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Ejector Expansion Refrigeration Systems

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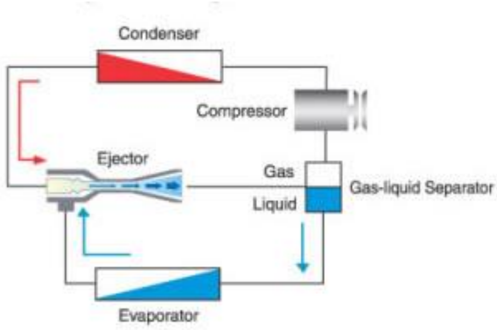
ABSTRACT : *Refrigeration forms the basic essence of living comfort. Ejector Expansion Refrigeration Cycle (EERC) is a not so commonly used method of refrigeration. The use of this method is quite understated. It increases the efficiency of the normal refrigeration cycle by almost 16% over the basic cycle by utilising the energy wasted otherwise in the expansion valve in form of expansion process losses. EERC system has high potential which if harnessed properly could prove to be a very efficient method of refrigeration. This paper aims to showcase the real features of this method in a hope that it finds its way out in the commercial industry today.*

I.INTRODUCTION

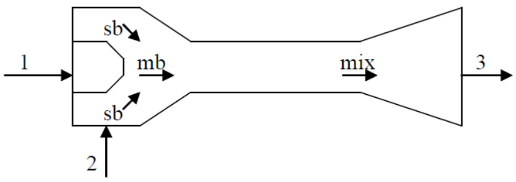
Refrigeration is one of the leading uses of electric power across the globe. The term "refrigeration" refers to air-conditioning for homes, businesses, and industry and the operation of refrigerators, freezers, and heat pumps. The technology most often in use today for refrigeration purpose is the vapour compression cycle which is 100 years old, inefficient, and environmentally unsound. Since the 1980's, the refrigeration industry has faced pressure to improve efficiency and reduce the emission of the chlorofluorocarbons (CFC) compounds which pose a serious threat to the environment. Attempts to decrease CFC emissions by using alternate compounds have typically made refrigeration devices less efficient.

Theoretically, in a refrigeration cycle, the pressure drop is considered as an isenthalpic process where the enthalpy remains constant. However, isenthalpic process causes a decrease in the evaporator cooling capacity due to energy loss in the throttling process. An efficiency-enhancing alternative was proposed to recover this energy loss, which uses an ejector that can be used to generate isentropic condition where the entropy remains constant in the throttling process. Such a cycle is called as ejector expansion refrigeration cycle. The ejector expansion refrigeration cycle is a variant of the standard vapour compression cycle in which an ejector is used to recover part of the work that would otherwise be lost in the expansion valve.



Schematic of an ejector expansion refrigeration cycle.

This method uses a two-phase ejector as an expansion device while the conventional refrigeration cycle uses an expansion valve. A typical ejector consists of a motive nozzle, a suction nozzle or receiving chamber, a mixing section and a diffuser. High pressure motive stream expands in the motive nozzle and its internal energy converts to kinetic energy. The high speed motive stream entrains low pressure suction stream into the mixing section. Both streams exchange momentum, kinetic and internal energies in the mixing section and become one stream with almost uniform pressure and speed. The stream converts its kinetic energy into internal energy in the diffuser to reach a pressure higher than the suction stream inlet pressure.



The working process of an ejector is shown. The motive stream expands in the motive nozzle from the high pressure P1 to the receiving chamber pressure P_b. The enthalpy reduces from h1 to h_{mb} and the velocity increases to u_{mb}. The suction stream expands in the suction nozzle from pressure P2 to P_b. The enthalpy reduces from h2 to $h_{sb}$and the velocity increases to u_{sb}. The two streams mix in the mixing section and become one stream with pressure Pm and velocity u_{mix}. This stream further increases its pressure to P3 in the diffuser by converting its kinetic energy into internal energy.

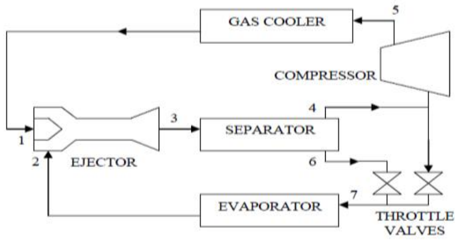
Menegay and Kornhauser (1994) proposed an improved design method for ejectors used as refrigerant expansion devices in vapour compression cycles. They showed that the assumption that the motive and suction nozzles should have the same outlet pressures for the ejector to reach optimal efficiency is only valid when both the motive and suction nozzles have efficiencies equal to one. Under all other circumstances, the motive nozzle discharge pressure is, optimally, greater than that of the suction nozzle. A one-dimensional, homogeneous equilibrium model with constant area mixing assumption was used for the ejector design. Refrigerant R-134a was used in their analysis.

Harrell and Kornhauser (1995) reported on performance tests of a two-phase flow ejector for ejector expansion refrigeration applications. Theoretically, a cooling COP improvement of approximately 23% is achievable for a typical refrigerating cycle and an ideal ejector. If the ejector has the same performance of typical single-phase ejectors, an improvement of 12% could be achieved. However, their preliminary data only showed an ejector performance corresponding to refrigeration cycle COP improvements ranging from 3.9% to 7.6%. They suggested that a more thorough understanding of the flow occurring within the ejector must be developed to achieve the operating potential of the ejector expansion refrigeration cycle. This is how the concept of two-phase ejector came into realization.

The two-phase ejector refrigeration cycle enables the evaporator to be flooded with refrigerant, resulting in a higher refrigerant-side heat transfer coefficient. EERC expands the liquid refrigerant in two steps. The first step is through a specifically designed nozzle where the liquid is used to increase the pressure of the gas returning to the compressor. After this stage, the liquid refrigerant is collected in a receiver where it is metered into the evaporator by conventional methods. The experimental study shows that this method gives a higher cooling capacity and a higher coefficient of performance. Moreover, the pressure ratio and the discharge temperature of the compressor of this method are lower than those of the conventional refrigeration cycle.

Study shows that, at condensing temperature of 40 °C and evaporation temperature of 5 °C, the coefficient of performance (COP) peaks up to 5.91, 21% over the conventional cycle. In addition, the volumetric cooling capacity (VCC) of the cycle which employs the use of ejector expansion jumps to 2590.76 kJ/m^3. Furthermore, the COP improvement over basic expansion cycle increases due to increase in pressure lift ratio with the increase in condenser temperature and decrease in evaporator temperature. In initial testing EERC performance was poor, mainly due to thermodynamic non equilibrium conditions in the ejector motive nozzle. Modifications were later made to correct this problem, and significant performance improvements were found.
This new technology will aid reduce energy use and greenhouse gas emission of vehicle air conditioning, refrigeration units and residential heat pumps. When this technology is installed in a refrigeration unit and combined with other complementary breakthroughs in components and controls, it improves efficiency by more than 50%, leading to a 70% reduction in refrigerant emissions and a 60% reduction in carbon dioxide emissions.
CALMAC Manufacturing Corporation, world leader in the product design and manufacture of Off-Peak Cooling (OPC) systems by utilizing Thermal Energy Storage (TES) to air condition buildings had come up with a proposal to popularize EERC method of refrigeration. Calmac had expended significant internal resources to overcome prior EERC failures in the industry. For example, industry efforts to achieve EERC had not generated sufficient pressure within the ejector nozzle to enhance refrigeration efficiency. Calmac, however, had developed techniques to achieve a six-percent improvement in energy expended for refrigeration through the use of the EERC.

That level of improvement was not high enough to make the technology cost effective, but, with further research and improvisations, Calmac expected a 10-percent improvement for air-conditioning and up to a 20-percent improvement for other, lower temperature applications. Moreover, more efficient refrigeration would reduce both the size of the equipment needed in the process and the potential release of CFC's into the environment. When improvements would reach the 10- percent threshold, cost savings would then be high enough to encourage original equipment manufacturers (OEMs) to use the EERC process. At that point, economic and environmental spill-over could be achieved.



Schematic of the Ejector Expansion Refrigeration Cycle

To simplify the theoretical model of the ejector expansion refrigeration cycle, the following assumptions are made:

1. Neglect the pressure drop in the gas cooler and evaporator and the connection tubes.
2. No heat losses to the environment from the system, except the heat rejection in the gas cooler.
3. The vapour stream from the separator is saturated vapour and the liquid stream from the separator is saturated liquid.
4. The flow across the expansion valve or the throttle valves is isenthalpic.
5. The compressor has a given isentropic efficiency.
6. The evaporator has a given outlet superheat and the gas cooler has a given outlet temperature.
7. The flow in the ejector is considered a one-dimensional homogeneous equilibrium flow.
8. Both the motive stream and the suction stream reach the same pressure at the inlet of the constant area mixing section of the ejector. There is no mixing between the two streams before the inlet of the constant area mixing section.
9. The expansion efficiencies of the motive stream and suction stream are given constants. The diffuser of the ejector also has a given efficiency.

Using these assumptions, the equations for the ejector expansion cycle were setup. Assuming that the pressure before the inlet of the constant area mixing section of the ejector is Pb and the entrainment ratio of the ejector is w, the following equations for the ejector section before the inlet of the constant area mixing section can be identified.

The motive stream follows an isentropic expansion process from P1 to Pb before it enters the constant area mixing section.

S mb,is =S mi (1)

The corresponding enthalpy of the motive stream at the end of the isentropic expansion process can be determined from the property relationship, f, derived from the equation of state.

hmb is□□f(S mb,is, Pb) (2)

Using the definition of expansion efficiency, the actual enthalpy of the motive stream at the inlet of the constant area mixing section of the ejector can be found.

Efficiency ŋ = (h_{mi} – h_{mb})/ (h_{mi}-$h_{mb,is}$) (3)

Applying the conservation of energy across the expansion process, the velocity of the motive stream at the inlet of the constant area mixing section is given by Equation (4).

Umb= Sq root of 2(h mi-h mb) (4)

The specific volume of the motive stream at the inlet of constant area mixing section can be found by a property relationship:

Vmb= f (h_{mb}, P_b) (5)

Using the conservation of mass, the area occupied by the motive stream at the inlet of constant area mixing section per unit total ejector flow rate is given by:

a_{mb}= v_{mb}/[u_{mb}(1+w)] (6)

The calculation sequence for the suction stream is analogous to the one for the motive stream as shown in Equations

(7) through (12).

$S_{sb,is} = S_{si}$ (7)

hsb is□□f(S sb,is, Pb) (8)

Efficiency $\eta = (h_{si} - h_{sb}) / (h_{si} - h_{sb,is})$ (9)

Usb= Sq root of 2(h si-h sb) (10)

Vsb= f (h_{sb}, P_b) (11)

$a_{sb} = [w^* v_{sb}]/[u_{sb}(1+w)]$ (12)

To calculate the mixing section outlet conditions, an iteration loop is applied. First, a value of the outlet pressure Pm is guessed. By assuming that the momentum conservation is s
atisfied for the mixing process in the constant areamixing section, the velocity of the mixing stream at the mixing section outlet is calculated by using in Equation (13).

$$P_b (a_{mb}+a_{sb}) + [1/(1+w)]u_{mb} + w/(1+w)u_{sb} = P_m (a_{mb}+a_{sb}) + U_{mix} \quad (13)$$

Using the conservation of energy, the enthalpy of the mixing stream at the mixing section outlet can be found.

$$h_{mi} + wh_{si} = (1+w)(h_{mix} + u_{mix}.u_{mix}/2) \quad (14)$$

From a property relationship, the specific volume of the mixing stream can be found.

$$V_{mix} = f\ (h_{mix} + P_m) \quad (15)$$

In the last step, the conservation of mass for the constant area mixing section requires that Equation (16) holds true.

$$(a_{mb} + a_{sb})u_{mix}/V_{mix} = 1 \quad (16)$$

The mixing pressure is then iterated until Equation (16) is satisfied.

In the next section, the calculations of the diffuser section of the ejector are presented. First, the entropy of the mixing stream at the outlet of the mixing section is found and set equal to the isentropic diffuser outlet entropy:

$$S_{mix} = f\ (h_{mix}, P_m) \quad (17)$$

$$S_{d,is} = S_{mix} \quad (18)$$

The stream enthalpy at the diffuser outlet can be found by applying the conservation of energy across the ejector:

$$(1+w)h_d = h_{mi} + wh_{si} \quad (19)$$

Given the efficiency of the diffuser, the isentropic enthalpy at the diffuser outlet can be found:

$$\eta_d = (h_{di,is} - h_{mix}) / (h_d - h_{mix}) \quad (20)$$

The diffuser outlet pressure and quality are then obtained from property relationships:

$$P_d = f\ (h_{d,is}, s_{d,is}) \quad (21)$$

$$X_d = f\ (h_{d,is}, P_d) \quad (22)$$

It should be noted that the entrainment ratio of the ejector and the ejector outlet quality must satisfy equation (23) in order to realize the cycle.

$$(1+w)x_d > 1 \quad (23)$$

Since it is assumed that the fluid streams leave the separator at saturated conditions, the gas and liquid enthalpies at the outlet of the separator can be found from property relationships:

$$H_{f,d} = f\ (P_d, x=0) \quad (24)$$

$$H_{g,d} = f\ (P_d, x=1) \quad (25)$$

Using a mass balance, the feedback vapour stream flow rate is given by:

$$m_{g,d} = (1+w)x_d - 1 \quad (26)$$

And the saturated liquid flow rate leaving the separator is given by:

$$M_{f,d} = (1+w)(1-x_d) \quad (27)$$

For a given superheat at the evaporator outlet, the enthalpy at the evaporator outlet can be found from a property relationship:

$$h_{e,o} = F\ (P_{e,}\ t_{e,o}) \qquad (28)$$

The evaporator capacity can be calculated as:

$$Q_{o,n} = wh_{e,o} - m_{f,d}h_{f,d} - m_{g,d}h_{g,d} \qquad (29)$$

The suction stream inlet enthalpy is equal to the evaporator outlet enthalpy:

$$h_{si} = h_{e,o} \qquad (30)$$

To find the compression work of the compressor, first the isentropic compressor outlet conditions are evaluated:

$$S_{g,d} = F\ (P_d,\ x{=}1) \qquad (31)$$

$$S_{comp,is} = S_{g,d} \qquad (32)$$

$$h_{comp} = f\ (S_{comp,is},\ P_c) \qquad (33)$$

From the isentropic efficiency of the compressor, the actual enthalpy at the compressor outlet can be found:

$$\eta_{comp} = (h_{comp,is} - h_{g,d})/\ (h_{comp} - h_{g,d}) \qquad (34)$$

Then, the compression work done by the compressor can be found as:

$$W_{comp,n} = h_{comp} - h_{g,d} \qquad (35)$$

The COP of the ejector expansion cycle can be determined by:

$$COP_n = Q_{e,n}/\ W_{comp,n} \qquad (36)$$

For the basic cycle, the specific evaporator capacity is given by:

$$q_e = h_{eo} - h_{mi} \qquad (37)$$

The isentropic compressor outlet conditions for the basic transcritical CO2 cycle can be found as follows:

$$S_{e,o} = f\ (\ P_e,\ t_{e,o}) \qquad (38)$$

$$S_{comp,is} = S_{e,o} \qquad (39)$$

$$h^b_{comp,is} = f\ (\ s^b_{comp,is},\ P_c) \qquad (40)$$

Using the definition of compressor isentropic efficiency, the actual enthalpy at the compressor outlet of the basic cycle can be found by:

$$\eta_{comp} = (h^b_{comp,is} - h_{e,o})/\ (h^b_{comp} - h_{e,o}) \qquad (41)$$

Then, the specific compressor work of the basiccycle is found by:

$$w_{comp} = h^b_{comp} - h_{e,o} \qquad (42)$$

Finally, the performance of the basic cycle is given by:

$$COP_b = q_e/\ w_{comp} \qquad (43)$$

The relative performance of the ejector expansion transcritical CO2 cycle to the basic transcritical CO2 cycle is defined as:

$$R = COP_n\ /\ COP_b \qquad (44)$$

Using the above theoretical model, the effects of the entrainment ratio w and the pressure drop in the receiving section of the ejector *Pe -Pb* on the relative performance of the ejector expansion cycle can be International investigated. In addition, using a given entrainment ratio w and a given pressure drop in the receiving section of the ejector *Pe -Pb*, the effect of different operating conditions on the relative performance of the ejector expansion cycle can be investigated.

II. RESULTS AND DISCUSSION

An ejector expansion refrigeration cycle is proposed to reduce the expansion process losses of the basic refrigeration cycle. A constant pressure-mixing model for the ejector was used to perform a thermodynamic cycle analysis of the ejector expansion refrigeration cycle. The effect of the entrainment ratio and the pressure drop in the receiving section of the ejector on the relative performance of the ejector expansion cycle was investigated using the theoretical model. The performance improvement of the ejector expansion cycle over the basic cycle for different operating conditions was investigated as well. It was found that the ejector expansion cycle improves the COP by more than 16% compared to the basic cycle for typical air conditioning application.

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