International Journal of Engineering Research and Development
e-ISSN: 2278-067X, **p**-ISSN: 2278-800X, www.ijerd.com
Volume 6, Issue 1 (February 2013), PP. 41-52

Combined Vapour Compression-Ejector Refrigeration System: A Review

Suhas D Kshirsagar[1], M M Deshmukh[2]
[1]M.tech student, Department of Mechanical engineering, Govt college of engineering , Amravati
[2]Assistant Professor, Department of Mechanical engineering, Govt college of engineering , Amravati

Abstract:- The latest developments of the ejector refrigeration and combined vapour compression-ejector refrigeration systems are presented. Also their operating conditions and the coefficients of performance, obtained by various researchers' theoretical and experimental studies, are given. The importance of the working fluid in the performance of the system is emphasized in conjunction with the intercooler, which allows the use of two different refrigerants at a time in the jet and compressor subsystems. Searching appropriate refrigerants, some theoretical and experimental studies show the advantages of using R134a in these systems. However, the use of hydrocarbon refrigerants like isobutene (R600a) is proposed as a good option, although research and some safety procedures have to be developed before applying these "nature friendly" refrigerants.

Keywords:- combined vapour compression-ejector refrigeration, ejector, vapour compression refrigeration

I. INTRODUCTION

Food conservation and the air conditioning of living spaces are indispensable for human beings in the modern life. Currently, the mechanical vapor compression systems used for this purpose, use large amounts of electrical power that is produced in great proportion by fossil fuel combustion, which is a cause of the global warming. Global warming makes imperative need to develop alternative technologies that will allow carrying out cooling applications reducing the use of electrical energy. Electrical energy can be remarkably saved by incorporating high efficiency devices or occupying other energy sources such as thermal energy. From the latter, the use of low grade energy sources such as residual energy from industrial processes or solar thermal energy has been proposed, not only for its ecological context but for its economical repercussion (Gonza´ lez Bravo, 2005[1]; Sun, 1998a,b[2][3]). Absorption, adsorption and jet compression systems are some technologies that use thermal energy.

As in the mechanical compression systems, the mentioned technologies are based on the refrigerant evaporation method, being its main difference in the way used to compress the working fluid. This paper is based on the ejector compression refrigeration systems and combined (or hybrid) cycles that will improves overall Coefficient of Performance (COP) by utilizing the low grade (in some cases waste heat) energy sources.

II. SIMPLE EJECTOR REFRIGERATION SYSTEM

The Ejector refrigeration systems (JCRS) were developed on the early 20th century reaching its highest point between 1970 and 1980 for air conditioning applications. These systems may be thermodynamically classified as 3 T thermal machines as well as the absorption an adsorption cycles. The thermodynamic cycle, as shown in Fig. 1

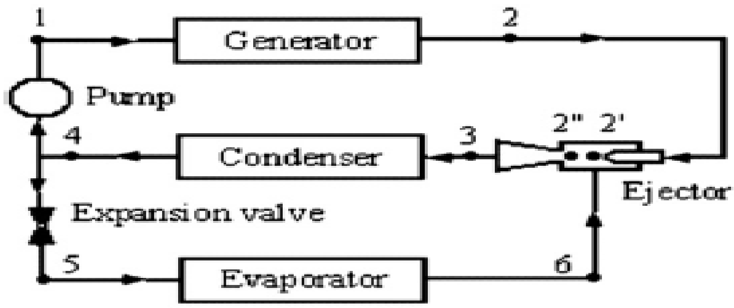

Figure 1 simple ejector refrigeration cycle

Only difference between conventional vapour compression refrigeration (VCR) system & ejector system is, in ejector system- compressor of VCR system is replaced by ejector and pump. Thermodynamic cycle, starts at the vapor generator exit (2), where the refrigerant is at a superheated vapor state. At these conditions, the internal geometry of the ejector allows the suction of the evaporator vapor (6) for its recompression at an intermediate pressure. The working fluid enters a condenser (3) where it is cooled to a saturated liquid state. The fluid is then divided in two streams (4); the first one is recirculated by a pump and transferred to the vapor generator (1). The other stream is driven through an expansion valve for its evaporation at the evaporator.

III. EJCTOR OPERATION

Depending on the application, injector is synonymously used for ejector. The main difference in this case is the discharge pressure at the diffuser exit. While the diffuser exit pressure of the ejector is closer to that of the suction flow than that of the motive fluid, the term injector is sometimes used for applications in which the diffuser discharge pressure can actually reach the pressure of the driving fluid. Other synonyms encountered in the literature are eductor, diffusion pump, aspirator, and jet pump.

In a ejector refrigeration system the ‘ejector’ plays two fundamental roles: the "entrainment" and recompression of the vapor leaving the evaporator to be discharged at the condenser. As the ejector behavior depends on thermal-mechanical factors and its geometry, its design requires great precision and its operation at steady state conditions.

An ejector is a device in which a higher pressure fluid (also called primary fluid) is used to induce a lower pressure fluid (called secondary fluid) into the ejector. Fluids from these two streams mix together and discharge to a pressure that lies between the pressures of these two fluids. In an ejector refrigeration cycle, the ejector and a pump are used instead of a compressor (in a vapour-compression system) for producing a cooling power. An ejector consists of 3 main parts: a suction chamber, a constant area and mixing chamber and a diffuser.

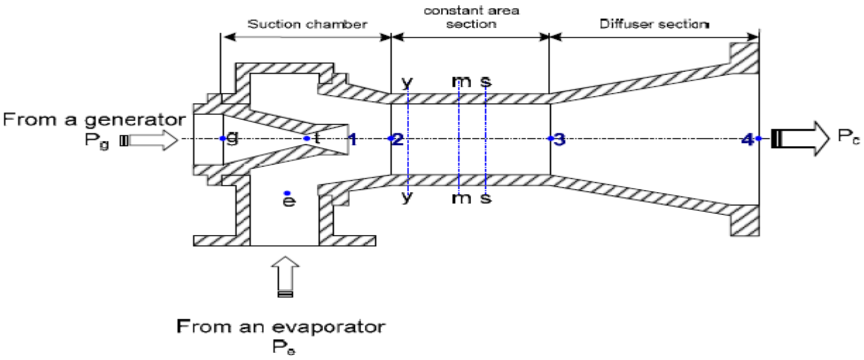

Figure 2 Ejector geometry & sections

When the primary flow goes through a converging- diverging nozzle in the ejector; vapour is drawn from the evaporator. The secondary flow is accelerated to a high velocity vapour stream and reaches subsonic velocity. Mixing starts at the onset of the constant-area section (section y-y, hypothetical throat, in Figure 2). In section y-y, both streams develop uniform pressure; choking of the secondary flow occurs. A combined stream develops into a transient supersonic stream and shocks at section s-s. The velocity of the mixing fluid must be high enough to increase the pressure after deceleration in the diffuser to a suitable condensing pressure.

Ejector refrigeration system poses inherent advantage of no moving parts. And thus less maintenance needed in operation. Profile of primary and secondary fluid stream inside the ejector at various cross section of ejector is as shown in figure 3.

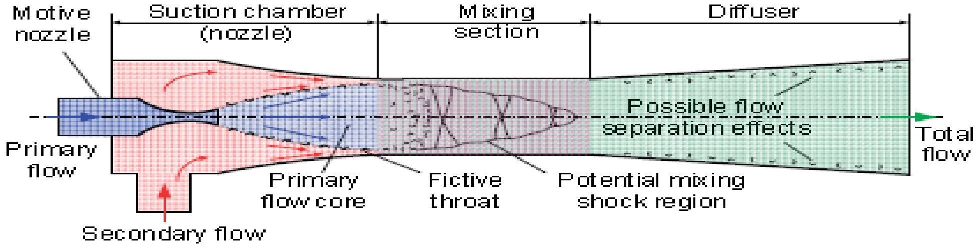

Figure 3 Behavior of primary and secondary fluid stream inside ejector

IV. DEVELOPMENT OF EJECTOR REFRIGERATION SYSTEM

Henry Giffard invented the condensing-type injector in 1858. The background of Giffard's invention was to find a solution to the problem of feeding liquid water to replenish the reservoir of steam engine boilers. Since then, ejectors have been studied intensively for a large number of different applications. In the past, ejectors have mostly been used in two different cycles for refrigeration purposes. In 1910, Leblanc introduced a cycle having a vapor jet ejector. His setup allowed producing a refrigeration effect by utilizing low-grade energy. Since steam was widely available at that time, the so-called steam jet refrigeration systems became popular in air-conditioning of large buildings and railroad cars. Nowadays, such cycles are used to harness solar heat or other low-grade heat sources. The patent by Gay (1931)[4] described how a two-phase ejector can be used to improve the performance of refrigeration systems by reducing the inherent throttling losses of the expansion valve.

Martynowsky (1954)[5] carried out the first investigation on the ejector refrigeration using a refrigerant different from the water, proposing the employment of R11 and R12 as working fluids Also, he considered the system powered by industrial waste thermal energy.

From the experimental data of a commercial ejector, Mizrahi et al. (1957)[6] found the behavior of an ejector system using different working fluids and considering a generator temperature of 60 ºC (Mizrahi et al., 1957). Among the refrigerants commonly used, they found that the system had the best behavior employing R22 and R12. They concluded that an ejector system is a feasible way of producing refrigeration with a low temperature heat source.

Heymann and Resnick (1964)[7] extended the work of Mizrahi et al. (1957) using in the ejector design the method derived by Keenan et al. (1950)[8]. They concluded that a generation temperature of 90 ºC was appropriate for the operation of an ejector system which could be provided by solar collectors.

Chen (1978)[9] employed the Elrod ejector theory in order to optimize an ejector system that was driven with the waste heat from the cooling system of an internal combustion engine. He used the R113 refrigerant as the working fluid. When varying the operating conditions it was found that system optimal behavior corresponded to its design point.

Hamner (1980)[10] chose R11, as the refrigerant in his ejector heat pump. The theoretical and experimental research considered the system global behavior, without involving the ejector performance. In the same way than Chen, Hamner suggested the use of an ejector system in order to provide automobile air conditioning. Faithfull (1984)[11] built an ejector system using R11 and pointed out that this system was the result of combining a Rankine cycle with a steam compression cycle, which was appropriate for the clarifying the thermodynamic principles and lecturing. Tyagi and Murty (1985)[12] and Chen and Hsu (1987)[13] carried out a parametric study of the ejector system. The first applied his system in cooling, which used refrigerants R11 and R113. They determined the coefficient of operation, the entrainment ratio and the condenser and evaporator cooling ratio, for different generation, condensation and evaporation temperatures, having been defined the efficiencies for main nozzle, mixing section and diffuser. They concluded that a higher cooling COP_c is obtained with higher T_{GE} and T_{EV} and smaller T_{CO}.

Lin-Tao et al. (1986)[14] arrived to the same conclusion when analyzed the Hamner system. These conclusions do not consider the experimental findings of the ejector characteristic in regard to its constant capacity, because of the ejector unavailability to reproduce theoretical or experimentally, its secondary flow choking phenomenon. Chen and Hsu studied theoretically the heat pump behavior operating with R11. They used the Elrod method to design the ejector and an optimal ejector was assigned for a certain design condition that considered the efficiencies of the main nozzle, diffuser and optimal mixing chamber, which is result of maximizing EV with respect to its area. They found the coefficient of performance for a range of operating conditions, included the one for its design, for the generation, condensation and evaporation temperature variation.

They found that T_{EV} does not affect the COP_h giving advantages to the heating mode over the cooling one, that higher COP_c corresponds to higherT_{GE} andT_{EV} andsmaller T_{CO}, just as Tyagi and Murty, that the addition of a regenerator and a precooler, as Huang and Jiang (1985) proposed, can increase the COP_c in a 17% operating at the generation, condensation and evaporation temperatures of 93.3 ºC, 43.3 ºC and 10 ºC, respectively. When they considered different efficiencies for the diffuser and main nozzle, found that the COP_c is more sensible to the changes of the efficiency of the main nozzle that to those of the diffuser. They concluded that a different optimal ejector corresponds to each operation mode.

Huang and Jiang (1985)[15] used the R113 as the working fluid in their experimental study and in the analysis of the ejector considered the Munday and Bagster choking theory. They included a regenerator and a pre-cooler to improve the behavior of the system. The first was used to preheat the liquid refrigerant returning to the generator by means of the hot refrigerant leaving the ejector, being reduced the heat transferred to the generator, increasing its COP_c. The second heat exchanger was used to pre cool the liquid refrigerant entering the expansion valve by means of the cold vapor leaving the evaporator. Huang et al. found that for a certain

condenser pressure -called critical- and below it, the cooling capacity and COP_c remained constant. They concluded that the system has to work at this critical pressure to avoid primary vapor waste.
Dorantes and Lallemand (1995)[16] proposed the use of the refrigerant mixtures in order to take advantage of the non azeotropic mixtures thermodynamic characteristics in the heat exchanger, also proposed the use of R142b. Later, Bounfarat and Lallemand (2009)[17] studied theoretically the effect of several mixtures of the refrigerants R22, R152a, R134a, RC318, R142b and R124 in a cooling ejector system, by means of the COP_c, the entrainment ratio U, the exergetic efficiency ε and $\Delta h_{EV}/\Delta h_{GE}$. The considered reference values are T_{GE} = 90 ºC, T_{CO} =25 ºC and T_{EV} =15 ºC. In regard to the pure refrigerants, R134a and R142b give the best system results. For the refrigerant mixtures, the moderate zeotropic and azeotropic provide the better system performance.

In 1996 Dorantes Rodrı´guez et al. (1996)[18] developed a mathematical model for a ejector refrigeration system using R142b as refrigerant and driven by solar energy, Their results were compared with the performance of an intermittent single effect absorption system named ISAAC of Energy Concepts, finding that the ejector refrigeration system COP values were very competitive and with the advantage of the simplicity of this kind of systems. In 1998 Nehad Al-Khalidy (1998)[19], carried out a theoretical study where the performance of a JCRS using different refrigerants was presented, selecting R113 as the best, although at present its use is prohibited. The author also evaluated some hydrocarbons. The work proposed certain refrigerant selection criteria, concluding that the molecular weight was an important parameter that had impact in the performance of the JCRS such as Holton concluded in a previous study (Kanjanapon and Satha, 2004). The following year Sun (1999)[20], realized a theoretical study comparing the COP of a JCRS using working fluids such as R718, R123, R134a, R11, R12, R113, R21, R142b, R152a, R318 and R500. He observed some similar behavior patterns but the best results were obtained with R152a and R500.

In 2000 Rogdakis and Alexis (2000)[21], theoretically studied a JCRS usingammoniaas refrigerant, however they did not find important performance improvements. Later, Riffat and Omer (2001)[22] used methanol as working fluid in an experimental system. Using Computational Fluid Dynamics (CFD) techniques they predicted the behavior of the ejector considering the distance of the ejector's main nozzle exit from the mixing section inlet. The same year, Nguyen et al. (2001)[23] evaluated and installed and evaluated a system at an office building in Loughborough, England. The system used R718 (water) as refrigerant for an air conditioning application that also provided heating in the winter season. An economical evaluation of the system was presented. They compared the system with a conventional technology of the same thermal capacity, finding that the investment payback period was 33 years which was very long for investors to be interested. Some economic indicators taken from this study are summarized in Table 1 to give an idea of market possibilities.

Another study realized in 2001 (Cizung et al., 2001)[24], theoretically compared the performance of a JCRS using R123, R134a, R152a, and the R717 and found R134a as the best working fluid not only from technical but also from environmental characteristics.
In 2004 Selvaraju and Mani (2004a, b)[25][26] presented very similar information comparing the system performance using R134a, R152a, R290, R600 and R717. The authors also confirmed R134a as the best working fluid, producing the highest system COP.

Table 1- Economic comparison between a mechanical and ejector refrigeration over a period of 30 years (table taken from Nguyen et al. (2001)

Economic indicators	Mechanical compression refrigeration system	Ejector refrigeration system
Estimated annual service cost(£)	240	-
Annual running cost(£)	348	12.6
Total spent in 30 years(£)	21,735	42,720
Equipment running cost	16,556	570
Total	38,291	43,290

The same year, Alexis and Katsanis (2004)[27] theoretically compared the performance of a JCRS using methanol with an ideal thermodynamic model for the same working conditions.
In 2005 Alexis (2005)[28] presented an exergy study of a JCRS finding the condenser and the ejector as the components with the largest exergy losses.

In 2006 Selvaraju and Mani (2006)[29] developed an experimental study of a JCRS using R134a as refrigerant finding the system's optimal operating conditions. Selvaraju and Mani concluded that the JCRS can be 20-30 percent more efficient than an absorption single effect system but this conclusion had no technical support. In the same period, Huang et al. (2006)[30] designed, installed and evaluated a JCRS using R141b and replaced the conventional mechanical refrigerant pump for a non moving thermal pump reducing the maintenance costs. Also Vidal et al. (2006)[31] theoretically evaluated a solar driven JCRS. They showed the effects of the collector area and the storage tank dimensions in the solar fraction. R141b was used as refrigerant obtaining a collection area of 80 m2 with a solar fraction of 42% and a thermal capacity of 10.5 kW.

By 2007 Jianlin et al. (2007)[32] in a theoretical study, presented a JCRS using R142b increasing the refrigerant sub cooling after the expansion valve, by another liquid-gas ejector.

The study concluded that the performance may be improved up to 10% with respect to another simple jet compression refrigeration cycles. However, the possible technical difficulties due to the control of the system are not mentioned. Sankarlal and Mani (2007)[33] realized an experimental evaluation of a JCRS using R717 as refrigerant. They studied the influence of the area, compression and expansion ratio in the performance of a system designed for an air conditioning application and a cooling capacity of 0.668 kW. Pridasawas and Lundqvist (2007)[34] developed a theoretical model of a solar JCRS based on TRNSYS. They found solar collection areas of around 80 m2 for a 75% solar fraction. The thermal cooling capacity was between 2.5 and 3 kW, for evaporating temperatures of 15 _C using isobutene as refrigerant. Yapici (2008)[35] realized a theoretical investigation of a JCRS using R123 as the working fluid. The evaporating temperatures were fort an air conditioning application with a cooling capacity of 1.2 kW. A mobile ejector nozzle was used and they pointed out the importance of the ejector design and its influence in the performance of the system. Boumaraf and Lallemand (2009)[17] developed software for the simulation and evaluation of the performance of a JCRS using R14b and R600a as working fluids. They concluded that R142b had better characteristics than R600a mainly due to the molecular weight of the former being twice than the latter. Referred to the parametric study they also concluded that the ejector design point must be selected according to the highest possible temperature of the heat source.

In 2010 Jianlin (Jianlin and Zhenxing, 2010)[36] presented a theoretical study of a single jet compression cycle behavior using R143a as working fluid under supercritical conditions. The study compared the proposed system with a subcritical one under the same operating conditions with R134a as working fluid. The authors inferred the potential increase in low grade energy when the system was operated under supercritical conditions. Using as operating conditions, generator, condensing and evaporation temperatures of 80 ºC, 30 ºC and 15 ºC respectively, the transcritical system presented an overall COP of 0.75 and a COP OF 0.45 for the subcritical one for a unitary cooling capacity. Theoretically, this cycle showed considerable advantages. However, as the author mentioned the problem in these cases is the high pressure of 8 MPa to be handled, having to be carefully designed and operated.

Although the single JCRS has an interesting range of applications, nevertheless the operating conditions or the ejector geometry as a single JCRS cannot reach compression ratios values higher than 4.0, reducing its application range only to air conditioning. At the same time, the COP values vary from 0.11 to a theoretical value of 0.6 for a temperature range between 70 and 90 ºC in the generator, 20-50 ºC in the condenser and 5 ºC in the evaporator (Gonza´ lez Bravo, 2005).

Table 2 presents the operating conditions and performance values of JCRS developments for the last fifteen years. Also the working fluids of every system are shown.

Table 2 Ejector refrigeration system developments for the last fifteen years.

Author(year)	Refrigerant	Temperature[ºC]	Cooling capacity(kW)	COP	Type of study
Dorantes Rodrı´guez et al. (1996)	R142b	T_{evap} = -10 T_{cond} = 30 T_{gen} = 105	2	0.34	Theoretical
Nehad Al-Khalidy (1998) R113	R113	T_{evap} = 18 T_{cond} = 42 T_{gen} = 87	0.42	0.55	Experimental
Da-Wen Sun (1999)	R718, R123, R134a R11, R12, R113, R21, R142b, R152a, RC318, R500	T_{evap} = 5 T_{cond} = 25 T_{gen} = 90	NA	0.5	Theoretical
Rogdakis and Alexis (2000)	R717 (ammonia)	T_{evap} = 12 T_{cond} = 34	NA	0.44	Theoretical

		T_{gen} = 100			
Riffat and Omer (2001)	Methanol	T_{evap} = -2 T_{cond} = 28 T_{gen} = 180	0.5	0.4	Theoretical
Nguyen et al. (2001	R718 (Water)	T_{evap} = 10 T_{cond} = 35 T_{gen} = 90	7	0.3	Experimental
Cizung et al. (2001)	R123, R134a, R152a and R717	T_{evap} = 14 T_{cond} = 35 T_{gen} = 90	NA	0.43	Theoretical
Selvaraju and Mani (2004a,b)	R134a, R152a, R290, R600a and R717	T_{evap} = 5 T_{cond} = 25 T_{gen} = 84	NA	0.33	Theoretical
Alexis and Katsanis (2004)	Methanol	T_{evap} = 5 T_{cond} = 42 T_{gen} = 150	NA	0.45	Theoretical
Alexis (2005)	R718 (Water)	T_{evap} = 8 T_{cond} = 44 T_{gen} = 160	100	0.6	Theoretical
Selvaraju and Mani (2006)	R134a	T_{evap} = 12 T_{cond} = 27 T_{gen} = 85	0.5	0.46	Experimental
Huang et al. (2006)	R141b	T_{evap} = 8 T_{cond} = 32 T_{gen} = 90	0.8	0.22	Experimental
Vidal et al. (2006)	R141b	T_{evap} = 8 T_{cond} = 32 T_{gen} = 80	10.5	0.39	Theoretical
Jianlin et al. (2007)	R142b	T_{evap} = 5 T_{cond} = 35 T_{gen} = 120	1	0.3	Theoretical
Pridasawas and Lundqvist (2007)	R600a (Isobutane)	T_{evap} = 15 T_{cond} = 35 T_{gen} = 120	3	0.48	Theoretical
Sankarlal and Mani (2007)	R717 (ammonia)	T_{evap} = 15 T_{cond} = 30 T_{gen} = 72	0.67	0.29	Experimental
Yapici (2008)	R123	T_{evap} = 10 T_{cond} = 35 T_{gen} = 98	1.2	0.39	Theoretical
Boumaraf and Lallemand (2009)	R142b and R600a	T_{evap} = 10 T_{cond} = 35 T_{gen} = 130	10	0.105	Theoretical
Jianlin and Zhenxing (2010)	R143a	T_{evap} = 15 T_{cond} = 30 T_{gen} = 80	1	0.75	Theoretical

V. COMBINED VAPOUR COMPRESSION-EJECTOR REFRIGERATION SYSTEM (VCR-VER)

In 1989 Sokolov and Hershgal (1989)[37] proposed a combined cycle between a ejector refrigeration system and a conventional mechanical compression system, as shown in figure 4, ameliorating the main limitations of each technology. On one side, the JCRS opens its application range and increases its efficiency. On the other hand, the mechanical compression refrigeration system (MCRS) reduces its electrical energy requirements. In 1990, Sokolov showed the design and evaluation of a combined VCR-VER JCRS obtaining significant improvements in the overall performance of the system (Sokolov and Hershgal, 1990).[38]

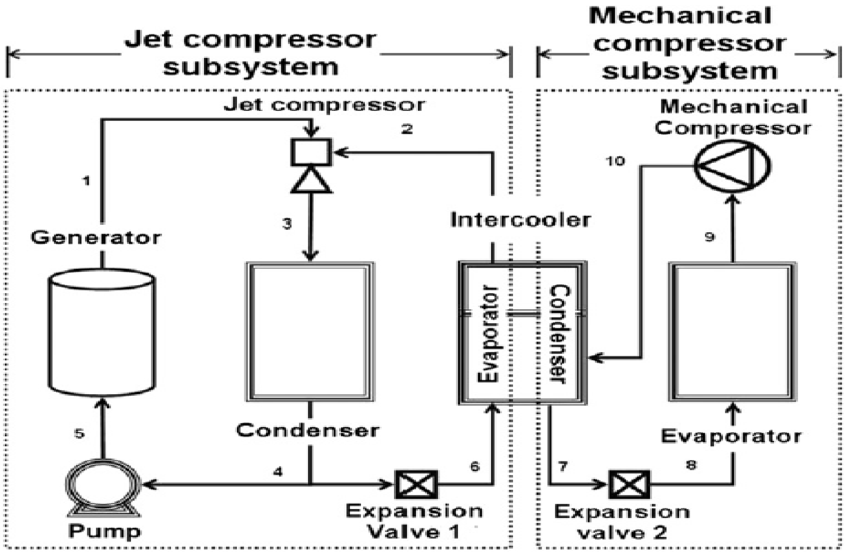

Figure 4 Improved jet refrigeration system

The interface between both systems is a heat exchanger called intercooler (Sokolov and Hershgal, 1990). The intercooler is a heat and sometimes also as a mass exchanger; it can be seen as the evaporator of the ejector system and as the condenser of the mechanical compression refrigeration. The intercooler allows the use of two different refrigerants at a time, selecting the most convenient thermo-physical refrigerant properties for each subsystem. This opens an important area of improvement for the overall system performance.

VI. DEVELOPMENT IN COMBINED VCR-VER SYSTEM

In 1996 Sun et al. (1996)[39] developed a theoretical study of a combined cycle, using a ejector system and an absorption cycle. The working fluid was a H_2O-LiBr mixture. The result of this study showed an improvement of the COP of around 20-40% with respect to a conventional single effect absorption cycle. The system was designed for air conditioning applications with COP values of between1 and 1.5. Despite the obvious improvements in efficiency, the required generator temperatures being between 180 and 240 ºC were too high to be handled by low grade energy sources. Also, the paper does not describe the practical operative disadvantages due to the nature of the system. In 1997, Sun realized a theoretical study of a hybrid ejector-compression refrigeration system (Sun, 1997)[40]. The system utilized a different refrigerant for each subsystem; water and R134a were used as working fluids for the ejector refrigeration system and the mechanical refrigeration system respectively. Designed for air conditioning applications, it used low grade energy sources with 80 ºC generator temperatures. The authors considered solar energy as an energy source with no cost; this resulted in very high COP values between 4 and 6.8, as they only considered the electrical energy utilized and not the required thermal energy. Also, they confirmed that the electrical energy requirements were reduced to half of that required regarding a conventional MCRS. The author proposed the use of the energy savings for the solar collector amortization. In 1998 Sun presented a comparative study of the effect of using different working pairs in a HJCRS system (Sun, 1998a, b). Refrigerant R718, the CFC's R11, R12, R113, the HCFC's R21, R123, R142b, the HFC's R134a, R152a, the organic compound RC318 and the azeotrope R500 were the refrigerants evaluated. The study utilized the Keenan ejector model and the Thermo-physical properties of the refrigerants were obtained using equations of state. The use of recovery heat exchangers was evaluated, concluding that the superheating of the fluid to the ejector reduces the entrainment rate as a consequence of the specific volume decreasing reducing the overall performance. The best working fluid pair was R718 (water) and R21for the JCRS and the MCRS respectively reaching an overall COP of over 0.7. By 2001 Huang et al. designed and evaluated an air conditioning HJCRS hybrid system using R141b in both mechanical an jet compression subsystems. They showed an interesting system configuration, taking the thermal energy from the gases at the exit of the mechanical compressor, as shown in figure 4. For the simulation and design, they used their own ejector model previously developed. The compressor behavior was predicted with an isentropic coefficient calculated for the particular case. The study concluded that the COP improvement was around 18% with respect to a simple JCRS. In 2004 Arbel and Sokolov (2004)[41] presented a theoretical study of a solar driven COMBINED VCR-VER using R142b as working fluid. The study compared the performance of the system with previous studies developed by Sokolov, where R113 was used. They showed not only technical but also ecological improvements by using R142b. At this time the use of R113 is prohibited.

The same year, Herna´ndez et al. (2004)[42] presented a theoretical study comparing the performance of R134a to R142b in a JCRS-MCRS cycle. The operating temperatures were selected considering an ice production application, driven by solar energy. The used the model developed by Lu to predict the behavior of ejector and the other devices were estimated by theoretical efficiencies. From this and previous studies an experimental testing system for JCRS was developed which is installed at the refrigeration and heat pumps laboratory at the Centro de Investigacio´n en Energı´a de la UNAM in Temixco, Morelos, Me´xico and shown in Fig. 5. At the present time experimental measurements are almost ready to begin. In 2005 Jaya et al. (2005)[43] theoretically compared a JCRS-MCRS using R124, R134a and R32 for evaporating temperatures between 5 and 15 ºC. They concluded that R32 gives the best COP value. Nevertheless R32 has drawbacks such as high generator pressures and high circulation ratios. They proposed the R134a as the preferred working fluid for low heat source temperature applications.

In 2007 Elakdhar et al. realized a theoretical study of a COMBINED VCR-VER for domestic refrigeration (Elakdhar et al., 2007)[44]. A simulation of the cycle was developed in FORTRAN and the thermo-physical properties were taken from REFPROP V8.0. The behavior of the system with different working fluids (R123, R124, R141b, R290, R152a, R717, R600a and R134a) was simulated. They obtained the best results for R141b. The system did not have an intercooler; the exit of the ejector was connected to the entrance of the compressor. They showed an inversely proportional relation between the COP and the decrement of the secondary evaporator temperature. However, as in other studies, some of the research developed should be renewed mainly because of the use of CFC's (chlorofluorocarbons) based refrigerants. After the development of the Kyoto protocol the use of environmental friendly refrigerants such as HCFC's (Hydro chlorofluorocarbons) and hydrocarbons has been promoted.

Table 3. Transition long term alternative refrigerants (Blitzar International, 2007)

Refrigerant	**Substitute for**	**ODP[Relative to R11]**	**GWP[Relative to CO_2]**
HCFC (Pure fluid in transition)			
R22	R502	0.05	1500
R142b	R114, R12B1	0.06	1800
HFF/HFC(Blends in transition			
R401b	R12 (R500)	0.035	1060
R402A	R502	0.02	2250
HFC Pure fluid without chlorine (Long term alternatives)			
R134a	R12 (R22)	0	1300
R152a	R12 (R22)	0	140
R125	R12 (R22)	0	2800
R143a	R12 (R22)	0	3800
R32	R12 (R22)	0	650
R227ea	R12B1	0	2900
R236fa	R12B1	0	6300
R23	R13 (R503)	0	11,700
HFC Blend without chlorine (Long term alternatives)			
R404A	R502	0	3260
R407A	R502	0	1770
R507	R502	0	3300
R407C	R22	0	1525
R410A	R22 (R13B1)	0	1725
R508A	R503	0	11,860
Halogen-free refrigerants (Long term alternatives)			
R717	R22 (R502)	0	0
R723	R22 (R502)	0	8
R600a	R114, R12B1	0	3
R290	R22 (R502)	0	3
R1270	R22 (R502)	0	3
R170	R13, R503	0	3
R744	Miscellaneous	0	1

As a reference, Table 3 presents the ozone depletion potential (ODP) and the Global Warming Potential (GWP) for some long term refrigerant alternatives and transition fluids according to Blitzar International (Blitzar International)[45]. ODP and GWP are referenced to R12 and CO2 respectively. As it can be seen the ODP problem is solved, nevertheless the research to reduce the GWP requires of greater contributions. In 2010, Petrenko et al. (2011)[46] presented a theoretical study of a trigeneration system that consisted of a cogeneration subsystem and a hybrid cooling subsystem. The combined VCR-VER cooling system included a mechanical compression and a jet compression refrigeration subsystem, using R744 (CO2) and R600 (butane) as working fluids, respectively. For the simulation they used an improved one-dimensional model which was validated with experimental data for several working fluids. The mechanical vapor compression subsystem was modeled using an isentropic compressor efficiency of 0.67 working under subcritical conditions. The cooling system was developed for a capacity of 10 kW and reached a total COP of 1.4 when operating under design conditions. The working fluids used met current environmental selection criteria.

The same year Vidal (Vidal and Colle, 2010)[31] carried out the simulation and thermo-economic optimization of a COMBINED VCR-VER cooling system. The working fluid for the mechanical compression subsystem was R134a whilstR141bwas proposed for the jet compression subsystem. The system used flat solar collectors to harness solar energy as the main power supply, having a gas burner as an auxiliary source. The authors discussed the importance of the proper selection of system components to obtain adequate payback periods. They pointed out the limitations of the JCRS and suggested the use of combined VCR-VER systems. The optimized system proposed had a cooling capacity of 10.5 kW and a COP of 0.89 when the generator, condenser, intercooler and evaporator temperatures were 80 ºC, 34 ºC, 19 ºC and 8 ºC respectively. The optimized solar collector area was assumed to be 105 m^2 for a solar fraction of 82%. The system was designed for an air conditioning application.

In 2012, Yinhai Zhu, Peixue Jiang (2012)[47] carried out the simulation of combined VCR-VER cooling system. They utilized the waste heat of basic compression system. Thus generator receives heat from basic compression system, thus creates vapours required to drive the ejector of ejector cycle. They found that hybrid refrigeration system with the parallel ejector cycle significantly improves the COP when the compressor discharge temperature is larger than 100 ºC. Simulations give an average COP increase for the hybrid system with R152a of 5.5% relative to the basic system and 8.6% with R22. The average COP increase of R134a system is about 0.7% due to its compressor discharge temperature is in the range of 70-90 ºC.
Table 4 summarizes the state of art for the COMBINED VCR-VER. It should be mentioned that other theoretical and experimental studies has been developed in the last years but they focused on the improvement of a specific system component, such as the ejector's geometry (Abdulateef et al., 2009[48]; Chunannond and Aphornratana, 2004[49]). However this work was focused on the improvements due to the use of different working fluids in simple and hybrid JCRS.

Table 4 Combined VCR-VER system from 1989 to 2007 [table taken from Gonza´lez Bravo et. al (2012)]

Author(year)	Refrigerant	Temperature [ºC]	Cooling capacity(kW)	Total COP	Type of study
Sokolov and Hershgal (1989)	R114	T_{evap} = -8 T_{int} = NA T_{cond} = 30 T_{gen} = 86	2.9	0.4	Experimental
Da-Wen Sun et al. (1996)	LiBr-R717	T_{evap} = 10 T_{int} = NA T_{cond} = 22 T_{gen} = 210	NA	1.8	Theoretical
Da-Wen Sun (1997)	R718 and R134a	T_{evap} = 5 T_{int} = 25 T_{cond} = 35 T_{gen} = 80	5	5	Theoretical
Da-Wen Sun (1998 a,b)	R21 and R718	T_{evap} = 5 T_{int} = 30 T_{cond} = 40 T_{gen} = 70	NA	0.65	Theoretical
Huang et al. (2001)	R141b and R22	T_{evap} = 5 T_{int} = 25 T_{cond} = 40	5.2	2.5	Experimental

		T_{gen} = 70			
Arbel and Sokolov (2004)	R142b	T_{evap} = 4 T_{int} = 38 T_{cond} = 50 T_{gen} = 100	3.5	5	Theoretical
Herna´ndez et al. (2004)	R134a, R1142b	T_{evap} = -10 T_{int} = NA T_{cond} = 30 T_{gen} = 85	1	0.48	Theoretical
Jaya et al. (2005)	R124, R134a and R32	T_{evap} = -5 T_{int} = NA T_{cond} = 20 T_{gen} = 100	NA	0.7	Theoretical
Elakdhar et al. (2007)	R123, R124, R141b, R290, R152a, R717, R600a and R134a	T_{evap1} = 5 T_{evap2} = -30 T_{cond} = 42 T_{gen} = NA	NA	1.38	Theoretical
Petrenko et al. (2011)	R744, R600	T_{evap} = -20 T_{int} = 20 T_{cond} = 36 T_{gen} = 120	10	1.4	Theoretical
Vidal and Colle (2010)	R134a, R141b	T_{evap} = 8 T_{int} = 19 T_{cond} = 34 T_{gen} = 80	10.5	0.89	Theoretical
Yinhai Zhu, Peixue Jiang (2012)	R134a, R152a, R22	T_{evap} = -5 T_{int} =NA T_{cond} = 50 T_{gen} = 82.55	5.99	2.40	Theoretical

Note- T_{evap}, T_{cond},T_{gen}, T_{int} are the evaporation, condenser, generator and intercooler temperatures respectively.

VII. CONCLUDING REMARK

The most appropriate working fluids for JCRS according to its performance are R134a, R141b, R142b, Methanol, R600a and finally R717 obtaining COP values from 0.3 to 0.48 for evaporator temperatures between 10 and 15 ºC, condenser temperatures between 25 and 42 ºC and generator temperatures between 72 and 180 ºC. Most recently, refrigerant R143a is seen as a good refrigerant proposal at least in theory as it has high working pressures which must be handled. Also R143a has a high GWP index. For the combined VCR-VER is possible to remark Sun's research in which for the same system higher COP values were obtained when two different working fluids were used. It is in this configuration where an important improvement opportunity for these systems exists, as the most adequate refrigerant thermo-physical properties for each subsystem can be selected. Referred to the state of art for COMBINED VCR-VER it can be concluded that R717, R134a and R142b are selected again when they are used in systems with evaporation temperatures around 4 ºC, condensation temperatures between 25 and 50 ºC, intercooler temperatures between 30 and 38 ºC and generation temperatures between 80 and 100 ºC.

As for simple ejector refrigeration system as well as for combined VCR-VER systems, the most suitable application is air conditioning because of the relatively higher evaporator temperatures required than for refrigeration applications. The search of new working fluids with very low or no environmental impact has not ended and it is clear that finding the working fluids with these characteristics will not be easy, except in the case of water, but limited for specific applications. As it can be seen in recent studies that hydrocarbons working fluids can be a good technical and environmental option for jet compression systems although they will require carefully developed security protocols due to their flammability. From a technical point of view, the use of combined VCR-VER systems allows to extend the ejector application range. This is because despite its geometry and configuration, the ejector cannot handle compression rates greater than 4, limiting its functionality for air conditioning applications. Another competitive advantage is that the combined VCR-VER cycles make possible the use of one working fluid at a time for each subsystem increasing the improving cycle performance opportunity. From the economical point of view, combined VCR-VER cycles require a high initial investment. Nevertheless large-scale systems can result profitable because of the electrical energy consumption decrease, in which the cost presents a clear growth trend. In environmental terms the energy diversification through combined VCR-VER cycles can help reduce the greenhouse effect gases production by handling them with solar energy or waste heat from some thermal devices.

REFERENCES

[1]. Gonza´ lez Bravo, H., 2005. Disen˜o te´rmico meca´nico de una ma´quina solar de refrigeracio´n por eyectocompresio´n de vapor para la produccio´n de 100 kg. de hielo. Universidad La Salle, Me´xico.

[2]. Sun, D.W., 1998a. Evaluation of a combined ejector-vapourcompression refrigeration system. Int. J. Energ. Res. 22, 333-342.

[3]. Sun, D.W., 1998b. Evaluation of a solar combined ejector-vapourcompression refrigeration system. International Journal of Energy Research 2, 333-342.

[4]. Gay N.H., 1931, Refrigerating system, *US Patent* 1,836,318.

[5]. Martynowsky, W., 1954. Use of waste heat for refrigeration. Refrigeration Eng. 62, 51.

[6]. Mizrahi, J., Solomiansky, M., Zisner, T., Resnick, W., 1957. Ejector refrigeration from low temperature energy. Bull. Res. Counc. of Israel 6C, 1-8.

[7]. Heymann, M., Resnick, W., 1964. Optimum ejector design for ejector-operated refrigeration. Israel J. of Tech. 2, 242.

[8]. Keenan, J., Newman, E., Lustwerk, F., 1950. An investigation of ejector design by analysis and. J. Appl. Mech., ASME72, 299-309.

[9]. Chen, L., 1978. A heat driven mobile refrigeration cycle analysis. Energy Convers. 18 (1), 25-29.

[10]. Hamner, R., 1980. An alternate source of cooling: the ejectorcompression heat pump. ASHRAE J. 22, 62-66.

[11]. Faithfull, D., 1984. A combined Rankine and vapour compression cycle heat pump for teaching purposes. Directly Fired Heat Pump for use in domestic and commercial, 3.1, 1-7.

[12]. Tyagi, K., Murty, K., 1985. Ejector-compression systems for cooling: utilising low grade waste. Heat Recovery Syst 5, 545-550.

[13]. Chen, F., Hsu, C., 1987. Performance of ejector heat pumps. Energy Res 11, 289-300.

[14]. Lin-Tao, L., 1986. PhD Thesis: Etudes The´orique et expe´rimentale de la production de froid par machine thrirherme a ejecteur de fluide frigorige´ne. Laboratorie d'Energetique el d'Automatique, Lyon.

[15]. Huang, B., Jiang, C.H., 1985. Ejector performance characteristics and design analysis of. Transact. ASME 107, 792-802.

[16]. Dorantes, R., Lallemand, A., 1995. Prediction of performance of a jet cooling system operating with pure. Int. J. Refrigerat. 18 (1), 21-30.

[17]. .Boumaraf, L., Lallemand, A., 2009. Modeling of an ejector refrigerating system operating in dimensioning and ofdimensioning conditions with the working fluid R142b and R600a. Appl. Therm. Eng 29, 265e274

[18]. Dorantes Rodrı´guez, R., Estrada Gazca, C., Pilatowsky Figueroa, I., 1996. Mathematical simulation of a solar ejector-compression refrigeration system. Appl. Therm. Eng. 16 (8/9), 669-675.

[19]. Al-Khalidy, N., 1998. An experimental study of an ejector cycle refrigeration machine operating on r113. Int. J. Refrigerat. 21 (8), 617-625.

[20]. Sun, D.W., 1999. Comparative study of the performance of an ejector refrigeration cycle operating with various refrigerants. Energ. Convers. Manag. 40, 873-884.

[21]. Rogdakis, E., Alexis, G., 2000. Design and parametric investigation of an ejector in an air-conditioning system. Appl. Therm. Eng. 20, 213-226.

[22]. Riffat, S.B., Omer, S.A., 2001. Cfd modelling and experimental investigation of an ejector refrigeration system using methanol as the working fuid. Int. J. Energ. Res. 25, 115-128.

[23]. Nguyen, V., Riffat, S., Doherty, P., 2001. Development of a solar powered passive ejector cooling system. Appl. Therm. Eng. 21, 157-168.

[24]. Cizung, K., Mani, A., Groll, M., 2001. Performance comparison of vapour jet refrigeration system with environment friendly working fuids. Appl. Therm. Eng. 21, 585-598.

[25]. Selvaraju, A., Mani, A., 2004a. Analysis of an ejector with environment friendly refrigerants. Int. J.Therm. Sci. 43, 915-921.

[26]. Selvaraju, A., Mani, A., 2004b. Analysis of an ejector with environment friendlyrefrigerants.Appl.Therm.Eng.24,827-838.

[27]. Alexis, G., Katsanis, J., 2004. Performance characteristics of a methanol ejector refrigeration unit. Energ. Convers. Manag. 45, 2729-2744.

[28]. Alexis, G., 2005. Exergy analysis of ejector refrigeration cycle using water as working fluid. Int. J. Energ. Res. 29, 95-105.
[29]. .Selvaraju, A., Mani, A., 2006. Experimental investigation on R134a vapours ejector refrigeration system. Int. J. Refrigerat. 29, 1160-1166.
[30]. Huang, B., Hu, S., Lee, S., 2006. Development of an ejector cooling system with thermal pumping effect. Int. J. Refrigerat. 29, 476-484.
[31]. Vidal, H., Colle, S., 2010. Simulation and economic optimization of a solar assisted combined ejectorevapor. Appl. Therm. Eng. 30, 478-486.
[32]. Jianlin, Y., Yunfeng, R., Hua, C., Yanzhong, L., 2007. Applying mechanical subcooling to ejector refrigeration cycle for improving the coeffcient of performance. Energ. Convers. Manag. 48, 1193-1199.
[33]. Sankarlal, T., Mani, A., 2007. Experimental investigations on ejector refrigeration system with ammonia. Renew. Energ 32, 1403-1413.
[34]. Pridasawas, W., Lundqvist, P., 2007. A year-round dynamic simulation of a solar-driven ejector refrigeration system with isobutane as a refrigerant. Int. J. Refrigerat. 30, 840-850.
[35]. Yapici, R., 2008. Experimental investigation of performance of vapor ejector refrigeration system using refrigerant R123. Energ. Convers. Manag 49, 953-961.
[36]. Jianlin, Y., Zhenxing, D., 2010. Theoretical study of a transcritical ejector refrigeration cycle with refrigerant R143a. Renew. Energ. 35, 2034-2039.
[37]. Sokolov, M., Hershgal, D., 1989. Compression enhanced ejector refrigeration cycle for low grade heat utilization. In: IEEE, CH2781-3/89, pp. 2543-2548.
[38]. .Sokolov, M., Hershgal, D., 1990. Enhanced ejector refrigeration cycles powered by low grade heat part1, 2 and3. Int. J. Refrigerat. 13, 351-356.
[39]. Sun, D.W., Eames, W.I., Aphornratana, S., 1996. Evaluation of a novel combined ejector absorption refrigeration cycle, I computer simulation. Int. J. Refrigerat. 19, 8.
[40]. Sun, D.W., 1997. Solar powered combined ejector-vapour compression cycle for air conditioning and refrigeration. Energ. Convers. Manag. 38 (5), 479-491.
[41]. Arbel, A., Sokolov, M., 2004. Revisiting solar-powered ejector air conditioner e the greener the better. Sol. Energ. 77, 57-66.
[42]. Herna´ndez, J., Dorantes, R., Best, R., Estrada, C., 2004. The behaviour of a hybrid compressor and ejector refrigeration system with refrigerants 134a and 142b. Appl. Therm. Eng. 24.
[43]. Jaya, P.V., Prakasha, R., Srinivasa, M., 2005. Studies on an Ejector absorption Refrigeration Cycle With New Working Fluid Pairs. In World Climate and Energy Event, pp. 113-122.
[44]. Elakdhar, M., Nehdi, E., Kairouani, L., 2007. Analysis of a compression/ejection cycle for domestic refrigeration. Ind. Eng. Chem. Res. 46, 4639-4644.
[45]. Blitzar International., 2007. Available on October 2008, de IIFIIR: www.iifiir.org/en/doc/1029.pdf.
[46]. Petrenko, V.O., Huang, B.I., Ierin, V.O., 2011. Design theoretical study of cascade CO2 sub-critical mechanical compression/butane ejector cooling cycle. Int. J. Refrigerat 34, 1649-1656.
[47]. Yinhai Zhu, Peixue Jiang, 2012. Hybrid vapor compression refrigeration system with an integrated ejector cooling cycle. Int. J. of refrigeration 35,68-78
[48]. Abdulateef, J., Sopian, K., Alghoul, M.A., Sulaiman, M., 2009. Review on solar-driven ejector refrigeration technologies. Renew. Sustain. Energ. Rev. 13, 1338-1349.
[49]. Chunannond, K., Aphornratana, S., 2004. Ejectors: applications in refrigeration technology. Renew. Sustain. Energ. Rev. 8, 129-155.