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ENHANCEMENT OF COEFFICIENT OF PERFORMANCE IN VAPOUR COMPRESSION REFRIGERATION CYCLE

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Abstract

To Improve the Coefficient of Performance (COP), it is require decreasing the Compressor Work and increasing the Refrigerating Effect. Experimental analysis on vapour compression refrigeration (VCR) system with R134A (Tetra Fluro Ethane) refrigerant was done and their results were recorded. The effects of the main parameters of performance analysis are mass flow rate of refrigerant, suction pressure of compressor, delivery pressure of compressor, temperature of evaporator and condenser

The results from vapour compression refrigerant plant was taken where the variables like suction pressure of compressor, delivery pressure of compressor ,temperature of evaporator and condenser were noted and coefficient of performance (COP) was calculated. The results obtained will be validated through CFD simulation.

Further diffuser has been introduced in between compressor and condenser so that power input to the compressor has been reduced there by enhancing COP. The enhancement will be done through CFD simulation; Modeling and meshing will be done in ICEMCFD, analysis in CFX and post results in CFD POST.

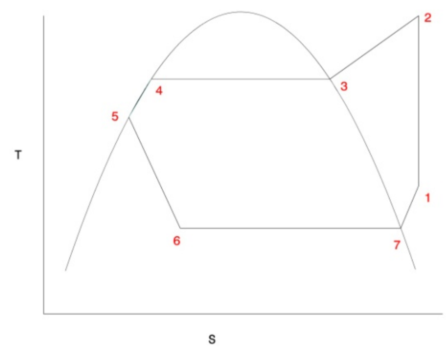
KEY WORDS: Tetra Fluro Ethane, ICEMCFD

1. Introduction

In thermodynamics Refrigeration is the major application area, in which the heat is transferred from a lower temperature region to a higher temperature region. The devices which produce refrigeration are known as Refrigerators and the cycle on which it operates are called refrigeration cycles. Vapour compression refrigeration cycle is the most regularly used refrigeration cycle in which the refrigerant is alternately vaporized and condensed and in the vapor phase it is compressed. Gas refrigeration cycle is the well-known refrigeration cycle in which cycle refrigerant remains in the gaseous phase throughout the cycle. Cascade refrigeration are the other refrigeration cycles discussed in this chapter; absorption refrigeration is the one more refrigeration cycle which is used where the refrigerant is dissolved in liquid before it is compressed.

2. Litreature Survey

Amit Prakash [1] worked on the topic of "Improving the performance of Vapor compression refrigeration system by using sub- cooling and diffuser". He found that COP of Vapor Compression Cycle is increased by lowering the power consumption /work input or increasing the refrigerating effect. By using sub-cooling and using a diffuser at condenser inlet refrigerating effect increases and power consumption or work input decreases. Thus performance of cycle is increased. High velocity refrigerant has various serious affect on vapor compression refrigeration system such as liquid hump, undesirable splashing of the liquid refrigerant in the condenser and damage To the condenser tubes by vibration, pitting and erosion.



Actual Vapour Compression Cycle (T-S Diagram)

3. Methodology

R. Reji Kumar et.al [2] worked on the topic of "Heat transfer enhancement in domestic refrigerator using R600a/mineral oil/nano-Al2O3 as working fluid" They found that the R600a refrigerant and mineral oil mixture with nanoparticles worked normally. Freezing capacity of the refrigeration system will be more with mineral oil + alumina nanoparticles oil mixture compared the system with POE oil. When the nanolubricant is used instead of conventional POE oil the power consumption of the compressor reduces by 11.5% when the conventional POE oil is replaced with nanorefrigerant the coefficient of performance (COP) of the refrigeration system also increases by 19.6 %.

$h_4 = h_f$ = Enthalpy at condenser temperature

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

$h_7 = h_g$ = Enthalpy at evaporator temperature

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

h_3 is h_g at condenser temperature

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

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

m_r = refrigeration capacity/refrigeration effect in kg/s

Refrigeration capacity=m (h_1-h_6)

Q=$m_R\, c_p\, dt$ Watts

Compressor power=m (h_2-h_1) kW

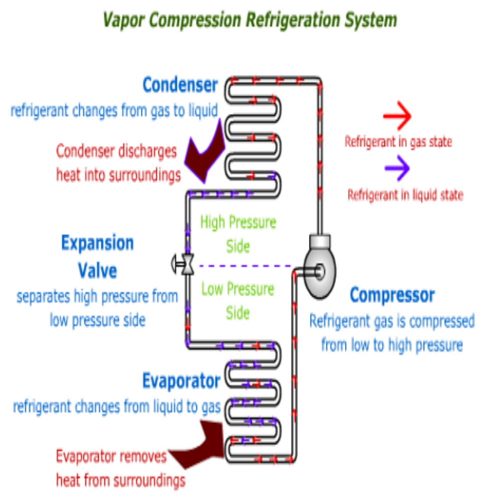
Result taken from refrigeration kit is validated through CFDFurther Diffuser has been used to decrease power input to the compressor which will enhance COP DIFFUSER LENGTH = (D1-D2)/tanθ

Different D/L ratio of 0.5 and 0.6 for divergence angle of 15o 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

4. Experimental setup

Experimental Set Up (Refrigeration Test Rig)



LINE DIAGRAM

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

Observations

T1 = Temperature at Compressor Inlet

T2 = Temperature at Compressor outlet

T3 = Temperature at Condenser outlet

T4 = Temperature at Evaporator Inlet

P1 = Upstream pressure of the Compressor

P2 = Downstream pressure of the Compressor

Refrigerant used - R134A (Tetra Fluoro Ethane)

Suction pressure of the compressor

= Low Pressure

Discharge pressure of the compressor

= High Pressure

Suction temperature of the compressor

= Outlet temperature of the evaporator

Discharge temperature of the compressor = Temperature at Compressor Outlet

Evaporator temperature = Temperature at Evaporator Inlet

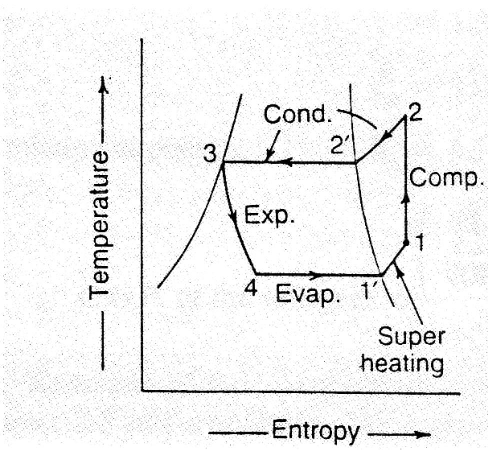
After condensation temperature

= Temperature at Condenser Outlet

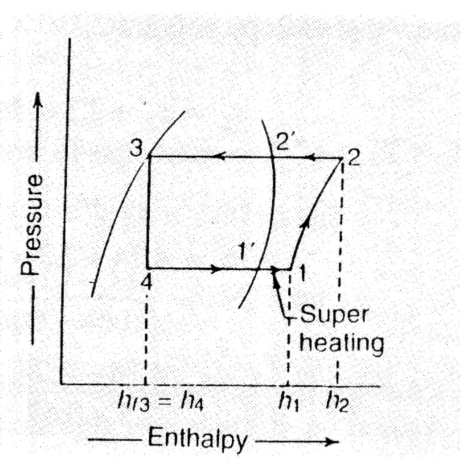
CALCULATIONS FOR COP

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	60.4	37.1	-6.2	7.8	764.9	20	137.9
2	4.7	61.2	34.1	-12.6	7.2	706.1	12.5	86.18
3	3.5	61.8	35	-13.8	7	686.5	10	68.95

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



(a) *T-s* diagram.



(b) *p-h* diagram.

Actual Vapour Compression Cycle

EXPERIMENT 1.

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**

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**

$h_3 = h_4 =$ **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) / (295.95 – 264.99)

= 5.55

Heat Flux

Refrigeration Capacity = 0.220 X 3.5 = 0.77 KJ/s

Refrigeration Effect (EXP -1) = 171.835 KJ/Kg

m = mass flow rate in = $\frac{Refrigertion\ Capacity}{Refrigeration\ Effect}$ = 0.77 / 171.835 = 0.00448 Kg/S

Heat transferred = Q = m C_p dT

= 0.00448 * 0.958 * (8-(-6.2) = 0.06094 KW = 60.94 W

Heat Flux

= Q/A = $\frac{60.94}{\pi \times 0.011 \times 1.539}$

= 1146 W/m²

For P_2 = 686.5 KPa saturation temp. is t_{s2} = 25.986 °C. But observed temperature is 61.8°C. Therefore the condition of the refrigerant after compression is Superheat.

$H_2 = h_{g2} + c_p (t_{sup}-t_{s2}) = 264.722 + 0.958 (61.8- 25.986) =$ **299.03 KJ/Kg**

$h_3 = h_4 =$ **89.202 KJ/Kg**

Coefficient of Performance = C.O.P.

= Refrigeration Effect / Work Done

$= (H_1 – h_4) / (H_2 – H_1)$

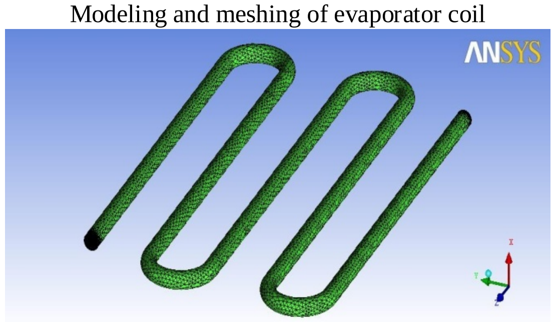
= (265.67 – 89.202) / (299.03 – 265.67)

= 5.29

5. Experimental results

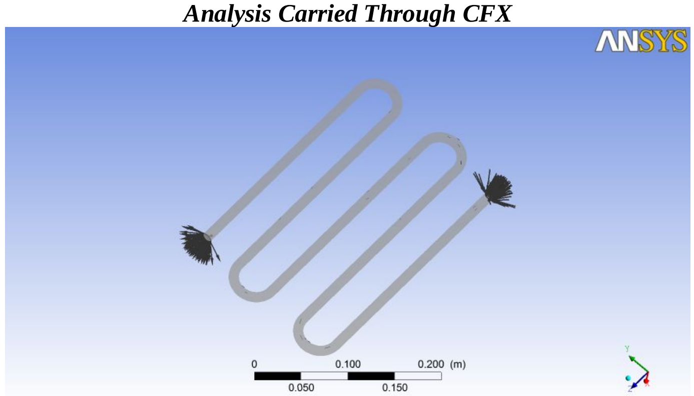
Evaporator Flow Analysis

Modeling and meshing of evaporator coil



Modeling and meshing

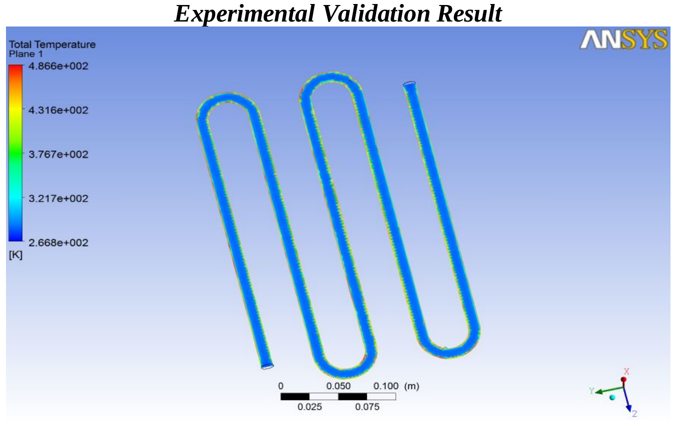
Analysis Carried Through CFX



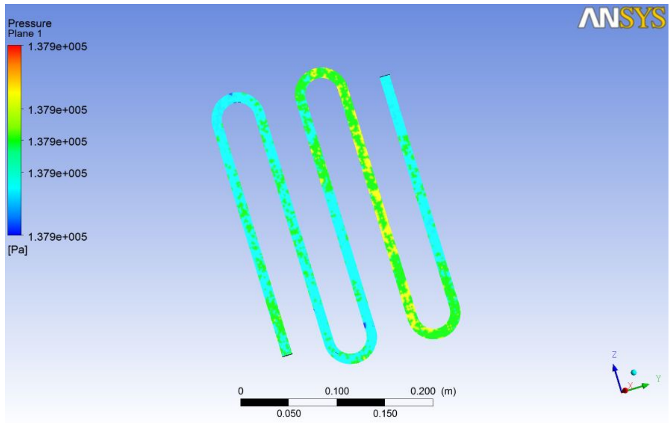
Model Imported From ICEM CFD

Boundary Condition

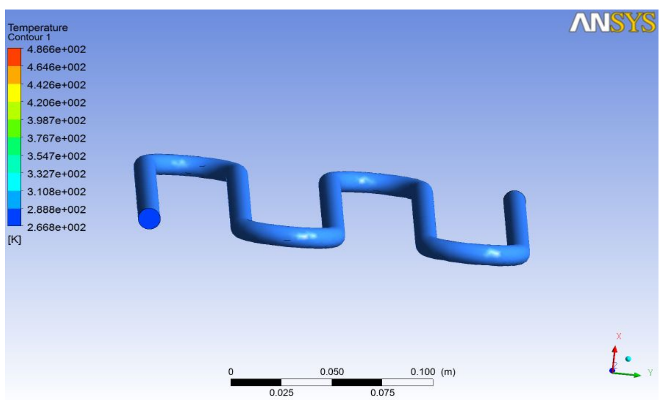
Experimental Validation Result



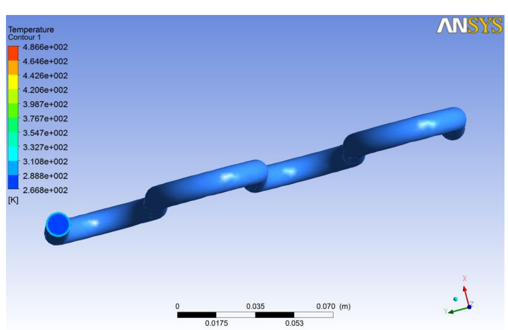
Temperature Plane



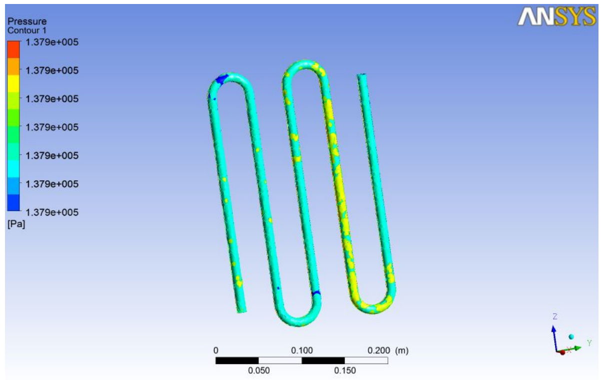
Pressure Plane

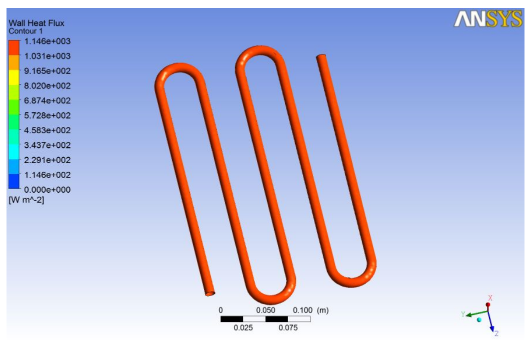


Temperature Contour inlet



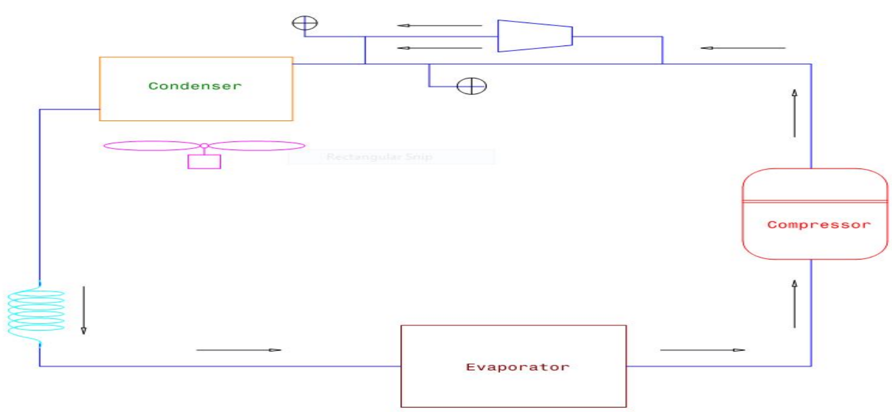
Temperature Contour outlet





Wall Heat Flux Contour

ENHANCEMENT OF COP BY PROVIDING DIFFUSER BETWEEN COMPRESSOR AND CONDENSER WITH DIVERGENCE ANGLE OF DIFFUSER 15^0



Vapour Compression Refrigeration Cycle Along With Diffuser Block Diagram

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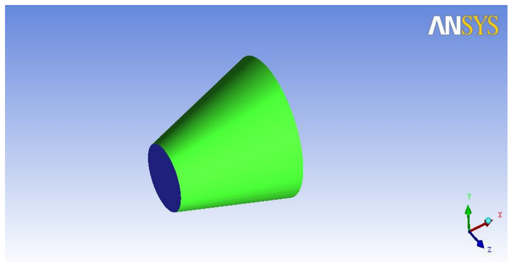
6. Results and Discussions

GEOMETRY OF DIFFUSER (D/L=0.5)

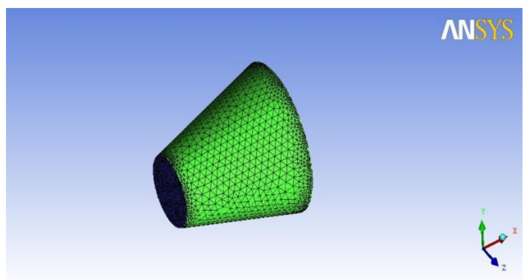
INNER DIA= 15 mm

ANGLE OF DIVERGENCE=15^0

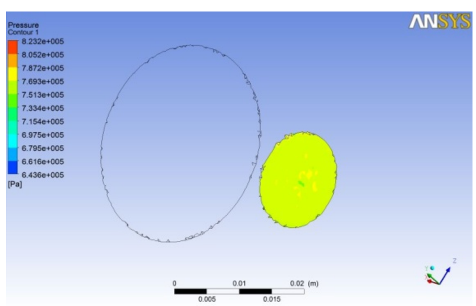
LENGTH OF DIFFUSER= 30 mm



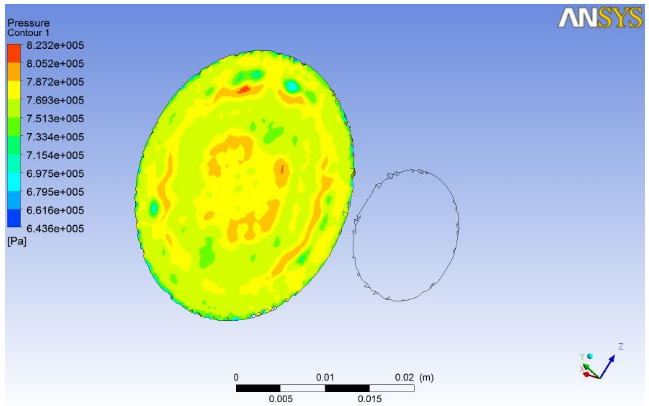
Diffuser Model



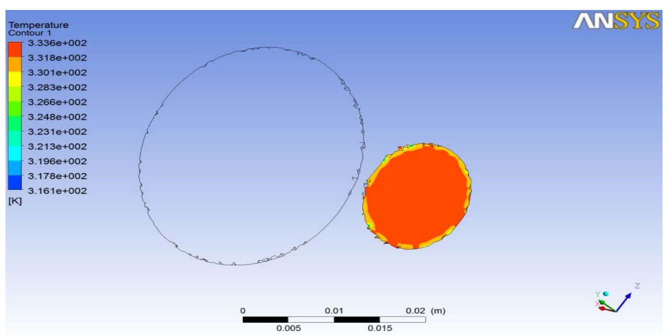
Meshing of Diffuser



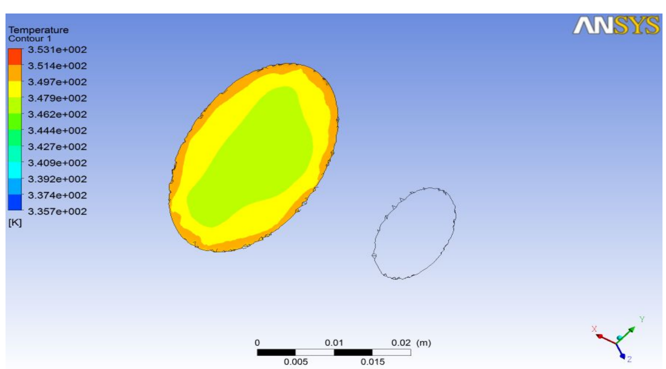
Inlet Pressure of the diffuser



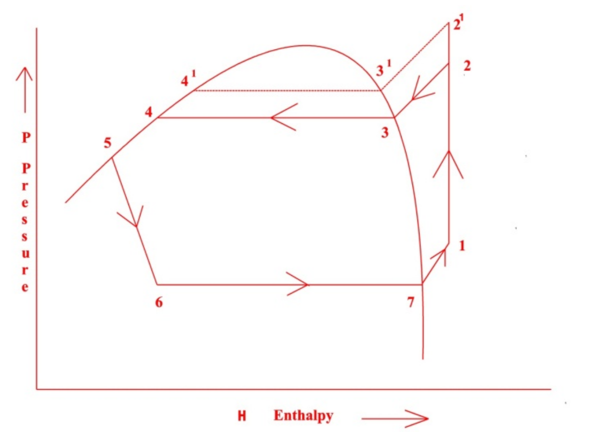
Outlet Pressure of the diffuser



Inlet Temperature of the diffuser



Outlet Temperature of the diffuser



<u>WITH DIFFUSER</u>

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.

$H_1 = h_{g1} + c_p\,(t_{sup}\text{-}t_{s1})$ = 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.

$H_2 = h_{g2} + c_p\,(t_{sup}\text{-}t_{s2})$ = 266.53 + 0.958 (60.4- 29.69) = **295.95 kJ/Kg**

$h_3 = h_4$ = **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) / (295.95 – 264.99)

= 5.55

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

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 = 348 K = 348 – 273 = 75 °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 = $\dfrac{COP\ with\ diffuser - COP\ without\ diffuser}{COP\ with\ diffuser} - \dfrac{5.55-3.83}{5.55}$ **= 0.31 = 31 %**

7. Conclusion and Discussion

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

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%

8. Conclusion

From above analysis following conclusion has been arrived.

- With the addition of diffuser pressure inlet of refrigerant to the condenser is increased.
- Work input to the compressor is decreased
- Since work input to the compressor decreased Coefficient of Performance (COP) of refrigerator has been increased.
- With the addition of diffuser Coefficient of Performance (COP) enhancement has been increased by 31 % when compared without diffuser.
- Power input to the compressor obtained without diffuser is 0.2 KW.
- Power input to the compressor with diffuser obtained is 0.14KW
- Finally conclusion drawn is Coefficient of Performance (COP) can be enhanced by placing diffuser.