Design and Simulation of an Ejector as an Expansion Device for Constant Speed Air Conditioner

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Abstract - This paper is a part in a series that reports on the theoretical study of the performance of the two-phase ejector expansion refrigeration cycle. In the present study, Ejector is design and simulate it and after simulation on different geometries suggesting practical design of constant area ejector and used as an expansion device in the refrigeration cycle. The effects of entrainment ratio (ω) on the throat diameter of the motive nozzle, on the coefficient of performance, primary mass flow rate of the refrigerant, secondary mass flow rate of the refrigerant, average evaporator pressure, discharge temperature and cooling capacity, Use of a two-phase flow ejector as an expansion device in vapor compression refrigeration systems is one of the efficient ways to enhance its performance. This present work aims to design a constant-area two phase flow ejector and to evaluate performance characteristics of the ejector expansion refrigeration system working with R22. In order to achieve these objectives, a simulation program is developed and effects of operating conditions and ejector internal efficiencies on the system performance are investigated using MATLAB software. Finally, correlations to size ejector main diameters as a function of operating conditions, system cooling capacity and ejector internal efficiencies are reported.

Keywords — Air Conditioning, Constant Area mixing, ejector Ejector Design, Ejector Simulation, Refrigeration, Geometry: Motive Nozzle, R22, MATLAB.

I. INTRODUCTION

Throttling loss is one of the thermodynamic losses in a conventional vapor compression refrigeration cycle. In order to reduce this loss, various devices and techniques have been attempted to use instead. Ejector is a device that uses a high-pressure fluid to pump a low-pressure fluid to a higher pressure at a diffuser outlet. Its low cost, no moving parts and ability to handle two-phase flow without damage make it attractive for being the expansion device in the refrigeration system.

The application of an ejector as an expansion device in

a refrigeration cycle has received comparatively little attention in literature. The productive studies have been continuously carried out by Kornhauser [1], Harrell and Kornhauser[2] and Menegay and Kornhauser [3].

Kornhauser [1] investigated the thermodynamic performance of the ejector expansion refrigeration cycle using R-12 as a refrigerant. He found that a theoretical COP could be improved up to 21% over the standard cycle at the evaporator temperature of_15C and the condenser temperature of 30C. This result is based on ideal cycle components and constant mixing pressure in the ejector. Harrell and Kornhauser [2] tested a two-phase ejector and estimated the COP of the refrigeration cycle using R-134a as a refrigerant. It was found that the COP improvement ranged from 3.9 to 7.6%. Menegay and Kornhauser [3] used a bubbly flow tube to reduce the thermodynamic non-equilibrium in the motive nozzle. An ejector using the bubbly flow tube which was installed upstream of the motive nozzle, improved up to 3.8% of the COP over the conventional cycle under standard conditions with R-12 as the refrigerant. However, this result was not as good as they expected. Moreover, Domanski[4] showed that the theoretical COP of the ejector expansion refrigeration cycle was very sensitive to the ejector efficiency.

Nakagawa and Takeuchi [5] concluded that the longer divergent part provided a longer period of time for the two-phase flow to achieve equilibrium. With this consequence, the longer length of the divergent part of the motive nozzle gave the higher motive nozzle efficiency.

It can also be noted that the effects of geometric parameters on the performance of the refrigeration system remain unstudied.

II. SYSTEM DESCRIPTION AND ANALYSIS

2.1 System Description

Throttling devices is used to reduce the refrigerant condensing pressure (high pressure) to the evaporating pressure (low pressure) by a throttling operation [11]. The main difference between the VCRS cycle and the EERC is that reducing of pressure and temperature in EERC is accomplished with the ejector instead of an expansion valve. Other components in the cycle are the same for both refrigeration cycles. A schematic view of the EERC and P– h diagram for the cycle are shown in Figure 1 shows

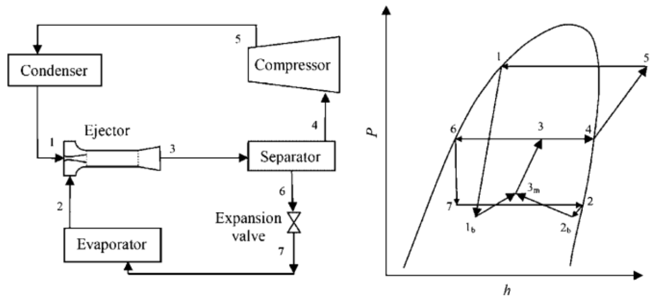

Figure 1. A schematic of the ejector expansion refrigeration cycle (EERC) and P– h diagram.

2.2 Difference Between Conventional and Ejector system

- For Expansion of refrigerant in the conventional Air-conditioning it uses capillary tube. In case of this research topic we'll replace this capillary with the constant area mixing ejector.
- Beside elimination of expansion device it requires to decide entrainment ratio for estimation primary and secondary flows in the ejector.

2.3 Various applications of ejector

To improve the overall performance of the air conditioning and refrigeration system there are number of opportunities, some of them are listed below:

1. Ejector for utilizing of low-grade energy
2. Ejector for recovery of expansion work
3. Ejector for driving liquid recirculation through the evaporator using expansion work.
4. Ejector for increasing the compressor discharge pressure.

3 EJECTOR ANALYSIS

The ejector expansion vapor compression refrigeration cycle has been modeled based on following conservation

laws and equations:

- Conservation of mass,
- Conservation of momentum,
- Conservation of energy

The following assumptions have been used to simplify the theoretical model and set up the equations per unit total ejector flow,

1. The refrigerant will be at all times in thermodynamic quasi-equilibrium.

2. There is negligible Pressure Drop

3. No wall Friction.

4. Steady state one-dimensional model.

5. Thermodynamic processes in compressor, expansion valve and ejector area assumed to be adiabatic

6. No heat transfer with the surrounding for the system except in the condenser.

7. The flow across the throttle valve is isenthalpic.

8. The refrigerant condition at the evaporator outlet is saturated vapour and condenser outlet is saturated liquid.

9. The vapour condition from the separator is saturated vapour and the liquid coming from the separator is saturated liquid.

3.1 Motive Nozzle

Ejector plays very important role as it recovers the expansion process loss. In the this section, the conservation equations of mass, energy and momentum are successively applied to each element of the primary, secondary and mixed fluids.

By using the definition of motive nozzle's isentropic efficiency (η_{mn}), the specific enthalpy of the primary fluid at the nozzle exit (h_{1b}) is given by the following expression;

$$h_{1b} = h_1(1-\eta_{mn}) + \eta_{mn}h_{1b,is} \quad (1)$$

$h_{1b,is}$ is the specific enthalpy of the motive stream at the

end of the isentropic expansion process. Energy equation for motive nozzle, the velocity can be calculated as,

$$u_{1b} = [2(h_{1b} - h_1)]^{0.5} \quad (2)$$

At the exit of the motive nozzle mass flux can be calculated as:

$$G_{1b} = \rho_{1b}u_{1b} \quad (3)$$

ρ_{1b} is fluid density at a motive nozzle exit where ρ_{1b} is function of back pressure.

Inlet constant area stream at the motive can be calculated from principle of conservation of mass is given by,

$$a_{1b} = \frac{\dot{m}_{tot}}{(\rho_{1b}u_{1b})(1+\omega)} \quad (4)$$

Ratio of the secondary mass flow rate $\dot{m}_s$ and primary mass flow rate $\dot{m}_p$ is known as ω and $\dot{m}_{tot}$ is the total mass flow. Flow rate through the motive nozzle, condenser and compressor is the mass flow rate and can be calculated as;

$$\dot{m}_p = \frac{\dot{m}_{tot}}{1+\omega} \quad (5)$$

3.2 Suction Nozzle

Equations for the suction nozzle exit which is similar to the motive nozzle and in the similar way as motive nozzle;

$$h_{2b}=h_2(1-\eta_{sn})+\eta_{sn}h_{2b,is} \quad (6)$$

$$u_{2b}=[2(h_2-h_{2b})]^{0.5} \quad (7)$$

$$G_{2b}=\rho_{2b}u_{2b} \quad (8)$$

$$a_{2b}=\frac{\omega m_{tot}v_{2b}}{(u_{2b})(1+\omega)} \quad (9)$$

$$\dot{m}_s=\frac{\omega\dot{m}_{tot}}{1+\omega}=\frac{Q_{eve}}{(h_2-h_7)} \quad (10)$$

Specific enthalpy h_2 of the suction stream at the end of the isentropic expansion process and suction nozzle which depends on the isentropic efficiency η_{sn} and the mass flow rate through the suction nozzle is secondary mass flow rate and evaporator.

3.3 Constant area mixing section

Mass flux (G_{3m}) of mixed stream at the exit of the mixing chamber (h_{3m}) and the velocity (u_{3m}) calculated by applying conservation of momentum following equations are obtained:

$$u_{3m}=\frac{P_b(a_{1b}+a_{2b})}{\dot{m}_{tot}}+\frac{u_{1b}}{1+\omega}+\frac{u_{2b}\omega}{1+\omega}-\frac{p_{3m}a_{3m}}{\dot{m}_{tot}} \quad (11)$$

$$G_{3m}=\rho_{3m}u_{3m} \quad (12)$$

Specific enthalpy of mixed stream at the exit of the chamber can be calculated by applying principle of conservation of energy;

$$h_{3m}=\frac{h_1+\omega h_2}{1+\omega}-\frac{{u_{3m}}^2}{2} \quad (13)$$

Total mass flow rate at the exit of a constant area mixing section, Eq(14) should be verified by (Li and Groll, 2005);

$$a_{3m}\rho_{3m}u_{3m}=\dot{m}_{tot} \quad (14)$$

3.4 Diffuser

Principle of conservation of energy helps to find out the

Specific enthalpy of the stream at the diffuser exit (h_3) through out the ejector;

$$h_3=\frac{h_{1+}\omega h_2}{1+\omega} \quad (15)$$

Isentropic specific enthalpy at the exit $(h_{3,is})$ is given by;

$$h_{3,is}=\eta_d(h_3-h_{3m})+h_{3m} \quad (16)$$

Isentropic efficiency (η_d) of the diffuser and quality of the refrigerant at the diffuser exit (x_3) is estimated at the diffuser exit pressure (p_3) and specific enthalpy (h_3).

To maintain required continuity for cycle, stream getting out of the ejector Eq. (17) should be approved (Deng et al., 2007);

$$x_3=\frac{1}{1+\omega} \quad (17)$$

3.5 Performance characteristics of EERS

Ejector efficiency (η_{ej}), COP, entrainment ratio (ω), compressor power, cooling capacity are the main performance characteristics of EERS.

Cooling capacity is;

$$Q_{eve}=\dot{m}_s(h_2-h_7) \quad (18)$$

$$P_{comp}=\frac{\dot{m}_p(h_{5b,is-}h_4)}{\eta_{comp}} \quad (19)$$

$$COP=\frac{Q_{eve}}{P_{comp}}=\omega\frac{h_2-h_7}{h_5-h_4} \quad (20)$$

$$\omega=\frac{\dot{m}_s}{\dot{m}_p} \quad (21)$$

$$\eta_{ej}=\omega\frac{h_C-h_2}{h_A-h_B} \quad (22)$$

3.6 Ejector Design

In two phase flow ejector motive nozzle, suction nozzle and diffuser are the main parts. Objective of the ejector design is to determine all diameters and length of these sections this ejector.

By using Henry and Fauske model (Henry and Fauske, 1971) we will determine the motive nozzle throat diameter is designed. Mathematical model of Henry and Fauske is consistently used in various experimental works (Disawas an Wongwise, 2004; Wongwise and Disaeas, 2005; chaiwongsa and Wongwises, 2008)it always better to consider metastable effect in throat diameters of nozzle which motive nozzle in this case this results finally as delayed flashing of refrigerant (R22) primary flow which is rooted through motive nozzle. For remaining design part of the ejector homogeneous equilibrium model (Sherif et al., 2000) is applied.

Critical mass flux for the mixture flow can be calculated by application of Henry and Fauske model

$$(1-x_o)v_{lo}(p_{0-}p_t)+\frac{x_o\gamma}{\gamma-1}(p_0v_{vo}-p_tv_{vt})=\frac{[(1-x_o)v_{lo}+x_ov_{vt}]^2}{2}G_c^2 \quad (23)$$

Different inputs which is required for Henry and Fauske model quality of refrigerant (x_0) and stagnation pressure (p_0), as mentioned by Henry and Fauske (1971). Condenser exit is referred as stagnation condition. That's why Henry and Fauske model is completely depends on condenser pressure.

Throat diameter of the motive nozzle($D_{mn,t}$), Motive nozzle exit diameter ($D_{mn,e}$), suction nozzle exit diameter ($D_{sn,e}$) and mixing section diameter (D_{ms}) can find out by following Eq. (24-27);

$$D_{mn,e} = \sqrt{\frac{4\dot{m}_p}{\pi G_{1b}}} \tag{24}$$

$$D_{sn,e} = \sqrt{\frac{4\dot{m}_s}{\pi G_{2b}}} \tag{25}$$

$$D_{ms} = \sqrt{\frac{4\dot{m}_{tot}}{\pi G_{3m}}} \tag{26}$$

$$D_{mn,t} = \sqrt{\frac{4\dot{m}_p}{\pi G_c}} \tag{27}$$

A simulation program based on Eqs (1) – (27) is developed using MTLAB (based on the approach by M. Hassanain et.al, 2015) sizing and performance characteristics of ejector expansion refrigeration system fig. shows the flow chart of the simulation program.

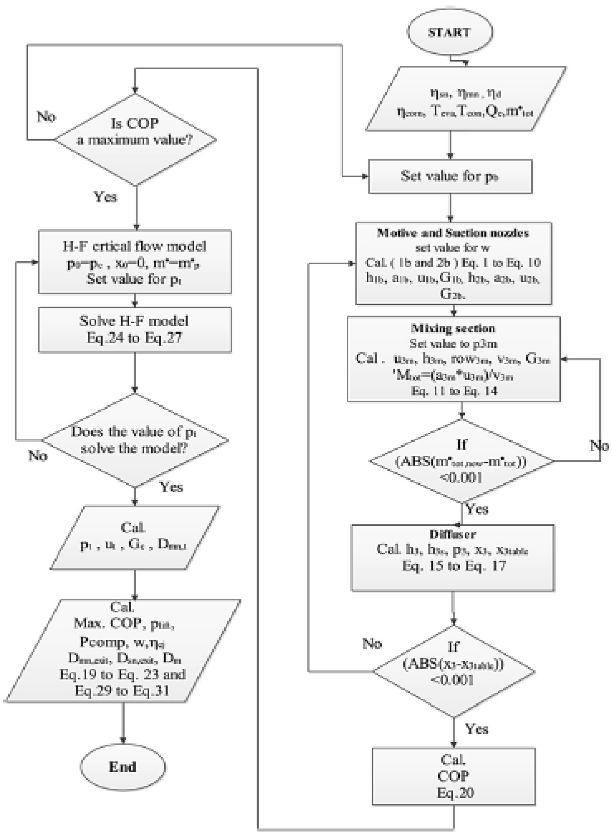

Fig. 3 Flow chart of the simulation programe

4 RESULTS AND DISCUSSIONS

First of all we need to calculate the ejector model by analytical method then this values is compared with simulation output from the program which is in the MATLAB value shows that analytical calculation is matching with the simulation hence, we can use trial and error method to find out the optimum dimensions of the ejector by using the program.

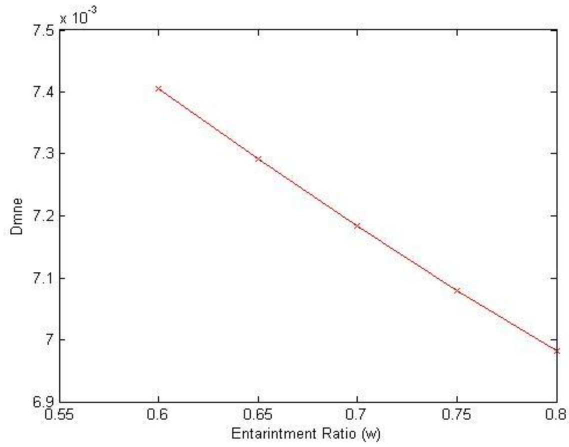

Fig 2 $D_{mn,e}$ Vs Entrainment Ratio (ω)

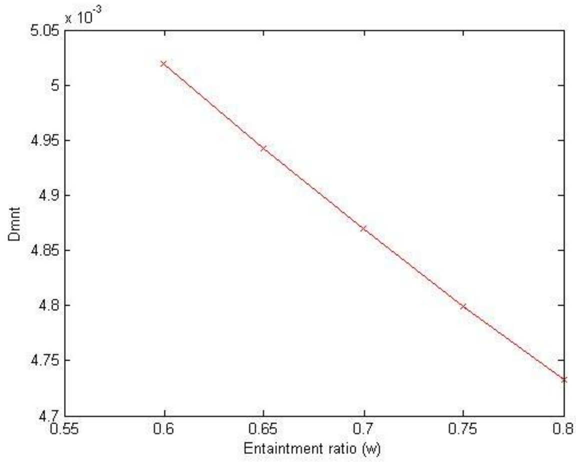

Fig 3 $D_{mn,t}$ Vs Entrainment Ratio (ω)

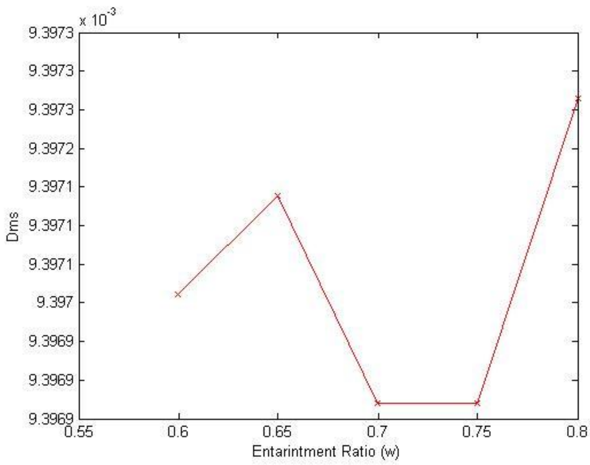

Fig 4 D_{ms} Vs Entrainment Ratio (ω)

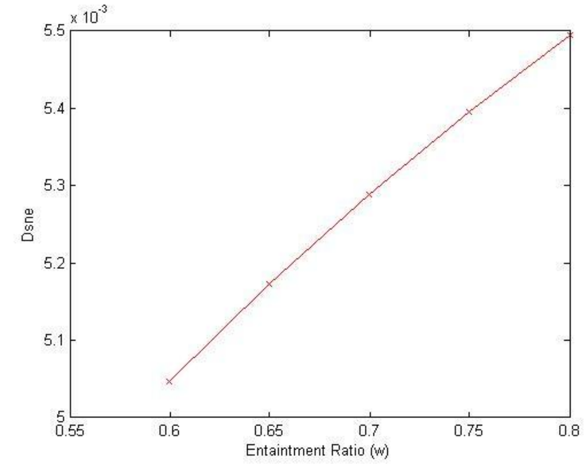

Fig 5 $D_{sn,e}$ Vs Entrainment Ratio (ω)

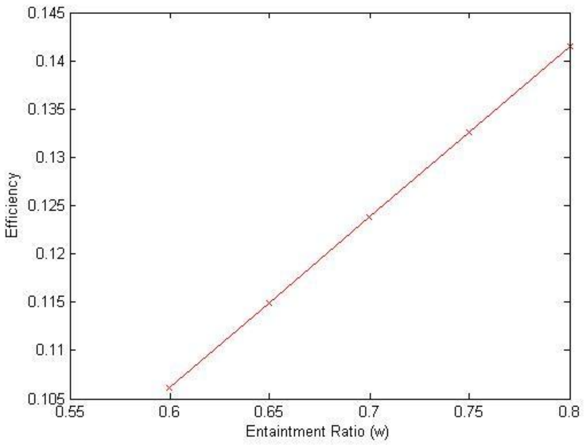

Fig 6 Efficiency Vs Entrainment Ratio (ω)

In constant area ejector there are some dimensions are essential based on that, we can decide the other geometrical aspects of the ejector. $D_{mn,e}, D_{ms}, D_{mn,t,} D_{sn,e}$ are the parameters based we are going decide the geometry of the ejector and lengths of it.

While, if we observed the eq. (24-27) then we realize this equations depends on the $\dot{m}_{tot}, \dot{m}_s, \dot{m}_p$ which is depend on the entrainment ratio (ω) decided by the designer hence we are varying this parameter and for reference we have demonstrate the graphs (Fig 2-6) at the rated capacity 5.5 KW at the constant speed of air conditioner.

There is one thing common in all the graphs that, as ω is decreasing concern diameters are increasing expect in case of D_{ms} the reason behind is when $\dot{m}_s, \dot{m}_p$ is mixed in the suction chamber and collectively enter in mixing section of the ejector which archives its total mass flow rate.

After applying the iterations in the program we can find the effective diameters at the different capacities, here we are discussing about the 5.5 KW capacity system and results is as according.

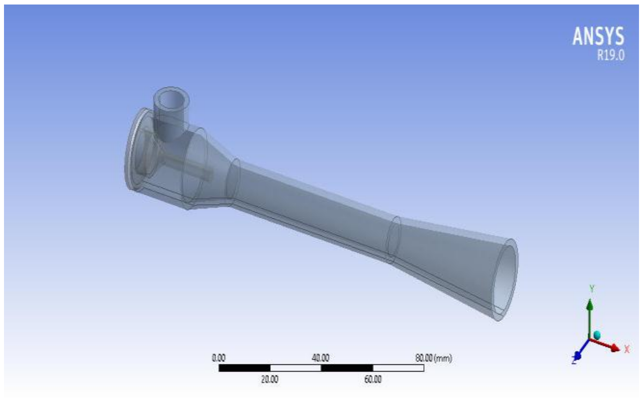

Fig 7. Schematic for Ejector

5 RESULTS AND DISCUSSIONS

In FLUENT we are going to simulating the model for required as well as extreme conditions by observing the results in FLUENT that shows that there is no substantial change in the pressure contours and velocity vectors, hence we decide to our focus on extreme conditions so we can articulate the results for some reliable results.

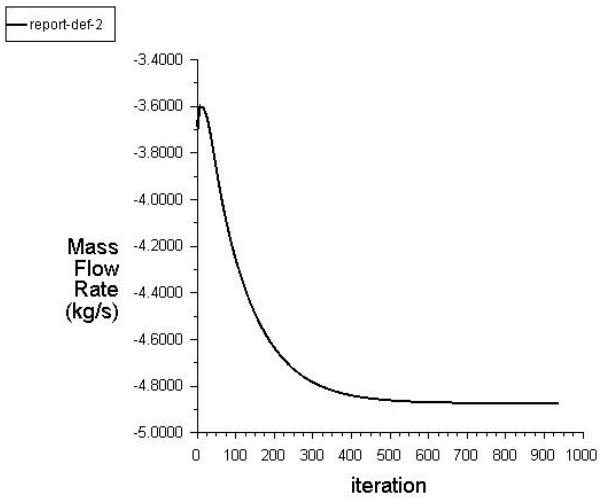

Fig 8. Mass Inlet

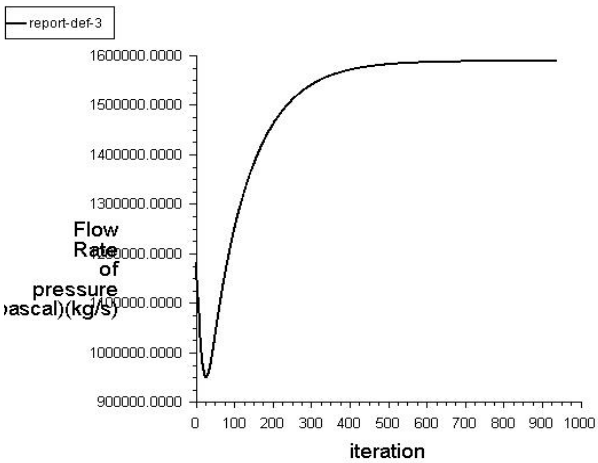

Fig 9. Mass 2nd Inlet

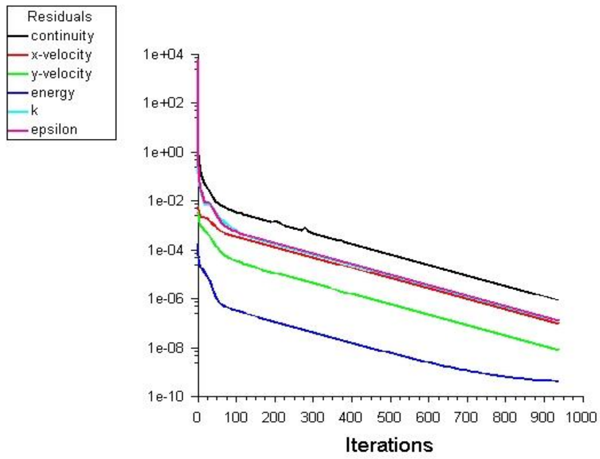

Fig 10. Residuals

Fig 8-10 shows mass flow rate graphs that passing through model which assures us that mass flow rate mixing at the suction chamber and getting mixed and at the outlet of the model we are getting total mass flow rate.

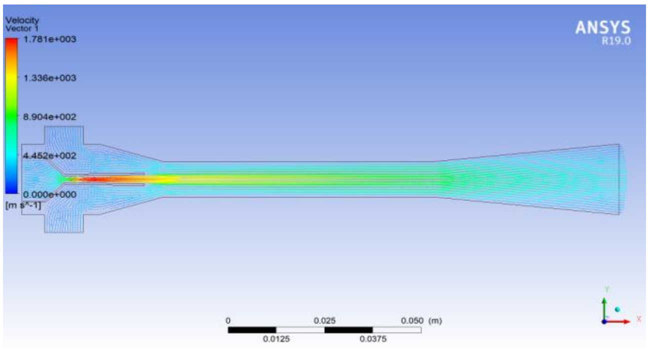

Fig 11. Velocity vectors

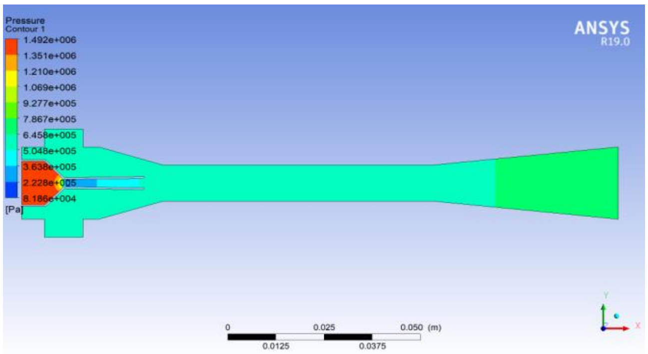

Fig 12. Pressure Contours

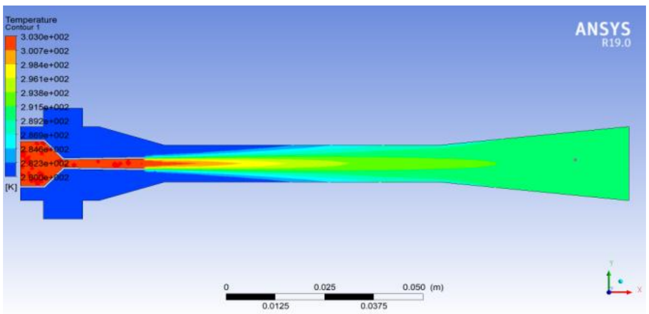

Fig 13. Temperature Contours

Fig 11 shows the velocity vectors for model, usually when the high pressure high temperature enters into the motive nozzle whilst from the second inlet takes the low pressure and low temperature, it is little unpredictable so, there is chances that flow can get reversed because tendency of high pressure to flow towards the low side, in the fig 11 we can se that flow is not getting revers.

Fig 12 showing information about pressure contours and we are getting desired pressure drop of 9- 12 bars.

Fig 13 articulating the information about temperature variation we are getting temperature drop of 323 K to 291 K.

6 CONCLUSION

In the present paper, the numerical results for 5.5 kW cooling capacity ejectors with variable area ratio are presented using R22 as working fluids. Working fluids were selected based on the criteria of low environmental impact and good performance in the range of operating primary nozzle. Simulation results were obtained with a CFD model using FLUENT. The results indicated that using a fixed geometry ejector is very sensitive to variations in operating conditions.

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