FINAL SCIENTIFIC REPORT
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Project Title: New Regenerative Cycle for Vapor Compression Refrigeration

DOE Award Number: DE-FG36-04GO14327

Document Title: Final Scientific Report

Period Covered by Report: September 30, 2004 to September 30, 2005

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Project Objective (as stated in the proposal): The main objective of this project is to confirm on a well-instrumented prototype the theoretically derived claims of higher efficiency and coefficient of performance for geothermal heat pumps based on a new regenerative thermodynamic cycle as comparing to existing technology. In order to demonstrate the improved performance of the prototype, it will be compared to published parameters of commercially available geothermal heat pumps manufactured by US and foreign companies. Other objectives are to optimize the design parameters and to determine the economic viability of the new technology.

Background (as stated in the proposal)**:** The proposed technology closely relates to EERE mission by improving energy efficiency, bringing clean, reliable and affordable heating and cooling to the residential and commercial buildings and reducing greenhouse gases emission. It can provide the same amount of heating and cooling with considerably less use of electrical energy and consequently has a potential of reducing our nations' dependence on foreign oil. The theoretical basis for the proposed thermodynamic cycle was previously developed and was originally called a "dynamic equilibrium" method. This theory considers the dynamic equations of state of the working fluid and proposes the methods for modification of T-S trajectories of adiabatic transformation by changing dynamic properties of gas, such as flow rate, speed and acceleration.

The substance of this proposal is a thermodynamic cycle characterized by the regenerative use of the potential energy of two-phase flow expansion, which in traditional systems is lost in expansion valves. The essential new features of the process are:

1. The application of two-step throttling of the working fluid and two-step compression of its vapor phase.
2. Use of a compressor as the initial step compression and a jet device as a second step, where throttling and compression are combined
3. Controlled ratio of a working fluid at the first and second step of compression.

In the proposed system, the compressor compresses the vapor only to 50-60% of the final pressure, while the additional compression is provided by a jet device using internal potential energy of the working fluid flow. Therefore, the amount of mechanical energy required by a compressor is significantly reduced, resulting in the increase of efficiency (either COP or EER). The novelty of the cycle is in the equipment and in the way the multi-staging is accomplished.

The anticipated result will be a new refrigeration system that requires less energy to accomplish a cooling task. The application of this technology will be for more efficient designs of: 1. Industrial chillers, 2. Refrigeration plants, 3. Heat pumps, 4. Gas Liquefaction plants, 5. Cryogenic systems.

Summary: The new regenerative cycle for vapor compression refrigeration was investigated within the scope of this project. For the purpose of Category I, we concentrated on one specific application – geothermal heat pumps, however the designed and fabricated system can be used successfully in other applications, especially for commercial and residential refrigeration and air-conditioning. The final result of this stage of the project was a well-instrumented prototype of approx. 10-12 kW heat pump unit, operating on the new cycle. The proposed method constitutes a significant improvement to presently used vapor-compression cycles, which relies on one step compression of a working fluid. The disadvantage of the traditional technique is its relatively low efficiency - the effect of heating or cooling is only 2-3 times larger than the consumption of electrical energy. While we were using a similar concept, to an extent, we also proposed a second step compression by a jet device rather than by a mechanical compressor. Such jet device combines compression of a vapor with simultaneous throttling of the liquid. In the proposed system, the compressor compresses the vapor to approximately 50-60% of the final pressure. Additional compression is provided in a jet device using internal potential energy of the working fluid flow. Therefore, the amount of mechanical energy required by a compressor is significantly reduced and a coefficient of performance (COP) can be theoretically increased by a ratio reaching 1.4-1.5.

The main objective of research within the first stage of this project was to demonstrate the feasibility of the new, regenerative cycle for application in refrigeration systems. We achieved it by designing and fabrication of the prototype heat pump with installed experimental ejector nozzle for the second step of compression. Even though ejectors have been known and used to improve the efficiency of refrigeration cycles since the beginning of the 20th century, our device is innovative and differs substantially from previous designs. The following can summarize major differences/innovations:

1. Our cycle is characterized by 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. The presented ejector/nozzle is working on the principle of critical (choked) flow of two-phase (liquid and vapor) mixture, therefore the velocity in the mixing chamber exceeds the sonic velocity. We are using previous discoveries indicating that the sonic speed in two-phase mixture is much lower than that in any of its components. The originality of our approach lies in the fact that we are achieving M>1 (Mach number) not by increasing the velocity of flow but by slowing down the sonic speed (instead of increasing the numerator, reducing the denominator in the formula) by adjusting the composition of the mixture.

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, by utilizing the properties of critical flow, can produce the outlet pressure much higher than the pressure of any of stream components. This is achieved by the creation of a "shock impulse", which was previously described in literature, but never used in practice for increasing the efficiency of ejector devices.

The first stage of the project was declared a success, since the prototype: 1) was working properly, i.e. delivered a cool on the evaporator and a heat on a condenser, in both options, i.e. with and without the ejector device and, 2) the ejector device has produced the jump in the pressure of approximately 15-20% above the compressor discharge pressure, thus saving the energy consumed by the compressor by a similar amount. Even though it is somewhat lower than predicted by theoretical

analysis, the work completed under this feasibility study represents a significant step towards the task of energy savings in refrigeration, air-conditioning and heat pump systems. Experiments presented in this report clearly demonstrate the possibility of obtaining even better results through systematic analysis and further experimentation. This will be the subject for the next stage of investigations with the final objective being the development of the commercial heat pump and/or refrigeration unit working on this novel thermodynamic cycle.

1.   Introduction

This report describes a novel approach to the Rankine vapor compression cycle for cooling and refrigeration. The specific innovation is the application of two-phase ejector as a second step of compression. The innovation has a potential of increasing the efficiency of the standard single-stage vapor compression cycle by up to 30% through a reduction of mechanical compression at the expense of harnessing kinetic energy of gas in the ejector device. The efficiency gain is achieved by separating the working medium to high and low density phases (liquid and vapor) and applying the liquid phase as an additional energy source when injected into accelerated flow of the vapor phase. This innovation was expected to substantially improve the efficiency and reliability of traditional vapor compression cycle (reversed Rankine cycle) for refrigeration and air-conditioning applications. In addition it will reduce the greenhouse gas emission by providing the same amount of refrigeration (cool) with less electric energy.

2.   State-of-the-Art Study

2.1. Use of Ejector 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. In a typical refrigeration cycle, the expansion valve lowers the refrigerant pressure by 5-7 times. The reason for lowering the pressure is to allow the refrigerant to evaporate at certain desired low temperature. However, 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. One method to accomplish this was developed by a group of scientists from City University (London) [7] The literature search has revealed that the principal method to accomplish this task in the past was using the ejector instead of the throttling valve.
The velocity increase in the throat of the ejector device is used to entrain the refrigerant exiting the evaporator by momentum exchange. The following diffuser section of the ejector re-compresses the refrigerant by slowing down the mixed stream. Through the action of an ejector, the compressor suction pressure is therefore higher than it would be in a standard cycle, resulting in less compression work thus improvement in cycle efficiency.
The first theoretical principles of the ejector were elaborated by Parsons in 1900 while the first prototype was built by Leblanc (1910). Further improvements were introduced by Gay in 1931 [9]. Ejectors were first applied for refrigeration cycles by Heller in 1955 for absorption systems and by Badylkes in 1958 for vapor compression systems [8]. In the USA, the first application was reported by Kemper in 1966, but only patent is in existence while no experimental or theoretical background have been published. Following up on this early work, Kornhauser [10] has conducted a theoretical analysis and showed that the ideal ejector cycle resulted in 21% efficiency as compared with standard vapor compression cycle. The prototype unit was built, however its performance was much less than the ideal and reached at maximum only 5% using working fluids CFCs/ HCFCs/ HFCs. This was attributed to shortcomings in the design of the ejector, specifically too simplified two-phase flow model assumed in the design.

Latest work on ejectors had concentrated on using them in transcritical CO_2 systems where high pressures allow for better recovery of the kinetic energy [11],[12], [13]. Detailed investigations were presented in [12], in particular a constant pressure mixing model for the superheated vapor ejector was established and the thermodynamic analysis of the ejector expansion for transcritical CO_2 was performed. It was found that the COP (Coefficient of Performance) of the transcritical CO_2 cycle with an ejector can be improved by as much as 16% over the basic transcritical CO_2 cycle for typical A/C operation conditions. However, only theoretical model is presented in the subject reference with no supporting practical experiments.

In all applications listed above, the ejector was designed as a classic Venturi nozzle. The basic design concept for a Venturi nozzle is that the outlet cross-section of the accelerating device (also called a motive nozzle) must be smaller than the cross section of the mixing chamber. This allows for transportation of one medium at the expense of the energy of the working medium (which can be a steam, gas or liquid). The outlet pressure in the traditional Venturi nozzles is the intermediate between pressures of the working and transporting medias. Further, the mixing chamber was in form of cylindrical channel of certain length, in order to allow for equalizing velocities of motive and suction streams.

In systems utilizing fluorocarbons as refrigerant, the motive stream was the high pressure liquid fed from the condenser or from the bypass on compressor discharge. As the suction stream, the low pressure vapor from evaporator outlet was used. The liquid and vapor phases were mixed and as a result, the two-phase mixture of intermediate pressure was obtained. The vapor phase was then separated from the mixture and fed into the compressor, while the liquid phase was directed via expansion valve to the evaporator inlet. The advantage of using the ejector was higher suction pressure on the compressor intake resulting in reduced compression ratio, and consequent increased cycle efficiency. In systems utilizing CO_2 as a working medium, both inlet streams are in gaseous state due to the transcritical nature of the cycle. The outlet consists of two-phase mixture of intermediate pressure.

In general, only few instances of practical use of ejectors in refrigeration cycles were found in the literature. Their application in transcritical CO_2 systems is certainly promising. In cycles working on traditional refrigerants, ejectors are best used in systems with multiple evaporators to equalize temperatures and pressures [8].

2.2. Theoretical Background of Two-Phase Flow

The innovation presented in this report relies on the principle of two-phase flow, therefore as a part of this project, we have conducted an extensive study of the literature related to this subject. It needs to emphasize that considerable portion of research on the subject has been conducted outside of the USA (Holland, Russia, China), therefore we included non-English literature search as well, especially in Russian, German and Polish languages. In addition to literature study, the Principal Investigator has attended seminars and conferences, in particular Refrigeration and A/C Conference at Purdue U. in July 2004 and International HVAC Conference in Orlando, FL, Feb. 2005. Several trips to universities and telephone discussions with scientists were also undertaken. In particular, PI has traveled to UMass to discuss the theoretical aspects of ejector design with Prof. D. Schmidt and to U. of RI to conduct discussions with Prof. A. Lucia, both recognized experts on the subject. Numerous phone conversations and e-mail correspondence were carried out with scientists in USA, Holland, Poland, Czech Rep. and Russia.

Our main objective in this study was to determine if any previous research was able to describe the conditions for sustained critical flow in two-phase mixtures. The advantages of critical flow are apparent – the independence of flow rate from the discharge conditions (downstream pressure and geometry), in a broad range of counter-pressures. The successful conclusion of this project could provide the fundamentals for design of hydraulic equipment with higher efficiency, for example pumps with flat characteristics, heat exchangers, ideal mixers and

proportioners. Similarly, the ejector design, which was proposed for this project, relies on the condition of sustained critical flow. Our conclusion was that even though many researchers present various results and data from their studies, the governing parameters associated with critical flow in two-phase medium, such as gas content, bubble radius, pressure ratios, flow ratios, etc. remain to be investigated.

It is known that gas-liquid flow has often a non-equilibrium character, which demonstrates itself by occurrence of pulsation of pressure, density, velocity and temperatures for both phases. Under certain conditions, these processes cause undesired effects, such as water hammer, vibrations of pipelines, perturbation of circulation modes and heat exchange, while under different conditions, occurrence of two-phase flow improves the heat exchange and increases the efficiency for many components of energy systems. Therefore, various theoretical schemes were considered in which the dynamic properties of two-phase stream or "bubbly liquids" could be profitably used in the design of propulsion devices. The underlying idea is that expansion of a compressed gas bubble-liquid mixture might be an efficient way to produce the momentum necessary for thrust.

Models exist for the prediction of unsteady two-phase flashing flows in variable cross section ducts and valves, including Homogeneous Equilibrium Model (HEM) and Homogeneous Relaxation Model (HRM) [2], [6]. None of those models however show the existence of a critical condition without making assumptions, which are unrealistic and cannot be justified experimentally. For example, HRM assumes small relative velocities in relation to speed of acoustic waves in medium. Despite this, the critical condition does exist in two-phase flow and can be very well observed in practice as shown by a number of studies, including the observation of this author on water-air mixture using the transparent nozzle (this experiments is described in details further in this report). The sudden change from misty and milky flow to bubbly flow is apparent at certain flow conditions, such as velocity, backpressure and volumetric content of each phase. One of fundamental publications on the subject, by Wallis [2] acknowledges the presence of critical flow but simultaneously confirms that the mathematical model for such condition is extremely complicated since it has to consider not only criticality in one location, but also may include parts of the upstream system. Concluding, it is possible to state that two-phase mixture, if it is sufficiently homogeneous, has completely different properties than each of its component. The most important is the reduction of speed of sound as first observed in 1941 (Wood) [1], which brings the possibility of supersonic flow at moderate velocities: 20-50 m/s.

The majority of research on two-phase flow has been devoted to nuclear reactor behavior under LOCA conditions. A multitude of computer codes were developed with different levels of accuracy and correctness of the underlying models and assumptions. The most comprehensible laboratory experiments to date were carried out in France under MOBY DICK program in 1980's and included study of steady state critical flow in nozzles at medium to high pressures ("steam-water choked flow")[6]. The nozzle model was selected due to its importance in simulation of small breaks in pipe. The results were widely disseminated throughout the scientific community; however, these experiments were not consequential in changing the way for two-phase flow engineering design. Based on our study, it appears that the detailed simulation of fast transient two-phase flow is yet an unresolved problem in spite of its practical importance and the progress in research in last several decades. Such fast transient appears in many industrial applications and processes with variety of initial and boundary conditions, different fluid and different thermodynamic conditions.

In the 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 [3]. Such situation is hard to understand since many publicized research results point to lower speed of sound in two-phase flow and consequently its higher compressibility.

The newer research shows that the two-phase medium is more compressive than the gas and consequently, the speed of sound in two-phase mediums is much lower than that in the homogeneous gas or liquid. 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 [1] derives his fundamental formula for the speed of sound in two-phase mixture:

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

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, etc. These results were confirmed by a number of researchers in USA and abroad. Consequently, a handful of articles published in 1980's and 90's [3], [5] present diagrams for speed of sound vs. β for two-phase mixture. The general character of such diagrams is shown in Figure 1 below.
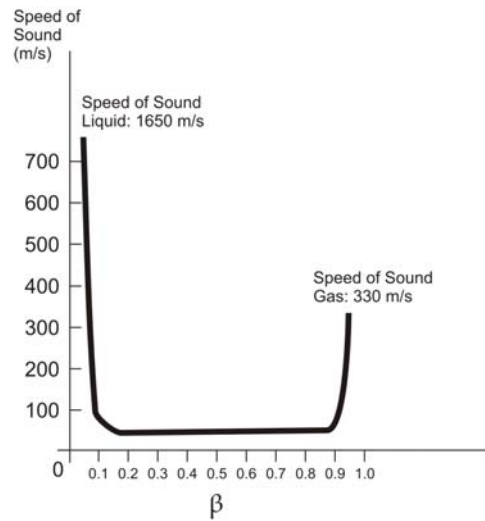

Figure 1. Sonic speed for two-phase flow

Van Wijngaarden [1] describes also the process when the vapor is accelerated in a jet device to a velocity to or slightly above the sonic velocity. With this increase in velocity, the energy of thermal motion of the molecules is converted into kinetic energy of flow and accompanied by a simultaneous decrease of internal energy. This increased kinetic energy is then converted in the jet device into potential energy in form of higher pressure. Such conversion is realized by deceleration of supersonic vapor flow by introducing a liquid component of higher density. The mixing of two phases, vapor and liquid, leads to the decrease in local sound velocity according to the diagram of Figure 1 and the concomitant creation of a "shock wave" with

consequent increase of pressure. The pressure ratio achieved by such shock wave can be calculated from the formula given by Campbell & Pitcher [1] for isothermal process:

$$p_2/p_1 = M^2 \hspace{3cm} (2)$$

where M is the Mach number.
For adiabatic process, the formula given by Fisenko [3] takes slightly different form:

$$p_2/p_1 = 1 + k \, \beta \, M^2 \hspace{3cm} (3)$$

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

The above theory brings about the possibility of obtaining the supersonic flow in Laval nozzles and this can be considered in propulsion devices. Among others, Witte [4] 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.

Reassuming, the state-of-an-art study presented above has concluded the following:
1. Critical flow in two-phase media such as air-water and steam-water were observed by a handful of researchers (including this author), however the theoretical model remains to be developed. Under critical flow conditions the flow rate from the system is independent of the conditions in the receiver (especially pressure).
2. The speed of sound in two-phase mixture is much lower than in any of its components, thus the supersonic flow can exist in such mixtures at relatively low velocities, i.e. 20-50 m/s.
3. The supersonic flow in Laval nozzles produces "shock waves", thus increasing the outlet pressure proportionally to the square of Mach number achieved in the nozzle. Shock waves of this nature were observed in experiments and called either "mixing shocks" or "condensing shocks", however the exact condition for their presence are not described in the available literature.
4. The amount of research on the above subject is limited and no practical applications were found. It is possible that this project might be the first attempt to bring the practical use of two-phase flow theory.
5. It appears that one of research directions in this and next phases of this project has to be definition of conditions for a sustained critical flow in two-phase mixture of liquid and vapor R22 refrigerant. No previous research was found on this specific subject.

3. Description of the New Refrigeration Cycle

3.1 Technical Concept

The nature of the innovation presented in this report lies in the application of supersonic ejector to increase the efficiency of vapor compression refrigeration cycle. We decided early in the stage of development that the working medium would be R22, the refrigerant that is widely available and relatively inexpensive. For that reason, all designing was performed for this refrigerant. Following are novel concepts and explanation how do they differ from previously reported ejectors for refrigeration cycles.

1. The ejector is used as a second stage compressor in the cycle. In this capacity it is lowering the compression ratio, but not by increasing the suction pressure as in known previous systems but by decreasing the discharge pressure from the compressor. The disclosed ejector is used as a second stage compressor in the cycle.
2. The design of the ejector is based on the theory of two-phase flow, which considers the previously explained lowering of speed of sound in two-phase mixture with subsequent creation of the shock, which increases the pressure on the outlet of the ejector.
3. In our ejector, the motive stream is the liquid refrigerant while the suction stream is the compressed vapor refrigerant from the compressor.

The Figure 2 below shows the design of the supersonic ejector:

Figure 2 was removed due to content of patentable material
Invention Disclosure Submitted to DOE
Docket Number: S-108,382

Figure 2. Supersonic Two-Phase Ejector: 1 – nozzle, 2 – needle valve, 3 – mixing chamber, 4 – diffuser, 5 – entrance branch pipes.

The principle of the ejector operation, presented above is utilized to construct the cooling/refrigeration system shown in Figure 3. In this new system, the mechanical compressor compresses the vapor to approximately 50-60% of the final pressure. Additional compression is provided by the ejector device explained above using internal potential energy of the working fluid flow. Therefore, the amount of mechanical energy required by a compressor is significantly reduced. The principle of the proposed system as shown in Figure 3 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 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 (10 and 11), and a pump (9). The evaporator (2) absorbs the heat from source (12), while the condenser (5) is connected to the heat sink – high temperature heat receiver (13). 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.

The working medium kept at low pressure vaporizes in the evaporator with using the heat energy of 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, located after the condenser. 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 application of this invention improves the efficiency of the thermal transformation by means of lowering the need for energy to run the compressor.
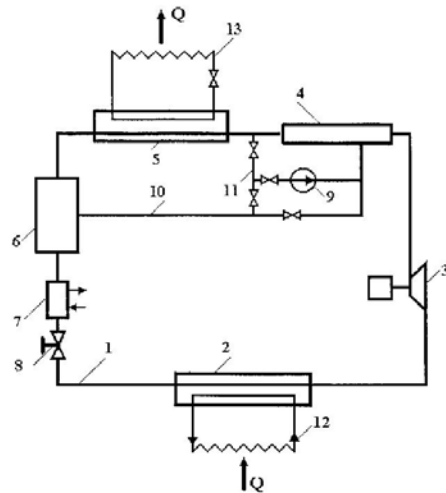
Figure 3. Schematic of the new refrigeration system/heat pump

The above purpose is realized by the method of heat transformation that includes evaporation of the part of working medium at lowered pressure with utilizing the thermal energy of the low-temperature source, mixing two parts of the working medium in a jet device, cooling the flow of working medium with transfer of its thermal energy to high-temperature receiver, and dividing the working medium to two parts.

The other specifics of the new device are:
- additional connection of ejector device to the circuit after the condenser in order to regulate the temperature of the ejector device.
- Installing a pump for liquid pumping into the side piping (feedback circuit)
- Installing a separation tank between cooler and expansion valve.
- Installing of the additional cooler (heat exchanger) before the expansion valve.

In the proposed method as opposed to known methods, the compression of the working medium is replaced by the compression of the vapor part of the working medium in the first stage in a compressor and then in the second stage in a vapor-liquid ejector device. In the ejector device, liquid-vapor mixture achieves supersonic velocity, which causes the sudden increase of the pressure (shock wave) with simultaneous condensation of the vapor and increase of the temperature.

The proposed method can be realized by utilizing the traditional working mediums such as low-temperature boiling fluids, the same as used by heat pumps and home refrigerators, for example R12, R22, R134a etc. or their mixtures with mineral or synthetic oils, water, etc.

Further explanation of the new proposed refrigeration cycle is presented on the p-h diagram in Figure 4 below. It needs to emphasize that the presentation as in Figure 4 is conditional and serves the purpose of illustration since the exact presentation of these processes is rather difficult because they are not stationary and have the variable masses of the working fluid.

In Fig.4 the following processes of change in the state of a working fluid are depicted:
- 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;

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7-8- expansion of this part of the cooled liquid working medium in the ejector;
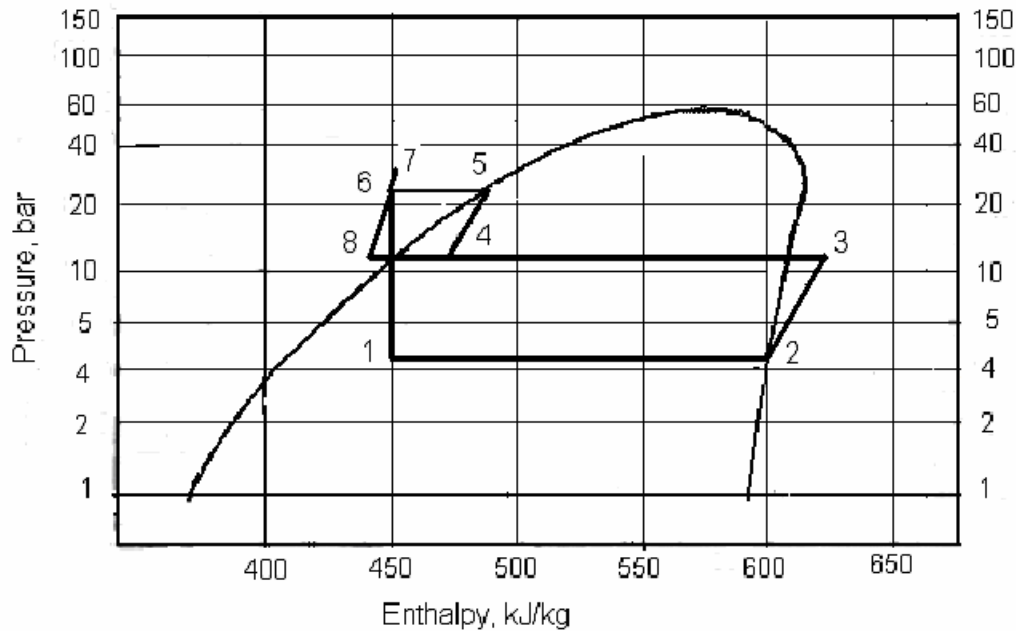6-1 – throttling of the evaporating part of the working fluid.

Figure 4. P-h diagram of the new refrigeration cycle with a two-phase ejector for R22 refrigerant

The technical concept proposed for the realization of the considered project is based on the following preconditions: - expediency of the creation of the geothermal heat pumps for the average climatic conditions as in USA and specifically New England; - the productivity of the created prototype should correspond to the equipment being in greatest demand in the market for similar services.

3.2.   Theoretical Model of the New Refrigeration Cycle with a Two-Phase Ejector

A theoretical model has been developed to evaluate the capabilities of the two-stage refrigerating system with a vapor-liquid ejector. The main distinguishing feature of such model from the similar models of the vapor-ejector or gas-ejector refrigeration cycles is the use of a liquid flow as ejecting (motive stream) agent and of the vapor phase as ejecting medium (suction stream). At present the issues of using the vapor-liquid flows in the refrigeration cycles appear not yet to be reflected sufficiently enough in the scientific and technical literature [3].

To simplify the theoretical model of the refrigeration cycle with a vapor-liquid injector, the following assumptions that are analogous to the ones in the paper [12] are made:
1. Neglect the pressure drop in the condenser and evaporator and in the connection tubes.
2. No heat losses to the environment from the system, except for the heat rejection in the condenser.
3. The vapor stream from the separator is a saturated vapor and the liquid stream from the separator is a 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 condenser 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 of the ejector expansion R22 cycle have been stated. Assuming that the pressure before the inlet of the constant area mixing section of the ejector is P_b and the ejection ratio of the ejector (ratio of mass flows of vapor m_v and liquid m_f) is

$$w = m_v / m_f \tag{4}$$

The motive stream follows an isentropic expansion process from pressure P_1 to pressure P_b before it enters the constant area mixing section, or otherwise the value of entropy S_i for the moving stream in the point 7 and in the point 8 are equal:

$$S_7 = S_8 \tag{5}$$

The corresponding enthalpy h_8 of the moving stream at the end of the isentropic expansion process can be determined using the P-h diagram for R22 or by equation

$$h_7 - h_8 = (p_7 - p_8) / \rho \tag{6}$$

Further, 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:

$$u_{mb} = \mu \sqrt{2(h_8 - h_7)} = \mu \sqrt{2gH} \tag{7}$$

where $g = 9.81$ m/ s^2, μ is a coefficient of discharge and H is pressure difference $(P_7 - P_8)$ of the motive stream expressed in meters of a liquid column.
With using a P-h diagram we can find the specific volume for both the motive stream in the point 8, V_8, and the suction stream in the point 3, V_3, as well as the same for their mixing in the point 4, V_4.
In the case of two-phase flows, the cross-section area of the ejector mixing section per unit total ejector flow rate, a_m, can be determined by equation

$$a_m = V_m / u_{mb} \tag{8}$$

in which $V_m = 2/ (\rho_4 + \rho_5)$ is the mean specific volume of the vapor-liquid mixture at the ejector mixing section and ρ_4, ρ_5 is the density of the vapor-liquid mixture in the states corresponding to points 4 and 5 on the P-h diagram of Figure 4.
The method employed here for calculating the cross-section area of the mixing channel is characteristic of similar techniques of two-phase ejector calculation given by Fisenko [3].
From the known values of the velocity of the mixing stream and across mixing section area (cross-section area of the cylindrical channel at the mixing chamber outlet), it is possible to calculate the pressures of the working fluid flow at the mixing chamber outlet, P_{mix}, and at the ejector outlet after the diffuser, P_d. In this event the following equations were applied:

$$P_b a_m + \frac{1}{1+w} u_{mb} + \frac{w}{1+w} u_{sb} = P_m a_m + u_{mix} \qquad (9)$$

$$P_d = P_{mix} + \frac{\rho(u_{mix}^2 - u_d^2)}{2} \qquad (10)$$

the former being the momentum conservation equation, whereas the latter is the energy conservation equation in the form of Bernoulli equation. In these equations $u_{mb\ and}\ u_{sb,}$ − velocities of the liquid and vapor flows (motive and suction) at the mixing section inlet, u_{mix}, u_d − mixture flow velocity at the diffuser inlet and outlet.

It needs to emphasize that in our case, the mixture velocity in the mixing chamber has to be somewhat higher than local sonic speed of this two-phase flow because in this case the efficiency of the vapor-liquid ejector increases [3]. In its turn, according to the known data [1], the speed of sound propagation α in a two-phase medium can be as low as only 20-50 m/s and for its estimate one can apply the equation

$$\alpha^2 = kP\ /\rho_{mix}\ or\ a^2 = P//\rho_f \beta(1-\beta) \qquad (11)$$

where k is isentropic coefficient, P, ρ_{mix} is pressure and density of the two-phase flow, ρ_f is density of the liquid phase and β is the volumetric content of vapor in the mixture. .
The quantity of energy ℓ_p, consumed by a pump in compressing a working fluid is calculated with the formula

$$\ell_p = m_f\,\Delta P_{7-6}/(\,\rho_{mix}\ \eta) = m_f(h_7-h_6\,)/(\,\rho_{mix}\ \eta) \qquad (12)$$

where η is efficiency (coefficient of efficiency) of the pump.
The quantity of energy needed for the compression of the vapor flow m_v by the compressor with the performance η_c is determined by the expression

$$\ell_c = m_v\,(h_3-h_2)\,/\,\eta \qquad (13)$$

Other energy characteristics of the cycle are defined as follows:

- Refrigerating capacity of the system Qo

$$Q_o = m_v\,(h_2-h_1) \qquad (14)$$

- Thermal performance Q_h

$$Q_h = (m_f + m_v)\,(h_5-h_6) = (m_f + m_v)\,c_p(T_5-T_6) \qquad (15)$$

- The compression work ℓ done by the compressor and the pump

$$\ell = \ell_c + \ell_p \qquad (16)$$

The coefficient of performance (COP) of the two phase ejector cycle can be determined by:

$$COP = Q_h\,/\ell \qquad (17)$$

For the basic one-step refrigeration cycle operating in the same temperature range, the evaporator heat capacity Q_{bo} and the condenser heat capacity Q_{bh} are given by:

$$Q_{bo} = m_v (h_{b2} - h_b) \qquad (18)$$

$$Q_{bh} = m_v (h_{b3} - h_{b6}) \qquad (19)$$

The compressor work of the same basic cycle operating without using the ejector is found by:

$$\ell_b = m_v (h_{b3} - h_{b2}) / \eta_c \qquad (20)$$

where h_{bi} are the enthalpies of the corresponding points in the P-h diagram cycle of Fig. 5 where a comparison is shown between one step compression conventional cycle and the new cycle. Then, the performance of the basic refrigerant cycle with the same temperature range is given by:
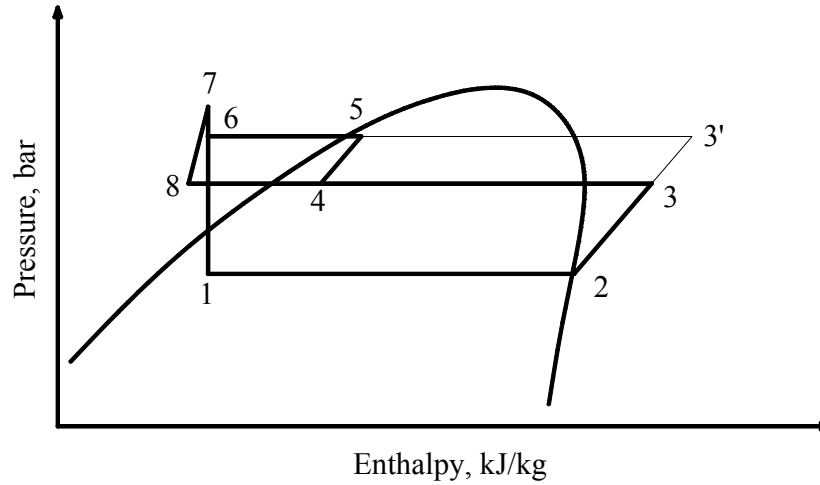
$$COP_b = Q_{bh} / \ell_b \qquad (21)$$

Figure 5. 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).

The relative performance of the two-phase ejector refrigerant cycle is defined as:

$$R = COP / COP_b \qquad (22)$$

   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. 5, while the properties of the refrigerant in the characteristic points of these diagrams are tabulated in Table 1. In carrying out calculations it has been assumed that coefficient of efficiency of the hydraulic pump and compressor is equal to 0.8, and corresponding values (magnitudes) of the evaporator capacity for both cycles under consideration are identical.

For the case studied which corresponds to the data of Table I, the ejector displays following characteristics:

- Velocity of outflow of the motive fluid from the ejector nozzle

$$u_{mb} = \mu\sqrt{2(h_7 - h_8)} = 0.86\sqrt{2 \cdot 2 \cdot 10^3} = 54.4 \, \text{m/s} \tag{23}$$

- Cross-section area of the mixing nozzle

$$A_m = 2/(\rho_4 + \rho_5) \, u_{mb} = 1/(860 \cdot 54.4) = 21.3 \times 10^{-6} \, \text{m}^2 \tag{24}$$

.
Then, considering that
$u_{mb} = 54.4$ m/s, $u_{sb} \approx 10$ m/s, $w = 0.11$, $\Delta P = 1.3$ MPa , $P_b = 1.19$ MPa and $u_{mix} = 20$ m/s, through the equation (9) we find the value of the mixture pressure $P_{mix} = 25.16$ MPa.
The specific energy characteristics of the cycle with a vapor-liquid ejector (cycle 1) in comparison with the traditional cycle (cycle 2) are given in Table II.

TABLE I
Qualities and Quantities of R22 Refrigerant in the Characteristic Points on the Diagram of Fig. 5

Point No.	Number of parts of R22		h_i, kJ/kg	P bar	t_i, ^{o}C	S_i kJ/kgK	x_i
	Total	Moved by the pump					
1	1	–	450	4.2	−5	4.07	0.22
2	1	–	602	4.2	−5	4.76	1
3	1	–	620	11.9	35	4.77	1
4	10	9	460	11.9	30	4.21	0.1
5	10	9	473	24.2	55	4.25	~0.01
6	10	9	450	24.2	40	4.15	0.
7	9	9	451	30	41	4.16	0.
8	9	9	449	11.9	30	4.14	~ 0.02
3_b	1	–	645	24.2	85	4.76	1

TABLE II
Specific energy characteristics of the cycle with a vapor-liquid 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	-
Coefficient of performance (COP)		4.9	3.53
COP1/ COP2 ratio		1.38	

The conducted theoretical analysis of the two-phase ejector for R22 cycle, despite the assumed simplifications of the model, permits to conclude that the application of vapor-liquid jet devices opens up new opportunities for a substantial improvement of the refrigeration cycle efficiency.

4. Fabrication and Preliminary Testing of Two-Phase Ejector

4.1 Fabrication

The design of the two-phase ejector was completed based on theoretical considerations given in Para. 3.2 of this report, and including further recommendations given in [3], pp. 112-116. The general design resembles the schematic of Figure 2 shown earlier in this report. The set of technical drawings was produced and it was kept on file at Magnetic Development, Inc. The majority of components were machined by a local machine shop, Profile Machine Products of Madison, CT. The most important part of the jet device is a nozzle with long conical opening, and two such parts were fabricated. One was produced by conventional machining process while the other by wire EDM (Electric Discharge Machining). It appears that the latter one has closer tolerances and it was used for assembly first, while the other nozzle was kept as a back-up. The problem of administrative nature was initially encountered in fabrication of the custom bellows to be welded to the needle part of the jet device. We contacted several US manufacturers of bellows and flexible joints but none of them was willing to make a one-time part of special design. Finally, the company Witzenmann of Pforzheim, Germany has agreed to fabricate the bellow and perform the specialized welding plus Helium leak test. The work was performed on the experimental basis and for the minimum charge at their fabrication shop in Opava, Czech Republic. Use of the foreign company was justified by the fact that no US manufacturer would engage in such experimental and low-volume (one piece only) production. Figure 6 below shows two main components of the ejector: the housing containing the motive nozzle and vapor entry channels (left side) and the ejector/diffuser, where mixing of both phases take place (right side).

Following the fabrication of all its components, the entire ejector was assembled. The view of the assembled ejector is shown in Figure 7. The entire device has two inlets for the vapor (gas) phase, fabricated out of copper tubing and clearly visible on the picture. The liquid phase was originally connected to the ejector via high-pressure plastic tube as shown behind the main body of the ejector. Shown in Figure 7 is a preliminary installation for initial testing of the ejector on air-water mixture. The detailed description of these experiments is given in the next paragraph.

Figure 6. Two main components of the vapor-liquid ejector: housing with motive nozzle and vapor inlet (left) and ejector with mixing chamber and diffuser
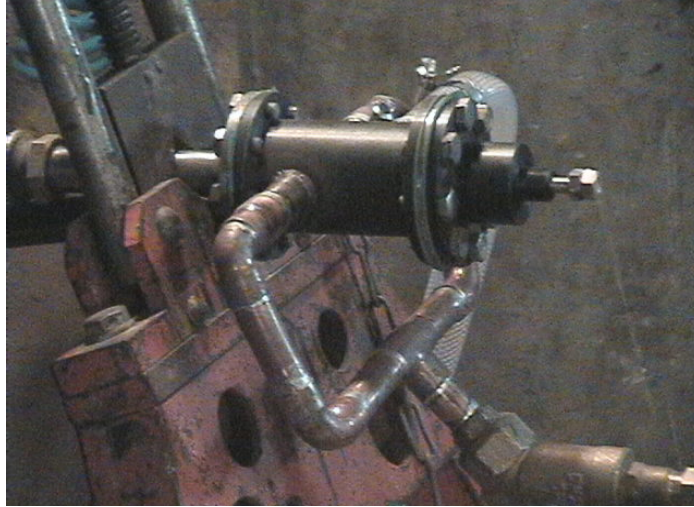
Figure 7. Assembled ejector (installed on trial stand for air-water mixture).

4.2   Laboratory Experiments with Air-Water Mixture

The published information about two-phase supersonic flow and associated shock waves, created by this phenomena is sketchy and incomplete, to say the least, therefore we decided to undertake some preliminary experiments in order to prove or disprove this theory. Our approach was justified by the fact that very limited information exists about any experiments conducted, and even if presented, the available literature does not show specific conditions under which those experiments were carried [1],[3],[4]. Most comprehensible description is shown in [1], where the author refers to experiments performed by others on the mixture of air and aqueous solution of glycerin. Reportedly, the pressure was raised by a factor of 2, and the author presents the oscilloscope profile of the pressure along the nozzle. Another author [3] shows the data from experiments carried on pure water with dissolved air, however only general schematic of the laboratory stand is presented. The pressure in this experiment was measured at several locations along the nozzle, while the valve was used at the outlet to regulate the counter-pressure. The diagrams are presented showing that the sudden pressure jump occurs at certain location along the nozzle and further, this location moves upstream with raising the counter-pressure. Witte [4] investigated the efficiency of a propulsion device based on injection of compressed air bubbles in the throat section of a Laval nozzle. He observed the pressure jump, which he called "mixing shocks", by different topology behind the shocks than in front. Per his description, before shocks, the mixture consisted of thin jets of water and entraining air, while after shocks, the mixture consisted of water with small air bubbles.

The decision was made that before building the full-scale prototype operating on R22 refrigerant, the initial experiments to be conducted on relatively simple laboratory stand and using air-water mixture as a working medium. We thought that this approach would result in considerable savings and in case of unsuccessful performance, we would try to re-design the ejector before building the prototype for R22 refrigerant. The major question was the method to measure or observe the pressure jump in the ejector. For this, two options were initially contemplated: one was to install pressure gauges along the length of the ejector and monitor the pressure with a number of pressure gauges (for example located every 1 inch along the ejector length) and another was to fabricate the ejector portion of the nozzle out of the transparent acrylic plastic and visually observe the flow process. The second option was selected as much more expedient and cost effective. The ejector was designed according to the recommendations given earlier in this report, considering the physical properties of air-water mixture. It has a 80mm long

cylindrical channel of 5mm diameter. At the outlet, the ejector is formed into a diffuser. While the technical drawing is on file, Figure 8 below shows the fabricated transparent ejector part.

Figure 8. Ejector fabricated of transparent material for observing air-water mixture flow.

The transparent ejector was then installed onto the housing part (made of steel) shown in Figure 6 above. The relatively simple and inexpensive laboratory stand was prepared to conduct those experiments as shown in Figure 9 below. Water from the pressurized tank (1) flows through the control valve (2) and flow meter (3) to the nozzle portion of the ejector (4), with further flow regulation provided by the needle valve. The water then flows to the mixing chamber in the ejector where it is mixed with metered amount of air from the compressor 8 and measured by the rotameter (9). The mixture then flows through the transparent ejector (5) to the open tank 7. The counter-pressure is regulated by valve 6. The water pump (10) closes the circuit by pumping the water back to the tank (1).
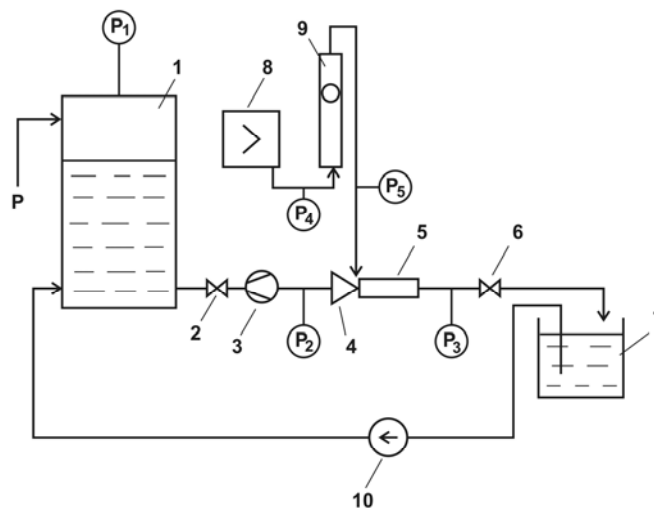

Figure 9. Schematics of laboratory stand to investigate air-water flow through the transparent ejector: 1- pressurized water tank, 2- control valve, 3- flow meter, 4- water nozzle, 5- transparent ejector with mixing chamber and diffuser, 6- valve controlling counter-pressure, 7- discharge water tank, 8- air compressor, 9- rotameter, 10- water pump, P_1 through P_5 - pressure gauges

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Photographs below show few details of the laboratory stand.

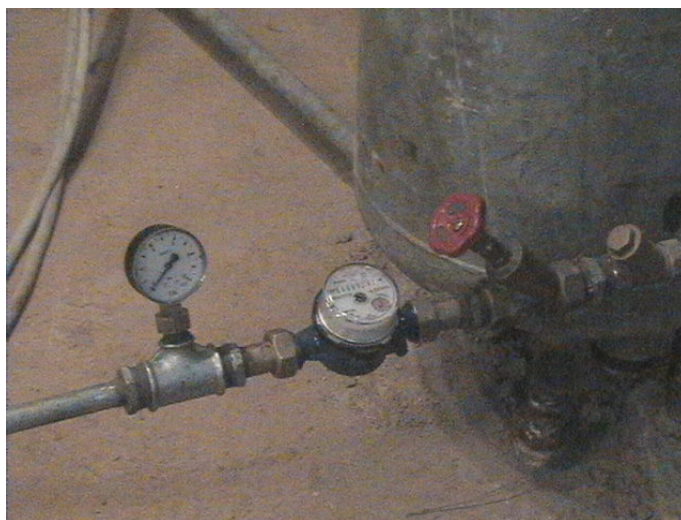Figure 10. Pressurized water tank (1) with pressure gauge p_1 on top.

Figure 11. Shows the control valve (2), flowmeter (3) and pressure gauge p_2

Figure 12. Nozzle (4) with transparent ejector (5) installed on the stand. Also, shows the compressor (8), rotameter (9) and pressure gauges p_4 (on the compressor) and p_5 (large). On the back visible open tank (7) and pump (10).

The initial experiments were conducted on the erected stand by varying the water and air flow, as well as water pressure p_1 air pressure p_5 and counter-pressure p_3. Prior to experiments, the calibration of the rotameter was performed by standard method. These first series of experiments were conducted with original ejector (steel, as shown in Fig. 7) in order to establish available ranges of pressures, water and air flow, velocities and volumetric ratios β. Pressures of water were kept between 7-8 bars to maintain adequate flow while pressures of air varied from 0 (pure water) to 2.6 bar. All results presented in Table III were obtained with valve (6) fully open, i.e. counter-pressure = 0.

TABLE III
Initial results for air-water mixture with no counter-pressure

Water Pressure p_1 (bar)	Rotameter reading	Air pressure p_5 (bar)	Water flow (l/min)	Air flow (m³/hr)	Water-air velocity (m/s)	Volume ratio β
7	0.3	0.7	8.26	1.05	42.5	0.68
7	0.4	1.1	7.79	1.3	48.6	0.73
8	0.4	1.1	8.45	1.3	49.7	0.72
8	0	0	8.92	0	14.7	0
7	1	2.4	6.42	2.84	88.7	0.88
7	0.6	1.6	6.87	2.04	67.4	0.83
8	0.3	0.75	7.85	1.1	43.2	0.7
8	0.6	1.8	7.34	2.14	71.0	0.83
8	1	2.6	6.9	3	93.9	0.88
7	0.3	0.7	7.53	1.05	41.3	0.7
7	0.5	1.35	7.05	1.72	58.9	0.8
8	0.3	0.75	7.99	1.1	43.4	0.7
8	0.6	1.8	7.35	2.14	71.0	0.83
8	0.9	2.5	6.9	2.92	91.7	0.88

The conclusions drawn from these first set of experiments were that even though flow velocities might be within the range of published sonic velocity for the air-water mixture, however, the β volumetric ratio might be too high, as the optimum should be around 0.5 [3]. Therefore, it was decided to make the following modifications: 1. fabricate new nozzle/housing with larger water nozzle diameter (5mm vs. 3mm) still controlled with adjustable needle valve as shown in Figure 2, and 2. transparent ejector with cylindrical channel of 5 mm diameter is acceptable, 3. use low air pressure to limit the air flow to minimum. After all these modifications, the lab stand was re-assembled and the new series of experiments were conducted with transparent ejector. It needs to emphasize that our objective was only to observe the flow and pressure shock and was not to establish governing parameters for its occurrence. The summary of the results is shown in Table IV below. The variable parameters included: water and air pressure (p_1 and p_5 respectively), counter-pressure (p_3) and needle valve adjustment, while water and air flow, their velocities and β ratio were the resultant values.

TABLE IV
Results for air-water flow in transparent ejector

p_1 (bar)	p_5 (bar)	p_3 (bar)	Water flow (l/min)	Air flow (m³/hr)	Velocity (m/sec)	β	Flow appearance
6	0	2	22.7	0	18.9	0	Clear water
6	0.8	4	16.0	1.1	28.6	0.53	Milky, transforming gradually into clear along the ejector length,
7	0.7	6	12.7	1.05	25.2	0.58	Bubbly water, transparent
6	3.0	4	3.44	2.78	41.4	0.93	Milky, not transparent

Based on above results it was determined that the supersonic flow and associated pressure jump does occur indeed in certain conditions. Specifically, we observed definite change in the appearance of the flow in the transparent ejector at certain conditions, especially with mixture velocity in the cylindrical channel exceeding 25 m/s and β ratio around 0.5. The appearance of the stream changed from milky and misty to transparent with air bubbles as shown in Table IV and in addition, a sufficient flow was still maintained even against relatively high backpressure. According to literature on the subject [1],[3], the transparent, bubbly flow is associated with critical flow with supersonic velocities. We observed such flow, even though there was no sharp transition from one type of flow to another. It is possible that transition occurred beyond the transparent ejector, i.e. closer to the water and air nozzles, where both components are mixed. The housing of those nozzles was not transparent (steel). The detailed experiments were however not the subject of this project and therefore we left it to possible later continuation, especially in university environment with well-equipped laboratory. At this time, we concluded that the optimum volumetric ratio of gas-to-liquid is around 1:1 (β = 0.5) and consequently, the R22 prototype was designed to achieve similar parameters.

The milky flow (thought to be non-critical) is shown on the photograph of Figure 13, while the transparent flow, on Figure 14.
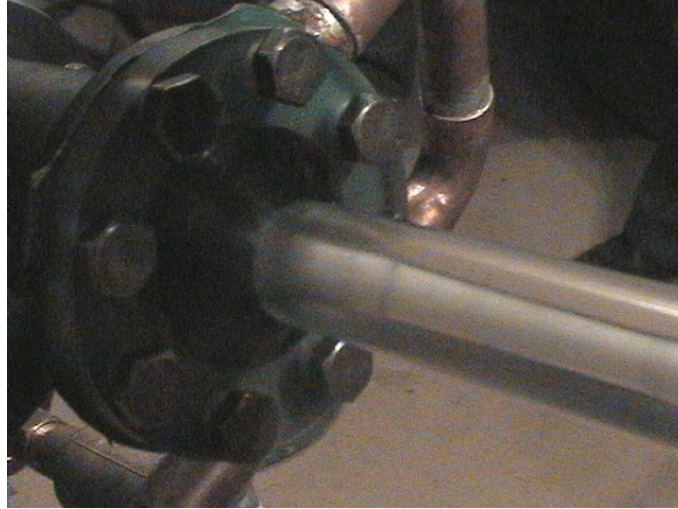
Figure 13. Non-transparent, milky and misty flow of air-water mixture

Figure 14. Transparent, bubbly flow

5. Design and Fabrication of Heat Pump Prototype with Two-Phase Ejector

The literature study, development of a theoretical model and initial laboratory investigations as described earlier in this report, have provided enough analytical and experimental data to design the prototype heat pump with two-phase ejector acting as a second stage compressor. The conceptual schematic with major components is shown in Figure 3. During the design of the prototype, our main objective was to be able to conduct a maximum amount of tests with varied parameters. For this purpose, numerous such possibilities were built into the design. The major assumptions were the size of the heat pump, approximately 3 Tons of refrigeration and the refrigerant: R22. The other was to have the option to run it as a conventional refrigeration cycle, through by-passing the ejector. It appears that most important parameter for proper ejector operation is the ratio of vapor-to-liquid and for that reason, the design of liquid refrigerant pump was of utmost importance. The pump output in our design is double regulated,

first with a Variable Frequency Drive (with maximum rpm ratio 1:4) and second, with by-pass valve, controlling the back flow of discharged liquid refrigerant to the suction side of the pump.

The final pressure in the system is controlled by the water temperature in the condenser, therefore, for optimum operation, we connected two heat exchangers in parallel to serve as the condenser. For best temperature regulation, the water flow can be directed either to both exchangers or to one only, and further, the flow is regulated by the control valve installed on the inlet side of the cooling water. Additional features were installed into the prototype, such as suction line heat exchanger with a by-pass and extra compressor protection by directing part of hot vapor from the compressor discharge into the suction accumulator, to assure that only vapor is flowing to the compressor suction.

Final design is shown in Figure 15 below, while Table V lists all major components used for fabrication of this prototype.

TABLE V
Major Components of the Fabricated Prototype

Item	Description of the component/Manufacturer	Type/Parameter	Value
1	Compressor Maneurop Hermetic, Water Cooled Condensing Unit	Reciprocating Type M/CMTW28-3 S/N QD104073141	2.5 HP 3-phase motor
2	Liquid Refrigerant Pump Blackmer	SGL 1.25 Sliding vane type	2HP, 1150 rpm, 3/60/230-460V
2a	Variable Frequency Drive for Item 2 AC Tech	Type ZZD-M1220C Freq. Ratio: 1:4	2 HP for 230 V 3-phase power
3	Condenser, coaxial, copper tube helical Packless Industries	Type COAX2301J	3 Ton
4	Evaporator, coaxial, copper tube helical Packless Industries	Type COAX2401J	4 Ton
5	Suction Line Heat Exchanger Packless Industries	Type HXR150	1.5 Ton
6	Expansion Valve Sporlan	Type EFV1C	
7	Suction Accumulator Refrigeration Research, Inc.	M/HX3738	7/8", 7.9 lbs R22
8	Liquid refrigerant Receiver Alco	Type AVRO6123MXP	3/8" connection, 10 lbs R22,
9	Ejector	Custom m'factured by Magnetic Dev. Inc	

For the assembly purpose, the steel frame was designed and fabricated in the local welding shop. The assembly was the conducted according to the following plan:

1. Install all major components on the frame: condensing unit with the compressor and condenser, liquid refrigerant pump assembly incl. pump, clutch, 3-phase motor and VF (variable frequency) drive, condenser, evaporator, liquid receiver and accumulator
2. Install the jet nozzle to the frame with previously designed clamps and holders

Figure 15 was removed due to content of patentable material
Invention Disclosure Submitted to DOE
Docket Number: S-108,382

Figure 15. Schematics of the Prototype Heat Pump
1- Evaporator, 2- Suction line heat exchanger, 3- Compressor, 4- Ejector device,
5- Condenser, 6- Liquid refrigerant receiver, 7- Liquid refrigerant pump, 8- Thermostatic
expansion valve, 9- Filter/dryer, 10- Suction accumulator, 11- Control valves (A through
P), 12- Check valves, 13- Sight glass with moisture indicator, 14- pressure gauges.

3. Make all necessary joints with silver solder and/or hard solder as required by the Code. Standard refrigeration tubing was used with diameters 3/8, 5/8 and ¾ as required by the design. In addition to soldered joints (sweat joints), a majority of valves were installed with flared connections.
4. Install water connections to condenser and evaporator
5. Make all electrical connections to pump and compressor. All equipment used was for 3-phase supply, 230V. System was approved by licensed electrician prior to operation.
6. Perform leak test using Nitrogen
7. Evacuate the system
8. Charge the system with R22 refrigerant (approx. 20 lbs)
9. Perform the trial run

The majority of assembly work was performed by the Technician, accompanied by the Principal Investigator. Items 7 and 8 above were performed by R&B Refrigeration, Inc. of Guilford, CT, a contractor with local HVAC License.

Photographs of Figures 16 through 20 show the assembled system:

Figure 16. Overall View of the Prototype

Figure 17. Front View of the Prototype

Fig. 17 shows the liquid refrigerant pump assembly with a variable frequency drive (on the left), condensing unit with the compressor (right bottom, blue color), two condenser heat exchangers connected in parallel (two black coils on the center-bottom) and the evaporator (coil on the top). Various measuring instruments are shown including pressure gauges on the suction and discharge of pump and compressor, pressure gauge on the outlet from the jet compressor, pressure gauge at the intermediate heat exchanger and the station of four temperature gauges (front center) for measuring the temperature of the refrigerant lines at different locations.

Figure 18. Rear View of the Prototype

Shows the suction accumulator (black vessel on the left) and liquid refrigerant receiver (black vessel on the right). Also shows the water connections to the condenser and evaporator and electric connections to the blue box on bottom left.

Figure 19. Top View of the Prototype

Figure shows the evaporator (large black coil), thermostatic expansion valve (valve with grey and green top located behind the water meter), intermediate heat exchanger (gold pipe on the back of the picture) and instrumentation to measure water temperature and flow through the evaporator.

Figure 20. Closer View of the Ejector Device

Figure above shows the closer view of the ejector nozzle/jet compressor. The device receives the liquid refrigerant from the pump through the pipe visible on the front (the one with installed temperature gauge) and vapor from the compressor via two pipes towards the front of the device. In the mixing chamber, both phases are mixed and accelerated to high velocity by forcing it through small orifices under pressure. The velocity in the mixing chamber must be above the speed of sound for the mixture in order to create a supersonic flow. The supersonic flow in turn produces the pressure jump in the ejector portion of the device.

Following features were built into the prototype in order to make it more versatile and to maximize the research possibilities.

1. By-pass for suction line heat exchanger:
   a) Valve A open, valve B closed: vapor from the evaporators directly enters the suction accumulator, by-passing the heat exchanger. This is the most traditional system for vapor-compression refrigeration cycle.
   b) Valve A closed, valve B open: vapor from evaporator enters the suction line heat exchanger where it is pre-heated by hot liquid from the condenser. The purpose is to improve the efficiency by raising the temperature of a vapor prior to entering the compressor and to protect the compressor from the possibility of small amount of liquid in the suction line.

2. Preheating of refrigerant vapor in the suction accumulator:
   a) Valves I and K open, valve L closed: hot compressed vapor from the compressor enters the heat exchanger coil in the suction accumulator   and boils out any liquid remained. This is used as an extra protection of the compressor from liquid in the suction line.
   b) Valves I and L open, valve K closed: the heat exchanger coil in the suction accumulator is by-passed.

3. Regulation of liquid refrigerant flow from the pump to ejector (in addition to regulation by pump rpm using variable frequency drive VFD);

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a)   Valves E and H fully open, valve G closed: flow control only with VFD, no backpressure regulation.

b)   Valves E and H fully open, valve G in various open positions: additional flow control by feeding the portion of discharge to the suction line.

4.   By-pass for the ejector device:

a)   Valve M closed, valve P open: prototype operates on the traditional vapor compression cycle with single stage compression by the compressor 3. Ejector device is bypassed and is not operational.

b)   Valve M open, valve P closed: this provides for second stage compression in the ejector device.

5.   Cooling water regulation to the condenser (not shown in Figure 15). The temperature in the condenser is a very important parameter as it controls the pressure in the system – pressure level 3 and 3' in Figure 4 above. Two heat exchangers connected in parallel are used as a condenser for maximum regulation. Both exchangers are coaxial, copper tube helical type. Valves are installed in order to disconnect one exchanger and in addition to regulate the amount of cooling water flowing through exchanger(s).

6. Laboratory Experiments

There were two main objectives of the laboratory experiments on the fabricated prototype with the first being to investigate the possibility of the pressure jump on the ejector and the other 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 following prototype setups were used (refer to Figure 15 for valve numbering):

-   For the conventional cycle with ejector by-pass: A- open, B-closed, C- open, D- open, E-closed, F-open, G- closed, H- closed, I- open, K- closed, L-open, M- closed, P- open.

-   For two-step compression with ejector as a 2nd step of compression: A- open, B- closed, C- open, D- open, E- open, F- open, G – partially open, and regulated during operation, H- open, I- open, K- closed at the start to be occasionally open for a few seconds during operation, L- to follow valve K: when K opens, L closes and when K closes, L opens, M- open, P- closed.

For this first round of experiments we by-passed the suction line heat exchanger and used the circuit for boiling liquid refrigerant in the suction accumulator (hot gas by-pass) only occasionally, a few seconds at the time and only in the ejector mode.

The operation in the conventional mode of vapor compression cycle was straightforward and standard results were obtained as shown in Table VI. The high pressure was regulated by the temperature in the condenser, therefore, we were adjusting the cooling water flow to obtain approximately 40 deg C water discharge. This required to completely by-pass one of two parallel heat exchangers and further to partially close the inlet valve. The heat output from the condenser was not measured as the water flow meter was installed on the evaporator side to measure the cooling capacity. In order to determine temperatures of refrigerant , the following assumptions have been made [14]:

1.   The temperature of the refrigerant in the evaporator is 5 deg C (8-12F) colder than the evaporator when compressor is running.

2.   The temperature of the refrigerant in a water-cooled condenser is 11 deg C (20F) warmer than the water temperature at the drain outlet.

TABLE VI
Results of Laboratory Experiments

Item	Parameter	Mode 1: Conventional cycle, ejector by-passed	Mode 2: 2-stage compression with ejector device	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, $\eta = 0.8$
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 Δp/0.8
18	Volumetric ratio β in ejector	N/A	0.82	
19	Fluid refrigerant velocity in the ejector, (m/s)	N/A	26.2	

In the conventional cycle, the refrigerant vapor was compressed to the final pressure by the compressor in one step, while in the cycle with the ejector, the vapor was initially compressed to intermediate pressure in the compressor and then to the final pressure in the ejector. For the purpose of most accurate comparison between both cycles, we tried to keep the temperature range of the refrigerant and the degree of its compression at the same level.

The pressure jump of approximately 0.35 MPa, was observed on the ejector at one certain position of the by-pass regulating valve (Valve G). The operation of the ejector was not sustained yet and lasted for few seconds. After short operation, first the pressure pulsation was observed on the gauges and then the "knocking" sound was heard in the pump, indicating that the vapor is present in the pump suction line. In order to protect the pump, immediately after "knocks" appeared, we have fully opened valve G in order to deliver discharged liquid to the suction line. It appears that the liquid receiver might be too small and more liquid is required to provide for the sustained operation. This will be investigated in the next phase of this project. The character of pressure jump in the ejector is unknown, and consequently, the velocity of the mixture in the mixing chamber of the ejector cannot be calculated at this time. It is possible that in order to produce the "shock", a portion of the vapor refrigerant must condense in the mixing chamber, rapidly changing the density of the mixture.

Table VI shows the comparison between parameters obtained in both cycles. Short explanation of the line items follows:

Items 1 and 2: Water inlet and outlet temperatures were measured by thermometers installed in the water lines.

Item 3: Flow was measured by water meter installed on the water inlet to the evaporator

Item 4: Cooling capacity was calculated from well known formula: $Q = mc\Delta t$

Item 5: This was read from the p-h graph for R22 – see distance between points 1 and 2 in Fig. 4. The compressor efficiency of 0.8 is assumed, therefore the result is divided by 0.8.

Item 6: Pressure was read on the pressure gauge installed on the compressor discharge line.

Item 7: As Item 6, Pressure gauge installed on the refrigerant line after the ejector, before the condenser.

Item 8: This was calculated by dividing the cooling capacity as calculated in Item 4 by the enthalpy difference between points 1 and 2 on Fig.4 – the value shown in Item 5 above

Item 9: Volumetric flow was calculated from mass flow as in Item 8, multiplying it by specific volume of R22 refrigerant (from tables) vapor in the temperature at compressor discharge.

Item 10: The compressor work was read from the p-h graph for R22 as an enthalpy difference between point 3" and 2 for conventional cycle and 3 and 2 for ejector cycle (points marked in Figure 5)

Item 11: This is obtained as a product of compressor work obtained in Item 10 and mass flow calculated in Item 8.

Item 12: For conventional cycle, this is a ratio of cooling capacity (Item 4) divided by a compressor work (Item 11). For ejector cycle, the cooling capacity (Item 4) is divided by the sum of compressor work (Item 11) and pump work (Item 17).

Item 13: The pressure was read on the pressure gauge installed on the pump discharge line

Item 14: Difference between discharge and suction pressure on the pump

Item 15: Mass output was read from the characteristic diagram for the pump used, for a given pressure difference (Item 14) and pump rotation, read from the display of VF Drive.

Item 16: Item 15 divided by the density of liquid R22 at given temperature (from tables).

Item 17: The pump power in kJ/min is calculated from the formula: $N = v \Delta p/\eta$. Pump efficiency of 0.8 is assumed.

Item 18: Volumetric ratio β was explained in details in Para. 2.2. It is calculated from the formula: $\beta = v_{vapor}/v_{vapor} + v_{liquid.}$ Volume of vapor is given in Item 9 while volume of liquid in Item 16.

Item 19: Only velocity of the liquid can be calculated as we discussed difficulties connected with calculating the velocity of the mixture in the mixing chamber. The velocity is the quotient of volumetric pump output (Item 16) and cross section of the liquid nozzle. The nozzle has the needle valve regulation and the velocity given here is for the valve fully open. The actual velocity is higher.

7. **Conclusions and Recommendations**

a) This report represents the results of the first phase of theoretical and experimental investigations on new regenerative vapor compression refrigeration cycle with special feature of the compression in two stages, initially by the compressor and then by vapor-liquid ejector.

b) The objectives of the first phase of this project were met by: 1) conducting the 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 traditional vapor compression cycle and 3) designing and fabrication of the heat pump prototype and practically demonstrating 16% energy savings.

c) Prior to fabricating the working prototype, initial investigations were conducted on a simplified stand using air-water mixture. Various appearances of the flow were observed, similar to those described in prior literature, which testifies that critical flow is possible under certain conditions.

d) The key scientific objective was to obtain the pressure jump on the ejector, which is thought to be associated with a critical two-phase flow. 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.

e) The prototype has achieved energy savings of 16%, based on experimental results (gauge readings) and certain reasonable assumptions and calculations for the amount of liquid and vapor flow. In the next phase, more realistic measurements will be performed by installing appropriate instrumentation into the liquid and vapor refrigerant lines.

f) The character of the pressure jump is not clear at this time. It could be a result of equalizing the velocities or of condensing a portion of vapor refrigerant in the mixing chamber. This is the first attempt to obtain such pressure jump, and no literature precedent has been found. In the next phase, use of transparent ejector might be considered.

g) Based on conducted experiments, recommendations were defined to sustain the critical flow process and to improve its efficiency. The most crucial will be a reduction of β ratio closer to 0.5, and increasing the velocity of the liquid in the ejector. This will be accomplished in the next phase by installing larger receiver tank, charging the system with more refrigerant and optimizing the ejector design, using computer simulation models.

h) The investigations under this project have provided the important scientific base for the development of new generation, more efficient refrigeration units, A/C and heat pumps. Considering the results of the initial phase, it will be feasible to develop the first commercial prototype in a relatively short time (2 years). These more efficient devices will contribute to significant reduction in energy consumption, consequently decreasing our country dependence on foreign oil.

i) The fabricated prototype can serve as an important tool for conducting a broad scope of experimental research programs on two-phase flow. In addition to our future work on this project, it can be used by academic, private industry or governmental research labs. The great majority of reported past research on this subject consists of theoretical studies and computer simulation, while lack of practical experiments is evident. Our prototype can contribute to better understanding of phenomena of two-phase flow, which so often occur in industry and in everyday life. In addition, it can provide the experimental basis for novel, not yet realized solutions to refrigeration, fluid transport and power generation.

8. Bibliography

1. Van Wijngaarden, L., *One-Dimensional Flow of Liquids Containing Small Gas Bubbles*, Annual Fluid Mechanics, 1972, vol. 4, pp. 369-396
2. Wallis, Graham B., *Critical Two-Phase Flow,* Int. J. Multiphase Flow, 1980, Vol. 6, pp. 97-112
3. Fisenko, V.V., *S'zhimayemost Teplonosityela i Effektivnost Raboty Konturov Tsirkulatsiy Ya.E.U.*, (book in Russian), Energoatomizdad, Moscow, 1987, 196 p.
4. Witte, J.H., *Mixing Shocks in Two-phase Flow,* J. of Fluid Mech., May 1969, Vol. 36, pp. 101-139
5. Zhao, J.F., *Investigation of the Compressibility of Extra-High Velocity Aerated Flow,* J. of Hydraulic Res., Vol. 38, 2000, No. 5
6. Downar-Zapolski P., et al., *The Non-Equilibrium Relaxation Model for One-Dimensional Flashing Liquid Flow,* Int. J. Multiphase Flow, Vol. 22, No.3, 1996, pp.473-483
7. Smith I.K., Stosic, N.R., *The Expressor: An Efficiency Boost to Vapour Compression Systems by Power Recovery from the Throttling Process,* AES, vol. 34, Heat Pump and Refrigeration Systems Design, Analysis and Applications, ASME 1995, pp.173-181
8. Bohdal, T., et al., *Urzadzenia Chlodnicze Sprezarkowe Parowe,* (book in Polish), WNT, Warsaw, 2003, 531 p.
9. Gay, N.H., 1931, *Refrigerating System*, US Patent No. 1,836,318
10. Kornhauser, A.A., *The use of and Ejector as a Refrigerant Expander,* Proc. of 1990 USNCR/IIR-Purdue Refrigeration Conference, West Lafayette, IN. p.10-19.
11. Hays L., Brasz J.J., *A Transcritical CO_2 Turbine-Compressor,* International Refrigeration and Air Conditioning Conference at Purdue, July 12-15, 2004, Paper No. C137
12. Li D., Groll, E.A., *Transcritical CO_2 Refrigeration Cycles with Ejector-Expansion Device,* International Refrigeration and Air Conditioning Conference at Purdue, July 12-15, 2004, Paper No. R153
13. Elbel, S.W., Hrnjak, P.S., *Effect of Internal Heat Exchanger on Performance of Transcritical CO_2 Systems with Ejector,* International Refrigeration and Air Conditioning Conference at Purdue, July 12-15, 2004, Paper No. R166
14. Althouse, A.D., Turnquist, C.H., Bracciano, A.F., *Modern Refrigeration and Air Conditioning*, The Goodheart-Willcox Company, 2004, 1211 p.