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Performance Improvement of Vapour Compression Refrigeration System Using Diffuser

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Abstract:
A modified Vapour Compression Refrigeration (VCR) System using diffuser was built to study the increase in coefficient of performance (COP) of the system. R600a (Isobutane) was taken as the alternative ecofriendly refrigerant and a comparative analysis with R134a (Tetrafluoroethane) was done. The diffuser was added to increase the COP by reducing the compressor work. The analysis were done for various diffusers and time intervals. The results showed that the modified VCR system had an increase in COP between 5% - 28%. Further it reduced the compressor power input by 13% - 17%. R600a showed better performance in both normal and modified VCR system.

Keywords: *VCR system, Diffuser, Coefficient of performance, Isobutane, Tetrafluoroethane.*

1. Introduction

The vapor compression refrigeration system uses a circulating liquid refrigerant as the medium which absorbs and removes heat from the space to be cooled and subsequently rejects that heat elsewhere. When the refrigerant leaves the compressor, it has comparatively high velocity and kinetic energy. This high velocity of fluid can cause splashing of liquid of refrigerant inside the condenser, and develop liquid hump. The presence of this liquid hump in the condenser will lead to decrease of surface area of heat transfer. The liquid hump also causes the insufficient supply of refrigerant to the evaporator a phenomenon called *starvation* which reduces the refrigeration effect. These effects can be overcome by using a diffuser at the condenser inlet [1-3]. It will increase the pressure and decrease the velocity of the refrigerant at condenser inlet thereby preventing liquid hump formation and hence prevents its effects. The diffuser also increases the performance of the system [4,5] and decreases the power input to the system.

R600a was chosen because of its ecofriendly characteristics of zero Ozone Depletion Potential (ODP) and negligible Global Warming Potential (GWP). It is an alternative for R134a which has huge GWP value. Various experiments proved that R600a shows better performance than R134a [6-8].

A diffuser is a simple static device that is used to raise the pressure of the fluid passing through it and reduce its kinetic energy. By Steady flow equation it can be proved that diffuser increases the pressure of the working fluid passing through it. It also shows that at larger cross section area, the velocity for steady flow of refrigerant decreases.

The diffuser smoothly decelerates the incoming refrigerant flow achieving minimum stagnation pressure losses and maximizes static pressure recovery. Due to pressure recovery, for the same refrigerating effect the compressor has to do less work. Hence, power consumption of the compressor will be reduced which results in improving the performance of the system.

As the refrigerant passes through the diffuser, pressure as well as temperature will increases. In air cooled condenser for constant air temperature, temperature difference between hot and cold fluid will be increased. So the amount of heat rejected from condenser will increase. To remove the same amount of heat less heat transfer area will be required will providing an opportunity to use a smaller condenser to achieve the same performance [5]. The diffuser also reduces the effect of starvation in vapor compression refrigeration systems.

1.1. Diffuser Design

The design parameters of a diffuser is shown in Figure 1.

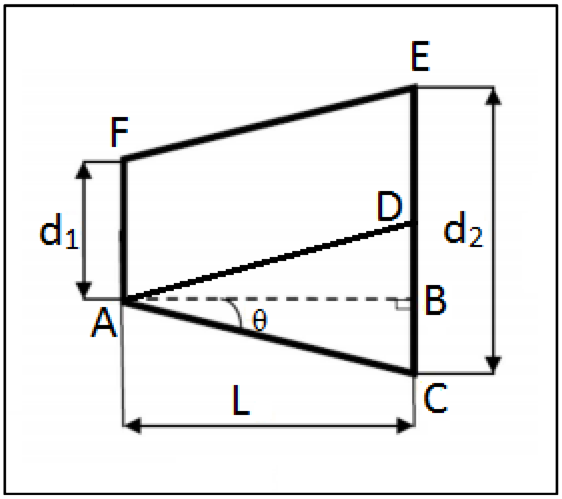

Figure 1: Line diagram of a diffuser

d_1 and d_2 correspond to inlet and outlet diameter respectively,
L =*'perpendicular'* length between the two areas of the diffuser,
θ = half angle of the diffuser
Given AB = L,

- ⇨ DC = d_2 – ED
- ⇨ DC = d_2 – d_1(Since AD ǁ FE, DE = d_1)
- ⇨ Since it is a *'symmetric'* diffuser,
- ⇨ BC = DB = $\frac{1}{2}$DC = $\frac{1}{2}(d_2 - d_1)$
- ⇨ $\tan\theta = \left(\frac{BC}{AB}\right) = \frac{1}{2}\left(\frac{d2-d1}{L}\right)$
- ⇨ $L = \frac{1}{2}\left(\frac{d2-d1}{\tan\theta}\right)$

This is the required relation between length (generally called as axial length) of the diffuser, inlet and outlet diameters of the diffuser and the angle of the diffuser

1.2. Steady Flow Energy Equation (S.F.E.E.) For Diffuser
In steady flow, the essential physical condition required is that mass flow rate and the energy flow rate should be constant. The given SFEE for diffuser is given below:

- ⇨ $\sum\dot{E}_1 = \sum\dot{E}_2$
- ⇨ $\dot{m}_1(e_1) + Q = \dot{m}_2(e_2) + W$

Since no work is done and no heat is transferred during the process of diffusion, Q = 0 and W = 0. (Isentropic process)
Now

- ⇨ $\dot{m}_1(e_1) = \dot{m}_2(e_2)$
- ⇨ $\dot{m}_1(h_1 + \frac{1}{2}C_1^2 + gz_1) = \dot{m}_2(h_2 + \frac{1}{2}C_2^2 + gz_2)$

For negligible loss in gravitational potential energy,
i.e.: $gz_1 - gz_2 \cong 0$, (Since the diffuser is in horizontal orientation)

- ⇨ $\dot{m}_1(h_1 + \frac{1}{2}C_1^2) = \dot{m}_2(h_2 + \frac{1}{2}C_2^2)$

Since it is a steady flow, therefore $\dot{m}_1 = \dot{m}_2$,
Now:

- ⇨ $h_1 + \frac{1}{2}C_1^2 = h_2 + \frac{1}{2}C_2^2$

For isentropic flow, $h = \frac{P}{\rho}$
$\left(\frac{P}{\rho}\right)_1 + \frac{1}{2}C_1^2 = \left(\frac{P}{\rho}\right)_2 + \frac{1}{2}C_2^2$.
Mass flow rate is given by:

- ⇨ $\dot{m} = \frac{d}{dt}(\rho.A.l) = \rho.A.C$
- ⇨ $\rho_1 A_1 C_1 = \rho_2 A_2 C_2$
- ⇨ $\rho_1 \cong \rho_2$, & $A_1 < A_2$, Therefore, $C_1 > C_2$.

Hence we can see the *velocity (or kinetic energy of the fluid) decreases* with increase in area, thus leading to *increase in pressure of the working fluid.*
[Note: In case of our study, the superheated refrigerant is *'subsonic'* as applied to the above calculations]

1.3. R134A and R600A

R134a is used as an alternative for R12 and R22 whereas R600a is used as an alternative for R134a itself. R134a is a long term environment friendly refrigerant that can be a possible replacement for many refrigerants due to its characteristics of zero ODP (Ozone depletion Potential), non-flammability, stability, and chemical inertness. Yet it imposes a threat because it has relatively high (GWP) global warming potential. Many studies are being carried out which are concentrating on the application of environmentally friendly refrigerants in refrigeration system. R600a comes as a boon in such cases having a zero ODP and also negligible GWP. Table 1 shows the comparison of refrigerants properties of R134a and R600a.

Refrigerant	R600a	R134a
Name	Isobutane	1,1,1,2-Tetrafluoroethane
Formula	$(CH_3)_3CH$	CF_3-CH_2F
Critical Temperature (in °C)	135	101
Molecular Weight (in kg/mole)	58.1	102
Normal Boiling Point (in °C)	-11.6	-26.5
Pressure at -25°C in bar (absolute)	0.58	1.07
Liquid Density at -25°C in kg/l	0.60	1.37
Vapour Density at -25°C in kg/m^3	1.3	4.4
Volumetric Capacity at -25°C in kJ/m^3	373	658
Enthalpy of Vaporization at -25°C in kJ/kg	376	216
Pressure at +20°C in bar (absolute)	3.0	5.7
Ozone Depletion Potential (ODP)	0	0
Global Warming Potential (GWP)	3	1430
ASHRAE Safety Group (2013)	A3	A1
ASHRAE Flammability	Yes (Highly flammable)	No
ASHRAE Toxicity	No	No
Replacement for	R134a	R12, R22

Table 1: Comparison of Refrigerant properties

2. Methodology

The system consists of two flow lines one is simple VCRS flow line without diffuser and other is modified flow line with diffuser. Two pressure gauges are installed *at diffuser outlet* and *at simple flow line* to measure the pressure of the refrigerant at diffuser outlet and pressure in simple VCRS flow line as shown in Figure 2. Thus we can calculate the pressure with and without diffuser. A constant refrigeration effect is maintained throughout the experiment. R134a and R600a were used as refrigerants.

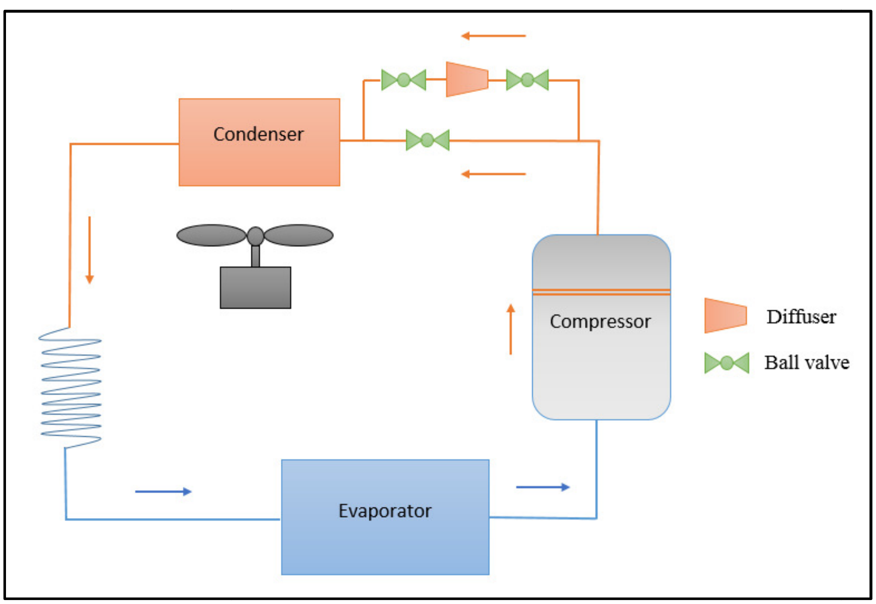

Figure 2: Schematic of vapour compression refrigeration system with diffuser at condenser inlet

The pH diagram of normal and modified VCR system is shown in Figure 3.

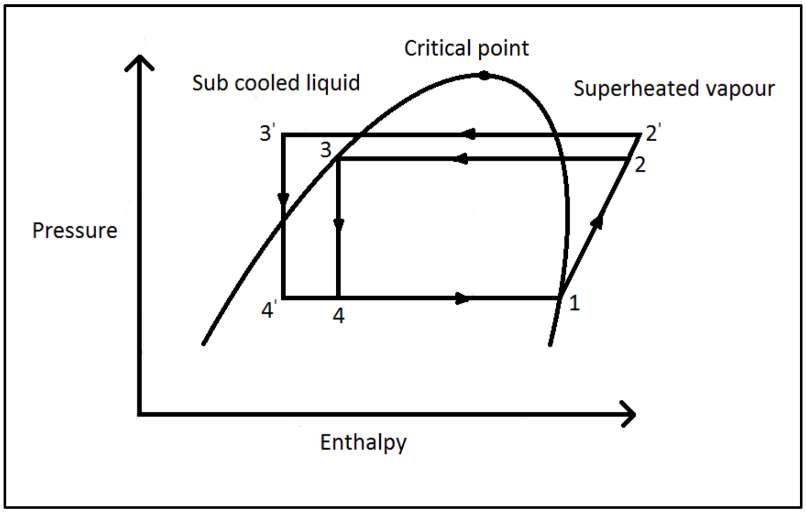

Figure 3: P-h Chart with and without diffuser

Given points 1, 2, 3, 4 are the points of significance of our study.
Point 1 corresponds to compressor inlet,
Point 2 is for compressor outlet (or diffuser inlet),
Point 2' (refer P-h graph shown below) is diffuser outlet in case the diffuser line valve is opened, otherwise 2 is considered as condenser inlet for condenser.
Point 3, (3') corresponds to condenser outlet or throttling valve inlet, (with diffuser)
Point 4, (4') corresponds to evaporator inlet, (with diffuser)

COP = $\frac{Netrefrigeration\ Effect}{Workdone}$

⇨ $\left(\frac{h1 - h4}{h2 - h1}\right)$

COP (when diffuser is not used) = $\frac{Net\ refrigeration\ Effect}{Workdone}$

⇨ $\left(\frac{h1 - h4}{h2' - h1}\right)$

After the use of diffuser, the pressure will decrease from point 2' to 2.
So the COP will become:

COP (when diffuser is used) = $\frac{h1 - h4}{h2 - h1}$

So the COP will increase by the factor:

⇨ $\left(\frac{h1 - h4}{h2 - h1}\right) - \left(\frac{h1 - h4}{h2' - h1}\right)$

3. Experimentation

Figure 4: Experimental Setup

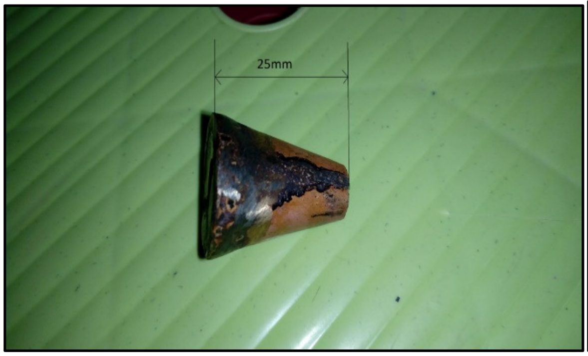

Figure 5: Diffuser 1

Figure 6: Diffuser 2

All the necessary components were bought and the refrigeration setup was created for the purpose of experiment as shown in Figure 4. Pressure gauges, Temperature indicator and Energy meter were added along with the setup for noting the readings. Two diffusers of different dimensions as shown in Figure 5 and 6 were fabricated with dimension shown in Table 2.

Parameters	Diffuser 1	Diffuser 2
Inlet Diameter	10mm	10mm
Outlet Diameter	25mm	35mm
Length of Diffuser	25mm	40.88mm
Angle of Diffuser	17^0	17^0

Table 2: Diffuser dimensions

The experiment was done in two stages. First the experiment was carried out using R134a. The setup was allowed to run for 45 minutes and the readings were taken for every 15 minutes' intervals. The readings were taken for both normal flow line and with diffuser separately. In the next stage the R134a was replaced by R600 and the readings were taken in a similar manner. Further the Diffuser 1 was replaced by Diffuser 2 and another set of readings were taken for R600a.
Finally, there were five readings for both R134a and R600a using normal flow line, Diffuser 1, Diffuser 2. The calculation for these readings were done as shown below.

4. Calculation

4.1. Actual COP Calculation

COP = $\frac{Heat\ Absorbed}{Work\ Done} = \frac{q}{W}$

Q = $\frac{mc_p\Delta T}{t}$

$\Delta T = T_1 - T_0$

$W = \left(\frac{1}{E_c}\right) \times \left(\frac{n}{t}\right)$

4.2. Theoretical COP Calculation

$h_1 = h_g$ at T_3

$s_1 = s_g$ at T_3

For isentropic process,

$s_1 = s_2$

$s_1 = s_2 = s_{g2} + c_{pg}\ln\left(\frac{T_{sup}}{T_{sat}}\right)$

$h_2 = h_{g2} + c_{pg2}\,(T_{sup} - T_{sat})$

$h_3 = h_f$ at T_3

Refrigeration Effect = $h_1 - h_4$

Work done = $h_2 - h_4$

⇨ COP = $\ln\left(\frac{Refrigeration\ Effect}{Work\ done}\right)$

Refrigeration Capacity = $\dot{m}_r \times$ Refrigeration Effect

Power consumed = $\dot{m}_r \times$ Work done per unit mass

5. Results and Discussions

Similarly, the calculations were done and the following results were obtained.

COP		
	R134a	**R600a**
Without Diffuser	1.053	1.012
Diffuser 1	1.1143	1.1873
Diffuser 2	-	1.3003

Table 3: COP of R134a and R600a

Power input to the compressor in kW		
	R134a	**R600a**
Without Diffuser	0.265	0.1900
Diffuser 1	0.2686	0.16196
Diffuser 2	-	0.16696

Table 4: Power input to the compressor

From Table 3, we can infer that the COP is increased with the use of diffuser for both the refrigerants. Diffuser 1 gave a 5.82% increase in COP when using R134a whereas a 17.32% increase when R600a is used. Also for R600a Diffuser 2 increased the COP by 28.48% and an increase of 9.51% than the Diffuser 1. From Table 4, we can infer that the Work done by the compressor is reduced gradually when diffuser is used. There is a 17.31% work reduction when Diffuser 1 is used with R600a and 13.79% work is reduced when Diffuser 2 is used with the same.

From the Figure 7-13 it is observed that the COP of the system is high at the beginning of the cycle and decreases as the time progresses. From Figure 7 & 11, at 45 minutes the COP of the normal system is almost 1 for both R134a and R600a whereas for the modified system it is slightly higher. For normal VCR system the COP of both the refrigerants is higher at the beginning but converge towards the end of the process as shown in Figure 12. But for the modified system the COP of R134a was higher than R600a only in the beginning of the process while at the end the COP of R134a drops below that of R600a as shown in Figure 13. From Figure 15 it is observed that when R600a is used, the work done by the compressor in normal system is almost constant. Whereas for the modified systems it decreases sharply after 15 mins. From Figure 14for R134a the work done in normal system decreases gradually until 30 mins but then becomes almost constant while for the modified system it decreases throughout the process.

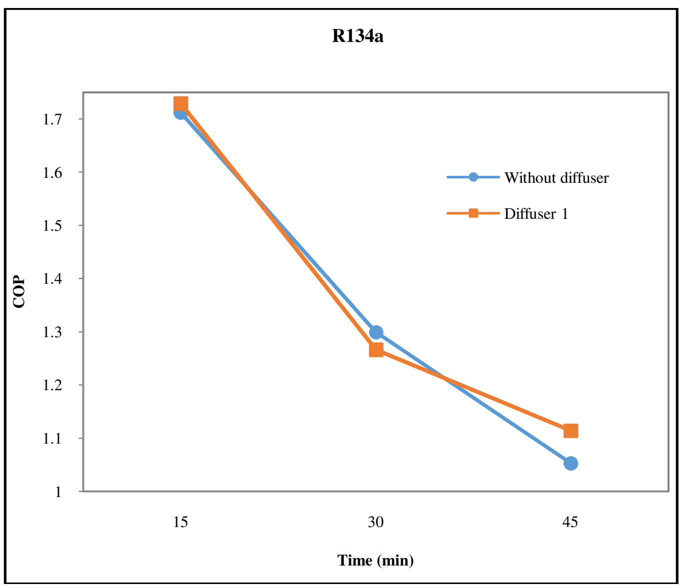

Figure 7: Performance characteristics of R134a with and without a diffuser

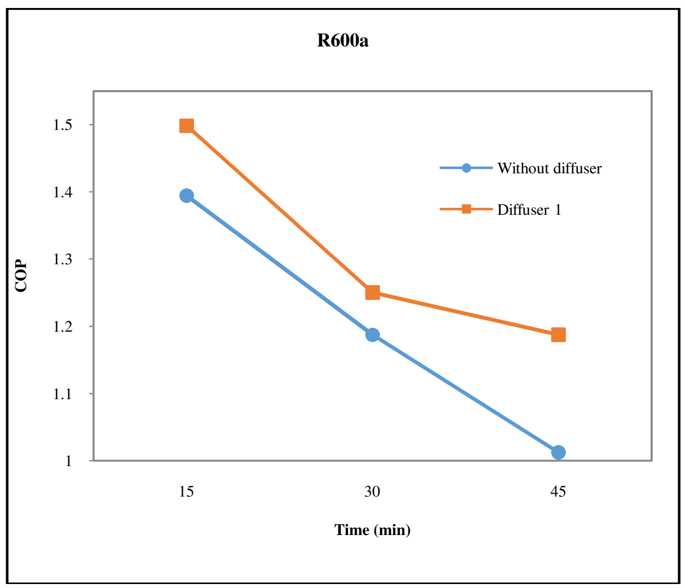

Figure 8: Performance characteristics of R600a without and with diffuser 1

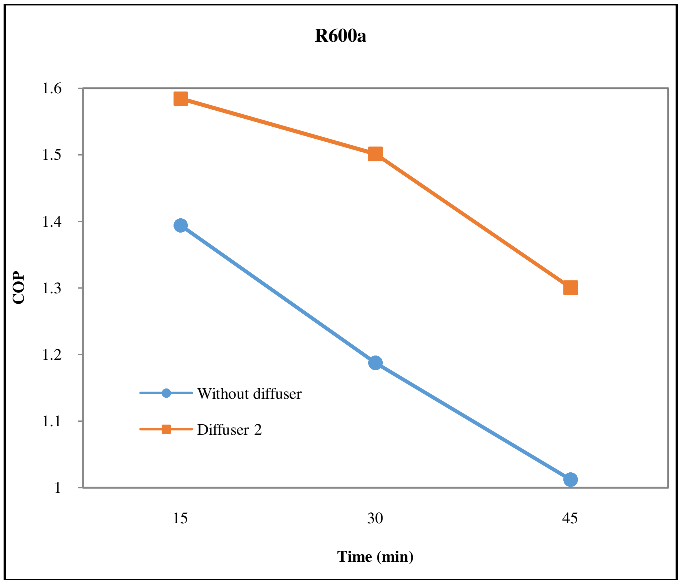

Figure 9: Performance characteristics of R600a without and with diffuser 2

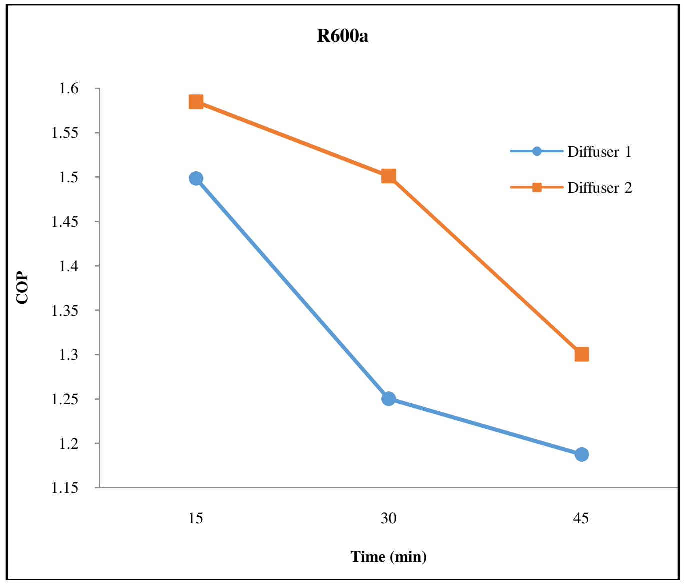

Figure 10: Performance characteristics of R600a with diffusers 1 & 2

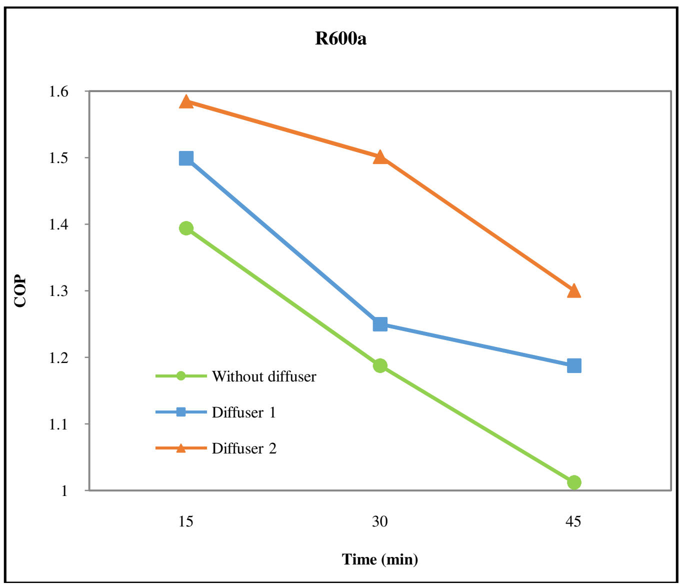

Figure 11: Performance characteristics of R600a without and with diffusers 1 & 2

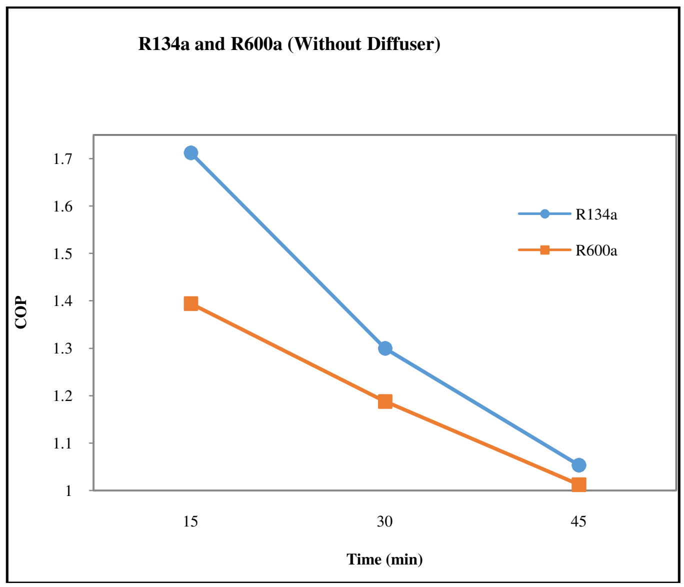

Figure 12: Performance Comparison of R134a and R600a without a diffuser

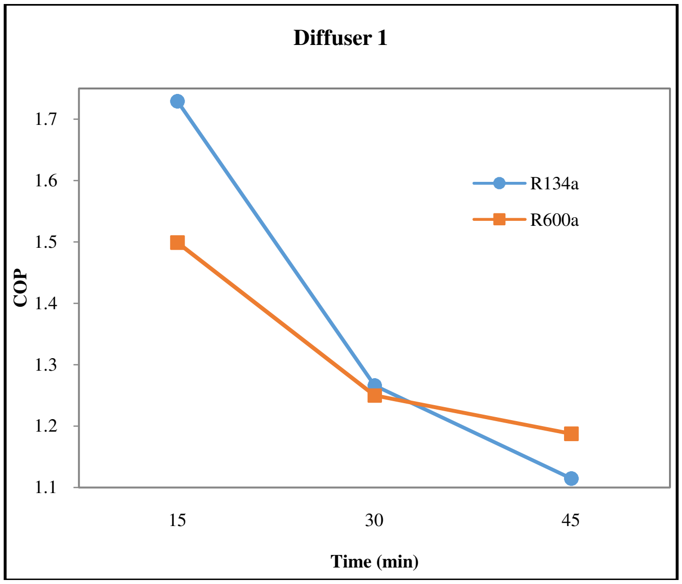

Figure 13: Performance comparison of R600a and R134a when a Diffuser 1 is used.

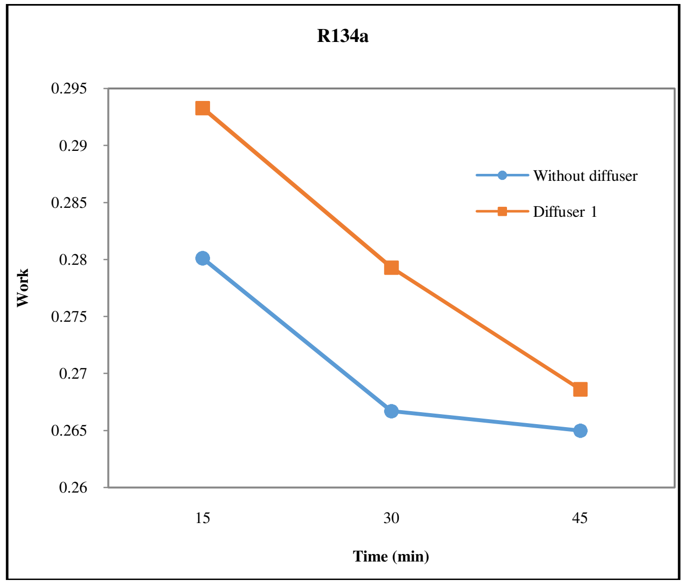

Figure 14: Work-Time graph of VCR system using R134a without and with diffuser 1

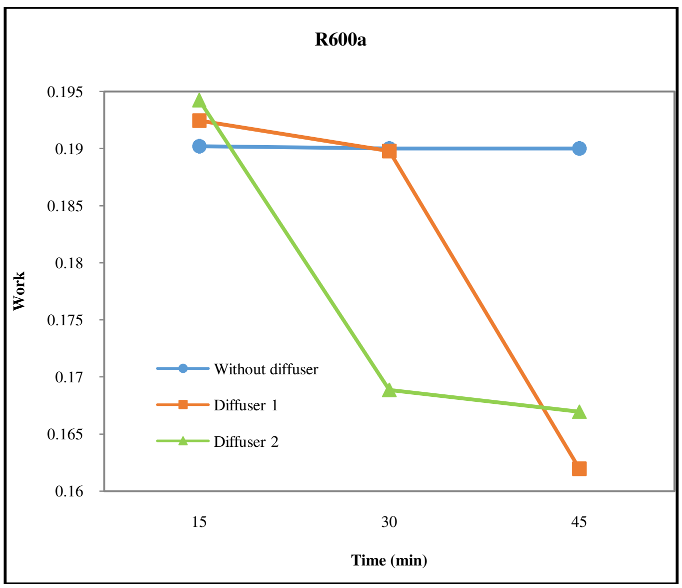

Figure 15: Work-Time graph of VCR system using R600a without and with diffusers

6. Conclusion

A modified VCR system was built using diffuser with the aim of increasing the COP of the system. R134a and R600a were used as refrigerants and two different diffusers were used for analysis. The results conclude the fact that the diffuser increases the performance of the VCR system and also reduces the compressor work. The COP of the refrigeration system increased considerably. The COP of the system increased by using the diffuser for both R134a and R600a. Diffuser 1 increased the COP by 5.82% for R134a and 17.32% for R600a. Also Diffuser 2 increased the COP of R600a by 28.48% which is 9.51% more than diffuser 1. It shows that with the increase in diffuser area ratio the COP increases. For R600a the power input to the compressor is reduced by 17.31 % using diffuser 1 and by 13.79 % using diffuser 2. Also it can be observed from the graphs that the COP and compressor power input decreased with the time for all the cases. Overall R600a showed better performance than R134a in all the cases. Hence it can be concluded that the use of diffuser increases the performance of VCR system, which increases with the increase in diffuser area. Also it is proved that R600a is a better refrigerant for the VCR system both with and without the diffuser.

7. Nomenclature

- → $\dot{E}_i$ - Rate of flow of energy at state i (kW)
- → e_i - Total specific energy at state i(kJ/kg)
- → $\dot{m}_i$ - Mass flow rate at state i (kg/s)
- → C_i - Velocity at state i (m/s)
- → ρ - Density of the fluid (kg/m^3)
- → COP - Coefficient of Performance
- → Q - Heat absorbed (kW)
- → W - Work done (kW)
- → m - Mass of water (kg)
- → c_p - Specific heat (kJ/kg)
- → ΔT - Difference between initial and final temperatures of water (^oC)
- → n - Number of energy meter revolution
- → E_c - Energy meter constant (imp/kWh)
- → P_i - Pressure at point i in the cycle (bar)
- → T_i - Temperatureat point i in the cycle(^oC)
- → h - Enthalpy (kJ/kgK)
- → s - Entropy(kJ/kgK)
- → T_{sup} - Superheated Temperature(^oC)
- → T_{sat} - Saturation Temperature(^oC)
- → $\dot{m}_r$ - Mass flow rate of refrigerant (kg/s)
- → P - Power consumed (kW)

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