law efficiency and COP of the cycle but little effects on the other components of the exergy losses. The total lowest irreversibilities are obtained by using R423a as a refrigerant in the refrigeration system.
for the increasing value of the condenser temperature while the highest total irreversibilities are obtained by using R152a as refrigerant in the cycle. Then the second lowest ones are obtained by using R134a as a refrigerant in the single stage vapor compression refrigeration system simulated by HGBP cycle.

- The increase in dead state temperature and relative humidity has a negative effect on COP, i.e. the COP goes down with increase in dead state temperature and relative humidity, respectively. R423a shows the higher COP than R134a and R152a. The COP and exergetic efficiency of both R134a and R152a are increased by degree of sub-cooling temperature of the refrigerant. However, reverse happens with R423a when the temperature of degree of sub-cooling is increased. The developed simulation model helped to establish the strong dependence between system performances and the compressor map.

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