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ABSTRACT

The paper describes the development of a novel vapor compression cycle for refrigeration with regenerative use of the potential energy of two-phase flow expansion, which in traditional systems is lost in expansion valves. The new cycle includes a second step compression by an ejector device, which combines the compression with simultaneous throttling of the liquid. The compressor compresses the vapor to approximately 2/3 of the final pressure and additional compression is provided in an ejector, thus the amount of mechanical energy required by a compressor is reduced and the efficiency is increased. Investigations described here were performed under the funding from the US Department of Energy. The thermodynamic model was developed for R22 refrigerant, showing a possible efficiency improvement of 38% as compared to the traditional vapor compression cycle. The theoretical work was followed by building a 10 kW prototype and practical demonstration of 16% energy savings in the first attempt.

1. INTRODUCTION

The investigation of a new cycle for vapor compression refrigeration with using a novel device for non-mechanical compression of refrigerants, called a "condensing ejector" was carried out within this project. The condensing ejector is a two-phase jet device in which a sub-cooled homogeneous working medium in a liquid state is mixed with its vapor phase, producing a liquid stream with a pressure that is higher than the pressure of either of the two inlet streams. The liquid, supplied by a mechanical pump, is mixed with a vapor of high temperature and velocity supplied by a compressor. The mixing takes place first in a convergent section and then in a constant area section of the ejector device. There is a large temperature difference and a high relative velocity between both streams, which results in a high rate of a heat and momentum transfer. The vapor phase is quickly condensed onto the liquid stream, producing rapid transformation from two-phase into single-phase flow with a resulting rise in pressure, called a "condensation shock." This is the two-phase analog to the ram effect used in supersonic aircraft engine design.

While a theoretical basis for the condensing ejector principle has been reasonably established in the past, only two practical applications have been reported—for underwater propulsion and liquid metal MHD power generation (Miguel *et al.*, 1964). Previous research has been concentrated on water-steam mixtures, and this paper presents what is believed to be the first attempt of using the principle with refrigerant as a working medium. This project involved the development of a condensing ejector to be used with refrigerants to further its application in a vapor compression cycle for refrigeration. The major improvement proposed hereby is a second step compression by an ejector device.

2. BACKGROUND OF EJECTOR APPLICATIONS IN REFRIGERATION CYCLES

Vapor compression refrigeration systems typically utilize expansion valves or other throttling devices to lower the pressure of liquid refrigerant and deliver it to the evaporator. The process of throttling is isenthalpic, which means that the kinetic energy produced during the pressure reduction is dissipated and eventually wasted. Therefore, it is desirable to recover this kinetic energy to increase the efficiency of the entire refrigeration cycle. The literature search has revealed that the principal method to accomplish this task in the past was to use the ejector instead of the throttling valve. Through the action of an ejector, the compressor suction pressure is higher than it would be in a standard cycle. This results in less compression work, thus improving in cycle efficiency.

The first theoretical principles of the ejector as reported by Bohdal *et al.* (2003) in his state-of-the art presentation were elaborated by Parsons (1911) while the first prototype was built by Leblanc (1910). Further improvements were introduced by Gay (1931). Ejectors were first applied for refrigeration cycles by Heller (1955) for absorption systems and by Badylkes (1958) for vapor compression systems. In the USA, the first application was reported by Kemper (1966), but only the patent is in existence while no experimental or theoretical background has been published. Following up on this early work, a theoretical analysis was conducted, showing the potential efficiency improvement of 21% when compared with a standard vapor compression cycle (Kornhauser, 1990). The prototype unit was built, however its performance was much less than ideal and reached a maximum of only 5% improvement using working fluids CFCs/ HCFCs/ HFCs. Recent work on ejectors has concentrated on using them in transcritical CO_2 systems where high pressures allow for better recovery of kinetic energy (Daqing, *et al.* 2004). Detailed investigations in particular a constant pressure mixing model for the superheated vapor ejector that was established and the thermodynamic analysis of the ejector expansion for transcritical CO_2 was performed (Elbel*, et al.* 2004). It was found that the COP (Coefficient of Performance) of the CO_2 cycle with an ejector can be improved by as much as 16% over the conventional CO_2 cycle for typical A/C operation conditions. However, only a theoretical model was presented with no supporting practical experiments.

In all applications listed above, the ejector was designed as a classic Venturi nozzle, which means that the outlet cross-section of a motive nozzle must be smaller than the cross section of the mixing chamber. The outlet pressure in Venturi nozzles is the intermediate between pressures of the working and transporting medias. The advantage of using the ejector was to raise the suction pressure to the compressor intake resulting in reduced compression ratio, and increased cycle efficiency. In general, only a few instances of practical use of the ejectors in refrigeration cycles were found. Their application in transcritical CO_2 systems is promising. In cycles working on traditional refrigerants, ejectors are best used in systems with multiple evaporators to equalize temperatures and pressures.

3. CONDENSING EJECTOR

The principle of condensing ejector operation is shown in Fig. 1 with a corresponding diagram of pressure distribution along the length of an ejector. As shown, two separate nozzles are used to accelerate the vapor and liquid streams. The vapor is fed from the compressor, while the liquid is delivered to the nozzle by an auxiliary pump. The stream enters the convergent mixing section (1), exits into a constant area mixing section (2) and leaves the ejector through a diffuser (4). The high relative velocity between the streams produces high values of the heat transfer coefficient, which causes a high rate of condensation. The remaining uncondensed vapor and a liquid enter the constant area mixing section where the “condensation shock” occurs (3) with a completely liquid state produced downstream of the shock. The pressure rises due to lowering the velocity of the stream and the mass flow balance is maintained by a sudden change of density (condensation).

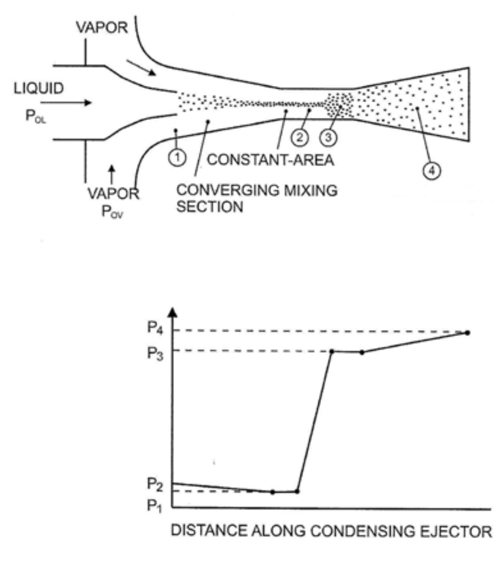


Figure 1. Principle of condensing ejector

A computer simulation is presently being conducted jointly with UMass and is under proposal for NSF funding. For this, a three-dimensional CFD model was modified to simulate flashing flow. This is an extension of past work by Schmidt, *et al.* (1999), which used a two-dimensional CFD code to predict the flashing flow of water through nozzles. The fundamental assumption is that the one-dimensional models that are successful for simpler geometries, such as converging/diverging nozzles can be extended to higher dimensions. The outstanding difficulty is to capture the internal heat transfer between phases. The two phases are presumed to be in hydrodynamic equilibrium (momentum exchange is infinitely fast) but not thermal equilibrium (heat transfer occurs at a finite rate). This makes flashing flow fundamentally different from traditional nucleate boiling or condensation, where the enthalpy for the phase change is exchanged with exterior surfaces. The past work by Schmidt was based on one-dimensional models for the internal heat transfer described by Downar-Zapolski *et al.* (1996).

The return to equilibrium model used by Downar-Zapolski (1996) is the starting point for the current model.

$$\frac{Dx}{Dt} = -\frac{x - \bar{x}}{\Theta} \tag{1}$$

In Eqn. (1) x is the quality and Θ is a local relaxation time, which the authors correlated to density and pressure. The instantaneous value of x is calculated from the local density, the vapor density, and the liquid density. The value of $\bar{x}$ is the quality that the fluid would attain after reaching thermodynamic equilibrium in the absence of work or external heat transfer. Thus $\bar{x}$ is a function of the local pressure and enthalpy, as shown in Eqn. (2). The subscripts "SL" refer to saturated liquid and "SG" refer to saturated vapor.

$$\bar{x} = \frac{h - h_{SL}(p)}{h_{SG}(p) - h_{SL}(p)} \tag{2}$$

The time scale is a function of the local thermodynamic quantities. One of the correlations suggested by Downar-Zapolski (1996) for Θ is given by Eqn. (3).

$$\Theta = \Theta_0 \varepsilon^{-0.54} \varphi^{-2.24} \tag{3}$$

This correlation was developed from measurements of void fraction in long, straight, tubes. The variable φ is a function of the local pressure.

$$\varphi = \frac{p_S(T_{in}) - p}{p_C - p_S(T_{in})} \tag{4}$$

In Eqn. (3), ε is the void fraction and is calculated from the density using

$$\varepsilon = \frac{\rho_L - \rho}{\rho_L - \rho_G} \tag{5}$$

Here p_S is the saturation pressure at the inlet temperature, and p_C is the critical pressure of the fluid. In order to use Eqns (2) through (5) in regions of condensation, where the local pressure is above the absolute value of saturation pressure, the absolute value of φ can be used in Eqn. (3). The experimental data for Θ were made during the transition from liquid to vapor and may not be accurate for condensing fluid. However, closure is required for this case and so some sort of assumption is required. As part of the current work, we are exploring the suitability of this approach for modeling condensation shocks.

In addition to "condensation shocks", the literature describes the other type of pressure raise in ejectors, called either "supersonic shock wave" or "cavitations shock". It has been observed in both homogeneous and non-homogeneous two-phase flow, for ex. in air-water mixture using a transparent nozzle (Bergander, 2005). It appears that this phenomenon depends upon reaching the critical flow in a two-phase mixture.

In standard engineering practice, the dynamic of fluids is described by two fundamental properties: viscosity and compressibility. Specifically for liquid, the viscosity and Reynolds number are determining properties, as speed of liquids is almost always slower than their sonic speed. On the other hand, for gases, which often move with speeds near their sonic speed, a Mach number or compressibility becomes the determining factor for calculations. The situation changes drastically for two-phase mixtures. To determine the dynamics of such flow the existing models still consider modified Reynolds number and viscosity, but traditionally compressibility is not utilized in these calculations. This is the great disadvantage of existing theoretical approach to analysis of two-phase flow because ignoring the influence of Mach number in two-phase flow leads to pipelines vibrations, intensification of waves, and possibly also inaccuracy in predicting LOCA conditions in nuclear reactors (Fisenko, 1987).

As reported by numerous researchers (Van Wijngaarden, 1972; Wallis, 1980,) the sonic speed in two-phase mixture is much lower than that in any of its components. By starting from the volumetric content of gas in gas-liquid mixture and introducing certain assumptions, i.e. no slip between phases and isothermal nature of the flow, Van Wijngaarden (1972) derives his fundamental formula for the speed of sound in two-phase mixture:

$$a^2 = p/\rho_f \beta(1-\beta) \tag{6}$$

where a is the speed of sound, p is the pressure, ρ_f is the density of the liquid phase and β is the volume occupied by the gas in a unit volume of the mixture. The formula contains the result that, unless β is very close to either zero or unity, the speed of sound in the two-phase mixture is lower than speed of sound in pure gas. A minimum exists for $\beta = 0.5$, in which case, at a pressure of 1 bar, a mixture of air and water has a sound velocity of 20 m/s. Similar results were obtained for two-phase homogeneous mixture, such as water-steam, liquid refrigerant-vapor refrigerant. Van Wijngaarden (1972) describes also the process when the vapor is accelerated in a jet device to a velocity at or slightly above the sonic velocity. The mixing of two phases, vapor and liquid, leads to the decrease in local sound velocity and the creation of a "shock wave" with consequent increase of pressure. The pressure ratio achieved by such shock wave can be calculated from the formula (Campbell, *et al.* 1958) for isothermal process:

$$p_2 /p_1 = M^2 \tag{7}$$

where M is the Mach number. For an adiabatic process, the formula given by Fisenko (1987) takes a slightly different form:

$$p_2 /p_1 = 1 + k \beta M^2 \tag{8}$$

Where k is the adiabatic coefficient: $k = c_p/c_v$

The above theory brings about the possibility of obtaining the supersonic flow in two-phase Laval nozzles and this can be considered in propulsion devices. Witte (1969) among others investigated the efficiency of a propulsion device based on injection of compressed air bubbles in the throat section of a nozzle and observed the pressure jump associated with the supersonic flow. Indeed, this was further confirmed by both computer modeling and laboratory experiments (Fisenko, 1987).

4. REFRIGERATION CYCLE WITH CONDENSING EJECTOR

The principle of the condensing ejector, presented above was utilized to construct the cooling/refrigeration system shown in Figure 2. In this new system, the mechanical compressor compresses the vapor to approximately 2/3 of the final pressure. Additional compression is provided by the ejector device, therefore the amount of mechanical energy required by a compressor is significantly reduced. The principle of the proposed system as shown in Figure 2 includes the main piping circuit (1), containing the evaporator (2), a compressor (3), an ejector device (4), a condenser (5), a separator tank (6), an optional intermediate heat exchanger (7) and an expansion valve (8). The circulation of a liquid phase of the working medium is provided by the additional liquid line (9), and a pump (10). The evaporator (2) absorbs the heat from source (11), while the condenser (5) is connected to the heat sink – high temperature heat receiver (12). It needs to note that the device as above can be used also for heating and in this capacity it can operate as a heat pump.

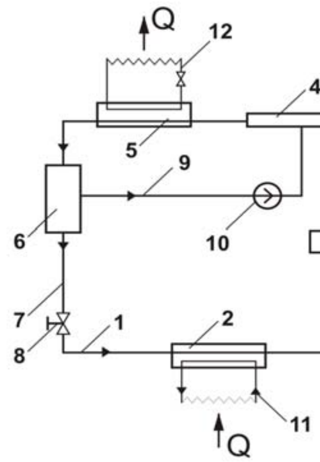


Figure 2. Schematic of the refrigeration system with the condensing ejector as a second-stage compressor

The working medium kept at low pressure vaporizes in the evaporator with using the heat energy of a low-temperature source. Further, the working medium is compressed in the compressor and is sent to the ejector where it mixes with the liquid flow coming from the separator. The flow of working medium is then directed to the condenser where it is cooled by transferring the heat to the high-temperature receiver. The ejector improves the efficiency of the cycle by decreasing the need for energy to run the compressor.

The theoretical energy savings for the new system can be established by analyzing the thermodynamic cycles for the new system vs. traditional single-stage compression cycle. Both cycles are presented in Figure 3 (Samkhan, 2005).

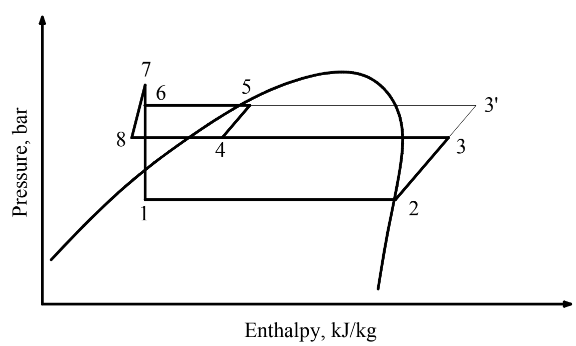


Figure 3. Comparison of p-h diagrams of the new refrigeration cycle with a two-phase ejector, Cycle 1 (points: 1-2-3-4-5-6-1 and 6-7-8-4) and the traditional cycle Cycle 2 (the point: 1-2-3’-6-1).
1-2: evaporation of a part of the working fluid; 2-3: compression of vapor in the compressor (the first step); 3-4-8: mixing of vapor and liquid parts of the working medium in the ejector; 4-5: compression of the working medium in the ejector (the second step); 5-6: isobaric cooling of the liquid working medium; 6-7: compression of a part of the cooled liquid working medium by the pump; 7-8: expansion of this part of the cooled liquid working medium in the ejector; 6-1: throttling of the evaporating part of the working fluid.

Using the above theoretical model, the relative performance of the two-phase ejector R22 cycle in comparison with a similar traditional cycle with the one-stage compression in the same temperature range can be estimated. The p-h diagrams of these cycles are presented in Fig. 3, while the properties of the refrigerant in the characteristic points of diagram in Fig. 3 were determined and further used to calculate the COP of both cycles. The coefficient of efficiency of the hydraulic pump and compressor was assumed to 0.8. As shown in Table I below, the application of the condensing ejector for R22 refrigeration cycle can achieve a 38% theoretical efficiency improvement.

TABLE I
Specific energy characteristics of the cycle with a vapor-liquid condensing ejector (Cycle 1) in comparison to the traditional cycle (Cycle 2)

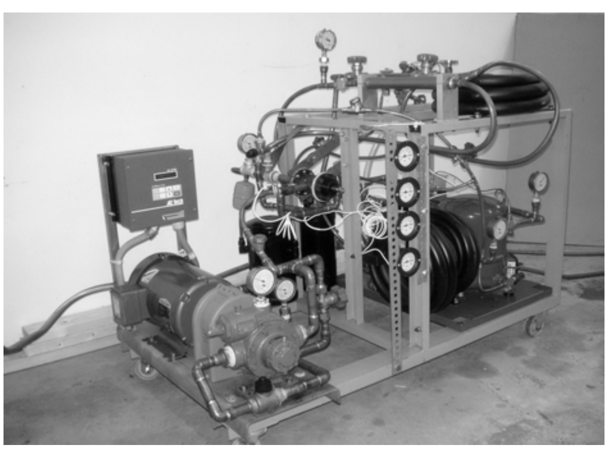
Quantity	Unit	Value	
		Cycle 1	Cycle 2
Heating capacity	kJ/ kg	187.5	190
Cooling capacity	kJ/ kg	152	152
Compressor work	kJ/ kg	22.5	53.7
Pump work	kJ/ kg	15.7	-
Coeff. of performance (COP)		4.9	3.53
COP1/ COP2 ratio		1.38	

Comparing to existing applications of ejectors in refrigeration cycle, our approach differs by the following:

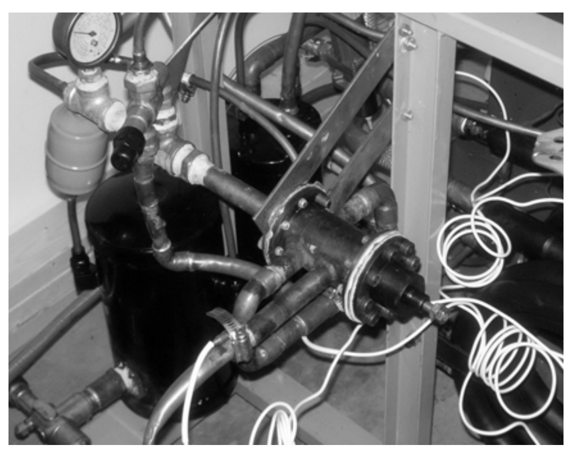
1. Our cycle is characterized by the location of the ejector device after the compressor discharge in order to increase the final cycle pressure (pressure at the inlet to the condenser), while all to-date designs used ejectors for increasing the suction pressure of the compressor,
2. Our ejector is working on the principle of two-phase flow and as such has a much higher efficiency than existing single-phase ejectors. It utilizes the principle of either the condensation shock or critical (choked) flow of two-phase mixture. In a latter case the velocity in the mixing chamber exceeds the sonic velocity and the M>1 condition is achieved not by increasing the velocity of flow but by slowing down the speed of sound (instead of increasing the numerator, we reduce the denominator in the Mach number).
3. All previous designs of the ejector relied on the increase of the pressure in the mixing chamber by the process of equalizing the velocities of both motive and suction streams. Consequently, the value of outlet pressure was intermediate between the pressures of motive and suction streams. Our design produces the outlet pressure much higher than the pressure of any of the stream components. This is achieved by the creation of a “condensation shock”, which was previously described in literature, but never used with refrigerants.

5. LABORATORY EXPERIMENTS

The laboratory stand of 10 kW capacity was fabricated under funding from the US Dept. of Energy according to schematic of Figure 2 and with observing basic principles of design and assembly of refrigeration systems. Its overall view is shown in Figure 4.



(a)



(b)

Figure 4. (a) Overall view of the laboratory stand, and (b) Closer view of the ejector device

The laboratory experiments were then carried out with two main objectives: the first being to investigate the possibility of the pressure jump on the ejector and the second to determine the energy savings for the cycle with the ejector vs. traditional cycle with single step compression. In order to operate in those two modes, the stand was designed with appropriate valves to by-pass the ejector and to operate in the standard vapor-compression regime. The operation in the conventional mode of vapor compression cycle was straightforward and standard results were obtained as shown in Table II. The high pressure was regulated by the temperature in the condenser**.**

When running in the ejector mode, the pressure jump of 0.35 MPa, was observed on the ejector at certain position of the liquid by-pass regulating valve (this corresponds to a specific liquid-to-vapor ratio). This effect was sustained for several seconds at our first attempt, after which time, pressure pulsation followed by "knocking" sound were observed, indicating that the vapor enters the pump suction line. At that moment, the entire amount of discharged liquid was immediately fed into the suction line in order to protect the pump. It appears that the liquid receiver might be too small and more liquid is required to secure the sustained operation. This will be investigated in the next phase of this project. Table II below shows the results for both modes of operation.

TABLE II
Results of Laboratory Experiments

Item	Parameter	Mode_1: Conventional, ejector by-passed	Mode_2: 2-stage compression with ejector	Comments
1	Temp inlet to evaporator (deg C)	18	18	Thermometer
2	Temp outlet from evaporator, (deg C)	4	4	Thermometer
3	Cooling water flow through evaporator (l/min)	8.4	8.4	Measured with flow meter
4	Cooling capacity kJ/min	492.0	492.0	$Q = mc\Delta t$
5	Cooling capacity from R22 p-h graph (kJ/kg)	145	145	Assuming t evap:-1C, pressure 0.46 MPa, x=0.3, η = 0.8
6	Compressor discharge pressure (MPa)	2.1	1.75	Pressure jump from 1.75 to 2.1 MPa on the ejector
7	Pressure at condenser (MPa)	2.1	2.1	Pressure gauge reading
8	Mass flow of refrigerant through compressor (kg/min)	3.4	3.4	
9	Volumetric flow of refrigerant through compressor, (m^3/min)	0.026	0.051	Different vapor density of R22 at various temperatures
10	Compressor work from R22 p-h graph (kJ/kg)	50.8	42.1	Compression in one stage from 0.46 to 2.1 MPa, efficiency 0.8
11	Total compressor work (kJ/min)	172.4	142.9	
12	COP for cooling	2.85	3.31	16% better efficiency
13	Pressure at pump discharge, (Mpa)	N/A	2.65	Pressure gauge reading
14	Δ p at pump (MPa)	N/A	0.55	
15	Mass pump output (kg/min)	0	11.2	Read from pump characteristics at Δp = 0.55 and rpm=285
16	Volumetric pump output (m^3/min)	0	0.011	
17	Energy used by pump, (kJ/min)	0	7.56	Calculated from the formula: $N=v\ \Delta p/0.8$
18	Volumetric ratio β in ejector	N/A	0.82	
19	Liquid refrigerant velocity in the ejector, (m/s)	N/A	26.2	

6. CONCLUSIONS

The objectives of the first phase of this project were met by: 1) conducting a state-of-an-art study, which confirmed that this project might represent the first attempt to practically use two-phase flow phenomena with refrigerant as a working medium, 2) developing the theoretical model that showed possible efficiency improvement of 38% as compared to the traditional vapor compression cycle and 3) designing and fabrication of the heat pump prototype and practically demonstrating 16% energy savings in an initial attempt (see line 12 in Table II).

The key scientific objective was to obtain the pressure jump on the ejector and such condition was indeed observed on the working prototype, where pressure on the ejector has increased by approximately 15-16%. At this time we were able to sustain this process for a short period of time, however conditions were defined to make it fully sustainable. The prototype has achieved energy savings of 16%, based on experimental results (gauge readings) and certain reasonable assumptions and calculations. In the next phase, more realistic measurements will be performed by installing flow meters into the liquid and vapor refrigerant lines. These experiments have defined the conditions to sustain the process and to improve its efficiency. This will be accomplished in the next phase by installing larger receiver tank, charging more refrigerant and optimizing the ejector design, using computer simulation models.

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