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Performance analysis of a vapour compression refrigeration system with a diffuser for theeco-friendly refrigerants R-134a, R-600a and R-152a

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ABSTRACT: A refrigeration system was developed which combines a simplevapour compression refrigeration with a diffuser. Main purpose of this combination is to reduce compressor work and to improve the coefficient of performance. It was found through the literature, that vapor leaving the compressor with high velocity will cause damage to the condenser by erosion and also the liquid sub cooling will be inefficient. So, the vapour refrigerant leaving the compressor is allowed to pass through a diffuser to convert the kinetic energy of refrigerant to pressure energy. Thus the compressor work can be reduced and the condenser can function more effectively. In this paper we are analyzing the cycle by taking three different refrigerants, R-134a (Tetrafluoroethane), R-600a (Isobutane) and R-152a(Difluoroethane) separately. Further, energy performance characteristic is found out for above refrigerants.

I. Introduction

Refrigeration is widely used in domestic applications such as preservation of food, ice making etc. It is a process of removing excess heat from a space by providing mechanical work. Many research works are going on this specific area for finding better system which consumes less power and produce better performance. The improved systems reduce the effect of global warmingas well as energy consumption. It is found that the performance of a vapor compression system can be improved by increasing the evaporator temperature and by decreasing the condenser temperature. But these values are dependent on atmospheric conditions [1].Several modifications are suggested by several investigators to the simple vapor compression cycle to enhance the co-efficient of performance.

Yari et al. [2] developed a new configuration of the ejector-vapour compression refrigeration cycle, whichuses an internal heat exchanger and intercooler to enhance the performance of the cycle. Obtained results showed that there was an increase of 8.6% and 8.15% in coefficient of performance and second law efficiency values respectively of the new ejector-vapour compression refrigeration cycle as compared to the conventional ejectorvapour compression refrigeration cycle with R125. It was also found that there was increase of 21% in the coefficient of performance of the new ejector-vapour compression cycle compare to the conventional vapour compression system.

Yinhai Zhu et al [1],developed a hybrid vapor compression system with an integrated ejector cooling cycle. The ejector cooling cycle is driven by the waste heat from the condenser in the vapor compression refrigeration cycle. The additional cooling capacity from the ejector cycle is directly input into the evaporator of the vapor compression refrigeration cycle. The results show that the COP is improved by 9.1% for R22 system.

Moo-Yeon lee et. al. [3] have studied the performance of a small capacity directly cooled refrigerator by using the alternative to R-134a.The compressor displacement volume of the alternative system with R600a/R290 (45/55) has modified from that of the original system with R-134a to the optimized R600a/R290 system was approximately 50% of that of the optimized R-134a system. The capillary tube lengths for each evaporator in the optimized R600a/R290 system were 500mm longer than those in the optimized R-134a system. The power consumption of the optimized R-134a system was 12.3% higher than that of the optimized R600a/R290 system. The cooling speed of the optimized R600a/R290 (45/55) system at evaporator 0^{o}C was improved temperature by 28.8% over that of the optimized R-134a system.

NeerajaUpadhyay*et. al.* [4] developed a new configuration of the diffuser and sub-cooling in vapour compression refrigeration cycle, for sub-cooling, a fan is used. By using diffuser, power consumption is less for same refrigeration effect, so performance is improved. The size of the condenser can be reduced, as the heat transfer is considerably increased due todiffuser application. So cost of the condenser will be reduced. The parameters like pressure and temperature were measured. After result analysis, it is found that the COP was enhanced from2.65 to 3.38 in the case when conventional vapour compression refrigeration system was used with diffuser.

In the present work, an isentropic diffuser is taken for the analysis which is able to produce 0.5bar pressure rise if the refrigerant leaving the compressor with sonic flow. Here the compressor, which increases the pressure of a low pressure vapour refrigerant, will consume less work than in the base cycle. So for getting same refrigeration effect as in the base cycle, less compressor work is needed. Diffuser is a device which converts kinetic energy of the fluid in to pressure energy. After leaving the diffuser the refrigerant enters the condenser with moderate velocity. This will prevent the splashing of liquid and therefore condenser size will be optimum, for producing sub-cooling[5]. Here we assume 5degree of sub-cooling and 5 degree of super heating at condenser outlet andcompressor inlet respectively. A heat exchanger is provided in between the condenser and evaporator for producing the same. Here, we are considering three different refrigerants (R-134a, R-600a & R152a) which does not contain chlorine atom, and an analysis is done for finding the rate of increase of co-efficient of performance with respect to the base cycle. The evaporator temperature varies from -16^{0}C to -6^{0}C and the condenser temperature is kept as 30^{0}C, constantin both cases (with and without diffuser). For each value of temperature, COP is calculated separately and a theoretical comparison is done for finding best refrigerant for the particular design.

II. System Description

In this, basic system is a simple vapour compression refrigeration system with a heat exchanger, coupled between condenser and evaporator to improve the refrigeration effect. The new system has a diffuser attached before the condenser as shown below

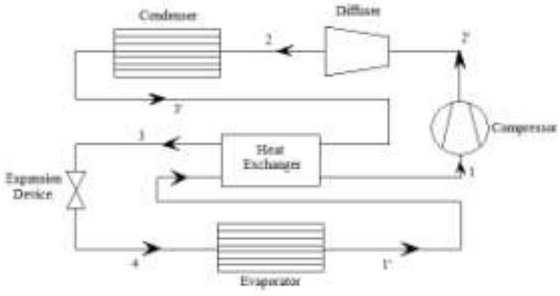

Fig. 1. Schematic diagram of vapor compression system with diffuser and heat exchanger

The vapour refrigerant leaving the evaporator (1^I) is assumed to be dry saturated, which absorbs sensible heat from the heat exchanger and the vapour becomes super-heated. Here the heat exchanger is designed so as to give a 5 degree of super heat i.e.,$T_{1}{}_{-}T_{1^I}$= 5°C. This super-heatedvapour enters the compressor, which increase the pressure from stage 1 to 2(if diffuser is not used) and from 1 to 2^I(if diffuser is used). The compression process is isentropic and the work done by the compressor to raise the pressure equals to the enthalpy difference between both stages.

$W_{compressor}$ = $h_{2}{}_{-}h_1$ (without diffuser)
$W_{compressor}$ = $h_{2^I}{}_{-}h_1$ (With diffuser)

Using the diffuser, the compressor work is reduced and the high pressure vapour refrigerant enters the condenser. Here therefrigerant rejects its latent heat of vaporization and attains liquid state(2 -3^I). The excess heat in the refrigerant is again rejected in the heat exchanger and the liquid get sub cooled. Here also T_3-T_{3^I} = 5°C

This sub cooled liquid is expanded through an expansion device to a lower pressure state 4 is an isenthalpic process (3 - 4). After the expansion, refrigerantabsorbs excess heat from the evaporator to becomevapour at the end. This is assumed to be a constant pressure process.
Refrigeration effect produced = h_1-h_4.
Corresponding Pressure- Enthalpy diagram has been shown in Figure 2.

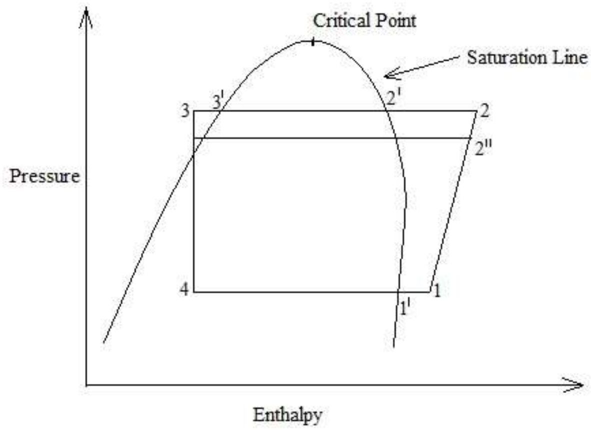

Fig. 2. Pressure - Enthalpy diagram for vapour compression refrigeration system with diffuser and heat exchanger

P-h diagram gives enthalpy at various points and the difference in enthalpy gives the amount of energy absorbed or rejected.
In the P-h diagram, path 1-2-3-4 is the base cycle without diffuser.
$h_{1^I}$is the enthalpy of saturated vapor refrigerant corresponding value of P_1
h_1 = h_{1^I} + $C_{pv}(T_{1}{}_{-}T_{1^I})$ (1)
Process 1-2 is an isentropic process so that S_1=S_2 (2)
S_1= S_{1^I} + C_{pv} ln $(T_{1}{}_{-}T_{1^I})$ (3)
S_2= S_{2^I}+ C_{pv} ln $(T_{2}{}_{-}T_{2^I})$ (4)
Here the only unknown value is T_2 which can be calculated from '2'
S_{2^I}andT_{2^I} are corresponding saturated values which will find out from the table.
By calculatingT_2, corresponding value of enthalpy can be calculated
h_2=h_{2^I} +C_{pv} $(T_{2}{}_{-}T_{2^I})$ (5)
h_3= h_{3^I} - $C_{pl}(T_{3^I}$-$T_3)$ (6)
h_{3^I}can be calculated from table and T_{2^I} = T_{3^I}
From process 3-4 enthalpy at the exit of the expansion device will be equal as the enthalpy at the exit of the heat exchanger (h_3=h_4)
So the refrigeration effect produced = h_1- h_4 (7)
Work done on the compressor = $h_{2}{}_{-}h_1$ (8)
Coefficient of performance of base cycle = (h_1- h_4)/($h_{2}{}_{-}h_1$) (9)

For same refrigerant, if diffuser is attached at the exit of compressor, the compressor work is changed from (h_2–h_1) to ($h_{2^{II}}$-h_1). Even though the vapour refrigerant exit from the compressor at the state 2^{II} and the remaining pressure rise is occur inside the diffuser. From the assumption the pressure rise in the diffuser is 0.5bar and it reject the refrigerant at the state 2 that of a base cycle. The evaporator temperature and condenser temperature are kept same and for same refrigerant the coefficient of performance is find out.
Coefficient of performance of modified cycle

$$COP = (h_1 - h_4)/(h_{2^{II}} - h_1) \quad (10)$$

Percentage increase in Coefficient of performance

$$\{(h_1 - h_4)/(h_{2^{II}} - h_1) - (h_1 - h_4)/(h_2 - h_1)\}/(h_1 - h_4)/(h_2 - h_1)\} * 100 \quad (11)$$

Using this relation, percentage of increase of coefficient of performance of each refrigerant at different evaporator temperature is plotted.

III. Results and Discussions

The present study used R-134a, R-600a and R-152a as the working fluids. The thermodynamic properties of the refrigerants were calculated from [6].
Figure 3 shows variation of coefficient of performance of each refrigerants with respect to the evaporator temperature. The condenser temperature is fixed as 30°C.

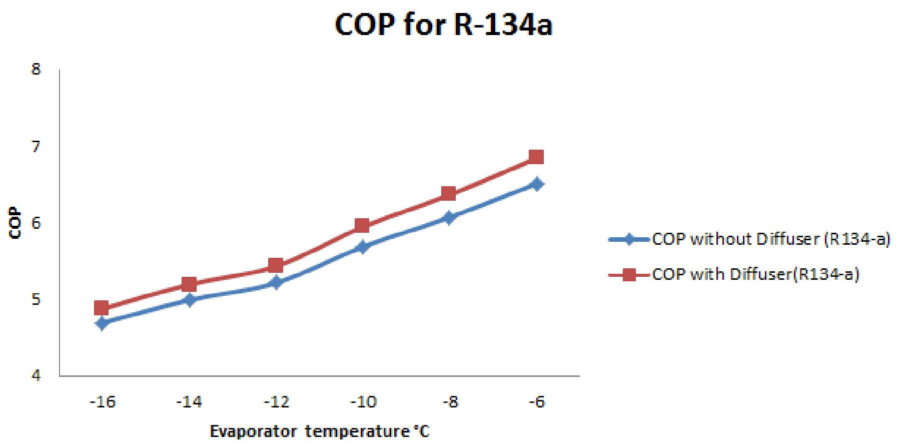

Figure 3 Coefficient of performance Vs Evaporator temperature of R-134a

From the graph we can observe the variation of COP at each evaporator temperature. It is clear that the COP at each evaporator temperature is more in the cycle which is having a diffuser. With increase in evaporator temperature, the COP of the cycle increases in both cases. It shows a linear relationship between the COP with and without diffuser up to the temperature -8°C and then there is a slight increase in difference in COP with both the systems. Maximum COP occurs at -6°Cof evaporator temperature which is 6.84 (with diffuser).
Figure 4 shows the same graph for R-152a as refrigerant. The operating conditions are same as that of previous one.

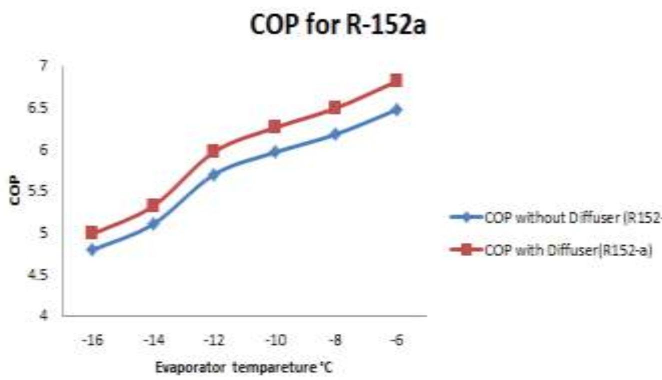

Figure 4 Coefficient of performance Vs Evaporator temperature of R-152a

When condenser temperature is constant,with the increase in evaporator temperature COP of the system also increases.For this refrigerant, the value of COP is almost same as that for R-134a. But the graph shows a more deviation as evaporation temperature increases. That means the rate of increase of COP in this case is more than that of R-134a. The maximum value of COP obtained is 6.811(with diffuser) when evaporator temperature is -6℃.

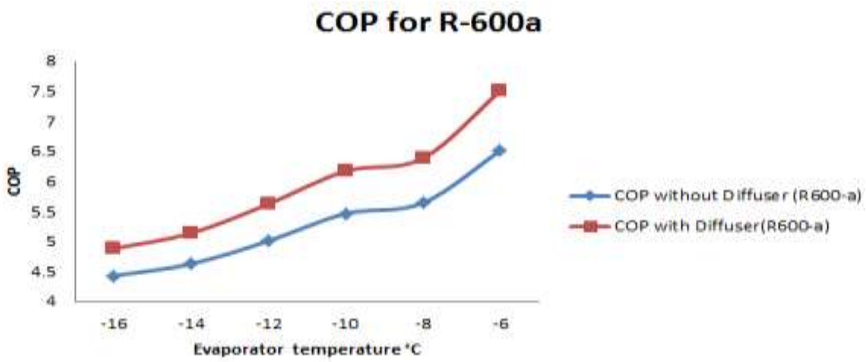

Figure 5 Coefficient of performance Vs Evaporator temperature of R-600a

For R-600a the figure shows a larger deviation compares to the previous graphs for R-134a and R-152a. Here also with an increase inevaporator temperature, the COP of the system increases.

On comparison of all the three refrigerants with refrigeration system having diffuser, R-600a gives more COP than the other two i.e., a COP of 7.5 at -6℃ evaporator temperature. At higher evaporator temperatures, the difference in COP is higher, so the percentage increase in COP will be higher for R-600a refrigerant compared to others, which is shown in Figure 6.

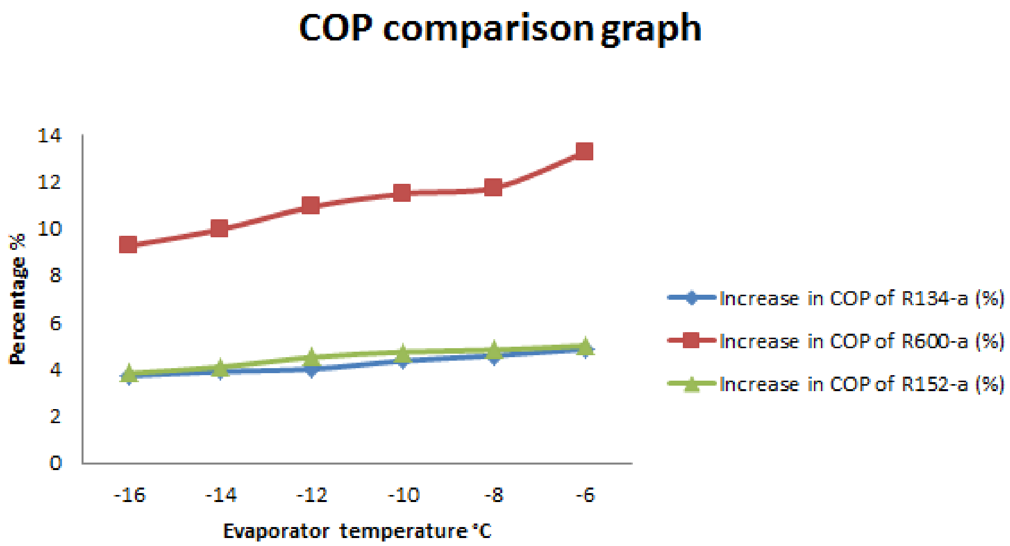

Figure 6 percentage increases in COP Vs. Evaporator temperature

This clearly shows a large increment in COP for R-600a refrigerant when a diffuser is mounted after the compressor. At each evaporator temperature, the increase in COP of R-600a is much higher than the other two refrigerants. It is found that if we use R-600a as refrigerant the COP can rise up to 13.25% than the base cycle. Even though other two refrigerantsR-134a and R-152a are performing well, they improve COP only upto 4.87, 5.01 respectively.

IV. Conclusion

COP of vapour compression refrigeration can be improved by either increasing the refrigeration effect or by decreasing the compressor work. Providing a heat exchanger, in between the evaporator and condenser will increase the refrigeration effect. The result shows that compressor work can be reduced by the application a diffuser in between compressor and condenser. A diffuser will convert the kinetic energy of the refrigerant to pressure energy without consuming any work input. This will also prevent condenser - erosion and the sub cooling occur effectively. Results obtained shows that, among the three different eco-friendly refrigerants, R-600a is the better performing one. It improves the COP by 13.25% that of the base system. This may be because of the saturation temperature of the refrigerant at the state of diffuser entry is lower than that of other two refrigerants. So the compressor work, which is enthalpy change, will be lower. Thus for same design conditions R-600a is more preferred because of its properties.

Nomenclature

T	Temperature ℃
Subscripts	
1	*Properties at the exit of evaporator (saturated)*
1^I	*Properties at the exit of heat exchanger(super-heated)*
2^{II}	*Properties at the exit of compressor*
2	*Properties at the exit of diffuser*
2^I	*Properties corresponding to the condenser temperature (saturated)*
3^I	*Properties at the exit of condenser (saturated)*
3	*properties at the exit of heat exchanger (Subcooled)*
4	*properties at the exit of expansion device*
h	*Enthalpy in kJ/Kg*
S	*entropy in kJ/kgK*
c_{pv}	*Specific heat of vapour refrigerant in kJ/kgK*
c_{pl}	*Specific heat of liquid refrigerant in kJ/kgK*
COP	*Coefficient of performance*

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