Enhancement of COP of Vapor Compression Refrigeration Cycle using CFD

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Abstract: - This Abstract presents the study on Vapor Compression Refrigeration system using diffuser to improve Coefficient of Performance. To Improve the Coefficient of Performance, it is to require that Compressor work should decrease. The purpose of a compressor in vapor compression system is to elevate the pressure of the refrigerant (refrigerant used is TETRA FLURO ETHANE), but refrigerant leaves the compressor with comparatively high velocity which may cause splashing of liquid refrigerant in the condenser, liquid hump and damage to condenser by erosion. It is needed to convert this kinetic energy to pressure energy, for which diffuser can be used. The diffuser of increasing cross-sectional area profile was designed, fabricated and introduced between Compressor and Condenser. By doing so, the power input to the compressor is reduced, thereby enhancing COP. The size of diffuser selected was of 15 degree divergence angle. After result analysis, COP was enhanced from 3.83 to 5.55. With the addition of diffuser Coefficient of Performance (COP) enhancement has been increased by 31% when compared without diffuser. The Experimental results obtained are validated using CFD; Modeling and meshing will be done in ICEMCFD, analysis in CFX and post results in CFD POST.

Keywords: ICEMCFD, Diffuser, COP, CFX, R134A,

I. INTRODUCTION

In thermodynamics Refrigeration is the major application area, in which the heat is transferred from a lower temperature part to a higher temperature part. The devices which develop refrigeration are known as Refrigerators. The cycle on which it operates are known as refrigeration cycles. There are many types of Refrigeration like Vapor compression refrigeration it is the most commonly used refrigeration, cascade refrigeration and thermo electric refrigeration.

1.1. Refrigerators and Heat Pumps

It is known heat always flows from higher temperature medium to a lower temperature medium without any aid of devices heat transfer occurs itself in nature. Reverse process, will not happen by itself. Special devices where the heat transfers from a lower temperature medium to higher temperature medium are called Refrigerators.

The working fluids which are used in the cycles of refrigeration are called refrigerants, and the refrigerators are cyclic devices. The schematic representation of a Refrigerator is displayed refer Fig 1.1a Where Q_L magnitude of heat removed at temperature T_L, Q_H is magnitude of heat rejected at temperature T_H to the surrounding space, the refrigerator net work input is $W_{net, in}$ and Q_H and Q_L represents magnitudes and they are positive quantities.

Heat pump transfers heat from a lower temperature medium to a higher temperature medium. Heat pumps, Refrigerators are basically the same devices; but are dissimilar in aim. Main aim of refrigerator is to keep the refrigerated space at a very less temperature by extracting the heat generated. Essential part of the operation, not the intent is discharging the heat to a higher-temperature medium. Aim of the heat pump is maintaining the space that is heated at a high temperature. It's capable of riveting heat from small-temperature medium. (Fig1.1b)

Functioning of heat pumps, refrigerators is shown by the coefficient of performance (COP) -

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling process}}{\text{Work input}} = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{\text{Desired result}}{\text{Required result}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}}$$

Above relations can be shown by exchanging Q_h, Q_L $W_{net,in}$ by $\dot{Q}_H$, $\dot{Q}_l$, $\dot{W}_{net,in}$. Both COP_{HP}, COP_R might be more than one. The similarity of equations 1-1, 1-2 shows

$COP_{HP} = COP_R + 1$

Since COP_R is a positive quantity. For set values of Q_l, Q_h, it gives us $COP_{hp} > 1$. That is, by supplying more energy to the house as it consumes work of heat pump, least, like electric heater. In reality, however, because of piping and many more devices bit of Q_h will be lost to surroundings, COP_{HP} may drop below unity. When it happens, normally system changes to fuel.

Rate of heat removal from the refrigerated space is shown by (TOR) Tons of refrigeration. Refrigeration system capacity will freeze 1ton (2000lbm) of liquefied water at 0℃ (32℉) into ice at 0℃ in 24h is told to be 1 ton. 1 ton of refrigeration is equal to 200 Btu/min or 211kj/min. The typical 200-m² residence cooling load is 3-ton (120-kW) range.

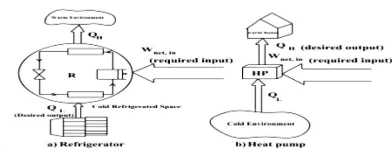

Fig 1.1 a) Refrigerator Fig 1.1 b) Heat pump

II. OBJECTIVE OF THE PRESENT WORK

- To Increase the COP of Vapor compression refrigeration cycle
- To decrease the compressor work
- To increase the pressure of the refrigerant entering the condenser

III. METHODOLOGY

The schematic diagram of the vapor compression refrigeration system with diffuser at condenser inlet the system consists of two flow lines one is simple VCR flow line without diffuser and other is flow line with diffuser. Thus we can calculate the pressure with and without diffuser. P-h diagram has been shown in figure 3.1. Figure 3.1 shows the pressure enthalpy chart of the system. The path 1-2-3-4-5-6-7 shows the VCR cycle with diffuser and path 1-2'-3'-4'-5-6-7 shows the VCR cycle without diffuser at condenser inlet.

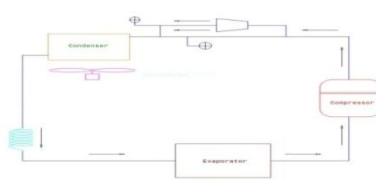

Fig.2. VCR system with diffuser

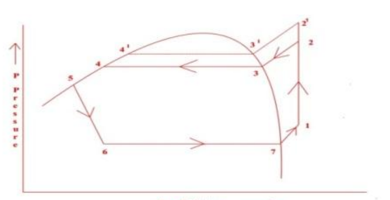

Fig 3. (P-H Diagram)

$h_4 = h_f$ = Enthalpy at condenser pressure

$h_5 = h_4 - c_p (T_4 - T_5)$

$h_7 = h_g$ = Enthalpy at evaporator pressure

$h_1 = h_7 + c_p (T_1 - T_7)$

h_3 is h_g at condenser temperature

$h_2 = h_3 + c_p (T_2 - T_4)$

COP = $\frac{\textit{Refrigeration effect}}{\textit{work input}}$ = $(h_7 - h_6) / (h_2 - h_1)$

Mr =Refrigeration capacity/refrigeration effect in kg/s

Refrigeration capacity=m $(h_1\text{-}h_6)$

Q=Mr Cp dt Watts

Compressor power=m $(h_2\text{-}h_1)$ kW

Result taken from refrigeration kit is validated through CFD

Further Diffuser has been used to decrease power input to the compressor which will enhance COP

DIFFUSER LENGTH = $(D_1\text{-}D_2)/\tan\theta$

Different D/L ratio of 0.5 and 0.6 for divergence angle of 15^0 has been carried so as to decrease power input of the compressor which will result in increasing COP

Modeling and meshing done in ICEM-CFD, analysis in CFX and post result in CFD POST

The above chapter gives an overview on the methodology of the experiment.

IV. EXPERIMENTAL SETUP

In this chapter we discuss about the experimental setup and calculations of the experiment conducted.

Fig 4. Experimental set up (Refrigeration Test Rig)

4.1 Specifications

Refrigerator Capacity = 220 lts.

Pipe Diameter of the Evaporator =11 mm = 0.011 m

Length of the evaporator coil = 1539 mm = 1.539 m

4.2 Observations

T_1 = Inlet Temperature at Compressor, (°C)

T_2 = Outlet Temperature at Compressor, (°C)

T_3 = Outlet Temperature at Condenser, (°C)

T_4 = Inlet Temperature at Evaporator, (°C)

P_1 = evaporator pressure (Kg/cm^2)

P_2 = condenser pressure (lb/in^2)

Refrigerant used - R134A (Tetra Fluoro Ethane)

Compressor Suction pressure = Low Pressure

Compressor Discharge pressure = High Pressure

Compressor Suction temperature = Evaporator Outlet temperature

Compressor Discharge temperature = Compressor Outlet Temperature

Temperature Evaporator = Temperature at Evaporator Inlet

After condensation temperature = Temperature at Condenser Outlet

TABLE I

Experiments	T_1	T_2	T_3	T_4	H.P (P_2)		L.P. (P_1)	
					Kg/cm^2	KPa	lb/in^2	KPa
1	8	75	29.92	-6.2	7.92	750	20	137.9
2	4.7	77.1	25.61	-12.6	7.6	720.5	12.5	86.18
3	3.5	77.9	25	-13.8	7.5	710.8	10	68.95

4.3Tabular Column:

Values of Enthalpy, Saturation temperature were taken from the table for R134A.

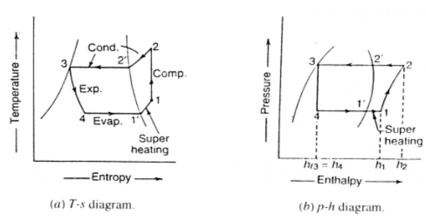

Fig.5. Actual Vapour Compression Cycle

4. 3.1. EXPERIMENT TRAIL 1. (Without Diffuser)

For P_1 = 137.89 KPa saturation temp. is t_{s1} = -19.15°C. But observed temperature is 8°C. Therefore the condition of the refrigerant before compression is Superheat.

$H_1 = h_{g1} + c_p (t_{sup}-t_{s1}) = 238.98 + 0.958 (8-(-19.15))$

= 264.99 KJ/Kg

Final pressure = P_2 = 770KPa

Temperature is 75°C.

For P_2 = 770 KPa saturation temp. is t_{s2} = 29.92°C.

The condition of the refrigerant after compression is Superheat.

$H_2 = h_{g2} + c_p (t_{sup}-t_{s2}) = 266.64 + 0.958 (75- 29.92)$

$= 309.83$ KJ/Kg

$H_6 = h_5 = 93.155$ KJ/Kg

Coefficient of Performance

C.O.P. = Refrigeration Effect / Work Done

$= (H_1 – H_4) / (H_2 – H_1)$

$= (264.99 – 93.155) / (309.83 – 264.99)$

$= 3.83$

4.3.2. TRAIL 2. (Without Diffuser)

For P_1 = 86.18 KPa saturation temp. is t_{s1} = -29.69°C. But observed temperature is 4.7°C. Therefore the condition of the refrigerant before compression is Superheat.

$H_1 = h_{g1} + c_p (t_{sup}-t_{s1}) = 232.4 + 0.958 (4.7-(-29.69))$

$= 265.35$ KJ/Kg

For P_2 = 720.5 KPa saturation temp. is t_{s2} = 28°C. But observed temperature is 61.2°C. Therefore the condition of the refrigerant after compression is Superheat.

$H_2 = h_{g2} + c_p (t_{sup}-t_{s2}) = 265.232 + 0.958 (77.1- 28) = 312.26$ KJ/Kg

$h_3 = h_4 = 89.202$ KJ/Kg

Coefficient of Performance = C.O.P. = Refrigeration Effect / Work done= $(H_1 – h_4) / (H_2 – H_1)$

$= (265.35 – 89.202) / (312.26 – 265.35)$

$= 3.75$

4.3.3. TRAIL 3. (Without Diffuser)

For P_1 = 68.95 KPa saturation temp. is t_{s1} = -34.29°C. But observed temperature is 3.5°C. Therefore the condition of the refrigerant before compression is Superheat.

$H_1 = h_{g1} + c_p (t_{sup}-t_{s1}) = 229.465 + 0.958 (3.5-(-34.29)) = 265.67$ KJ/Kg

For P_2 = 700.8 KPa saturation temp. is t_{s2} = 26.318°C. But observed temperature is 77.9°C. Therefore the condition of the refrigerant after compression is Superheat.

$H_2 = h_{g2} + c_p (t_{sup}-t_{s2}) = 264.722 + 0.958 (77.9- 26.318) = 314.13$ KJ/Kg

$h_3 = h_4 = 89.202$ KJ/Kg

Coefficient of Performance = C.O.P. = Refrigeration Effect / Work Done

$= (H_1 – h_4) / (H_2 – H_1)$

$= (265.67 – 89.202) / (314.13 – 265.67)$

= 3.64

CFD VALIDATION OF THE ABOVE RESULTS

4.5. Evaporator Flow Analysis

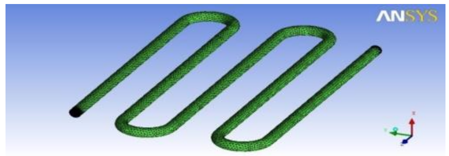

Fig 6. Modeling and meshing

4.5.1. Analysis Carried Through CFX

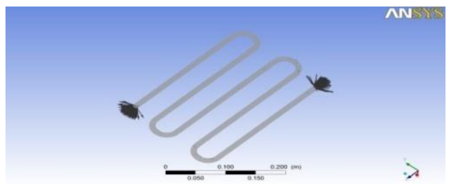

Fig 7. Model Imported From ICEM CFD

V. RESULTS AND DISCUSSIONS

We discuss about the results obtained from the experiment.

5.1 GEOMETRY OF DIFFUSER (D/L=0.5)

INNER DIA= 15 mm

ANGLE OF DIVERGENCE=15^0

LENGTH OF DIFFUSER= 30 mm

DIFFUSER ANGLE SELECTED ON THE BASIS OF REFERENCE PAPER

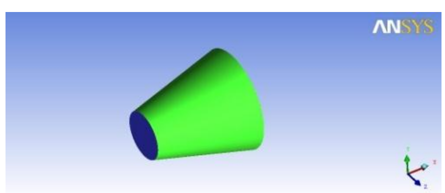

Fig. 8. Diffuser Model

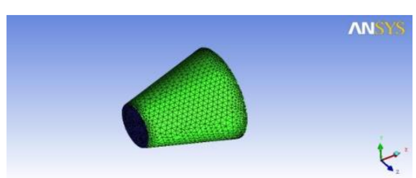

Fig 9. Meshing of Diffuser

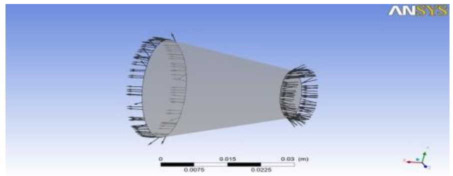

Fig 10. Imported from ICEMCFD to CFX for Analysis

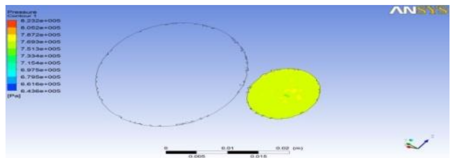

Fig 11. Inlet Pressure of the diffuser is validated

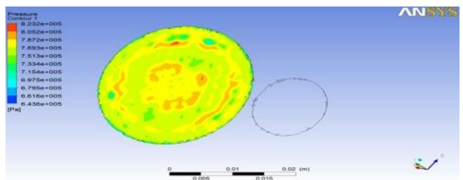

Fig 12. Outlet Pressure of the diffuser is validated

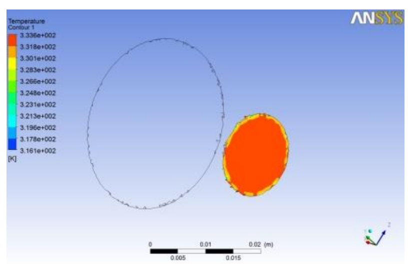

Fig 13. Inlet Temperature of the diffuser is validated

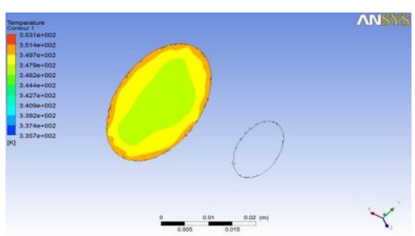

Fig 14. Outlet Temperature of the diffuser is validated

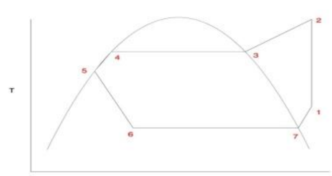

Fig 16. Actual Vapour Compression Cycle (T-S diagram)

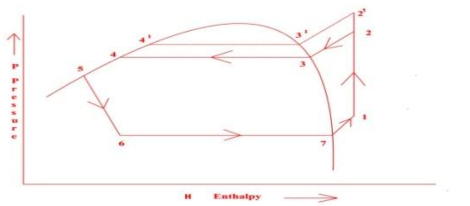

Fig 17. Pressure Enthalpy diagram

When diffuser added pressure from 2 to 2^1 i.e. from bar to bar.

But, if diffuser is not there compressor take additional power input to reach bar.

WITH DIFFUSER

For P_1 = 137.89 KPa saturation temperature is

t_{s1} = -19.15°C. But observed temperature is 8°C. Therefore the condition of the refrigerant before compression is Superheat.

$H_1 = h_{g1} + c_p (t_{sup}-t_{s1}) = 238.98 + 0.958 (8-(-19.15)) = 264.99$ KJ/Kg

For P_2 = 764.9 KPa saturation temp. is t_{s2} = 29.69°C. But observed temperature is 60.4°C. Therefore the condition of the refrigerant after compression is Superheat.

$H_2 = h_{g2} + c_p (t_{sup}-t_{s2}) = 266.53 + 0.958 (60.4- 29.69) = 295.95$ KJ/Kg

$H_5 = h_6 = 93.155$ KJ/Kg

Coefficient of Performance

C.O.P. = Refrigeration Effect / Work Done

$= (H_1 – h_6) / (H_2 – H_1)$

$= (264.99 – 93.155) / (295.95 – 264.99)$

= 5.55

WITHOUT DIFFUSER

Therefore Compressor input = $h_2{}^1 - h_1$

Refrigeration effect = $h_1 - h_6$

For P_1 = 137.89 KPa saturation temp. is t_{s1} = -19.15°C. But observed temperature is 8°C. Therefore the condition of the refrigerant before compression is Superheat.

$H_1 = h_{g1} + c_p (t_{sup}-t_{s1}) = 238.98 + 0.958 (8-(-19.15)) = 264.99$ KJ/Kg

Final pressure = $P_2{}^1$ = 770KPa

Final Temperature = 74.09°C.

For $P_2{}^1$ = 770 KPa saturation temp. is $t_3{}^1$ = 29.92°C.

The condition of the refrigerant after compression is Superheat.

$H_2{}^1 = h_3{}^1 + c_p (t_{sup}-t_3{}^1) = 266.64 + 0.958 (75- 29.92) = 309.83$ KJ/Kg

$H_6 = h_5 = 93.155$ KJ/Kg

Coefficient of Performance

C.O.P. = Refrigeration Effect / Work Done

$= (H_1 - h_6) / (H_2{}^1 - H_1)$

$= (264.99 - 93.155) / (309.83 - 264.99)$

= 3.83

Enhancement of COP =

$$\frac{COP\ with\ diffuser - COP\ without\ diffuser}{COP\ with\ diffuser} = \frac{5.55 - 3.83}{5.55}$$

= 0.31 = 31%

Table5.1 Results

SL NO	REFRIGERATION EFFECT WITHOUT DIFFUSER KJ/Kg	COMPRESSOR INPUT WITHOUT DIFFUSER KJ/Kg	REFRIGERATION EFFECT WITH DIFFUSER KJ/Kg	COMPRESSOR INPUT WITH DIFFUSER KJ/Kg
1	171.84	44.84	171.84	30.96

Table 5.2 Results

SL NO	POWER INPUT TO COMPRESSOR WITHOUT DIFFUSER KW	POWER INPUT TO COMPRESSOR WITH DIFFUSER KW	COP WITHOUT DIFFUSER	COP WITH DIFFUSER	% OF ENHANCEMENT OF COP
1	0.2	0.14	3.83	5.55	31%

VI. CONCLUSIONS

From above analysis following conclusion has been arrived.

1. With the addition of diffuser pressure inlet of refrigerant to the Condenser is increased.
2. Compressor Work input is decreased
3. Since compressor work input decreased COP has been increased.
4. With the addition of diffuser Coefficient of Performance (COP) Enhancement has been increased by 31% when compared without diffuser.
5. Compressor Power input obtained without diffuser is 0.2 KW.
6. Compressor Power input with diffuser obtained is 0.14KW
7. Finally conclusion drawn is Coefficient of Performance (COP) can be enhanced by placing diffuser.

FUTURE SCOPE OF WORK

1. With varying divergence angle to the diffuser analysis can be carried out
2. By adopting passive method for diffuser enhancement of pressure can be done
3. By providing heat exchanger in between evaporator and compressor temperature outlet can be raised.
4. Bleed vapour to the condenser can be sent directly from evaporator COP can be raised.

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