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# Design and analysis of ejector refrigeration system using R-134a refrigerant

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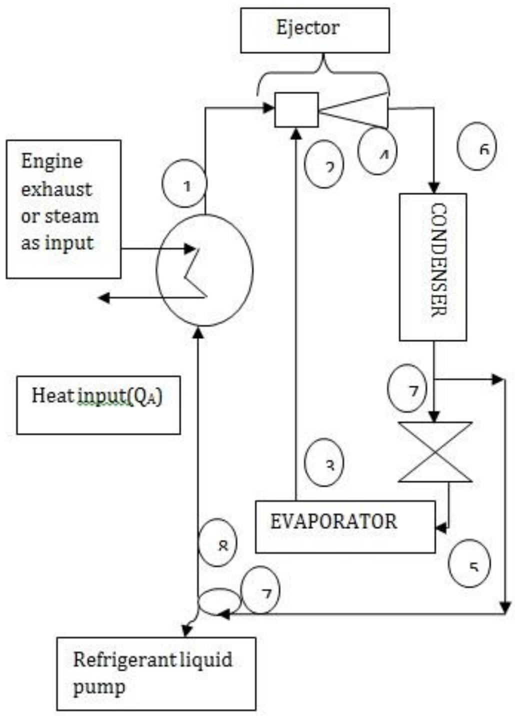
**Abstract** - *heat-driven ejector refrigeration system offers the advantage of simplicity and can operate from low-temperature heat energy sources. There by, it proves to be a good substitute to the conventional compressor-driven refrigeration systems. In this project iam determining theoritical COP values by using R134A ,R410A, R22, R12 refrigerants'. And a care full design and analysis of ejector using CFD package and variation of entrainment ratio by changing length of mixing section and by changing back pressure i.e outlet pressure of diffuser. and also observing the pressure recovery and entrainment ratio.*

**Key Words : R134A, R410A, R22, R12 , COP, entrainment ratio, CFD**

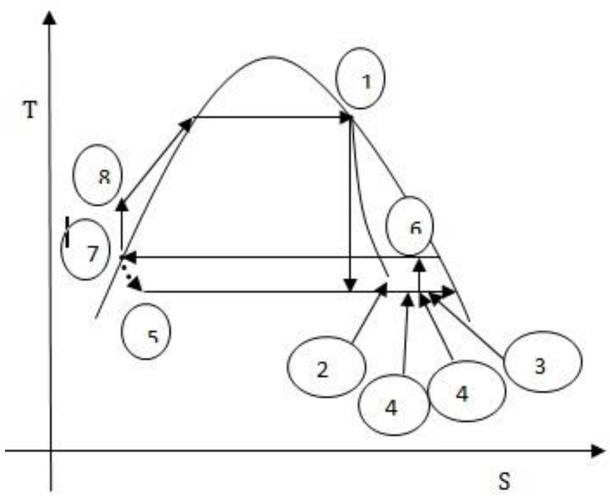
## 1.INTRODUCTION

Over the past few years, the concerning on energy saving and environment protecting has become an increasingly predominant .Developments of industries and increase in population have caused a great demand of cooling and refrigeration application which use the mechanical vapour compression systems. These systems are generally powered by electricity generated by burning fossil fuels. The fossil fuel consumption can contribute to the global warming – the "greenhouse effect". This compels more and more researchers turn to make use of low grade thermal energy. The ejector refrigeration system provides a promising way of producing a cooling effect by utilizing waste heat from industrial process and internal combustion engine or using renewable energy, such as solar power and geothermal energy. With the development of technology, these systems can operate using low-temperature heat Source below 100 °C and even less. Many theoretical and experimental studies of the ejector refrigeration systems are presented in literatures for various working fluids, 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 Utilization of low grade industrial waste heat reduces fossil fuel consumption and corresponding green house gas emission. Organic Rankin cycles are capable of producing power utilizing low grade waste heat [1-3]. However supercritical and transcritical $CO_2$ power cycles would be better options for utilization of low grade waste heat at different temperatures due to better matching of temperature profiles of supercritical CO2 and the flue gas during heat exchange process [4]. Several studies are conducted to improve the performance of $CO_2$ based power cycles [5-9]. Organic flash power cycle is another option that allows even more heat recovery from the given mass of flue gas [10-12]. Low grade heat source is also capable of producing refrigeration effects by using either vapour absorption refrigeration cycle or ejector based refrigeration cycle. It may be noted that one major advantage of ejector based refrigeration cycle is the capability of utilizing wide range of working fluids as refrigerants. Mazzelli and Milazzo [13] conducted both numerical and experimental analyses of a heat source driven supersonic ejector chiller using R245fa as the working fluid. Analysis conducted by Mansour et al. [14] revealed that an ejector assisted mechanical compression system could improve the COP of a conventional vapour

### 1.1 SCHEMATIC MODEL OF EJECTOR REFRIGERATION SYSTEM



### 1.2 .T-S diagram



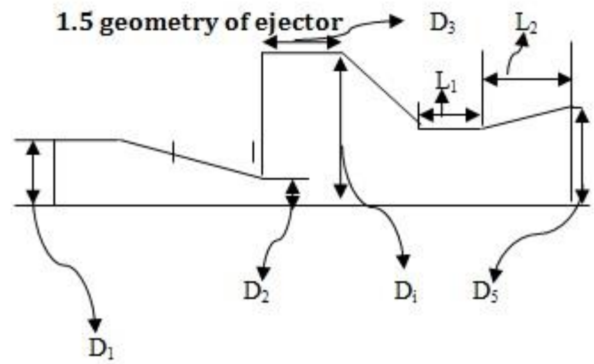

### 1.3 Various thermodynamic process in system:

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 refregerant
4 : condition of mixture of high velocity refrigerant from nozzle & the entrained Refrigerant before compression
6-7 : condensation process in condenser.
7-5 : throttling process

### 1.4 .SAMPLE PROBLEM DESCRIPTION

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

We got COP for R134a = 1.105




### 2. Dimensions of 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 constant area mixing section ($L_1$) | 25 mm |
| Inlet diameter of constant pressure mixing section($D_i$) | 5.69mm |
| Exit diameter of constant pressure mixing section($D_e$) | 4.65 mm |
| Inlet diameter of diffuser($D_4$) | 4.0959mm |
| Out let diameter of diffuser($D_5$) | 5 mm |
| Length of the diffuser($L_2$) | 8.2 mm |

Note: As we designed the ejector using some analytical and experimental results but it is not accurate so we can vary or change it according to the performance of ejector by using CFD tool so the design parameters are just for an idea of geometry

### 2.1 Thermal Design data of condenser

| Type | Shell and tube |
|---|---|
| Flow configuration | Counter flow |
| Shell side mass flow rate | 0.0568kg/sec |
| Tube side mass flow rate | 0.5 kg/sec |
| Shell side heat transfer coefficient($h_o$) | 1534.79w/m² k |
| Tube side heat transfer coefficient($h_i$) | 31177.46 w/m² k |
| Over all heat transfer coefficient(U) | 360 w/m² k |
| No of tubes($N_T$) | 66 |
| No of passes($N_P$) | 2 |
| Tube outer diameter | 9.5 mm |
| Tube inner diameter | 7.01mm |
| Shell diameter | 0.115 mtr |
| Tube length | 3.97 mtr |
| Pitch ratio(PR) | 1.25 |
| Pitch type | square |
| Baffle spacing | 70mm |
| No of baffles | 56 |
| Baffle cut | 38mm |
| Tube pitch($P_T$) | 11.875mm |
| Shell side pressure drop( $\Delta P_S$) | 0.422 Kpa |
| Tube side pressure drop( $\Delta P_{Total}$) | 9.1603 Kpa |

### 2.2 Thermal Design data of refrigerant boiler or pressure vessel

| Material | Steel |
|---|---|
| Volume of vessel | 0.03 m³ |
| design pressure | 25 bar |
| Diameter of vessel | 0.276 mtr |
| Length of vessel | 0.5 mtr |
| Thickness of vessel | 2 mm |
| Amount of refrigerant in vessel@16.04bar | 31.89 kg |

### 2.3. Thermal design data of refrigerant Pump

| Type | centrifugal |
|---|---|
| Design power | 100 W |
| Mass flow rate | 0.05 kg/sec |
| Velocity in suction pipe($V_S$) | 34.0915 m/sec |
| Modified velocity in suction pipe($V_S$) or $V_{f1}$ | 0.5 m/sec |
| Suction pipe diameter($D_S$) | 10.224 mm |
| Velocity in discharge pipe($V_d$) | 28.124 m/sec |
| Modified velocity in discharge pipe($V_d$) | 5.5 m/sec |
| discharge pipe diameter($D_d$) | 4 mm |
| Manometric head($H_M$) | 194.046 mtr |
| Manometric efficiency($\eta_M$) | 95% |
| Shaft diameter($d_{SH}$) | 3mm |
| Hub diameter($d_{hub}$) | 4mm |
| Inlet tangential velocity($U_1$) | 4.22m/sec |
| Inlet blade angle($\alpha$) | 6.75 deg |
| Breadth of impeller at inlet($B_1$) | 2.522 mm |
| Vane angle at inlet($\theta$) | 90 deg(radial flow) |
| Speed of pump(N) | 7875.80 rpm |
| Outlet tangential velocity($U_2$) | 61.856m/sec |
| Outlet blade angle($\beta$) | 9.654 deg |
| Breadth of impeller at outlet($B_2$) | 0.02 mm |
| Vane angle at outlet($\phi$) | 10.55deg(radial flow) |
| No of vanes(Z) | 2 |

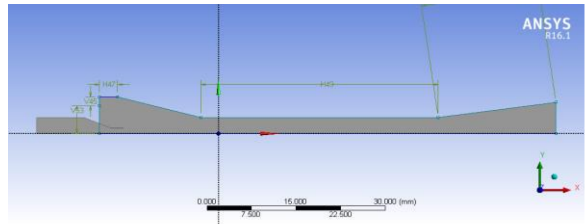
### 2.4 . Thermal design data of evaporator coil

| | |
|---|---|
| Type | Compact finned tube heat exchanger |
| Cooling load on coil | 5.34kw |
| Mass flow of air($m_a$) | 0.1153 kg/sec |
| Mass flow of refrigerant($m_r$) | 0.032 kg/sec |
| Surface matrix configuration | 8.0-3/8 T |
| Tube OD ,cm | 1.02 |
| Tube ID,cm | 0.9525 |
| Fin thickness ,cm | 0.033 |
| Fin area/total area | 0.839 |
| Air passage hydraulic diameter ($D_h$), cm | 0.3633 |
| Free flow area/frontal area($\sigma$) | 0.534 |
| Heat transfer area/ total volume($\beta$) $m^2/m^3$ | 587 |
| $A_{min\ free\ flow}$area,$m^2$ | 6.08e-3 |
| Frontal area($A_f$), $m^2$ | 0.01138 |
| Length of coil, mtr | 0.4 |
| Dryness fraction at inlet to coil | 0.0878 |
| Tube side heat transfer coefficient($h_i$),w/m-k | 39466.702 |
| Air side heat transfer coefficient($h_o$), w/m-k | 150 |
| Pressure drop on tube side($\Delta p_t$),kpa | 0.347 |
| Pressure drop on air side($\Delta p_a$),kpa | 1.5 |
| Overall heat transfer coefficient(U) , w/m-k | 300 |
| No of tubes($N_t$) | 15 |
| No of passes($N_p$) | 4 |
| No of rows($N_r$) | 3 |
| No of coils($N_c$) | 5 |
| Depth of coil(D),mm | 130.8 |
| Heat transfer area(A), $m^2$ | 0.909 |

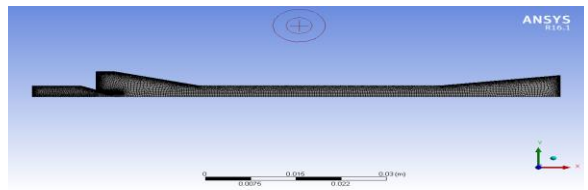
### 2.5 . Thermal design data of capillary

| | |
|---|---|
| Diameter (D) | 1.016mm |
| Length (L) | 30mm |

### 3. Basic CFD model of ejector



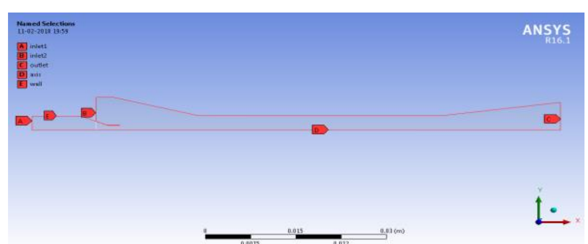

### 3.1 meshed model



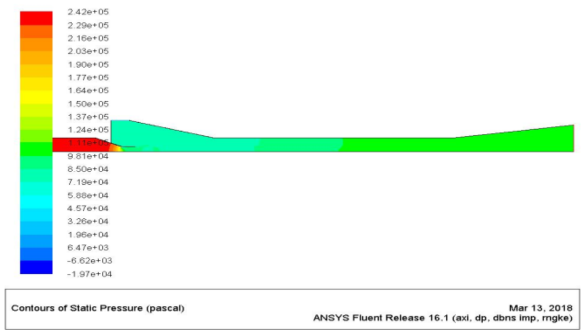

No of elements : 11682
No of nodes : 12615

### 3.2Boundary conditions

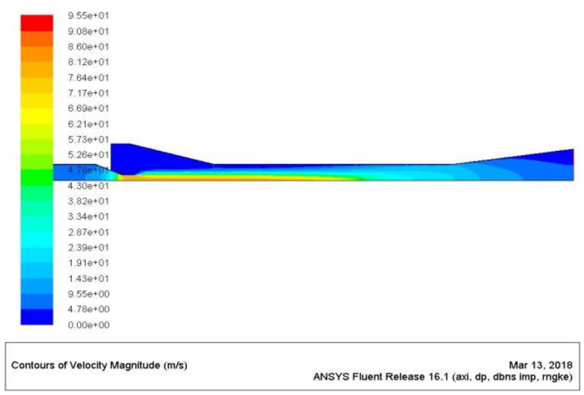
Inlet 1 : pressure in let (24334Pa, 273K)
Inlet 2 : pressure inlet (84380Pa, 248K)
Outlet : pressure out let (1.1bar,248K)
Wall : wall
Axis : axis
Model : Viscous k- ε model



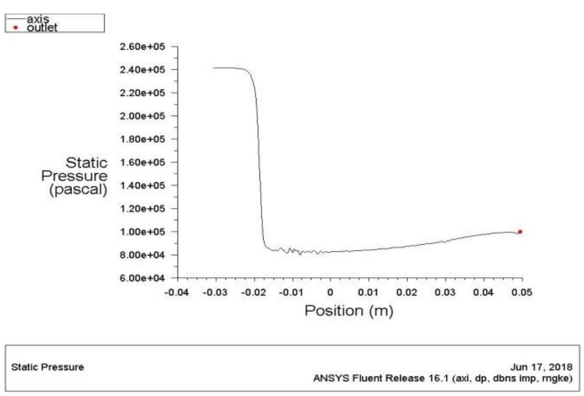
### 3.3 .Contours of pressure




### 3.4 Contour of velocity

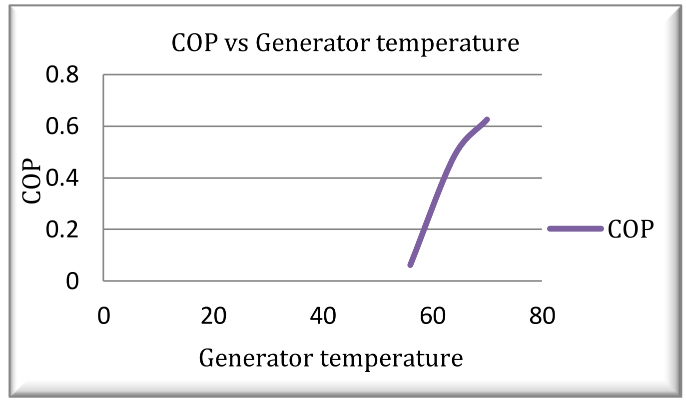



### 3.5 .Pressure vs axial distance of ejector



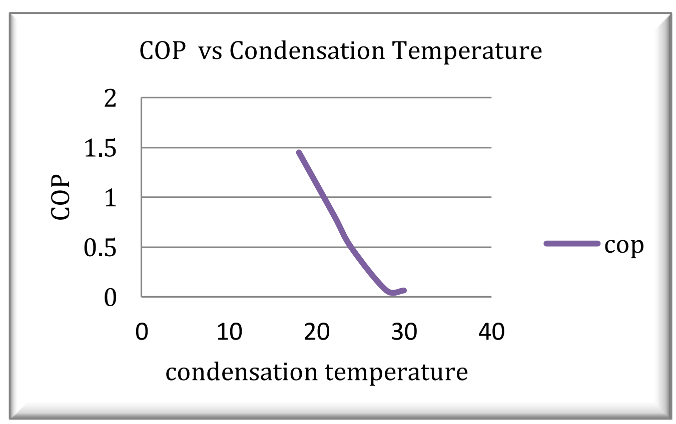

## 4.VARIATION OF COP VALUES

### 4.1 Variation of COP by changing generator temperature and keeping condenser temperature(22⁰c) evaporator temperature (6⁰c) as constant




The COP value has increased by increasing generator temperature because, as the generator temperature increases its saturation pressure increases and due to this the primary or motive nozzle inlet pressure increases and by nozzle action the static pressure at the outlet of the nozzle decreases and due to this the entrainment ratio increases. As we know COP value is directly propotional to the entrainment ratio(ER) so COP value increases due to increase in the generator temperature

### 4.2. Variation of COP by changing condensation temperature and keeping generator temperature(70⁰c), evaporator temperature(6⁰c) as constant



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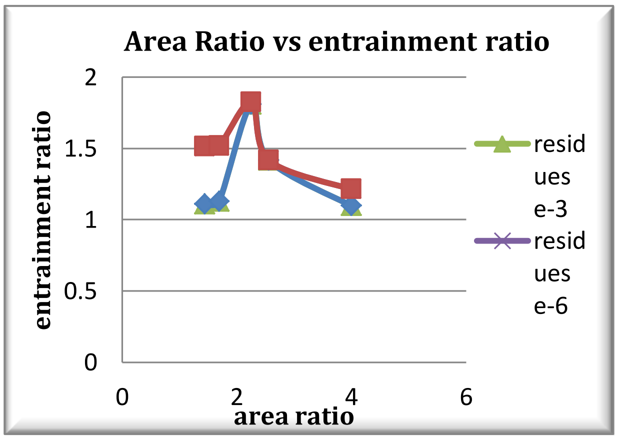
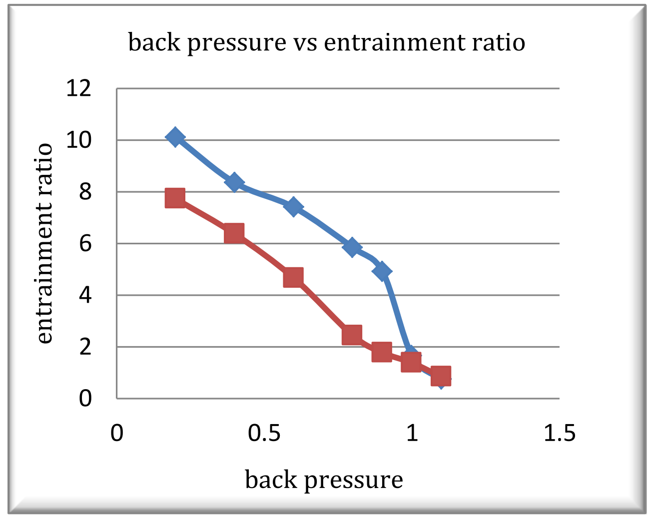
The COP value has increased by decreasing condensation temperature because as the condensation pressure decreases by decrease in the temperature and due to decrease in the condensation pressure an additional pressure potential is available for driving mass flow rate and this increases the secondary mass flow rate and in turn increases the entrainment ratio(ER), Which in turn increases the COP of the system

### 4.3. Variation of entrainment ratio by changing area ratio of the ejector and keeping constant pressure mixing length(7mm) and constant area mixing length (40mm) as constant



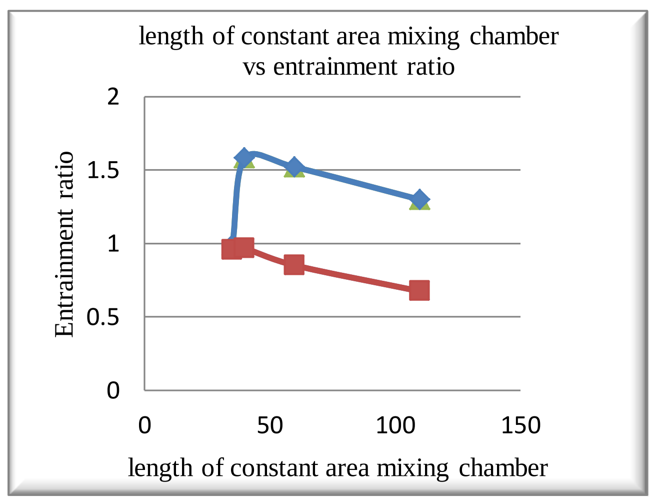

The entrainment ratio (ER) has increased by increasing area ratio( ratio of secondary to the primary area) because as mass flow is directly propotional to cross sectional area so duet o increase in secondary area, mass flow increases and due to this entrainment ratio increases by increasing in the area ratio. But this increase is up to certain limit and after that limit the entrainment ratio decreases by increase in area ratio and this decrease is due to formation of vortocities and flow separation.here we got that limit point as 2.56

### 4.4.Variation of entrainment ratio by changing back pressure and keeping primary inlet pressure(243340 pa), secondary inlet pressure(84380 pa),constant pressure mixing length(14mm) and constant area mixing length (25mm) as constant




The entrainment ratio(ER) increases as the back pressure decreases because due to decrease in the back pressure the is an additional pressure potential for the driving of mass flow rate due to this additional pressure potential there is an increase in secondary mass flow rate and due to this entrainment ratio increases

### 4.5. Variation of entrainment ratio (ER) by changing the length of constant area mixing chamber($L_{am}$) by keeping constant pressure mixing chamber as constant($L_{pm}$=14mm)




As the length of mixing chamber increases the entrainment ratio increases up to certain length and after that due to frictional effect and formation of eddies the back flow increases and due to this entrainment ratio decreases

## 5.CONCLUSION

Ejector chillers may enter the market of heat powered refrigeration as soon as their cost per unit cooling power becomes equal or lower than that of absorption chillers systems. However, market competitiveness of ejector chillers may be reached only after an increase of the system COP, here in this project a complete design of all components of ejector refrigeration system for a 1.5 ton capacity has been designed

## ACKNOWLEDGEMENT

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