Performance analysis of solar assisted vapour Jet refrigeration system with regenerator (CRMC method)

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Abstract. There are various forms of low-grade thermal energies that could all be useful in powering an ejector refrigeration system. Effective utilization of low-grade thermal energy is feasible from many available sources from industrial areas, solar and the automobile exhausts etc. The normal Vapour Jet Refrigeration system is compared with the efficient Vapour Jet Refrigeration system fitted with a solar assisted regenerative heat exchanger unit. The ejector is designed using Constant Rate Momentum change (CRMC) methodology to enhance the Coefficient of Performance by the usage of the refrigerant R134a as the working medium. The ejector cycle has been recognized as the promising technology for utilization of solar energy for cooling. The heat absorbed by the solar flat plate collector is partially used to heat refrigerant in the ejector cycle. Study on the effect of refrigerant temperature due to heat provided by boiler and solar flat plate in the Vapour Jet Refrigeration system on Coefficient of Performance, pumping power, entrainment ratio, diffuser outlet temperature & heat rejected at condenser is carried out. Simulation of ejector designed through Constant Rate Momentum change (CRMC) methodology is done using FLUENT software.

Keywords: Ejector, Refrigerant R134a, Entrainment ratio, Mach number, Constant Rate Momentum change (CRMC), regenerator, COP.

1. Introduction
The low-grade thermal energies which are available in various forms such as vehicle exhaust heat, industrial processes, waste heat, and the hot water which is generated by solar collector could all be useful in powering the system ejector refrigeration system. The technology investigated herein in addition to energy savings could result in the reduction of harmful greenhouse emissions. It is understood that fossil fuels burnt in power plants to generate electricity is associated to be linked to environmental climate shifts. Over several decades, the application of low-grade thermal energy has gained interest for many researchers. The availability of low grade energy is more in industrial processes, solar collectors and automobile exhausts. Since a huge part of industrial waste heat is deserted to the atmosphere, many reported researchers work carried out in the area of low grade energy for economic significance[1]. Refrigeration is found to be an emerging area to use the low-grade energy, since it is economically reasonable and ecological pleasant applications to utilize the thermal energy, which is otherwise go as waste. Refrigeration is carried out in producing necessary cooling effects in many processing industries, such as pharmaceutical, food, petrochemical and other related industries. Among the available refrigeration system mechanical vapour compression and vapour absorption refrigeration are very popular. Ejector refrigeration is a also emerging as a promising technology that uses waste heat energy as main energy source for refrigeration with different types of refrigerants such as R134a, R717 etc[4][5].

2. Ejector refrigeration
The essential components of ejector refrigeration system are ejector, a generator and a pump and they perform the function of thermal compression. Ejector system, replaces the mechanical Vapour-compressor by utilizing low-grade energy. This permits the system to be driven by thermal energy. The key component of the system is ejector, which is used to entrain, mix and recompress both the fluids by transfer of momentum and energy using a primary jet with high velocity. Traditionally in ejector refrigeration, water is used as refrigerant. In early 1900 steam is used in first cooling and refrigeration jet systems. In order to recognize and increase the performance of steam jet systems extensive research has been carried out. The major drawback of this systems is the low COP values and not able to produce the cooling effect below zero degree Celsius. Even though refrigeration system using ejector is a valuable alternative to traditional refrigeration systems of vapour compression, the refrigeration system of vapour compression remains dominant in the market.

The performance of the refrigeration system-using ejector is improved by the use of refrigerants like halocarbon compound. To increase the ejector efficiency and reduce the unit cost production more research and enhancement are required. In the recent years vapour absorption refrigeration system are commercially available. To make the ejector refrigeration system commercially available product the benefits of the ejector refrigeration system should be compared to that of absorption refrigeration system.

3. Ejector

The essential components of jet ejector are a supersonic nozzle, chamber for secondary stream suction, a mixing section and a diffuser. To achieve very high kinetic energy of motive stream, a typical converging diverging nozzle is used. The kinetic energy of the motive stream is increased by interior energy and related flow work due to pressure.

3.1 Working Principle

A supersonic flow is created at the outlet of the motive nozzle. The condition of the fluid changes from liquid phase to liquid vapour phase when it passes through the nozzle, due to the drop in pressure in the nozzle, which is even lower than the saturation pressure. The gleaming of the motive fluid is delayed in the interior of the nozzle by the effects of hydrodynamic non equilibrium and thermodynamics. Due to lower pressure created by the primary nozzle, the secondary fluid is drawn in to the jet ejector through the secondary nozzle. In the suction chamber the high-speed motive fluid begins to interact with the entrained fluid. The secondary fluid is accelerated due to the momentum transferred from the primary flow. To pre accelerate the flow of secondary an extra suction nozzle can be used. This helps to reduce the difference in velocity between the motive and entrained fluid streams and thereby excessive shearing losses is reduced.

The mixing of the motive and entrained fluid takes place at constant pressure. The process is repeatedly affected by the shock wave phenomena, which results in a significant rise in pressure and drop in velocity. The mixed flow then passes through the diffuser. The role of the diffuser is to recover the available kinetic energy at the mixing exit and transform it into the pressure energy. The total outlet pressure at the exit of the diffuser is always in between the inlet flow pressures of the primary and secondary fluid. The two main characteristics, which regulates the ejector working performance, are the compression ratio and the entrainment ratio, where the compression ratio is the ratio between the discharge and entrained fluid pressure.

3.2 Advantages of ejector system

- Since the installation and the operating costs of the Ejector systems are low they are considered the most reliable technology in refrigeration
- Because of its high reliability and easiness, they are extensively used in power plants, aerospace, propulsion and cooling applications
- The system is with low noise and virtually no vibration since no moving parts are involved.
The ejector systems have a great potential application in saving energy and give security to the environmental effects by utilizing the waste and thermal energy of low grade such as in air-conditioning and refrigeration.

By removing compressor, related power loss disappears.

Problems such as vibration, noise of compressor and driving belt have been eliminated.

3.3 Disadvantages of ejector system

- The limiting factor is that the efficiency of the conventional ejectors are typically low.
- It has very low performance values in comparison with the systems of vapor absorption and compression.

4. Vapour jet refrigeration system

The system consists of a boiler, an evaporator, a condenser, a liquid pump, an expansion device and an ejector. There are two loops in this refrigeration system. One is power loop, which consist of liquid pump, boiler, condenser and jet ejector, where primary fluid is pumped to higher pressure by liquid pump and heated in the boiler. The high-pressure and high temperature refrigerant is fed to the jet ejector which draws the refrigerant of low pressure and temperature, mixes with primary refrigerant and compresses to high pressure and high temperature gas. The refrigerant then passes through the condenser and it gets converted to liquid refrigerant. A portion of liquid refrigerant flows back to the pump and the remaining refrigerant is sent to the refrigeration cycle.

Other loop is refrigeration cycle where the refrigerant is expanded in the expansion device and the low temperature refrigerant is allowed to pass through the evaporator and refrigeration occurs in the evaporator coil.

5. Design of ejector

The various assumptions for the design of jet ejector are

- The flow is one dimensional and steady.
- Velocities are negligible at all inlet and exit conditions of ejector such as in motive and secondary inlet and at the diffuser exit.
- The losses due to the friction and mixing in the nozzle and the diffuser are distinct in terms of isentropic efficiency.
- The walls of the nozzle and the diffuser are adiabatic.
- The motive and the entrained fluids are identical and they should have the same molecular weight and specific heat values.
- The two streams mixing is assumed to be finished before the occurrence of normal shock wave and carried out at constant static pressure.
- Normal shock due to mixing takes place at end of mixing chamber at constant area.
- The motive and the entrained fluids are dry saturated vapours and obey the ideal gas behaviour.

The primary fluid from the boiler enters the ejector with a high pressure and low velocity which is called as the initial stagnation condition. As the motive fluid flow enters the ejectors converging part, its pressure gets decreased due to the reducing area and its velocity is increased.

5.1 Mass flow rate

\[ Q_E = m(h_{g,e} - h_{f,c}) \]  

(1)
Where $Q_E =$ cooling load in kW and it is the heat absorbed at the evaporator

$h_{g,e} =$ vapour’s specific enthalpy at the evaporator outlet temperature

$h_{f,e} =$ liquids specific enthalpy at the condenser saturation temperature

5.2 Entrainment ratio

$$ER = \left( \frac{m_2}{m_1} \right)$$ (2)

Where $ER =$ Entrainment ratio

$m_1 =$ mass flow of primary fluid

$m_2 =$ mass flow of secondary fluid

5.3 Static pressure and temperature

The ratio between the fluid to sound velocities is called Mach number

$$M = \frac{\gamma}{a}$$ (3)

$$a = \sqrt{\gamma R T}$$ (4)

Where $M =$ Mach number

$R =$ gas constant of the refrigerant in J/kgK

$\gamma =$ specific heat ratio

$a =$velocity of sound in m/s

$V =$velocity of the fluid in m/s

Total temperature and total pressure can be calculated by using Mach number and static conditions at different section in the nozzle.

$$T_0 = T \times \left[ 1 + \left( \gamma - 1 \right) \times M^2 \right]$$ (5)

$$P_0 = P \times \left[ 1 + \left( \gamma - 1/2 \right) \times M^2 \right]^{(\gamma-1)/\gamma}$$ (6)

$$\frac{P_0}{P} = \left[ \frac{T_0}{T} \right]^{(\gamma-1)/\gamma}$$ (7)

Where $T_0 =$ Total temperature of the fluid

$P_0 =$ Total pressure of the fluid

$P =$ Static pressure of the fluid

$T =$ Static temperature of the fluid

5.4 Mixing section

The motive and entrained flows entrainment process is governed by momentum equation
\[ \oint A_{dp} = (1 + R_m)V_m - m_1V_1 - m_2V_2 \]  
(8)

\[
V_m = \frac{(m_1V_{1,1} + (m_2V_2)}{m} \]  
(9)

\[
T_{0,m} = \frac{(m_1T_{0,1,1} + (m_2T_{0,2})}{m} \]  
(10)

\[
D = 2 \times \sqrt{\frac{mXRXT}{\pi X PV}} \]  
(11)

Where \( V_m, T_m \) is velocity & temperature at the mixing region of ejector

\( D \) is the diameter of the diffuser.

\( m_1, m_2 \) is the mass flow rate of primary and secondary fluid.

### 5.5 Losses in ejector

The major losses in a jet ejector are frictional loss, mixing and kinetic energy loss and loss due to shock waves. The loss due to friction of the ejector occurs throughout the length, whereas the mixing occurs in the mixing section where the both streams interact. Kinetic energy losses are those, which occur within the convergent part of the nozzle and within the diffuser.

These result from frictional effects caused by flow separation. Flow separation occurs mostly on the divergent passages when flow area increases rapidly. This leads to strong turbulence near the wall. Shock wave loss is an irreversible process that occurs at the end of the mixing section. This loss leads to decrease in kinetic energy of the fluid flow and drastically the ejectors pressure lift ratio gets reduced.

### 6. Constant Rate of Momentum Change method

This method is designed with the diffuser geometry which eliminates the thermodynamic shock process that occur in the mixing chamber. At constant rate the fluid flow momentum is changed, which resulted in the elimination of shock wave and, which results in the gradual increase of static pressure from inlet to that of exit[3]. The momentum change at constant rate improves the entrainment ratio. It increases the pressure lift ratio and improves the performance of ejector[7].

CRMC design is based on the equation:

\[
\frac{dM}{dX} = \gamma \]  
(12)

Where \( \frac{dM}{dX} \) = rate of change of momentum constant

\( \gamma \) = specific heat ratio

### 7. Vapour jet refrigeration system with regenerator

The schematic of Vapour jet refrigeration cycle is shown in figure 1. The system consists of a boiler, solar flat plate collector, an evaporator, a condenser, a liquid pump, an expansion device, an ejector and a regenerator. Heat addition to the primary fluid is carried out in three stages. In the first stage waste heat available in the refrigerant that flows from the ejector outlet to the condenser is conserved by transferring the heat to low temperature primary fluid using a counter flow heat exchanger called regenerator. The super-heated refrigerant from the diffuser section of the ejector is condensed in the
condenser. A large amount of coolant is required to condense super-heated vapour to saturated liquid. In order to avoid this, a heat exchanger is incorporated between diffuser and the condenser. The liquid refrigerant from the pump can be used to cool this super-heated vapour wherein it preheats the fluid entering the boiler. Consequently, the heat supplied at the boiler can be reduced. The temperature of the primary refrigerant increases from $T_2$ to $T_3'$. In the second stage heat addition occurs in the solar flat plate collector. The temperature of primary refrigerant further increases from $T_3'$ to $T_3''$. The third stage of heat addition occurs in the boiler. Temperature is increased from $T_3'$ to $T_4$, i.e. till the refrigerant is heated to the required temperature.

**Figure 1.** Schematic diagram of Vapour Jet refrigeration system with regenerator

**Figure 2.** T-S diagram of Vapour Jet refrigeration system with regenerator

7.1 Heating loads
The equations of energy balance under the steady state conditions are:

\[ Q_S = m_1 \times (h_{g,b} - h_{f,p}) \]  
(13)

Where \( Q_S \) = Heat supplied at the boiler in and flat plate collector kW
\( h_{g,b} \) = specific enthalpy of vapor at the boiler outlet
\( h_{f,p} \) = specific enthalpy of liquid at the pump outlet

\[ Q_C = m_1 \times (h_{g,d} - h_{f,c}) \]  
(14)

Where \( Q_C \) = Heat rejected at the condenser in kW
\( h_{g,d} \) = vapour’s specific enthalpy at the ejector outlet
\( h_{f,c} \) = liquids specific enthalpy at the condenser outlet

\[ P_p = m_1 \times W_P \]  
(15)

\[ W_P = V_C \times (P_b - P_c) \]  
(16)

Where \( P_p \) = pump power required in kW
\( W_P \) = specific work done by the pump
\( m_1 \) = mass flow rate of primary fluid(kg/s)
\( V_C \) = specific volume (m³/kg)
\( P_b \) = boiler pressure in bar
\( P_c \) = condenser pressure in bar.

7.2 Performance Evaluation

Coefficient of performance

COP is defined as the ratio of refrigeration effect obtained to the work done by the refrigerant.

\[(\text{COP})_{\text{th}} = \frac{CL}{Q_S + W_P} \ \text{(Without regenerator)} \tag{17}\]

Where \( CL \) = Cooling load
\( Q_S \) = Heat supplied to the boiler
\( W_P \) = Pump work

\[(\text{COP})_{\text{th}} = \frac{CL}{Q_R + W_P} \ \text{(With regenerator)} \tag{18}\]

\( Q_R \) = Heat supplied to the preheated fluid coming from the regenerator

\[ Q_R = Q_S - Q_{\text{reg}} \]  
(19)

\( Q_{\text{reg}} \) = heat gained by the refrigerant in the regenerator

The refrigerant properties of R134a are taken from standard refrigerant property tables.
Assuming the nozzles, diffuser, pump are cent percent efficient and neglecting the pressure drop in the condenser, evaporator and boiler calculations are made.

7.3 Inference
It is observed that the incorporation of regenerator in a system with Conventional ejector increases the COP of the cycle. It is evident that with regenerator having 10% effectiveness, the COP of the system is increased by 6.25%. For every 10% increase in effectiveness of regenerator, it is seen that COP is increased by 11.7%.

It is observed that the incorporation of regenerator in the system with CRMC ejector increases the COP of the cycle. It is evident that with regenerator having 10% effectiveness, the COP of the system is further increased by 3.23%. For every 10% increase in effectiveness of regenerator, it is seen that COP is increased by 13.5%. Graph.1 shows the Variation of COP with regenerator effectiveness.

| REGENERATOR EFFECTIVENESS (%) | (COP)TH(WITHOUT REGENERATOR) | (COP)TH(WITH REGENERATOR) |
|-------------------------------|-----------------------------|-----------------------------|
|                               | CONVENTIONAL EJECTOR SYSTEM | CRMC EJECTOR SYSTEM         |
|                               | 0.16                        | 0.3                         | 1.86                        | 3.24                        |
| 90                            | 0.16                        | 0.3                         | 0.91                        | 1.54                        |
| 80                            | 0.16                        | 0.3                         | 0.59                        | 1.06                        |
| 70                            | 0.16                        | 0.3                         | 0.44                        | 0.82                        |
| 60                            | 0.16                        | 0.3                         | 0.35                        | 0.66                        |
| 50                            | 0.16                        | 0.3                         | 0.29                        | 0.55                        |
| 40                            | 0.16                        | 0.3                         | 0.25                        | 0.48                        |
| 30                            | 0.16                        | 0.3                         | 0.22                        | 0.42                        |
| 20                            | 0.16                        | 0.3                         | 0.19                        | 0.37                        |
| 10                            | 0.16                        | 0.31                        | 0.17                        | 0.32                        |
8. Parametric study on different boiler temperature

8.1 Boiler temperatures effect on COP (theoretical)

The main objective of this study is to evaluate the performance of the system for various boiler temperatures. The operating conditions are condenser temperature 313K and evaporator temperature 278K. The ratio of refrigeration capacity to boiler heat and pump work is used to calculate the COP. It is seen from the graph that the coefficient of performance is found to be decreasing as the heat that is supplied to the boiler increases with increase in temperature.

8.2 Effect of boiler temperature on (COPth)RE

It is found that when the boiler temperature exceeds an optimum value, an increase in primary mass flow rate leads to an increase in the heat supply in the boiler consequently which decreases the coefficient of performance. When the effectiveness of the regenerator varies the coefficient of performance also varies. The boiler temperature is a function of heat supplied. Hence it affects the coefficient of performance.
8.3 Effect of boiler temperature on pumping power

Pumping power is plotted against different temperatures of boiler. As the temperature is increased, then there is an increase in the entrainment ratio hence the primary mass flow decreases, which in turn increase the pumping power. Pump work is negligible on comparing to heat supplied. The coefficient of performance is not much varied by the variation of the pumping power.

8.4 Effect of the temperature of boiler on ER

The entrainment ratio obtained on the various boiler temperatures is plotted. It is observed that the increase in boiler temperature leads to decrease in the flow rate through the motive nozzle. Hence power required to pump the working fluid becomes less. As boiler temperature is increasing, the heat that has to be supplied is high thereby it decreases the co-efficient of performance is decreased.
Graph 5. Effect of boiler temperature on Entrainment Ratio

8.5 Effect of solar flat plate collector temperature on Diffuser Outlet Temperature
It is observed that the increase in boiler temperature leads to decrease in amount of mass flow rate through primary nozzle. Hence the temperature of the fluid entering the ejector is found to be high which entrains the secondary flow from evaporator. The fluids mix together and diffuse in the ejector. The temperature at the outlet of diffuser is directly proportional to the boiler temperature.

Graph 6. Effect of boiler temperature on Diffuser Outlet Temperature

8.6 Effect of boiler temperature on heat supplied at boiler
It is observed that the increase in boiler temperature leads to the heat supply at the boiler is high to attain such high temperatures. More amount of heat supply decreases the coefficient of performance.

Graph 7. Effect of boiler temperature on heat supplied at boiler
8.7 Effect of solar flat plate collector Temperature on Heat Rejected at Condenser
It is seen that the heat rejection at the condenser increases because of the increase in diffuser outlet temperature, which demands large condenser. For the same size of condenser, the heat that has to be given out to liquefy the total mass of the refrigerant, increases.

Graph 8. Effect of boiler temperature on Heat Rejected at Condenser

9. Computational fluid analysis using FLUENT
Using the design dimensions of the jet ejector theoretical simulation is carried out using FLUENT software. The jet ejector geometry is created using GAMBIT and a mesh size of 0.125 mm is used for meshing. Pressure inlet boundary condition is specified for the primary and secondary flow inlets. Pressure outlet boundary condition is specified for the exit of the jet ejector. To define the working fluid used in the simulation since R134a is not available in fluent database, the properties have been incorporated from NIST real gas properties.

NIST Real Gas:
Material name: R134a--1, 1, 1, 2-tetrafluoroethane Molecular Weight: 102.032
Critical properties:
  • Temperature: 374.21(K)
  • Pressure : 4.05928e+006 (Pa)
  • Density : 5.01705 (mol/L) 511.9 (kg/m3)
The simulation is carried out and the results of FLUENT simulation matched well with the theoretical analysis of the jet ejector.

10. Conclusion
The vapour ejector refrigeration system for 0.75ton for the purpose of automobile air-conditioning been designed theoretically. A regenerator has been incorporated to increase the COP. A parametric study on the effect of variables by varying the boiler temperatures at constant pressure has been reported. The computational fluid dynamics analysis on the designed system had been carried out using fluent software. It is concluded that the CRMC ejector is found to be more efficient than the conventional ejector and COP of the system increased with addition of regenerator in the cycle.
11. References

[1] Zhu Y and Jiang P 2012 Hybrid vapor compression refrigeration system with an integrated ejector cooling cycle International Journal of Refrigeration 35 68–78.

[2] Sun D-W 1998 Evaluation of a combined ejector-vapour-compression refrigeration system International Journal of Energy Research 22 333–42.

[3] Kumar V, Singhal G and Subbarao P M V 2013 Study of supersonic flow in a constant rate of momentum change (CRMC) ejector with frictional effects Applied Thermal Engineering 60 61–71.

[4] Selvaraju A and Mani A 2006 Experimental investigation on R134a vapour ejector refrigeration system International Journal of Refrigeration 29 1160–6.

[5] Cizungu K, Mani A and Groll M 2001 Performance comparison of vapour jet refrigeration system with environment friendly working fluids Applied Thermal Engineering 21 585–98.

[6] Kashyap S and Gupta R C 2011 Theoretical study of ejector refrigeration system with working fluid R410a International Journal of Engineering Science and Technology 3 6508-6513.

[7] Kitratana B, Aphornratana S, Thongtip T 2017 comparison of traditional and CRMC ejector performance used in a steam ejector refrigeration Energy Procedia 138 476-481.