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"Replacement of vapor compression system of domestic refrigerator by an ejector refrigeration system"

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Abstract: Ejector Refrigeration System as name indicated that it uses the ejector as the main replacement for the compressor refers to conventional refrigeration systems.

Ejector Refrigeration system is heat driven process; it uses low grade heat as power source for refrigeration systems. Waste process heat and waste steam is common in industry and it use for heating, cleaning and absorption cooling cycles along with other uses.

Ejector is the critical component of the Ejector Refrigeration Systems. So, Design of ejector play an important role in ERS. Ejector has fewer moving parts compare to the compressor.

An ejector is a device in which a high-velocity jet of fluid mixes with a second fluid stream. The mixture is discharged into a region at a pressure higher than the source of the second fluid. So, there will less wear of components.

ERS help in reduction in Global warming and Greenhouse Effect Emission., So now days we using Natural Refrigerant such as carbon dioxide, air, water and ammonia, have been used as Refrigerants in cooling systems.

It is been required for cooling is common in many Industries and home refrigeration to space cooling.

The main aim of ERS is to use low grade heat to drive an ERS Cooling Systems. This is cycle is recently drawn renewed attention due its simplicity of construction, ruggedness and few moving parts.

1. INTRODUCTION:

An ejector refrigeration system can be considered as a modification of a conventional vapor compression cycle (VCC). An ejector takes the place of a compressor to pressurize the refrigerant vapor flowing from an evaporator and discharge it to a condenser. The conventional ejector refrigeration cycle. Working fluid is heated at a high pressure and temperature in the generator. High-pressure refrigerant vapor enters the nozzle. Working fluid is then accelerated to a high velocity and entrains motive steam from the evaporator, resulting in a cooling effect. After that, mixed vapor steams are discharged from the nozzle to the condenser where they are cooled down and condensed to liquid fluids. A part of the liquid refrigerant returns to the evaporator through an expansion valve whereas the other part is pumped to the generator. Ejector cooling technology can be used for air conditioning in trains and large buildings.

Ejector is the critical component of the ejector. So, design of ejector plays a significant role in ERS. Ejector has fewer moving parts compared to the compressor. The main aim of the ERS is to use low grade heat to drive an ERS Cooling System. ERS helps in reducing Global warming and Greenhouse Effect Emission. However, ejector refrigeration systems always have a smaller coefficient of performance (COP) compared with that of vapor compression systems, but it can be more practical and economical when waste heat, solar energy, or exhaust heat are used to provide heat to the generator of an ejector system.

1.1 Problem Statement:

In a conventional vapor refrigeration system, Refrigerant leakage problems are more and the refrigerant used in vapor compression cycle is costly and toxic. The compressor used in the conventional system uses electrical energy for compressing the refrigerant gas. The Compressor used is very noisy. The compressor may fail due to electrical failure and also due to high temperature and pressure.

1.2 Objective:

The main part of the ejector refrigeration system is ejector. In conventional refrigeration system we use compressor that utilize electrical energy. Ejector works like a mechanical compressor and can be used to increase the fluid pressure without any moving parts.

An ejector-based cooling system offers several advantages such as no moving parts, efficient utilization of the waste heat and low cost. So, the improved system performance will reduce energy consumption as well as reduce CO2 emissions.

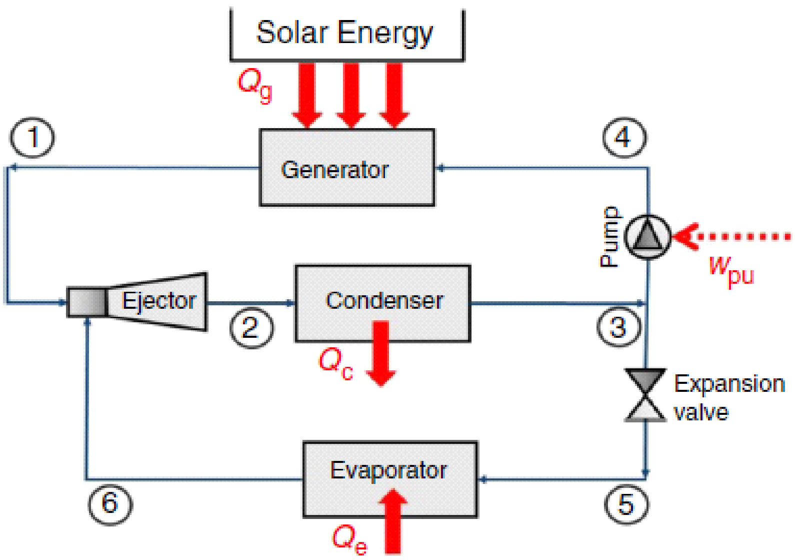
1.3 Scope:

There are a lot of benefits associated with the ejector refrigeration system that uses waste heat. To reduce the emission of greenhouse gases, specifically CO2 emissions. The system is very reliable, so the running and maintenance cost is less. This ejector refrigeration system provides a benefit that traditional refrigerators do not, a non-toxic, inflammable working fluid (R718 i.e., water).

1.4 Methodology:

An ejector refrigeration system can be considered as a modification of a conventional vapor compression cycle (VCC). An ejector takes the place of a compressor to pressurize the refrigerant vapor flowing from an evaporator and discharge it to a condenser.

Working fluid is heated at a high pressure and temperature in the generator. High-pressure refrigerant vapor enters the nozzle. Working fluid is then accelerated to a high velocity and entrains motive steam from the evaporator, resulting in a cooling effect. After that, mixed vapor steams are discharged from the nozzle to the condenser where they are cooled down and condensed to liquid fluids. A part of the liquid refrigerant returns to the evaporator through an expansion valve whereas the other part is pumped to the generator.



2. Why is an ejector preferred?

The air-conditioning test performance of the ejector refrigerator-based air-conditioner (ERAC) was proposed under various hot water temperatures, primary nozzle sizes, cooling loads and air-conditioned space temperatures.

This was to demonstrate the practical use of the ERAC in a real air-conditioning application under the warm ambient while the system was driven by a relatively low temperature heat source. The optimal performance of the ERAC working under the thermal comfort condition (**T**air-con = 25C) was demonstrated.

The findings of this present work are summarized as follows:

- For a certain hot water temperature and primary nozzle used, an increase in the cooling load caused the air-conditioned space temperature to be increased. However, the cooling load must not be too high when the thermal comfort condition is considered.
- For a certain hot water temperature, when the primary nozzle throat was enlarged, a decrease in the Tair-con was found at identical cooling load. However, using a bigger primary nozzle caused a decrease in the system COP.
- For operating at the thermal comfort condition, initially, increasing the hot water resulted in increasing the system COP until it reached its maximum value (optimal performance). Later, the system COP decreased when the hot water continued to increase. Therefore, the optimal heat source for this particular working condition was determined.
- When operating at the thermal comfort condition, enlarging the primary nozzle throat required a lower Thot to produce the optimal point of operation. However, a slight decrease in the system COP was found.
- A change in the air-conditioned space temperature significantly affected the optimal performance. It was evident that the optimal performance of the ERAC was achieved with a lower Thot and vice versa.

The air-conditioning tests indicated the high potential of the ERAC for real operation.

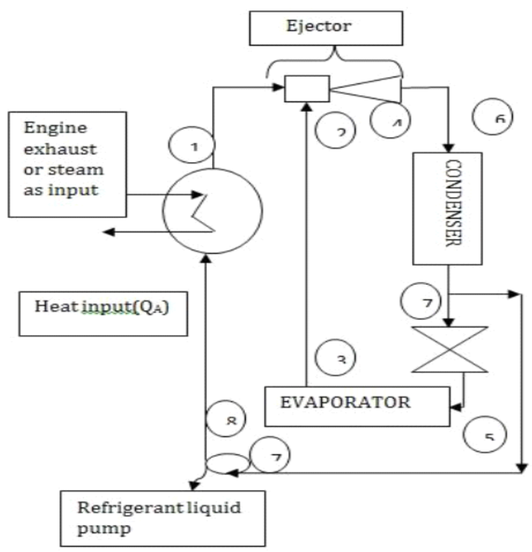
The results are useful for being a reference case to further develop the ejector refrigeration system for practical use.

In “Energy Analysis of a Novel Ejector-Compressor Cooling Cycle Driven by Electricity and Heat” we saw that Low-grade heat is abundantly available as solar energy and as industrial waste heat which can be converted into cooling by the proposed EEVCRC system which has been modeled, compared, analyzed, and optimized using EES.

The ejector model used in EES simulations of the system is a new analytical model which gives an on-design optimal performance of ejectors for the available working conditions.

The system gives better performance than all the three systems it has been compared with.

The sensitivity analysis found that the COP of the proposed system increases exponentially at lower condensation temperatures and higher evaporator temperatures, making it very suitable for industrial water-cooled systems and higher temperature cooling applications.



At 50C condenser temperature, the electrical COP of EEVCRC is 50% higher than conventional VCC while at 35C, the electrical COP of EEVCRC is 90% higher than conventional VCC. For the higher temperature heat source, and hence the higher generator temperatures, the electrical COP of EEVCRC increases linearly while there is no increase in the electrical COP for ECS.

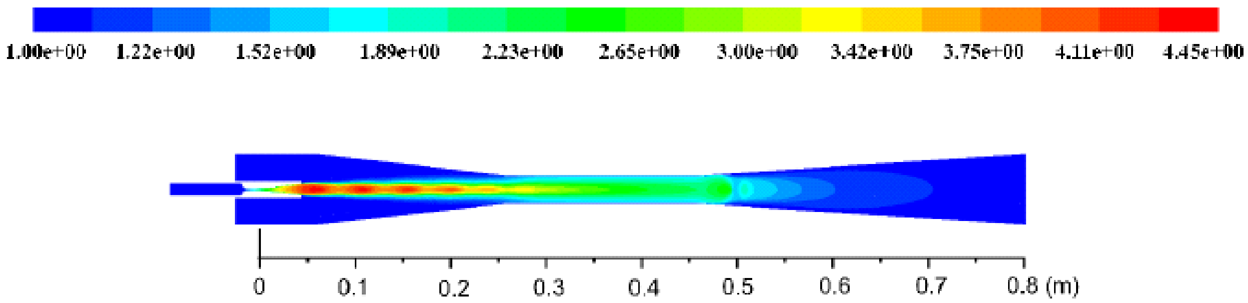
It is found that by using the second ejector at the upstream of the electrical compressor, the electrical COP is increased by 49.2% as compared to a single ejector system. At 50C condenser temperature, the electrical COP of EVCC is 67.7% higher than conventional VCC while at 35C, the electrical COP of EVCC is 85.6% higher than conventional VCC.

For the higher temperature heat source, and hence the higher generator temperatures, the electrical COP of EVCC increases linearly.

The relatively better global COP indicates that a small solar collector will be needed if this system is driven by solar thermal energy.

In “A Steam Ejector Refrigeration System Powered by Engine Combustion Waste Heat” the engine combustion waste heat as the heat source is driven by the steam ejector refrigeration system, which can improve the energy utilization and save cost, as is proposed.

The effect of the shock wave structure on the pumping performance of the steam ejector is thoroughly studied. The shock train and co-velocity region locating in the mixing chamber and throat sections is defined as the pseudo-shock region. While the other part is the oblique-shock region consisting of a single normal shock wave and a series of the oblique shock waves locating in the subsonic diffuser section. The shock train, the single normal shock wave and the oblique shock wave are the three types of shock waves, respectively.



In conclusion, the main findings of this study are as follows:

(i) The shock wave structure is thoroughly studied by the experimental-CFD method. An experimental system of the ejector refrigeration was established and the CFD model was verified. Based on the ideal gas model, the k-w SST turbulence model is chosen, as it can predict the complex flow in the ejector.

(ii) The structure of the shock wave is divided into two regions: pseudo-shock region and the oblique-shock region. Three types of shock waves are determined as the shock train, the normal shock wave and the oblique shock wave, respectively.

(iii) The length of the pseudo-shock region increases with the increasing of the primary fluid pressure when the pressure is less than 0.34 MPa, then maintains constant. Due to a part of the energy of the mixed fluid is consumed, the entrainment ratio and the pumping performance of the steam ejector decreases. The bifurcation shock waves increase, and the energy of the normal shock wave decreases with the increase of the primary fluid pressure when the primary fluid pressure is more than 0.34 MPa.

(iv) As the primary fluid pressure increases, the entrainment ratio declines and the critical back pressure and the length of shock waves increasing of the steam ejector increase.

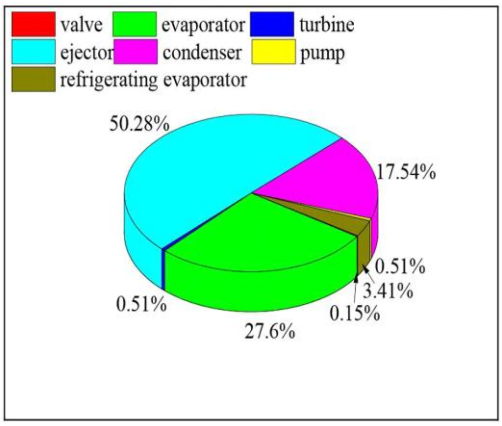
(v) The idealized primary fluid pressure is between 0.32 MPa and 0.36 MPa and the idealized Mach number is between 1.6 and 2.1. The growth rate of the critical back pressure with the primary fluid pressure at 0.40 MPa–0.46 MPa accelerates 61% compared with the pressure at 0.32 MPa–0.40 MPa.

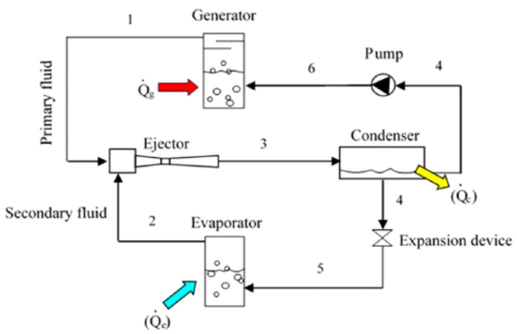
With the utilization of CFD analysis, the critical operating primary fluid pressure of the steam ejector system was identified. It can enhance the effectiveness of converting combustion waste heat to recyclable and useful energy sources in a steam ejector refrigeration system. Future works can be carried out to further enhance the understanding of the shock structure via experimentation using high-speed photography techniques.

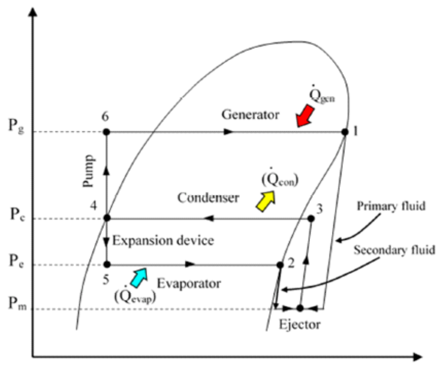
In “Design and analysis of ejector refrigeration system using R-134a refrigerant” Ejector chillers may enter the market of heat powered refrigeration as soon as their cost per unit cooling power becomes equal or lower than that of absorption chillers systems.

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.

In “Performance Analysis on a Power and Ejector-Refrigeration System and the Involved Ejector” In order to make better use of the thermal energy at low and medium temperature and improve the organic Rankine cycle performance, the power, and ejector refrigeration system has been put forward and a lot of research has been carried out. This article presents a new study of the combined system and the key component ejector using the zeotropic mixture R134a/R123 as working fluid.







First, the influence of heat source temperature, turbine outlet pressure, and different mixture compositions on the performance of the combined system and ejector were analyzed. It can be found that entrainment ratio is not sensitive to the change of heat source temperature.

Through energy analysis of the combined system, it can be found that ejector, evaporator, and condenser take up for most energy destruction of the system.

The result illustrates that energy destruction mainly occurs in the component of ejector, which can reach 50.28%. Then, the relationship between input heat of ejector, net power change value and power saved by ejector was compared. It can be found that net power reduction is less than power saved by ejector and refrigeration output.

Finally, the energy efficiency of ejector was defined and the effects of other parameters on it are analyzed. The results show that the energy efficiency of ejector and COP are oppositely changing.

In the combined system, scholars have done a lot of research on evaporator, condenser, turbine and pump. This article mainly focuses on factors that affect ejector performance. Compared with compressors, ejector consumes less power.

Therefore, the relationship between saving power and energy destruction in ejector requires more research. Many parameters affect net power and refrigeration output of the combined system. According to the research situation, this article studies the influence of turbine outlet pressure, heat source temperature and mass fraction on the performance of the combined system.

In "Performance of ejector refrigeration cycle based on solar energy working with various refrigerants" The conventional compressor-based refrigeration system operates with CFCs causing ozone layer depletion and has large energy-consuming moving parts.

This energy mostly extracts from non-renewable sources like fossil fuels.

On the other hand, the cooling demand is increasing day by day, and continuous dependence on natural fuel contributes to CO_2 emissions into the atmosphere.

Solar thermal refrigeration system (STRS) is a better alternative way which converts heat energy from the renewable energy source and waste heat source and provides a cooling effect. Solar thermal energy emerges as a smart energy system that is cost-effective and 100% renewable energy system.

Utilization of solar thermal energy in the refrigeration system conserves energy as well as reduces the global CO_2 footprint, and hence, this technology draws the interest of both the scientific community and industries [1]. Moreover, TRS is capable of operating a wide range of working fluids that are environmentally safe, non-toxic, nonflammable, and non-corrosive.

	T_e (°C)	T_c (°C)	T_g (°C)	Q_{ref} (kW)	COP_e	COP_g
Conventional VCC	5	45	90	10	3.77	-
Cascade ejection-compression system (CECS) [17]	5	45	90	10	5.58	1.18
Ejection-Compression system (ECS) [9]	5	45	90	10	4.67	2.88
Enhanced Ejector Refrigeration System (EERS) [15]	5	45	90	10	5.5	2.16
Our proposed system (EVCC)	5	45	90	10	8.206	2.32

A one-dimensional theoretical analysis of the ejector refrigeration system working with low-grade energy resources is presented in this present work.

A computer program based on a series of isentropic relations of ejector flow is developed in SciLab and interfaced with cool prop, a thermodynamic library, for material properties of different working fluids.

Initially, the performance of the ejector refrigeration systems is calculated under the overall operation mode. Later, the system performance is evaluated with different refrigerants under critical mode.

Before the subcritical mode of operation, at a constant area ratio and condenser pressure, the coefficient of performance increases with an increase in the generator temperature until the chocking condition.

At the chocking condition, the generator temperature attains an optimum value and gives the highest COP. This COP value ranges from 0.25 to 0.17 at a given area ratio of 11.2 under overall mode. In the critical mode, further increment in generator temperature diminishes the performance.

Under the critical mode operation, at a constant generator temperature as well as a constant evaporator temperature, increasing the area ratio widens the hypothetical throat area and allows more secondary flow and enhances the performance.

In contrast, increasing the generator temperature increases the primary flow and stalls the secondary flow. Thus, the increment in evaporator temperature enhances the system performance while the increment in generator temperature diminishes the system performance.

Among the six considered refrigerants, R717 exhibits substantially better performance in terms of entrainment ratio, coefficient of performance, and cooling load. Even though the R718 shows the least coefficient of performance and entertainment ratio, it shows moderate cooling capacity than R141b, R245fa, R123, and R365mfc.

3. ANALYTICAL CALCULATIONS (NH3)

3.1 Design of Nozzle:

- First of all we have to decide approximate value of pressure for our system;

So let us consider:

- Condenser Pressure = 10 Bar
- Evaporator Temperature = 1 Bar
- Boiler Pressure Temperature = 20 Bar

3.1.1 Throat Design:

- Gamma ("γ") = Cp/Cv = 1.337
- Cp = 2.08
- for water.
- Throat Area (At):

$$A_t = w_t / P_t \sqrt{RT_t / \gamma g_c}$$

- Wt = flow rate through boiler to entry nozzles is 20 kg/s
- Pt = Pressure at throat, so we have to match the primary vapor pressure

  with secondary pressure, i.e., pt. = 1bar
- Tt = Temperature of throat is given by Formula

$$T_t = T_c \left[\frac{1}{1 + \frac{\gamma - 1}{2}} \right]$$

- Tt = 85.554 °C from above Formula
- After Calculation At = 347.2 mm2.

- Diameter of Throat dt = (4*At/π)^1/2 .
- Diameter of Throat By calculation = 21mm

3.1.2 Inlet Design:

- From the study it is found that the inlet diameter of nozzles should be 3-5 Times of Nozzle Throat, so the ejector will have usable face.
- Dc = Inlet Diameter of nozzle = 3*Dt
- Dc = 63mm.
- Area Ac = π*dc^2/4
- Ac = 3117.25mm^2.

3.1.3 Exit Design:

- Me-Mass flow rate at Exit is given below Formula

$$M_e^2 = \frac{2}{\gamma - 1}\left[\left(\frac{P_c}{P_{atm}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

- Me^2 = 4.669kg/s therefore Me = 2.16
- Ae = Area Exit of Nozzle is given Below Formula

$$A_e = \frac{A_t}{M_e}\left[\frac{1 + \frac{\gamma-1}{2} M_e^2}{(\gamma+1)/2}\right]^{\frac{\gamma+1}{2(\gamma-1)}} - ($$

- Ae = 700mm^2.
- So now we find the diameter of Exit part, Formula is
- De = (4*Ae/π) ^1/2.
- De = 29.86mm.

3.1.4 Angle and Length:

- Alpha (α) = The nozzle angle divergence
- half angle is α<15 degree.
- Beta (β) = The Nozzle convergence half angle is β = 60 degree.
- Vc = volume of inlet Chamber = 1.1*Ac*lc
- Vc= 38500mm^3
- Lc=50 mm

3.1.5 Efficient of working of nozzle:

- There are Some Parameter Has to be followed to increase the efficiency of Systems
- Suction Head Should be mounted on the Top Part.
- Pressure at Primary Nozzle exit should be less than the Evaporator Pressure.
- Reduced the Back pressure of vapor

3.1.6 Dimension of Nozzle:

- Dc = Inlet Diameter of Nozzle = 63mm
- Dt = Throat Diameter = 21mm.
- De = Exit Diameter of Nozzle = 34mm.
- Alpha (α)<15 degree.
- Beta (β) = 60 degree.
- Lc = Characteristic Length = 50 mm

3.2. Design of Ejector: -

3.2.1 Throat Design:

- Gamma ("γ") = Cp/Cv =1.337
- Cp for water =2.08
- Throat Area (At):

$$A_t = w_t / P_t \sqrt{RT_t / \gamma g_c}$$

- Wt = flow rate through boiler to entry of Ejector which combination of primary and secondary flow is 40kg/s
- Pt = Pressure at throat, so we have to match the primary vapor pressure
- with secondary pressure, i.e. Pt = pt + ps = 1.5bar
- Tt = Temperature of throat is given by Formula
- Tt = 85.55 deg from above Formula
- After Calculation At = 462mm^2.
- Diameter of Throat By calculation = (4*At/π)^1/2 .
- Diameter of Throat dt = 24mm.

3.2.2 Converging Part Design

- From the study it is found that the inlet diameter of Ejector should be

2.5 - 5 Times of Ejector Throat.

- Di = Inlet Diameter of Ejector = 2.5 * Dt
- Di = 60.7mm.
- Area Ai = π * di^2/4
- Ai = 2893.8mm^2.

3.2.3 Diffuser Design

- Mass flow rate at Exit is given below Formula

$$M_e^2 = \frac{2}{\gamma - 1}\left[\left(\frac{P_c}{P_{atm}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

- Me^2 = 4.669kg/s therefore Me = 2.16
- Ae = Area Exit of Nozzle is given Below Formula
- Ae = Ad = 933.35mm^2.
- So now we find the diameter of diffuser part, formula is De = (4*Ad/π) ^1/2.
- Dd = 34.47mm.

SR. NO.	SYMBOL	NOMENCLATURE
1.	Wt	Flow rate through boiler to entry nozzle
2.	Pt	Pressure at throat
3.	Tt	Temperature of throat
4.	Dt	Diameter of Throat
5.	Dc	Inlet Diameter of nozzle
6.	Me	Mass flow rate at Exit
7.	Ae	Area Exit of Nozzle
8.	α	The nozzle angle divergence
9.	β	The Nozzle convergence
10.	Vc	Volume of inlet Chamber

4. ANALYTICAL CALCULATIONS (R134a)

ANT = (m1/PB)*(RTt/ϒg)^0.57

ANT = (0.01275/(1.0666*10^6))*((81.5*309.5)/(1.303*9.81))^0.5

ANT = 0.531 sq mm

DNT = (ANT*4/Π)^0.5 = 0.82 mm

DNI = 3*DNT = 3*0.82 = 2.46 mm

ANI = (Π*DNI^2)/4 = 4.753 sq mm

MPF = v/c = 471/181.3 = 2.597

ANE = (ANT/MPF)*((1+((Υ-1)*(MPF^2)/2))/((Υ+1)/2))^((Υ+1)/(2*(Υ-1)))

ANE = (0.531/2.597)*((1+((1.3031)*(2.597^2)/2))/((1.303+1)/2))^((1.303+1)/(2*(1.303-1)))

ANE = 1.737 sq mm

DNE = (ANE*4/Π) = (1.737*4/3.14) = 1.49 mm

AET = (mm/PC)*(RTm/Υg)^0.5

AET = (0.02231/(0.653*10^6))*((81.5*293)/(1.303*9.81))^0.5

AET = 1.4776 sq mm

DET = (AET*4/Π)^0.5 = 1.372 mm

DEI = (5)*DET = 5*1.372 = 6.858 mm

AEI = (Π*DEI^2)/4 = 37 sq mm

Mm = MPF + MSF = 2.597+1.43 = 4.027

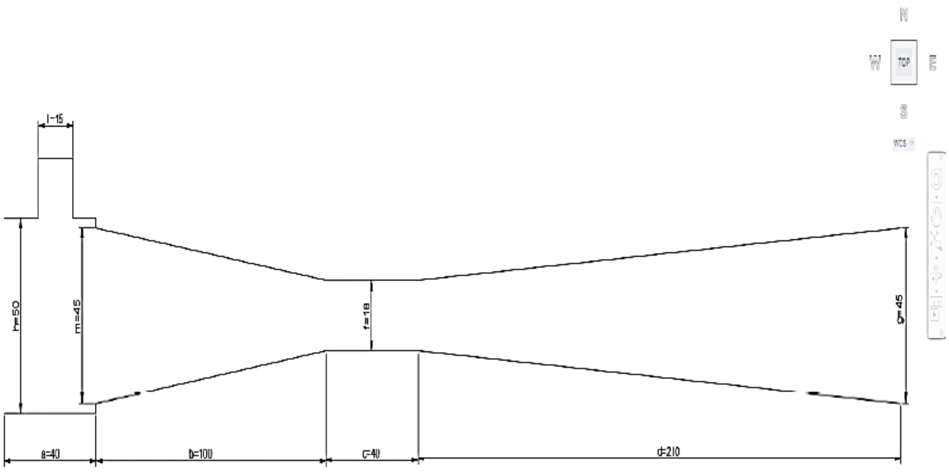
AEE = (AET/Mm)*((1+((Υ-1)*(Mm^2)/2))/((Υ+1)/2))^((Υ+1)/(2*(Υ-1)))

AEE = (1.4776/4.027)*((1+((1.3031)*(4.027^2)/2))/((1.303+1)/2))^((1.303+1)/(2*(1.303-1)))

AEE = 1.686 sq mm

DEE = (AEE*4/Π)^0.5 = 1.465 mm

SR. NO.	SYMBOL	NOMENCLATURE
1.	A	Area
2.	D	Diameter
3.	P	Pressure
4.	M	Mach Number
5.	Υ	Heat capacity ratio
6.	NT	Nozzle throat
7.	NI	Nozzle inlet
8.	NE	Nozzle exit
9.	ET	Ejector throat
10.	EE	Ejector exit
11.	SF	Secondary fluid
12.	PF	Primary fluid
13.	v	Velocity of fluid
14.	c	Local sonic speed



5. COLLECTION OF RESOURCES

a. **Research papers** – gathered the latest research papers which showed the advantages of Ejector Refrigeration System over a standard VCC system.

b. **R134a refrigerant** – a refrigerator serviceman was called in our campus to examine the VCC system in the RAC laboratory. He recommended R134a as a refrigerant for our system. He had tin cans of R134a available and called him to replace the refrigerant available in the system.

b. **Digital thermometer** – instead of using thermocouple, thermostat or a subzero thermometer, we used a digital thermometer was used to measure the temperature of the cooling chamber. The standard VCC system was tested for 24 hours and the lowest temperature measured was degree Celsius.

c. **Multimeter** – a multimeter from the electronics laboratory was used to measure the voltage and current consumed by the VCC system instead of using a clamp meter.

d. **Manufacturing of ejector** – local manufactures of refrigerator parts were contacted to design the ejector of desired dimensions and standards from Indiamart and Google web.

6. COMPARISON WITH STANDARD VCC SYSTEM

6.1 COP of VCC System

Co-efficient of performance of VCC

COP = output/input

= heat rejected/energy supplied

COP = MCvdt/VIT

= (0.20826*0.718*(32.5-6)) / (1.1*250*3*10^-3)

COP = 4.803

Where,

M = mass of air in the cooling chamber

= density of air*volume of cooling chamber

= 1.157*0.18

= **0.20826 kg**

Cv = specific heat capacity of air at constant volume

= **0.718 kJ/kgK**

dt = room temperature – cooling chamber temperature

= 32.5 – 6

= **26.5 deg. C**

V = voltage

= **250 V**

I = current

= **1.1 Amps**

T = time required to reach the temperature of 6 deg. C

= **3 hrs**

6.2 COP of ER system:

Co-efficient of performance of ERS

COP = output/input

= heat rejected/energy supplied

COP = MCvdt/PT

= (0.20826*0.718*(32.5-6)) / (250*3*10^-3)

COP = 7.35

Where,

M = mass of air in the cooling chamber

= density of air*volume of cooling chamber

= 1.157*0.18

= **0.20826 kg**

Cv = specific heat capacity of air at constant volume

= **0.718 kJ/kgK**

dt = room temperature – cooling chamber temperature

= 32.5 – 6

= **26.5 deg. C**

T = time required to reach the temperature of 6 deg. C

= **3 hrs**

Thus, COP of ERS is 150% better than that of VCC.

Sr. No.	Symbol	Abbreviation
1.	COP	Coefficient of performance
2.	M	Mass of air in the cooling chamber
3.	Cv	Specific heat capacity of air at constant volume
4.	dt	Temperature difference
5.	V	Voltage
6.	I	Current
7.	T	Time required to reach the temperature of 6 degrees Celsius

7. FUTURE SCOPE AND CONCLUSION

Hence, with help of calculations done above we can conclude that the ERS is more efficient than the VCC system used in the current domestic refrigerator.

- An ejector is based upon Bernoulli's Principle which states: 'When the speed of a fluid increases its pressure decreases and vice versa'.
- The system gives better performance than all the three systems it has been compared with. The sensitivity analysis found that the COP of the proposed system increases exponentially at lower condensation temperatures and higher evaporator temperatures, making it very suitable for industrial water-cooled systems and higher temperature cooling applications.
- No moving parts, hence low maintenance requirement.
- Environmentally friendly option and easy to control using standard techniques.
- 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.

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