Enhancement of Coefficient of Performance by Analysis of Flow through Vapour Compression Refrigeration Cycle using CFD

Shrikrishna Bhat[#1], Gunesh Ravanikar[#2], Dr. Matrthi B H[#3], Ravindra A R[#4]
Mechanical Engineering Department, EWIT, Bengaluru-91
KARNATAKA, INDIA
#1 P G STUDENT,shrikrishnabhat28@gmail.com, #2 ASSISTANT PROFESSOR,#3 PROFESSOR & HOD.
#4P G STUDENT,ravindra.0905@gmail.com

Abstract— **To Improve the Coefficient of Performance (COP), it is required to diminish the Compressor Work and build the Refrigerating Effect. Test examination on vapor pressure refrigeration (VCR) framework with R134A (Tetra Fluro Ethane) refrigerant was done and their outcomes were recorded. The impacts of the fundamental parameters of execution examination are mass stream rate of refrigerant, suction weight of compressor, conveyance weight of compressor, temperature of evaporator and condenser. The outcomes from vapor pressure refrigerant plant was taken where the variables like suction weight of compressor, conveyance weight of compressor ,temperature of evaporator and condenser were noted and coefficient of execution (COP) was figured. The outcomes got will be accepted through CFD recreation. Further Passive strategy has been embraced for evaporator and diffuser has been presented in the middle of compressor and condenser so control contribution to the compressor has been diminished there by upgrading COP. The upgrade will be done through CFD recreation; Modeling and lattice will be done in ICEMCFD, examination in CFX and post results in CFD POST.**

Keywords— **Thrust, Altitude, Pressure, supersonic, sonic, throat.**

1. Introduction

In thermodynamics Refrigeration is the real application range, in which the warmth is exchanged to higher temperature from a lower temperature district. The gadgets which deliver refrigeration are known as Refrigerators, the cycle which it works is called refrigeration cycles. Vapor pressure R- cycle is the most frequently utilized cycle, refrigerant is then again vaporized and dense and in the vapor stage it is compacted. Gas R-cycle is the surely understood R-cycle in which cycle refrigerant stays at the vaporous stage all through the cycle. Course refrigeration are the other refrigeration cycles talked about in this section; retention refrigeration is the one more refrigeration cycle which is utilized where the refrigerant is broken down in fluid before it is compacted. One more refrigeration in which refrigeration is delivered by passing the electric current through two unique materials is called as the thermoelectric refrigeration.

1.1Refrigerators and Heat Pumps

We as a whole realize that warmth dependably spills out of higher temperature medium to a lower temperature medium i.e. toward diminishing temperature. This warmth move process happens in nature without help of any gadgets. The converse procedure, be that as it may, can't happen without anyone else. The unique gadgets by which the warmth exchanges from a lower temperature medium to a higher temperature medium are called Refrigerators.

The working liquids utilized as a part of the refrigeration cycles are called refrigerants, and the iceboxes are cyclic gadgets. Here Q-L is the extent of warmth expelled at temperature the refrigerated space from TL, Q-H is the size of warmth rejected at temperature TH to the warm space, the net work contribution to the fridge is Wnet, in and Q-H and Q-L speaks to sizes and they are sure amounts.

Heat pump is the gadget that exchanges heat from a lowest temperature area to a highest temperature locale. Heat pumps and Refrigerators are fundamentally the same gadgets; yet they are divergent in their targets as it were. The primary goal of a cooler is keeping up the space at a lower temperature by expelling warm temperature from it. A vital portion of the operation, not the intention is releasing this warmth to a highest-temperature medium. The target of a warmth pump is keeping up the warmed space at a high temperature
Execution of warmth pumps and fridges is communicated regarding the coefficient of execution (COP), characterized as –

$$COP_R = \frac{Desired\ output}{Required\ input} = \frac{Cooling\ effect}{Work\ input} = \frac{Q_L}{W_{net,in}} \quad (1.1)$$

$$COP_{HP} = \frac{Desired\ output}{Required\ input} = \frac{Heating\ effect}{Work\ input} = \frac{Q_H}{W_{net,in}} \quad (1.2)$$

Above relation may be expressed in the form by replacing the quantities $Q_{H,}$ Q_l and $W_{net,in}$ by $\dot{Q}_{H,}$ $\dot{Q}_L$, and $\dot{W}_{net,in}$ respectively. Note that both COP_{HP} and COP_R can be greater than 1. The comparison of equations 1.1 and 1.2 reveals that

$$COP_{HP} = 1+COP_R \quad (1.3)$$

Since COPR is a positive amount. For settled estimations of Q-L and Q-H, this connection suggests that COP-HP >1. That is, by supplying as much vitality to the house as it devours a warmth pump capacities, at any rate, as a resistance radiator. In actuality, be that as it may, through funneling and different gadgets a portion of Q-H is lost due to the outside air, when the outside air temperature is too low, COP-HP may drop underneath solidarity. When this happens, the framework ordinarily changes to fuel (oil, regular duct, hydrocarbon gas and so forth) or resistance warming mod.

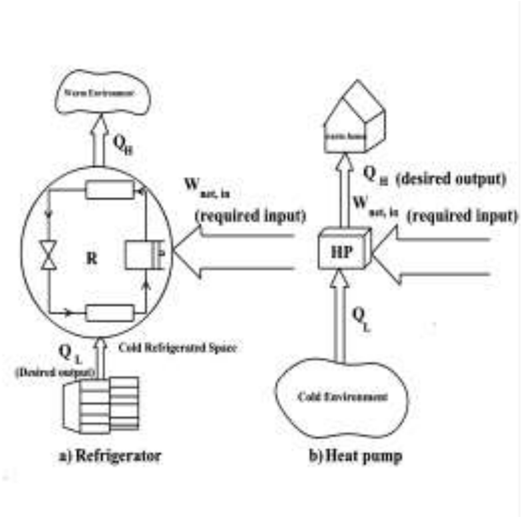


Fig 1.1 a) Refrigerator **Fig 1.1 b) Heat pump**

The ideal V-C Refrigeration cycle

A considerable lot of im-practicalities related with turned around, Carnot cycle can be expelled through vaporizing the refrigerant totally before it compacted and by supplanting the turbine with a throttling gadgets, for example, an extension duct or fine tube. The cycle that outcomes known as perfect vapour pressure R-cycle, and demonstrated schematically and T-s graph. The vapour-pressure R-cycle is the most comprehensively utilized cycle for coolers, aerating and cooling frameworks, and warmth pumps. It comprises of four procedures:

Process 1-2 Isentropic pressure in a compressor.

Process 2-3 Constant-weight heat dismissal in a condenser.

Process 3-4 Throttling in a development gadget.

Process 4-1 Constant –pressure heat retention in an evaporator.

In the perfect vapor-pressure refrigeration cycle, the refrigerant enters as a soaked vapor to the compressor at state 1 and is compacted isentropically to the condenser weight at state 2. Amid this isentropic pressure handle the temperature of the refrigerant increments to the temperature which is well over that of the encompassing medium. In the wake of leaving the compressor the refrigerant then enters the condenser as superheated vapor at state 2 and leaves the condenser as immersed fluid at state 3 by dismissing the warmth to the environment. The temperature of the refrigerant at the state 3 is still over the temp of the environment.

In T-s graph, as appeared in Fig 1.2, the warmth exchange for inside reversible procedures is spoken to by the range under the procedure bend. The warmth retention by the refrigerant in the evaporator is spoken to by the region under the procedure bend (4-1), and the warmth dismissal in the condenser is spoken to by range under the procedure bend (2-3). A thumb guideline is that the COP enhances by 2 to 4 percent for each °C for expansion in vanishing temperature or decline in the gathering temperature.

P-h chart as appeared in Fig1.3 is often utilized as a part of the investigation of vapor-pressure refrigeration cycles,. Three of the four procedures show up as straight lines on this graph. Furthermore, the warmth move in the evaporator and the condenser is with respect to the lengths of the equal procedure bends.

The perfect vapor-pressure refrigeration cycle includes an irreversible (throttling) process, consequently is not an inside reversible cycle dissimilar to the perfect cycles examined some time recently. Make it more sensible model for real vapor-pressure refrigeration cycle this (throttling) procedure is kept up in the cycle. On the chance that the throttling gadget is supplanted by the isen-tropic turbine. Accordingly, net work information would decrease and the refrigeration limit would increment (by the region under procedure bend 4^1-4 in Fig 1.2). In the event that the extension valve is supplanted by a turbine, it is not a practical, in any case, subsequent to the additional expense and unpredictability can't legitimize by the additional advantages.

The every one of the 4 segments related with vapour-pressure R- cycle is unfaltering –flow gadgets, and accordingly all the 4 forms that build the cycle can be examined as enduring stream forms. The refrigerant potential vitality and dynamic vitality changes are generally little with respect to the warmth and work exchange terms, and subsequently they can be overlooked. At that point the consistent stream vitality condition on a unit-mass premise lessens to :

$$(q_{in} - q_{out}) + (W_{in} - W_{out}) = h_e - h_i$$

The evaporator and the condenser do not involve any work, and the compressor can be estimated as adiabatic. Then the COP's of refrigerators and heat pumps working on the vapour- compression refrigeration cycle can be expressed as:

$$COP_R = \frac{q_L}{W_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1} \quad (1\text{-}7)$$

And

$$COP_{HP} = \frac{q_H}{W_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1} \quad (1\text{-}8)$$

$h_1 = h_g$ at pressure p_1 and $h_3 = h_f$ at pressure p_3 for ideal case.

In 1834, the Englishman Jacob Perkins got a patent for Vapor pressure refrigeration, shut cycle ice machine utilizing other unstable liquids or ether as refrigerants. The operational model of this machine was constructed, be that as it may it was never made by financially. In 1850, Alexander Twining planned and construct the vapor-pressure ice machines utilizing ethyl ether. Ethyl ether is a financially utilized refrigerant as a part of vapor pressure refrigeration frameworks. Fundamentally, vapor pressure refrigeration frameworks were substantial and for the most part utilized for fermenting, cool stockpiling, and ice making. They needed programmed control and steam-motor driven. In 1890s, the more established units were begun to supplanted by electric engine driven littler machines outfitted with programmed controls and the refrigeration frameworks started to show up in family units and butcher shops. In 1930, nonstop changes made it plausible to have vapor pressure refrigeration frameworks were relatively little, ependable, productive, & economical.

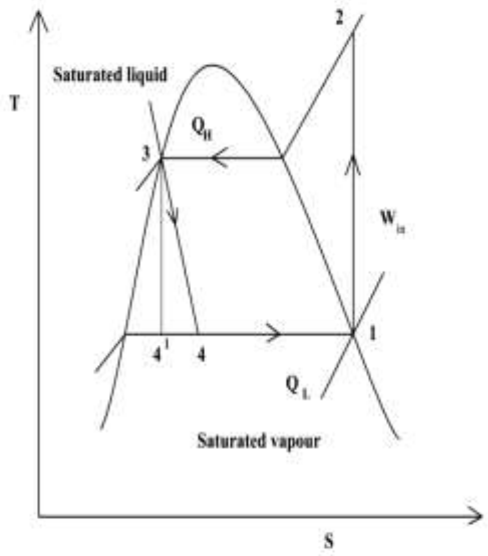


Fig T-S Diagram For The Ideal Vapour Compression Refrigeration Cycle

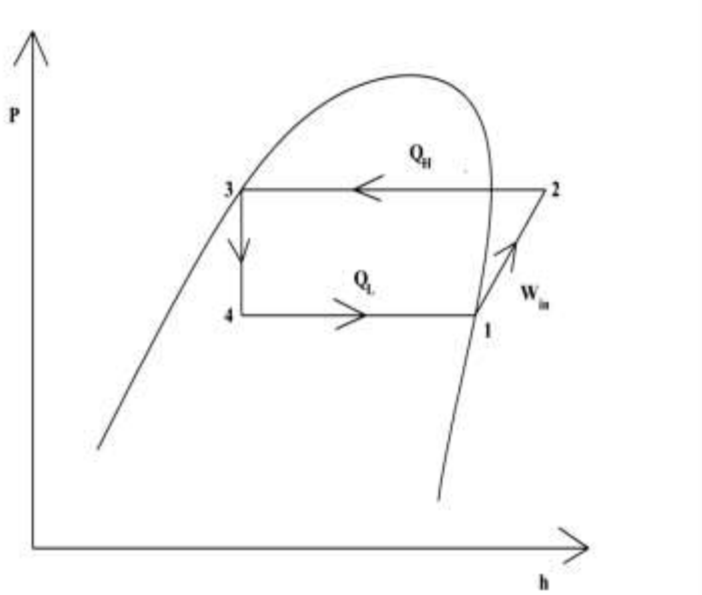


Fig P-h Diagram Of An Ideal Vapour Compression Refrigeration Cycle

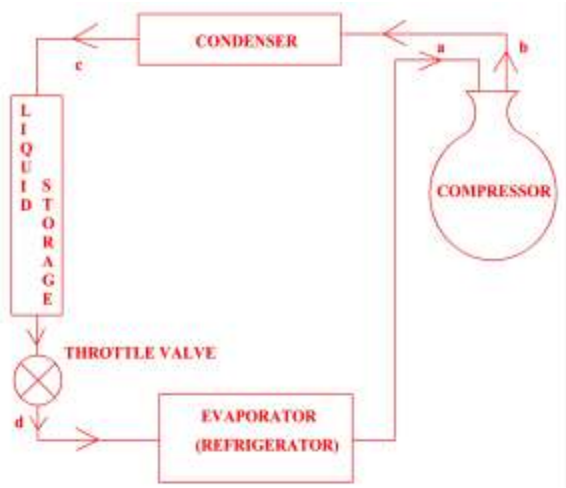


Fig Vapour Compression Refrigeration Systems

METHODOLOGY

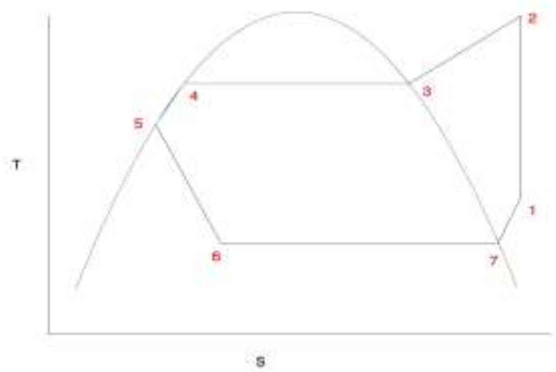


Fig Actual Vapour Compression Cycle (T-S Diagram)

$h_4 = h_f$ = 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
$h_2 = h_3 + c_p (T_2 - T_4)$

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

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 CFD
Further Diffuser has been used to decrease power input to the compressor which will enhance COP
DIFFUSER LENGTH = $(D_1-D_2)/\tan\theta$
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

Fig Experimental Set Up (Refrigeration Test Rig)

Fig Refrigeration Test Rig

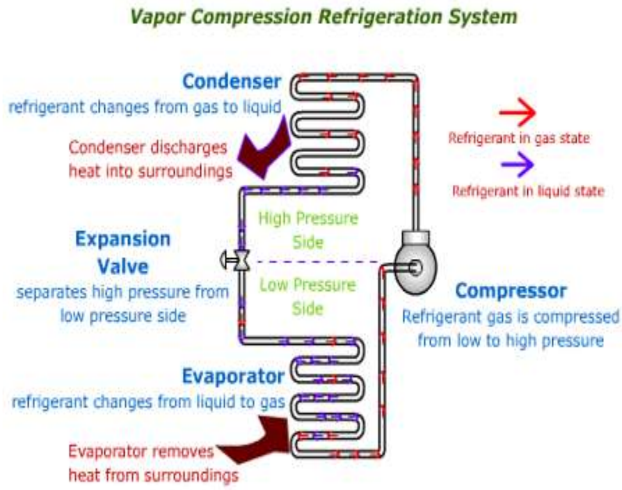


Fig Line Diagram of Vapour Compression Refrigeration System

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
T_1 = Temperature at Compressor Inlet, (^oC)
T_2 = Temperature at Compressor Outlet , (^oC)
T_3 = Temperature at Condenser Outlet , (^oC)
T_4 = Temperature at Evaporator Inlet , (^oC)
P_1 = Upstream pressure of the Compressor (Kg/cm^2)
P_2 = Downstream pressure of the Compressor (lb/in^2)
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

Exp	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

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 X C_p X dT
= 0.00448 X 0.958 X (8-(-6.2) = 0.06094 KW = 60.94 W

Heat Flux = Q/A = $\frac{60.94}{\pi X\ 0.011\ X\ 1.539}$ **= 1146 W/m²**

EXPERIMENT 2.

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 = 706.1 KPa saturation temp. is t_{s2} = 26.976 °C. But observed temperature is 61.2°C. Therefore the condition of the refrigerant after compression is Superheat.
$H_2 = h_{g2} + c_p (t_{sup}-t_{s2}) = 265.232 + 0.958 (61.2- 26.976)$ = **298.02 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) / (298.02 – 265.35)
= 5.39

EXPERIMENT 3.

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 = 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

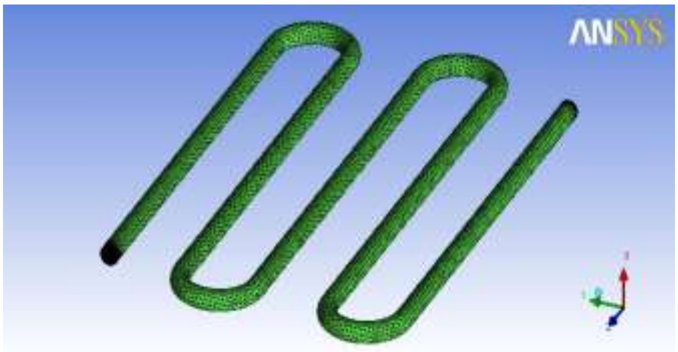
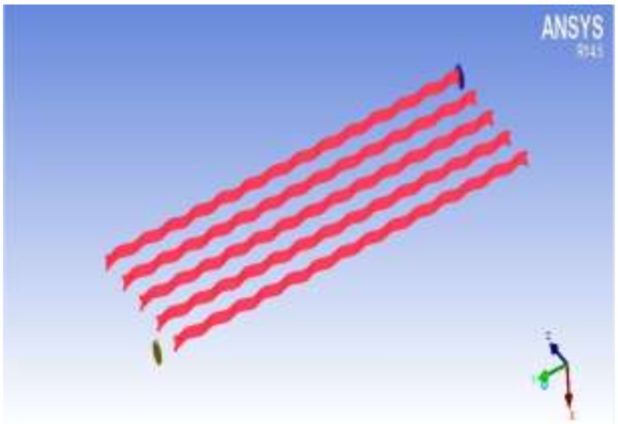
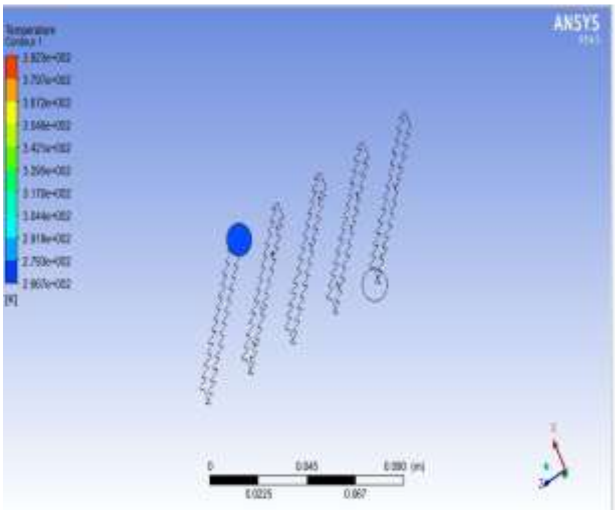


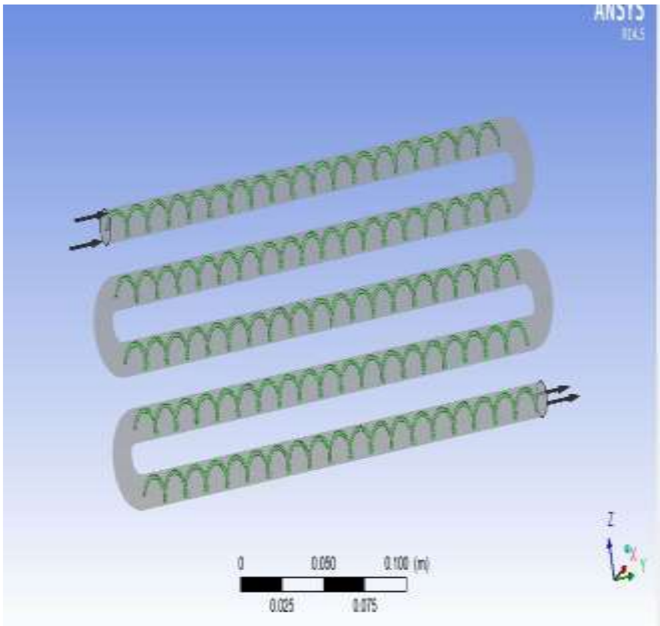
Fig Modeling and meshing



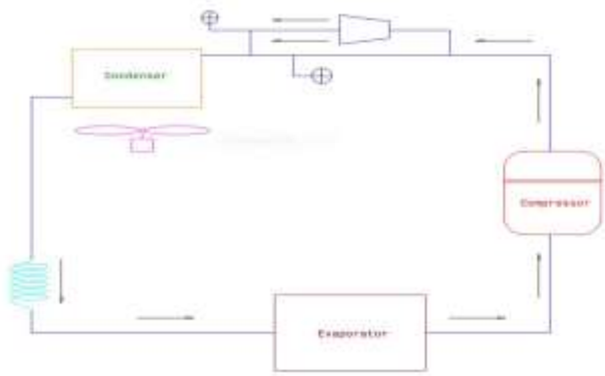
STRIP MODEL PROVIDED INSIDE ICEM CFD



TEMPERATURE INLET



Thread model


Vapour- Compression Refrigeration Cycle Along With Diffuser Block Diagram

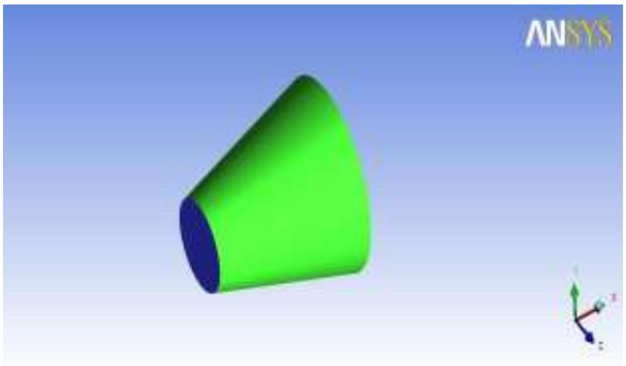


Fig. Diffuser Model

<u>WITH DIFFUSER</u>

For P_1 = 137.89 KPa saturation temp. is $t_{s1} = -19.15^oC$. But observed temperature is 8^oC. 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 ^oC. But observed temperature is 60.4 ^oC. 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

<u>WITHOUT DIFFUSER</u>

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 =

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

= 0.31 = 31 %

CONCLUSION

From above investigation taking after conclusion has been arrived.

1. With the expansion of diffuser weight bay of refrigerant to the condenser is expanded.
2. Work contribution to the compressor is diminished
3. Since work contribution to the compressor diminished Coefficient of Performance (COP) of cooler has been expanded.
4. With presenting aloof strategy Coefficient of Performance (COP) upgrade has been expanded by 24.44 % when contrasted and try.
5. Finally conclusion drawn is Coefficient of Performance (COP) can be upgraded by receiving latent strategy

References

1. Amit prakash “TO IMPROVE THE PERFORMANCE OF VAPOUR COMPRESSION REFRIGERATION SYSTEM BY USING SUBCOOLING AND DIFFUSER” IJEBEA 13-129 2013
2. R. Rejikumar et.al “ENHANCEMENT OF HEAT TRANSFER IN DOMESTIC REFRIGERATOR USING R600a/mineral oil/ nano-AL2O3 AS WORKING FLUID” IJCETR APRIL 2013 VOL 03,ISSUE 04.
3. M. Krishna Prasanna M.E. student, and P.S Kishore “ENHANCEMENT OF COEFFICIENT OF PERFORMANCE (COP) IN VAPOUR COMPRESSION REFRIGERATION SYSTEM”
4. J.K.Dabas et.al “PERFORMANCE CHARACTERISTICS OF VAPOUR COMPRESSION REFRIGERATION SYSTEM UNDER REAL TRANSIENT CONDITIONS”
5. Lucia VILCEANU and Mihaela FLORI “PERFORMANCE CHARACTERISTICS OF VAPOUR-COMPRESSION SYSTEMS”
6. R.T.Saudagar and Dr . U.S. Wankhede “INTRODUCING DIFFUSER AT COMPRESSOR INLET IN A VAPOUR COMPRESSION REFRIGERATION SYSTEM ”
7. P.Thangavel “ SIMULATION ANALYSIS OF COMPRESSION REFRIGERATION CYCLE WITH DIFFERENT REFRIGERANTS”
8. A . Baskaran and P.Koshy Mathews “THERMAL ANALYSIS OF VAPOUR COMPRESSION REFRIGERATION SYSTEM WITH R152a and its blends R429A, R430A, R431A, and R435A”

