Analysis and Design of ejector freezing system

Alapati Venkateswarlu[1*], A. Adilakshmi[2] & V.K. Sharma[3]

[1]Professor, Mechanical, VR Siddhartha College of Engineering, Vijayawada, India.
[2]Professor, PVP Siddhartha Institute of Technology, Vijayawada, India.
[3]Professor and Vice Chancellor, Bhagawanth University, Ajmeer, India.
**Corresponding author E-mail:alapativ.l1@gmail.com*

Abstract

The heat driven ejector refrigeration framework offers the benefit of straightforwardness and can work from low-temperature heat vitality sources. There by, it turns out to be a good substitute to the conventional compressor-driven refrigeration system when any or a combination of heat sources is available. Chillers may speak to a contender for absorption chillers, when their cost per unit cooling power ends up equivalent or lower. This objective isn't a long way from our present accomplishments. If input energy is waste or renewable heat, the system operating cost is mainly due to the investment in heat exchangers .Therefore, a cost reduction requires an increase of COP. This last might be enhanced by a careful design of the ejector, which requires a profound understanding into the thermodynamics and fluid flow and their complex physics. The devices and the learning for a propelled configuration are as of now accessible and the change potential is huge. Since ejector utilizes low grade thermal energy for refrigeration, so in this way as we have high thermal energy rejection in automobiles exhaust so this thermal energy can be used as heat input to the generator. By doing this the extra power developed by engine to run compressor of A/C can be eliminated and which in turn reduces the fuel consumption. here the whole performance of system depends on the efficient design of the ejector (thermocompressor). The refrigerant gas used here is R134A,R410A for which the gamma (c_p/c_v) value is 1.292 & according to this the ratio of inlet pressure(p_o) to the exit pressure(p_e) i.ep_o/p_e=0.562 therefore for different inlet pressure & back pressure values the primary motive nozzle is designed whether it should be convergent or convergent-divergent. In the same way the generator, condenser, refrigerant liquid pump capacities are calculated. If the suction pressure of ejector is high then the refrigerant in the evaporator evaporates at low temperature (t_e) therebywe can get more COP.

Keywords: Ejector, freezing system,renewable heat, Mixing Chamber, CFD, R134A, R410A.

1. Introduction

Over the past few years, the concerning on energy saving and environment protecting has become an increasingly predominant businesses and increment in populace have caused an awesome request of cooling and refrigeration application which utilize the mechanical vapor compression system frameworks. These systems are generally powered by power created by consuming fossil fuels.

The fossil fuel consumption can contribute to the global warming–the “greenhouse effect”. This compels more and more researchers turn to make utilization of low grade thermal energy The ejector refrigeration framework gives a promising method for delivering a cooling impact by using waste heat from modern process and inside burning motor or utilizing sustainable power source, for example, sunlight based power and geothermal vitality. With the improvement of innovation, these frameworks can work utilizing low- temperature warm Source beneath 100 °C and evenless.

Numerous hypothetical and exploratory investigations of the ejector refrigeration frameworks are introduced in writings for different working liquids, including R113, R123 , R134a , R141b , R152a , R245fa, R290 , R600 , R600a, and ammonia. Here in this project I am using R410A gas as working medium and automobile engine exhaust as a heat source for generator.

2.Design and Methodology

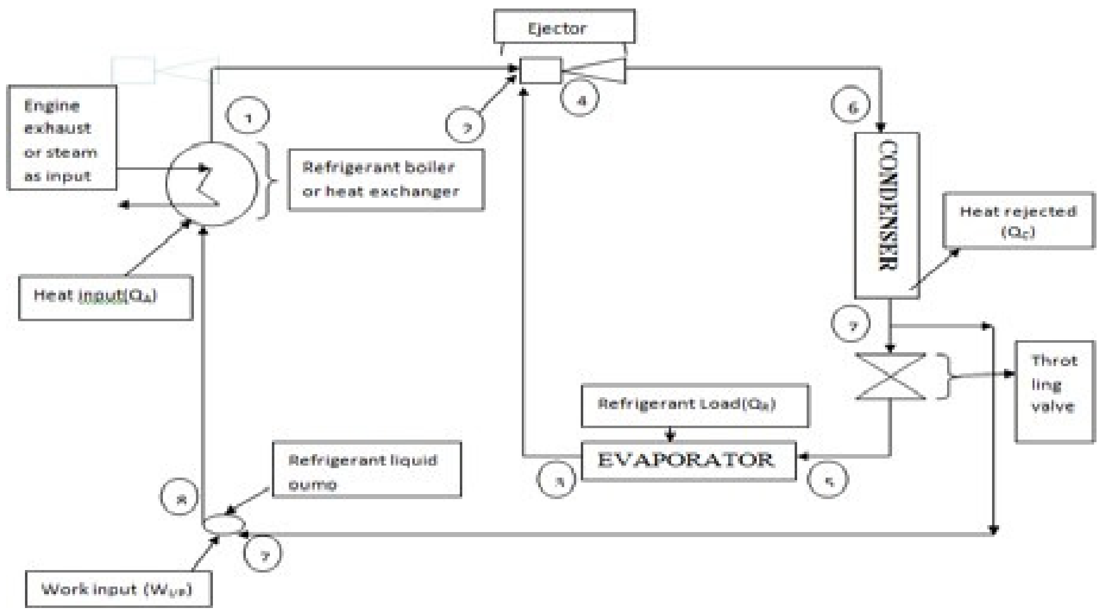


Figure 1: Schematic model of ejector freezing system

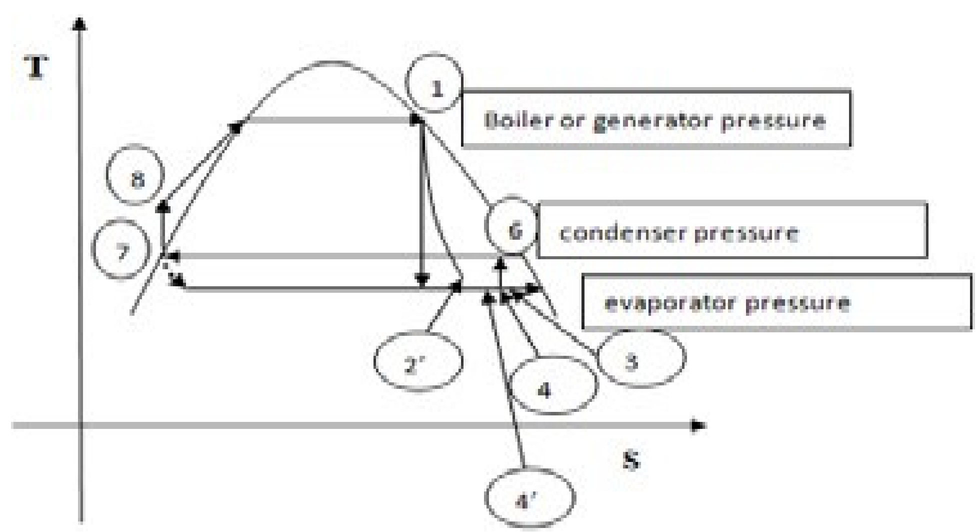


Figure 2 :T-S diagram of ejector freezing system

2.1 Various thermodynamic processes in a system:

1-2 : Isentropic expansion in ejector nozzle.
1-2' : Actual expansion in ejector nozzle.
7-8 : Refrigerant pump work.
8-1 : Constant pressure heat addition in boiler.
4-6 : Isentropic compression in diffuser part of ejector.
4' : Condition of refrigerant vapour before just mixing with motive refrigerant.
4 : Condition of mixture of high velocity refrigerant from nozzle & the entrained Refrigerant before compression
3 : Condition of refrigerant vapour at exit of evaporator.
6-7 : Condensation process in condenser.
7-5 : Throttling process

2.2 Problem description in a system:

Calculate the necessary preliminary calculations for a ejector refrigeration system for 1.5 TR Capacity and 58^0C generator temperature & 5^0C evaporator temperature with R134A gas as a refrigerant (take $\eta_{\text{nozzle}} = 90\%$, $\eta_{\text{entrainment}} = 63\%$, $\eta_{\text{diffuser}} = 78\%$) and quality of vapour at beginning of compression is $x_4 = 0.94$

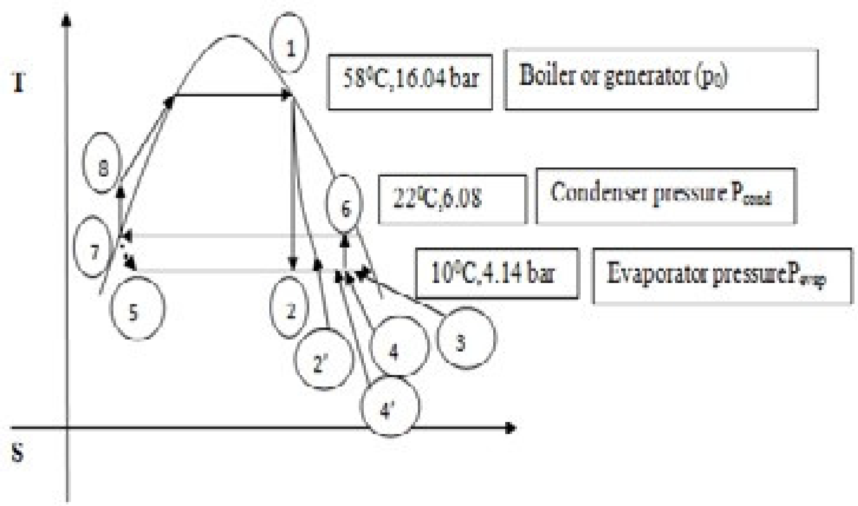


Figure 3: T-S diagram of ejector freezing system with temperature points

2.3 From R134A tables we have

For 58^0C sat pressure is 16.04 bar.
For 10^0C sat pressure is 4.14 bar.

From process 1-2

$s_1 = s_2 = s_g@50^0C = 1.6873$

$1.705 = s_{f2} + x_2 s_{fg2}$

$1.705 = 1.048 + x_2 0.7244$

$x_2 = 0.906$

From process 1-2 nozzle expansion we have

$h_1 - h_2 = \frac{(v_2)^2}{2}$----------- 1

From data book we have $h_1 = h_g = 426.5$ kJ/kg

From data book and diagram

$h_2 = h_{f2} + x_2 h_{fg}$

$= 213.6 + 0.906(191)$

$h_2 = 386.646$

From eq1 we have

$v_2 = 399.2$ m/sec

From velocity of sound we have

$a = \sqrt{\Upsilon RT}$

<for R134A gas at 58^0C $\Upsilon = 1.19$

R = 0.0815 kJ/kg-k>

a = 180 m\sec

Mach number $(M_a) = \frac{v}{a}$

$M_a = 2.217$ (supersonic)

From the relation of isentropic flow through nozzle we have

$$\frac{p_0}{P} = \left[1 + \frac{\Upsilon - 1}{2} \times M_a{}^2\right]^{\frac{\Upsilon}{\Upsilon - 1}}$$

On substituting values we get

$$\frac{p_0}{P} = 1.7793$$

P = 9.0909 bar (Critical pressure)

From assumption we can take

$P_{cond} = 6.08$ bar @ Tsat = 22^0C

$P_{evap} = 4.14$ bar @ Tsat = 10^0C

For process 7- 8 we have

$h_7 - h_8 = -\int v\,dp < v_{sat}$@$30^o$ c = 0.008m^3/kg

$230.4 - h_8 = -\int 0.0008(16.04 - 6.08)$ $h_7 = h_f$@ 30^o c = 230.4 kJ/kg >

$h_8 = 232$ kJ/kg For process 8-1 we have heat addition

$Q_{added} = h_1 - h_8$

= 426.8-232

= 194.8 kJ/kg

Now finding the dryness fractions at locations of 2',4,6,3

$$\eta_{nozzle} = \frac{h_1 - h_2'}{h_1 - h_2}$$

$$0.90 = \frac{426.5 - h_2'}{426.5 - 386.646}$$

$h_2' = 390.6314$ kJ/kg

But $h_2 = h_{f2} + x_2 h_{fg2}$

$390.631 = 213.6 + x_2\,(191)$

$x_2 = 0.906$

similarly

$\eta_{entrainment} = \frac{h_1 - h_4}{h_1 - h_2'}$

$$0.63 = \frac{426.5 - h4}{426.5 - 390.63} \quad h_4 = 403.902 \text{ kJ/kg}$$

but $h_4 = h_{f4} + x_4 h_{fg4}$

$407.45 = 213.6 + x_4\ 191$

$x_4 = 0.986$

Given $x_4' = 0.99$

So $h_4 = h_{f4} + x_4' h_{fg4}$

$= 213.6 + 0.99(191)$

$h_4' = 402.69$ kJ/kg

Dryness fraction after isentropic compression x_6 is

$s_4 = s_{f4} + x_4' s_{fg4}$

$= 1.0485 + 0.94(1.7229 - 1.0485)$

$s_4 = 1.758292$ as $s_4 = s_6$

$1.7182 = s_{f6} + x_6 s_{fg6}$

$1.7182 = 1.048 + x_6(1.7229 - 1.048)$

Ratio of mass of motive fluid (m_s) to mass of vapour produced in evaporator (m_v) is given by

$$\frac{m_s}{m_v} = \frac{(h6 - h5)}{[(h1 - h2) \times \eta nozzle \times \eta ent \times \eta comp - (h6 - h5)]}$$

$$= \frac{411.1 - 402.69}{39.86 * 0.9 * 0.63 * 0.78 - (411.1 - 402.69)]}$$

$$= \frac{8.31}{17.62 - 8.31}$$

$= 0.89$

Quality of vapour from evaporator is given by point 3

Let us find enthalpy at point 3

We know that $M_v h_3 + M_s h_4 = (M_s + M_v) h_5$

$$h_3 + \frac{Ms}{Mv} h4' = \left(\frac{Ms}{Mv} + 1\right) h_4$$

on substituting values we get

$h_3 = 404.97$ kJ/kg

but we know that

$h_3 = h_{f3} + x_3 h_{fg3}$

$404.97 = 213.6 + x_3(191)$

$x_3 = 1$ <Quality of vapour from evaporator is >

Given capacity = 1.5 TR

= 1.5 * 3.56 = 5.34KW

$m_v(h_5 - h_3) = 5.34$

$m_v(404.5 - 213.6) = 5.34$

$m_v = 0.0279$kg/sec<This is nothing but mass flow of refrigerant in evaporator>

We know

$\frac{ms}{mv}$ = 0.899=0.89.

m_s = 0.0248 kg/sec

Coefficient of performance (COP)=$\frac{refregeranteffect}{workinput}$

Now heat added to system is = m_v x 194.8 KW

= 0.0248 x 194.8

= 4.831KW

Therefore COP = $\frac{1.5*3.56}{4.831}$

=$\frac{5.34}{4.831}$

COP = 1.105

Similarly by doing analysis using forR410A Gas we got COP= 0.97

3.Designing Of Primary Nozzle Of Ejector

3.1 From the above calculations we have

Mach number at nozzle exit (M_a) = 0.724

Mass flow rate through evaporator coil (M_V) =0.032 kg/sec.

Therefore mass flow rate through motive nozzle (M_S) is given by

$$\frac{Ms}{Mv} = \frac{(h6 - h4)}{[(h1 - h2) \times \eta nozzle \times \eta ent \times \eta comp - (h6 - h5)]}$$

Where h6 = enthalpy of gas at outlet of ejector = 411.1 kJ/kg

h4 = enthalpy of mixture before compression = 402.69 kJ/kg

h1 = enthalpy of gas at inlet of motive nozzle = 426.5 kJ/kg

h2 = enthalpy of gas at outlet of motive nozzle = 386.64 kJ/kg

And we assumed ηnozzle , ηent, ηcomp as 0.9,0.63,0.78.

Therefore $\frac{Ms}{Mv} = \frac{411.1-402.69}{[(426.5-386.69)\times0.9\times0.63\times0.78-(411.1-402.69)]}$

= 0.89

M_S=0.89 *Mv

= 0.89*0.0274

$\boxed{M_S = 0.024389 \text{ kg/sec}}$

From isentropic flow through nozzles we have (here we are using only convergent nozzle so even though there is supersonic flow we restrict to only sonic so we put Mach number($Ma = 1$).

$$\frac{M_S}{A} = \sqrt{\frac{\Upsilon}{R}} * \frac{P0}{\sqrt{T0}} * \frac{Ma}{\left[1 + \frac{\Upsilon-1}{2} \times Ma^2\right]^{\frac{\Upsilon+1}{2(\Upsilon-1)}}}$$

$$= \sqrt{\frac{1.196}{0.0815 * 1000}} * \frac{16 * 10^5}{\sqrt{331}} * \frac{1}{\left[1 + \frac{1.196-1}{2} \times 1^2\right]^{\frac{1.196+1}{2(1.196-1)}}}$$

$$\frac{M_S}{A} = 4780.198$$

A = 5.248e-6

$$\frac{\pi}{4} D^2 = 5.248\text{e-}6$$

D =2.57mm= D_2

$$\boxed{D_2 = 2.57\text{mm}}$$

(This is out let diameter of converging nozzle)

From continuity equation we have:

ρAV= constant

$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$ where ρ_1 = Density of refrigerant at entryof motive nozzle

A_1 = Area of cross section at entryof motive nozzle

V_1 = Inlet velocity of motive Nozzle (5m/s)

Similarlyρ_2, A_2,V_2 are corresponding parametersat exit

We have

$$\frac{\rho_1}{\rho_2} = \left[1 + \frac{\Upsilon - 1}{2} \times Ma^2\right]^{\frac{1}{\Upsilon-1}}$$

$$=\left[1 + \frac{1.19-1}{2} \times 1^2\right]^{\frac{1}{1.19-1}}$$

$$= 1.6122$$

$1.6122 * A_1 V_1 = A_2 V_2$

$1.6122 * 5 * A_1 = A_2 * 180$

$$\frac{A1}{A2} = 22.329$$

$$\boxed{D_1 = 12.144 \text{ mm}}$$

D_1=Diameter of the nozzle at entrance

3.2 Diameter of Secondary Inlet(SUCTION PIPE)

The pressure at the secondary inlet (suction pipe) is 6bar

We have the formula for pressure is $P = \rho g h$

where

600000 = 82.71*9.81*h ρ=density of fluid = 82.71kg/m^3

V = vel

A = cro

$$\boxed{h = 739.476\ m}$$

We know that velocity of flow

$$V = \sqrt{2gh}$$

$$= \sqrt{2 * 9.81 * 739.476}$$

V = 120.4513 m/s

Therefore from formula of mass flow rate we have

$$M = \rho A V$$

$$0.032 = 82.71 * \frac{\pi}{4} D^2 * 120.4531$$

$$\boxed{D3 = 2.02mm}$$

3.3 Diameter of Mixing Chamber

From continuity equation we have

$\rho_2 A_2 V_2 = \rho_3 A_3 V_3 + \rho_4 A_4 V_4$

Where as we know that $\rho_2 = \rho_3 = \rho_4 \rho_4$= Density of refrigerant inmixing section

Take V_4=100m/s i.e average of nozzle outlet and secondary inlet velocities A_4= Area of cross section ofMixing section

V_4 =Velocity in the mixingsection

Similarly ρ_3, A_3, V_3 are corresponding parameterssecondary inlet

$(2.57 * 10^{-3})^2 * 180 = (2.02*10^{-3})^2*120 + D^2{}_4*100$

$$\boxed{D_4 = 4.0959\ mm}$$

3.4 LengthOfthe Mixing Chamber

From correlation established by R.A.Tyler and R.G.Williamson by experimentation, we have

L_0 (initial mixing length) = $4.2*(1-\bar{D})^{5/3}*(1-\frac{V0}{Vi})^{-1}$

Where V_0 = secondary inlet velocity ofFluid

V_i = velocity of fluid at nozzle

$$L(\text{total length}) = L_0 + \frac{1}{K} * \ln\left[\frac{\beta * K}{2f}\right]$$

$$K = 0.75*(1-\frac{V0}{Vi})^{2.5}*\left[\frac{1}{\bar{D}^2(1-\bar{D}^2)}\right]^{1/6}$$

$L_0 = 4.2*(1-0.3508)^{5/3}*(1-\frac{120}{180})^{-1}\overline{D} = \frac{Di}{D0} = \frac{nozzle\ oulet\ diameter}{mixing\ chamber\ diameter}$

= 6 mm

β = momentum coefficient = 1.01

$K = 0.75*(1-\frac{120}{180})^{2.5}*\left[\frac{1}{0.3508^2(1-0.3508^2)}\right]^{1/6}$ K = Constant

= 0.0585

f = friction factor=$0.184*Re^{-0.2}$

= $0.184 * 2406600^{-0.2}$

$L = 6+\frac{1}{0.0585}*\ln\left|\frac{1.01*0.0585}{2*9.739*10^{\wedge}-3}\right| = 9.739*10^{-3}$

L= 25 mm

Length of the mixing chamber L=25 mm

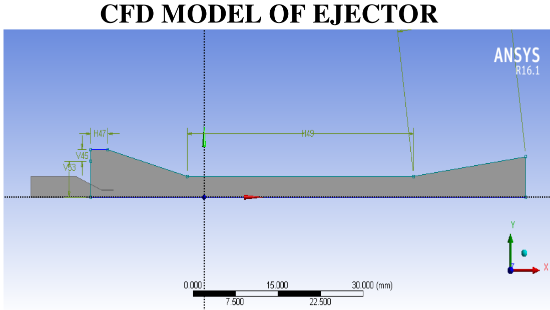
3.5 Calculated DesignParametersOf Ejector

Inlet diameter of motive nozzle(D_1)	12.144 mm
Outlet diameter of motive nozzle(D_2)	2.57 mm
Suction tube diameter(D_3)	2.02 mm
Diameter of mixing section(D_4)	4.0959 mm
Total length of mixing section (L)	25 mm

Table 1

4.1CFD AnalysisOfan Ejector:

CFD MODEL OF EJECTOR



MESHED MODEL OF EJECTOR

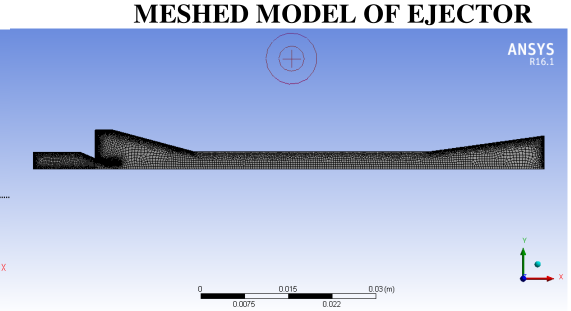


Figure 4a :Diagram of CFD model of ejector

Figure 4b : Diagram of meshed model of ejector

No of elements : 11682

No of nodes : 12615

<u>Boundary conditions</u>

Inlet 1 : pressure in let (16bar,340K)

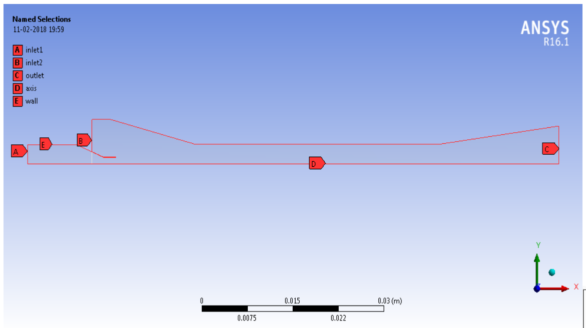
Inlet 2 : pressure inlet (4bar,256K)

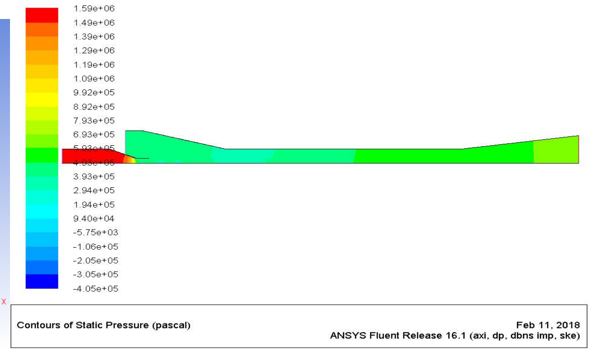
Outlet : pressure out let(6bar,300K)

Wall : wall

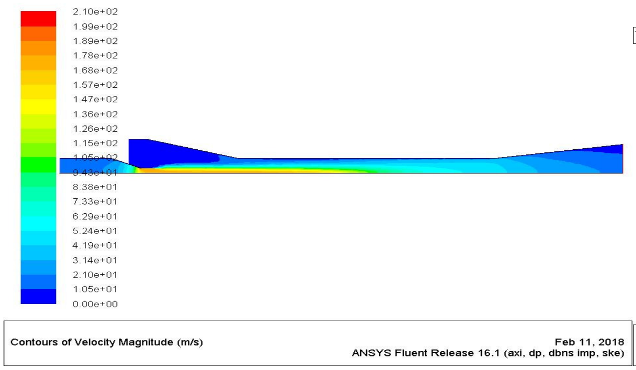
Axis : axis

Viscous K- ε model **Contours of pressure**





Contours of velocity **Pressure vs axial distance of ejector**



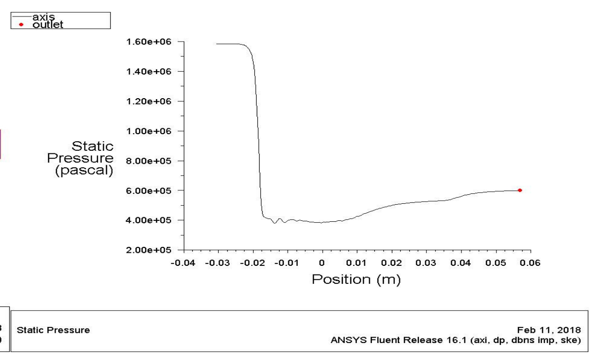


Figure 5 :A .Diagram of Viscous K- ε modelof ejector,**B.** Contours of pressure, C.Contours of velocity D. Pressure vs axial distance of ejector

4.2 Variation of mass flow(M_s) according to percentage change of back pressure(P_b) with respect to secondary inlet pressure(P_s)

S.No	Inlet Pressure1 (bar)	Inlet Pressure 2 (bar)	Outlet pressure (bar)	Primary mass flow(kg/s) (Mp)	Secondary mass flow(kg/s) (Ms)	Entrainment ratio(Ms/Mp)
1	16	4	6	0.0424	0.039	0.928
2	16	4	5	0.0426	0.1056	2.38
3	16	4	3	0.0429	0.2346	5.468
4	16	4	2	0.043	0.2914	6.776

Table 2

4.3 Entrainment ratio(ER) vs(Pb-Ps)/Ps

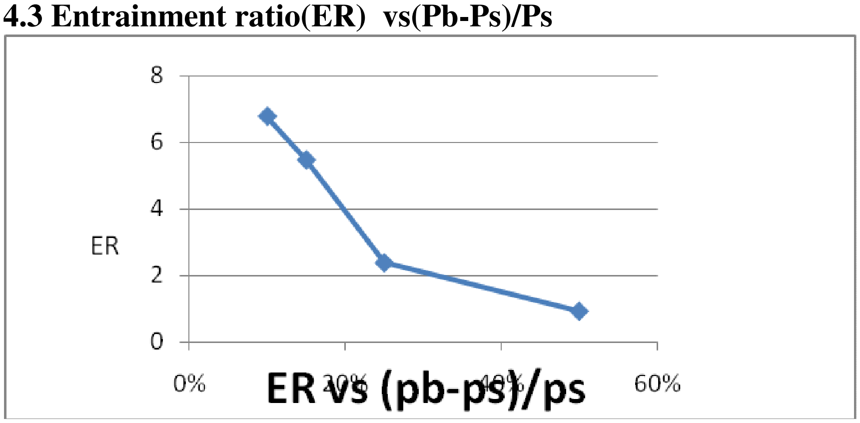


Figure 6 :Graph showsEntrainment ratio(ER) between (P_b-Ps)/Psof an ejector

5. Conclusions:

Chillers are very effective components in the process of refrigeration. While absorption chillers have dominated the market, ejector chillers are also picking up. Most important constraint in choosing the chiller is the cost component. In case of absorption chillers, they have been found to be cost effective. While the ejector chillers performance has been found to be smooth, there has to be cost rationalisation. In any project analysis, cost component plays an important role hence the need for economising the input components.

Once cost rationalization is achieved, the ejector chiller can occupy the market space slowly being vacated by absorption chiller. But for this to be achieved ejector chillers need to become cost competitive through an increase in the system COP, that involves reduced size of heat changers and the plant. Once this happens ejector chillers can make a niche for themselves, in the heat powered refrigeration market.

In the refrigeration process the delivery efficiency depends upon the ejector performance. While cost efficiency needs to be achieved, performance efficiency should not be compromised. While the performance requires low cost robust operation and absolute safety standards,

It is in this context that efforts to enhance ejector efficiency were initiated and analysed in the paper. Results indicate that the performance of refrigeration system is exponentially influenced by incremental improvements in ejector performance. The process has been done duly considering cost effectiveness, robustness of operational performance, standard environmental practices. The research indicates steam-based systems to be more suitable for large scale industrial environments, while small systems could be powered synthetic fluids.

There is a need for further advanced research to develop analytical tools and mathematical models to understand thr complexity and dynamics of thermodynamics. Precessual dynamics of various boundary layers also need to be understood to draw reliable and repeatable conclusions.

To conclude, improvisation of systemic designs focusing on Supersonic Steam ejectors and vapour compression systems coupled with multi-dimensional flow analysis could be a viable testing ground for examining further applications.

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