A Review On Historical And Present Developments In Ejector Systems

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Abstract

Ejectors are simple pieces of equipment. Nevertheless, many of their possible services are overlooked. They often are used to pump gases and vapours from a system to create a vacuum. However, they can be used for a great number of other pumping situations. This paperprovides reviewon the development in ejectors, applications of ejector systems and system performance enhancement. Several topics are categorized provides useful guidelines regarding background and operating principles of ejector including mathematical modelling, numerical simulation of ejector system, geometric optimizations. Research works carried out recently are still limited to computer modelling, forthe real industrial applications more experimental and large-scale work are needed in order to provide better understanding.

1. INTRODUCTION

Ejectors are co-current flow systems, where simultaneous aspiration and dispersion of the entrained fluid takes place. This causes continuous formation of fresh interface and generation of large interfacial area because of the entrained fluid between the phases. The ejector essentially consists of an assemble comprising of nozzle, converging section, mixing throat and diffuser. According to the Bernoulli's principle when the motive fluid is pumped through the nozzle of a jet ejector at a high velocity, a low pressure region is created at just outside the nozzle. A second fluid gets entrained into the ejector through this low pressure region. The dispersion of the entrained fluid in the throat of the ejector with the motive fluid jet emerging from the nozzle leads to intimate mixing of the two phases. A diffuser section of the mixing throat helps in pressure recovery. The motive fluid jet performs two functions one, it develops the suction for the entrainment of the secondary fluid and the second; it provides energy for the dispersion of the one phase into the other. This process has been largely exploits in vacuum systems in which high speed fluid stream is used to generate vacuum.Refrigeration is recognized as an indispensable method of improving human beings' living conditions since early twentieth century. Refrigeration systems, in the various applications including food storage and provision of thermal comfort, have contributed significantly to the industrial and health sectors. Conventional vapour compression refrigeration cycles are driven by electricity with the consumption of fossil fuels. However, this results in air pollution and emission of greenhouse gases, and consequently poses a threat to the environment. Hence, improvement on the refrigeration system's working performance will result in less combustion of primary energy, and mitigation of the environmental pollution. Ejector refrigeration systems (ERS) are more attractive com-pared with traditional vapour compression refrigeration systems, with the advantage of simplicity in construction, installation and maintenance. Moreover, in an ERS, compression can be achieved without consuming mechanical energy directly. Furthermore, the utilization of low-grade thermal energy (such as solar energy and industrial waste heat) in the system can helps to mitigate the problems related to the environment, particularly by reduction ofCO_2 emission from the combustion of fossil fuels.

However, due to their relatively low coefficient of performance (COP) [1–3], ERS are still less dominant in the market place compared with conventional refrigeration systems. Therefore, in order to promote the use of ERS, many researchers have been engaged in enhancing the performance of ejector system and combining ERS with other refrigeration systems in order to improve the overall system performance. Building on other published review papers [1–3], this paper aims to update the research progress and development in ejector technology in the last decade. This paper will emphasize on the various combination of ejectors and other cycles. Linkages and comparisons between different research cases are presented, and similar study concepts are grouped and briefly described as overall summaries.

2. DEVELOPMENTS IN EJECTOR MODELS

Ejector refrigeration systems were first invented by Sir Charles Parsons around 1901 for removing air from a steam engine's condenser. It was later used in the first steam jet refrigeration system by Maurice Leblanc et al. [50] in 1910. Since then, considerable efforts have been concentrated on the enhancement and refinement of ERS.

2.1 Single Phase Models

A one dimensional model described by Keenan et al. [51] in 1950 was the first application of continuity, momentum and energy equation in ejector design principle. This model has been used as a theoretical basis in ejector design since then. Keenan's model, however, cannot predict the constant-capacity characteristic and was later modified by Munday and Bagster [52]. Based on their theory, it is assumed that the primary fluid flows out without mixing with the secondary fluid immediately induces a converging duct for the secondary fluid. This duct acts as a converging nozzle such that the secondary flow is accelerated to a sonic velocity at some place, known as effective or hypothetical throat. After that both fluids mix with a uniform pressure. Eames et al. [44] studied a small-scale steam-jet refrigerator and presented a theoretical model that included irreversibilities associated with the primary nozzle, the mixing chamber and the diffuser. This model was based on constant-pressure mixing process, but without considering the choking of the secondary flow. In order to take this in to account, Huang et al. [36] presented a one-dimensional critical model (double-chocking) by assuming that mixing of two streams occurs inside constant area section with uniform pressure. The model was experimentally verified with 11 different ejectors using R141b as the working fluid. In order to simplify the model, more models [53] were proposed to calculate the performance of ERS. In these models, the thermo-physical and transportation properties need to be obtained from data base, which limits their application.

Zhu et al. [54] proposed an ejector for a real time control and optimization of an ejector system, which was based on one-dimension analysis. Though the model is simplified, the expressions were more complex and some parameters needed to be determined experimentally. In order to give a more accurate prediction of the ejector performance in the mixing chamber, Yapici and Ersoy [18] derived a local model based on constant-area mixing process. The ejector consisted of a primary nozzle, a mixing chamber in cylindrical structure and a diffuser. Compared with a similar model designed by Sun and Eames [55] under same operating temperatures, Yapici's model showed better COP. Elakhdar et al. [20] developed a mathematical model in order to specifically design a R134a ejector and predict the performance characteristics over different operating conditions. Simulation results showed that the present model data were in good agreement with experimental data in the literature with an average error of 6%. A constant-area 1-D model was recently presented by Khalil et al. [30]. Governing equations were developed for the ejector's three different operating regimes, supersonic regime, the transition regime and the mixed regime. Environmental friendly refrigerants were used as working fluids in the simulation. Results were compared with that of experimental data available in the literature, and good agreement was demonstrated.All the above models are based on ideal gas assumption which does not reflect the actual process occurring in the ejector. Rogdakis and Alexis [56] improved the model proposed by Munday and Bagster [52] by using the thermodynamic and transportation properties of real gases. When considering the friction losses, a constant coefficient was assumed to simplify the model.

However, the friction losses were closely related to the velocity, and the velocity varied considerably along the ejector. Taking this into account, Selvaraju and Mani [19] developed a model based on Munday and Bagster's theory for critical performance analysis of the ejector system. This model applied an expression to describe the friction losses in the constant area section. A 1-D model avoiding the ideal gas assumption was proposed by Grazzini et al. [57]. Heat exchanger irreversibilities were taken into consideration, and real gas behaviour was simulated. A comparison between different refrigerants was presented and R245fa was selected as a working fluid. However, validation with experimental data in literature was not available. In order to check the validity of the ideal gas assumption, Grazzini et al. [58] evolved another model with the key concept of metastable state. To set the border for metastable region, a spinodal curve was introduced. The modelling results were compared with experimental data. The author concluded that in order to avoid complexity, the metastable behaviour of steam can be implemented in a single 1-D model giving stable results.

2.2 Two Phase Models

The abovementioned models are based on the assumption that the flow in the ejector is in compressible single phase and recompression occurs across a normal shock wave. However, under many real applications, phase change can occur and a condensation shock may develop. Thus, some researchers are engaged in ejector simulation with two-phase flow. By introducing dryness of the fluid in the calculation of the specific volume, enthalpy and entropy, Sherif et al. [59] derived an isentropic homogeneous expansion/compression model to account for phase change due to expansion, compression and mixing. In this model, the primary fluid was a two-phase mixture and the secondary fluid was either a sub-cooled or saturated liquid having the same chemical composition as the primary fluid. Cizungu et al. [45] derived a two-phase thermodynamic model to calculate the entrainment ratio. This model can be used both for single-phase and two-phase ejector with single component or two components working fluids. He et al. [60] investigated the usefulness of a multivariate grey prediction model, which incorporated grey relational analysis to predict the performance of ERS. The importance of influencing variables was first evaluated, and then the variables were ranked according to the grey relational method. It also compared the performance of the combined grey model with that of conventional one-dimension theory model as well as experimental data. The simulation results showed that the grey system theory can be used to analyze the ERS.

2.3 CFD models

Despite the remarkable progress, made in thermodynamic modelling, these models were unable to reproduce the flow physics locally along the ejector. It is the understanding of local interactions between shock waves and boundary layers, their influence on mixing and re-compression rate that will produce a more reliable and accurate design, in terms of geometry, refrigerant type and operation conditions. Computational Fluid Dynamics (CFD) modelling can provide more accurate simulations of the ejector in accordance with experiment results. Early CFD studies can be traced back to late 1990s. However, they failed to overcome some of the fundamental problems, especially regarding the simulation of shock-mixing layer interaction and ejector operation under different working conditions. Compressibility or turbulence was hardly taken into consideration. Even when turbulence was considered, only k-epsilon based models were used. No experimental validations or justification, except for CPUcost were carried out. Recently, Rusly et al. [32] stimulated the flow through an R141b ejector by using the real gas model in the commercial code, FLUENT. The effects of ejector geometries on system performance were investigated numerically.

The CFD results were validated with experimental data and good agreement was found. The selection of correct turbulence model plays an important role in predicting the mixing process in the ejector for CFD studies. Turbulence effects in the ejector have been modelled using the standard k-epsilon turbulence model by Scott [61] using CFD. The CFD results were later verified with an experimental investigation of an ejector with R245fa as a working fluid [62]. Comparisons were made between results from experiments, CFD model and a theoretical 1-D model by Ouzzane and Aidoun [63]. It was concluded that CFD model provided better agreement (difference of less than 16%) than 1-D model. Aiming at validation the choice of a turbulence model for the computation of supersonic ejectors in refrigeration applications, Bartosiewicz et al. [64] compared experimental distribution data with results of simulation using different turbulence models. However, the choice of air as working fluid and other test conditions were not very in accordance with cooling cycles. Later they extended their work using R142b as refrigerant et al. [65]. With consideration of shock-boundary layer interactions, this ejector model contributed to the understanding of the local structure of the flow and demonstrated the crucial role of the secondary nozzle for the mixing rate performance.

Pianthong et al. [5] employed the CFD with realizable k-epsilon turbulence model to predict the flow phenomena and performance in steam ejectors with application in refrigeration system. The result indicated that CFD can predict ejector performance very well and reveal the effect of operating conditions on the effective area that was directly related to its performance. In order to consider the sensitivity of the turbulence model over several conditions, Hemidi et al. [66] carried out CFD analysis of a supersonic air ejector with single and two phase operation. Entrainment ratio based on K-epsilon model and k-o-sst model were compared with experimental data. The results demonstrated that even with the same prediction level, both models could provide very different local flow structures.

2.4 Non Steady Flow Models

Since the mid-1990s, some researchers have focused on the theory and implementation of non-steady or pressure-exchange ejector. Compared with conventional steady ejectors, non-steady ejectors allow energy transfer between two directly interacting fluids but maintaining them separable. By utilizing the reversible work of pressure forces acting at fluid interfaces between primary flow and secondary flow, non-steady ejectors have the potential of much greater momentum transfer efficiency.

Recently, Hong et al. [67] presented a novel thermal driven rotor-vane/pressure-exchange ERS. Unlike other pressure-exchange ejectors which had canted primary nozzles on their rotors, the rotor of this ejector had vanes directly on it and a primary nozzle separated from it. Computational study and experimental work were included in order to optimize the ejector geometry. However, the concentrations were only placed on the overall shape of rotor vane, without considering Mach number of incoming flow, the geometry in the interaction zone and the diffuser geometry.

Ababneh et al. [68] studied the effects of the secondary fluid temperature on the energy transfer in a non-steady ejector with a radial-flow diffuser. The flow field was analyzed at Mach numbers 2.5 and 3.0, with a range of temperatures from À 10 1C to 55 1C. The results revealed that the actual energy transfer to the secondary fluid, which included the effects of irreversibilities, decreased with the increase in ambient temperature. However, due to mechanical difficulties, the experimental work was halted, only numerical simulation was presented. Gould et al. [29] carried out theoretical analysis of a steam pressure exchange (PE) ejector in automotive air conditioning (AC) system. Waste heat from the engine of vehicle was utilized as the main heat source. Comparisons were made between a conventional R134a AC system and the steam PE ejector AC system at idling and 50 mph conditions. The results showed that the steam PE ejector system consumed at least 68% less energy than R134a AC system. And COP of PE ejector AC system was 2.5–5.5 times that of R134a AC system at both conditions. However, the theoretical data were not verified with experimental results.Table 1 lists the references with different model types and their key simulation results.

Table 1 – Working conditions and Simulation Results for Selected Models

Reference	Simulation model	Refrigerant	Evaporator temperature (°C)	Condenser temperature (°C)	Boiler temperature (°C)	System performance	
						COP	Condition
Eames et al. [44]	Constant pressure model with irreversibilities	water	5:10	26:37	120:140	0.239	$T_b = 120\,°C$, $T_c = 27\,°C$, $T_e = 5\,°C$, $\gamma_A = 102$
Yapıcı and Ersoy [18]	Constant area model	R123	5	30	60:100	0.295	$T_b = 100\,°C$, $T_c = 30\,°C$, $T_e = 5\,°C$, $\gamma_A = 11.46$
Khalil et al. [30]	Constant area model takes into account of R134a vapour state and friction loss	R134a	6:10	25:40	65:85	0.355	$T_b = 70\,°C$, $T_c = 35\,°C$, $T_e = 10\,°C$, $\gamma_A = 0.838$
Grazzini et al. [57,58]	1-D model with real gas behaviour and heat exchanger irreversibilities Diffuser design on a modified CRMC criterion accounts for friction loss	R245fa	12	35	115	0.325	$T_b = 115\,°C$, $T_c = 35\,°C$, $T_e = 12\,°C$
Cizungu et al. [45]	A two-phase thermodynamic model takes into account the duct effectiveness, wall friction, momentum loss, ejector geometry, shock waves (with no assumption of constant area/pressure mixing)	R11	3.5:8.5	30:35	80:130	0.415	$T_b = 90\,°C$, $T_c = 30\,°C$, $T_e = 8.5\,°C$, $\gamma_A = 6$
Pianthong et al. [5]	CFD modelling	water	5:15	15:35	120:140	0.42	$T_b = 120\,°C$, $T_c = 30\,°C$, $T_e = 10\,°C$
Aidoun and Ouzzane [12]	1-D model accounts for changes in refrigerant properties with the flows axial position	R141b	5	NA	75	0.322	$T_b = 75\,°C$, $T_e = 5\,°C$,
P_c= 120:380 Kpa Boumaraf and Lallemand [40]	Constant area model includes a correlation of the ejector entrainment ratio established in different operating conditions	R142b	10	35	120:130	0.128	$T_b = 120\,°C$, $T_e = 10\,°C$
		R600a	10	35	120:130	0.089	$T_b = 120\,°C$, $T_e = 10\,°C$

3.DEVELOPMENTS IN EJECTOR GEOMETRIC OPTIMIZATION

In order to make the ejector system more economically attractive, a number of researches have been investigated the optimization of the ejector geometry on system performance.

3.1 Area Ratio

An important non-dimensional factor affecting ejector performance is the area ratio γ_A between primary nozzle and constant area section. It is known that flow emerges from the primary nozzle and maintains its definition as primary fluid for some distance. The secondary fluid is entrained into the region between the primary fluid and the ejector wall. If an ejector of fixed primary pressure, secondary pressure and nozzle geometry is considered, increasing the mixing section area will result in a greater flow area for the secondary stream. The entrainment ratio will therefore increase but since the compression work available from the primary flow is unchanged, the ejector is unable to compress to higher discharge pressures. In this case, according to Varga et al. [42] increasing γ_A increases entrainment ratio and decreases the critical back (condenser) pressure and therefore an optimal value should exist, depending on operating conditions. Yapici et al. [15] studied the performance of R123, using six configurations of ejector over a range of the ejector area ratio from 6.5 to 11.5. It was concluded that the optimum area ratio increased approximately linearly with generator temperature in the ranges of 83–103 1C. Instead of using water-cooled condenser, Jia et al. [21] presented an experimental investigation on air-cooled ERS using R134a with 2 kW cooling capacity. Replaceable nozzles with varying ejector area ratios from 2.74 to 5.37 were used, and the best system performance was shown for area ratio from 3.69 to 4.76.

Cizungu et al. [45] modelled a two-phase ejector with ammonia as working fluid, and found out a quasi linear dependence between γ_A and the driving pressure ratio (pressure ratio of boiler to condenser). This result was suitable for the rough draft of sizing and operational behaviour of the refrigerator. Area ratio, however, can be identified as a single optimum that would bring the ejector to operate at critical mode for a given condenser temperature. Obviously, this would require different ejectors for different operating conditions. In order to overcome this problem, a new feature a spindle was implemented and tested numerically and experimentally by Ma et al. [6] and Varga et al. [69]. By changing the spindle position, the area ratio γ_A can be changed. As the spindle tip travels forward, the primary nozzle throat area decreases, and consequently γ_A increases. CFD simulation was carried out by Varga et al. [70] to analyze the effect area ratio on the ejector performance. The authors indicated that ejectors with area ratios varying from 13.5 to 26.4 could achieve entrainment ratios from 0.18 to 0.38. They also pointed out that by changing the spindle position, an optimal γ_A can be achieved with a single ejector.

Experimental investigation of this spindle system was carried out by Ma et al. [6] using water as refrigerant. The results showed that when spindle position was 8 mm inwards the mixing chamber, an optimum entrainment ratio of 0.38 could be achieved, which were less than the maximum value of Varga's CFD modelling [70] at almost same designed working conditions. The group [69] later summarized and compared the experimental results with CFD data. It was concluded that CFD and experimental primary flow rates agreed well, with an average relative error of 7.7%. Table 2 shows the various studies on the ejector's area ratio with different working conditions.

Table 2 Results on the Ejector's Area Ratio with Different Working Conditions

Reference	Method	Working fluid	Area ratio	COP	T_e (°C)/P_e	T_c (°C)/P_c	T_b (°C)/P_b	Conclusion
Yapici et al. [15]	Experiment	R123	6.5:11.5	0.29:0.41	10 °C	125 kpa	83:103 °C	(1) The optimum area ratio nearly linearly increased with the generator temperature in the studied range. (2) For a given ejector area ratio, there existed an optimum generator temperature at which maximum COP was obtained from the ejector refrigeration system.
Jia et al. [21]	Experiment	R134a	2.74:5.37	0.14:0.35	4.43 bar	7.8 bar	17.5:16.5 bar	(1) Optimum area ratio was found between 3.69 and 4.76, with maximum COP in the range of 0.24–0.30. (2) For given operating conditions, the cooling capacities were related to area ratios as well as nozzle diameters while COPs depended only on area ratios.
Cizungu et al. [45]	Numerical modelling	Ammonia	4:12	0.24:0.34	4.5 °C	32 °C	80:130 °C	There exited a quasi linear dependence between γ_A and the driving pressure ratio (pressure ratio of boiler to condenser).
Varga et al. [69,70]	CFD modelling	Water	13.5:26.4	0.18:0.38	10 °C	4.25: 5.63 kpa	70.1:101 kpa	By changing the spindle position, the effective nozzle area can be adjusted and an optimal γ_A can be adjusted with a single ejector
Ma et al. [6]	Experiment	Water		0.17:0.32	10 °C	25 mbar: 65 mbar	84:96 °C	

3.2 Nozzle Exit Position (NXP)

The nozzle exit position (NXP) inwards or outwards the mixing chamber is known to affect both the entrainment and pressure lift ratio performance of ejectors. In the experimental studies [7,10,16] and CFD simulations [32, 64, 71–74], it was demonstrated that moving the nozzle exit into the mixing chamberreduced COP and cooling capacity. Recently, Eames et al. [48]found a clear optimum of the entrainment ratio (40% increases) at5 mm from the entrance of the entrainment chamber. In this casethe ejector tail was designed by the constant momentum ratechange (CMRC) method and R245fa was used as working fluid.Similar conclusions can be found from numerical investigationscarried out by Varga et al. [70] and Zhu et al. [75]. CFD modellingresults from Varga et al. [70] indicated that an optimum entrainment ratio of 0.33 can be achieved when NXP was 60 mmdownstream. Zhu et al. [75] reported that the optimum NXPwas not only proportional to the mixing section throat diameter,but also increased as the primary flow pressure rises. The authorsalso pointed out that the ejector performance was very sensitiveto the converging angle y of the mixing section. When NXP waswithin its optimum range, the optimum y was in the range of1.45–4.21. A relative larger y was required to maximize ejectorperformance when the primary flow pressure raised. In contrast, CFD analyses of Rusly et al. [32] and Pianthonget al. [5] showed that NXP only had a small influence onentrainment ratio. In the first case, a 20% variation in NXP wasconsidered in an ejector using R141b as working fluid. Comparedto the base model, moving the nozzle towards the constant areasection caused l to decrease, while moving it in the otherdirection l remained practically unchanged. The authors claimedthat the optimum NXP of 1.5 diameters of the constant areasection produced better performance. Pianthong et al. [5] varied NXP in the range from 15 to 10 mm from the mixing chamberinlet. The entrainment ratio increased slightly as NXP was movedfurther from the inlet section.Optimum primary nozzle position or converging angle cannot bepredefined to meet all operating conditions. When the operatingconditions are different from the design point, the NXP should beadjusted accordingly to maximize the ejector performance. An ejectorwith movable primary nozzle can provide a flexible NXP when theconditions are out of the design point. This was first presented byAphornratana and Eames [76].

Recently Yapici et al. [13] carried outan experimental investigation on an ejector refrigerator with movableprimary nozzle. The author concluded that the optimum primary nozzle exit should be 5 mm from the mixing chamber inlet.Due to the varying nature of the operation conditions as wellas the different ejector geometries, no general agreement can beachieved among various researches.

3.3 Primary Nozzle Diameter

The relationship between primary nozzle diameter and the boiler temperature was reported by Cizungu et al. [45]. Using ammonia as working fluid, the author stated that the optimum primary nozzle diameter decreased with increase in the boiler temperature. Similar results were obtained by Sun [16] with an ejector driven by the working fluid R123. Chaiwongsa et al. [10] analyzed the effects of various nozzle outlet diameters D_{nt} (D_{nt} 2 mm, 2.5 mm and 3 mm) of a motive nozzle on the system performance. The Nozzle with outlet diameter of 2 mm was found to yield the highest COP.

3.4 Constant Area Section Length and Diffuser Geometry

Constant area section length is commonly believed to have no influence on the entrainment ratio [26, 40]. However, Pianthong et al. [5] reported that the critical back pressure increased with Lm and thus allowed to operate the ejector in double chocking mode in a wider range of operating conditions. As seen from Fig. 1, a thermodynamic shock wave can cause a sudden fall in Mach number as the flow changes from supersonic (Ma>1) to subsonic (Ma<1). This process results in a fall in total pressure and this effect reduces the maximum pressure lift ratio, which a conventional ejector refrigerator can achieve. With the aim to overcome this shortfall, Eames et al. [77] developed the constant rate of momentum change (CRMC) method to produce a diffuser geometry that removes the thermodynamics shock process within the diffuser at the design-point operating conditions. Theoretical results described in this paper indicated significant improvements in both entrainment ratio and pressure lift ratio, above those achievable from ejector designed using conventional methods. Experimental data were presented by Worall et al. [78] that supported the theoretical findings.

4. DEVELOPMENTS IN EJECTOR PERFORMANCE IMPROVEMENT

The ability of making use of renewable energy and theadvantages of simplicity in construction, installation and maintenance make ERS more cost-effectively competitive comparedwith other refrigeration system. The system performance for ERS,however, is relatively low. Hence, the engineers and researchersare making efforts to improve system efficiency for ERS. Pastdecade has seen many research innovations of enhancing systemperformance, including reduction of the mechanical pump workin ERS, utilization of the special refrigerants, and utilization andstorage of available renewable energy. Many research groupshave widely carried out theoretical calculations, computer simulations and experimental works in these areas.

4.1 EJECTOR REFRIGERATION SYSTEM WITHOUT PUMP

The pump, with the function to convey liquid condensate inthe condenser back to the generator, is the only moving part intheERS. This equipment, however, not only requires additionalmechanical energy, but also needs more maintenance than otherparts. Hence, many researchers have tried to utilize other methods to eliminate those shortcomings.

4.1.1 Gravitational Ejector

Kasperski et al. [79] presented a gravitational ERS (as shown inFig. 1) in a simulation model. Unlike the pump version ejectorrefrigerator, the heat exchangers are placed at different levels.Thus, with the help of the refrigerant hydrostatic pressure, thevertical arrangement of the heat exchangers enables pressuredifferences between the exchangers to be equalized. The lowestpressure in the refrigerator installation was obtained in theevaporator. It caused the inflow of liquid to the highest installation level. The highest pressure was obtained in a steam generator, which forces the lowest liquid level.The limitation of this system lies in its requirement of greatheight differences and the length of pipe work, which increasesfriction and heat losses.

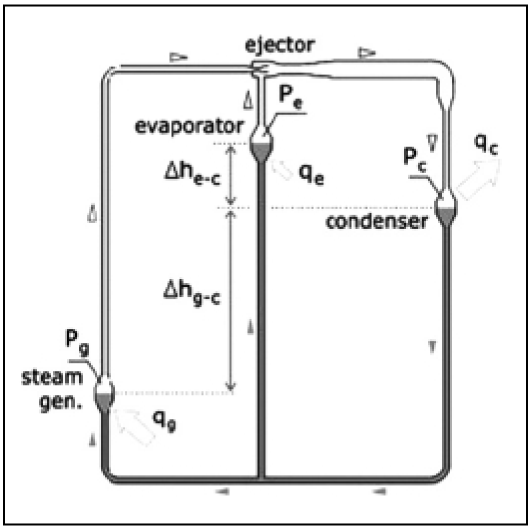


Fig. 1 Schematic Diagram of a Gravitational Ejector Refrigerator [79]

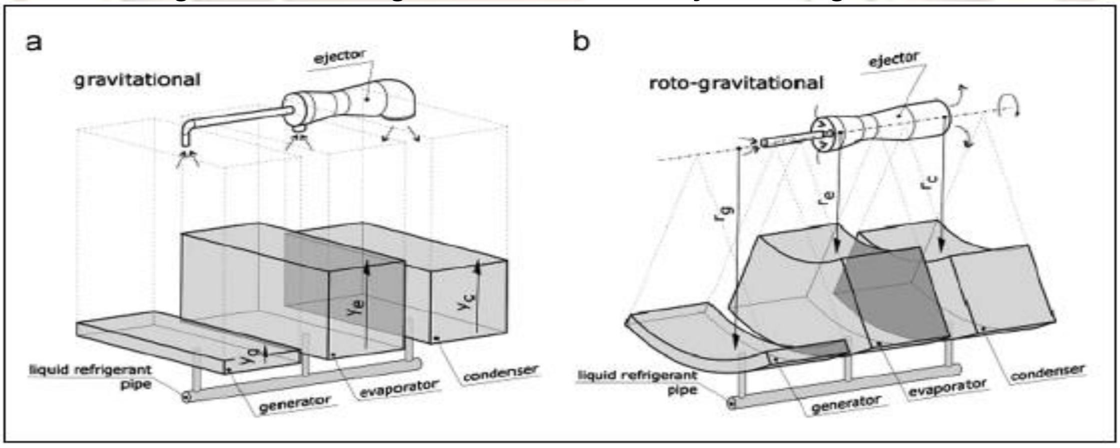


Fig. 2 Schematic Diagram of Liquid Refrigerant Levels in Gravitational (a) and Roto-Gravitational (b) Ejector Refrigerators [80]

Therefore, the conception of the gravitational refrigerator (Fig. 2a) was later developed into a rotatingrefrigerator (Fig. 2b) by Kasperski et al.[80]. With lager accelerations of rotary motion, this roto-gravitational refrigerator significantly decreased the size of the gravitational refrigerator.However, the author only proposed a mathematical model, no experimental results were presented.

4.1.2 Bi-Ejector Refrigeration System

A schematic view of a solar-powered bi-ERS designed by Shenet al. [81] is shown in Fig. 3. In this system, an ejector (injector)replaces the mechanical pump to promote pressure of the liquidcondensate and conveys the condensate back to the generator.Ideally, the system will lead to zero electricity consumption.The authors studied the performance of this system withdifferent refrigerants using numerical modelling. The resultshowed that the overall COP of the system was mainly affectedby the gas–gas ejector entrainment ratio in the refrigeration loop.Compared with other refrigerants, under the same operatingconditions, the gas–liquid ejector (injector) entrainment ratio ofR718 was relatively high.

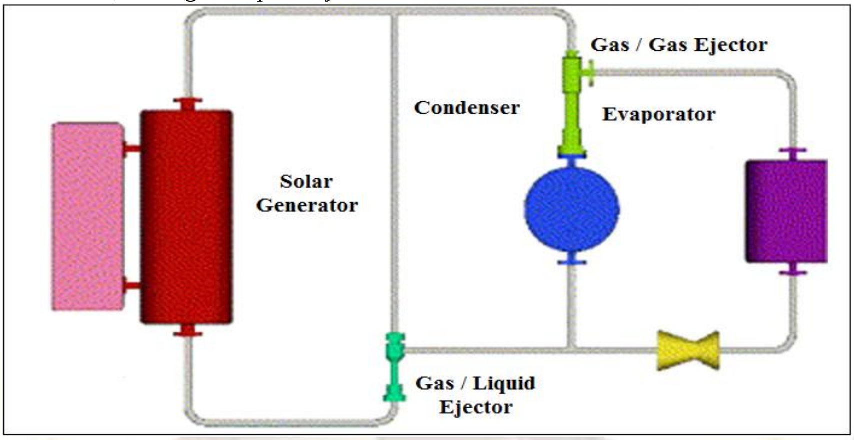


Fig. 3 Schematic Diagram of Solar-Powered Bi-ERS [81]

However, the best overall system COPachieved was 0.26 using R717 as the refrigerant.In the proceeding work, Wang and Shen [82] took considerations of the effect of injector structures on the system performance. The real fluid's thermal properties were considered in thenew injector thermodynamic model. The authors concluded thatwith increasing generator temperature, the entrainment ratio ofthe injector and the thermal efficiency of the solar collector werereduced, whilst the entrainment ratio of the ejector and COP ofbi-ER sub-system were improved. The overall COP of the systemreached an optimum value of 0.132.

4.1.3 Ejector Refrigeration System with Thermal Pumping Effect

An ERS that utilizes a multi-function generator (MFG) to eliminate the mechanical pump was presented by Huang and Wang[83,84]. The MFG serves as both a pump and a vapour generator. Therewere two generators in the ECS/MFG. Each generator consisted ofa vapour generator and an evacuation chamber. The vapourgenerator was a heat exchanger like a conventional boiler forpressurizing and to generating vapour. The evacuation chamberwas composed of a cooling jacket and a liquid holding tank. Thecooling jacket provided a cooling effect to depressurize thegenerator in order to intake the liquid from condenser. Detailedsystem description can be referred to Huang and Wang [83, 84].This system makes use of the pressure change in the generatorto create backflow of liquid condensate. However, the system iscomposed of too many elements, which will lead to unavoidableconsumption of available thermal energy.

4.1.4 Heat Pipe and Ejector Cooling System

Integration of the heat pipe with an ejector will result in a compact and high performance system, which does not require additional pump work. This system can also utilize solar energy or hybrid sources and so reduces the demand for electricity and thus fossil fuel consumption.

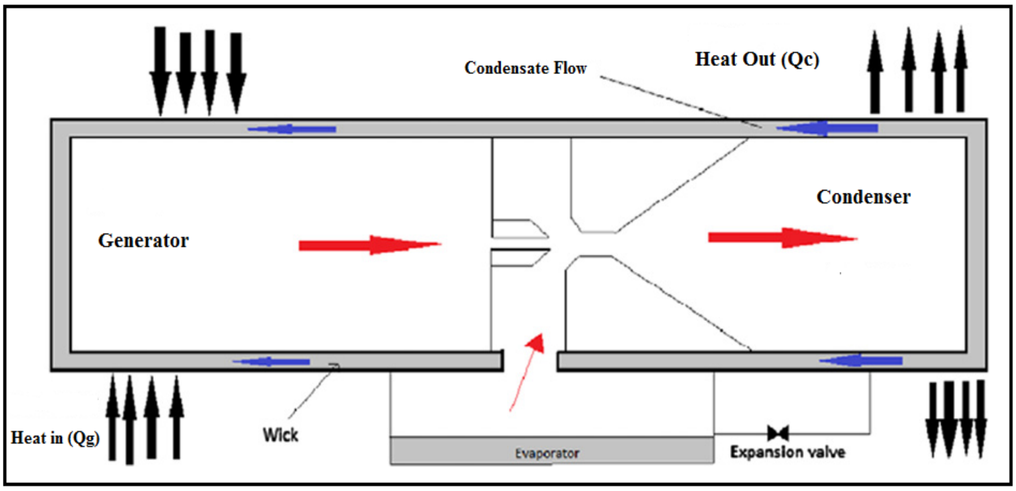


Fig. 4 Schematic Diagram of Heat Pipe / Ejector Refrigeration System [85]

The basic cycle of the heat pipe/ERS is shown in Fig. 4. The system consists of a heat pipe, ejector, evaporator and expansion valve. The low potential heat is added to the system in the generator section. Then the working fluid evaporates and flows through the primary nozzle of the ejector. Therefore it expands and contributes to the decrease of the pressure in the evaporator. Thus, the refrigeration cycle can be completed. In the condenser, some of the working fluid was returned to the generator by the wick action, while the remainder was expanded through the expansion valve to the evaporator. Unlike other vapour compression refrigeration system, which is powered by mains electricity generated by large plants, the heat pipe/ERS does not require any electricity input. With the aim of finding the optimum operating conditions for a heat pipe/ERS, Ziapour et al. [85] carried out an energy and exergy analysis based on the first and second laws of thermo- dynamics. The simulation results were compared with available experimental data from literature for steam ejector refrigerator.

4.2.EJECTOR REFRIGERATION SYSTEM WITH MULTI-COMPONENTS

Although the single stage ERS is simple, it is difficult to keep the system running at optimum conditions due to the variation of working conditions. Ambient temperatures above design conditions or lower generator temperature often lead to operational difficulties. Attempts have been made to solve this problem by using multi-components ejectors.

4.2.1Ejector Refrigeration System with an Additional Jet Pump

Yu et al. [25] proposed an ERS with an additional liquid–vapour jet pump, as shown in Fig. 5. This additional jet pump is applied to entrain the mixing vapour from the ejector, which acts as secondary flow for jet pump. In this case, the backpressure of the ejector can be reduced by the jet-pump, and then the entrainment ratio and COP of the system could be increased. Simulation results showed that, compared with conventional ERS at same working conditions, the COP of NERS was increased by 57.1% and 45.9%, with R152a and R134a, respectively as refrigerants. The group [86] later presented another system with similar configuration (as shown in Fig. 6). In this system, the auxiliary jet pump was designed to accomplish the effects of both entrainment and regeneration. Different from conventional ERS, the exhaust of the ejector in this system was divided into two parts. One part was discharged at the normal condenser pressure, another part was discharged at a higher pressure than the condenser pressure, and thus this part with higher temperature was rejected as heat for regeneration. Compared with the conventional system, the simulation results showed that the COP of this system increased from 9.3% to 12.1% when generator temperature was in arrange of 80–160 1C, the condensing temperature was in arrange of 35–45 1C and the evaporating temperature was fixed at10 1C.

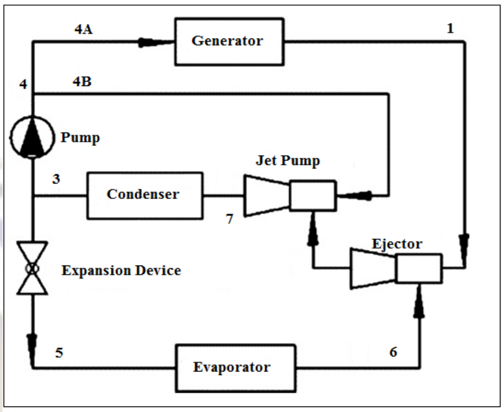


Fig. 5 Schematic diagram of an Ejector Refrigeration System with Additional Jet Pump [25]

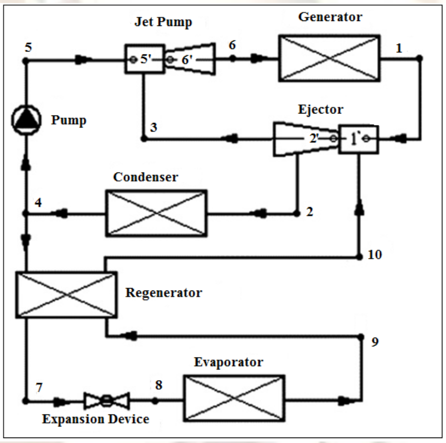


Fig.6. Schematic Diagram of a Refrigeration System with Additional Jet Pump [86]

4.2.2 Multi-Stage Ejector Refrigeration System

An example of the multi-stage ejector refrigeration system arrangement is shown in Fig. 7. Several ejectors are placed in parallel before the condenser. One ejector operates at a time and the operation of each ejector is determined by the condenser pressure. Ejector 1 operates when the condenser pressure is below Pc1ejector2 operates at a condenser pressure between Pc1 and Pc2; and ejector 3 operates at a condenser pressure between Pc2 and Pc3. This arrangement was proposed by Sokolov and Hershgal [87]. However practical work was not available.

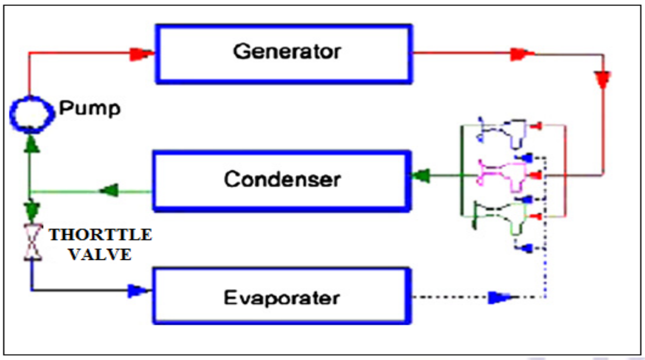


Fig. 7 Schematic Diagram of Multi-Stage ERS [87]

4.2.3 Multi-Evaporator Compression System

Multi-evaporator compression systems (MECS) are generally used in transport refrigeration applications. Kairouani et al. [27] studied a multi-evaporator refrigeration system utilizing ejector for vapour pre-compression. As shown in Fig. 8, the ejectors are positioned at the outlets of evaporators, which can increase the suction pressure. In the diffuser, the kinetic energy of the mixture is converted into pressure energy. The specific work of the compressor is reduced and then the COP of the system is improved. A comparison of the system performances with environment friendly refrigerants (R290, R600a, R717, R134a, R152a, and R141b) is made. R141b proved to give the most advantageous COP among all working fluids.

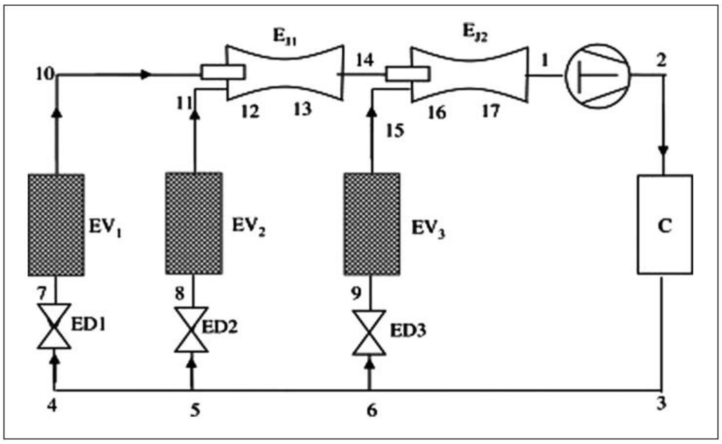


Fig. 8 Schematic Diagram of a Multi-Evaporators Compression System [27]

Liu et al. [88] presented three different configurations for two- evaporator refrigeration cycle. As shown in Fig. 13 the working principles of series (Fig. 9a) and parallel (Fig. 9b) systems can be easily recognized from the schematic views. The combined circulatory cross-regenerative thermal system (Fig. 9c) is an improvement of the hybrid circulatory system, where the evaporators of the cooling chamber and freezing chamber are in parallel.

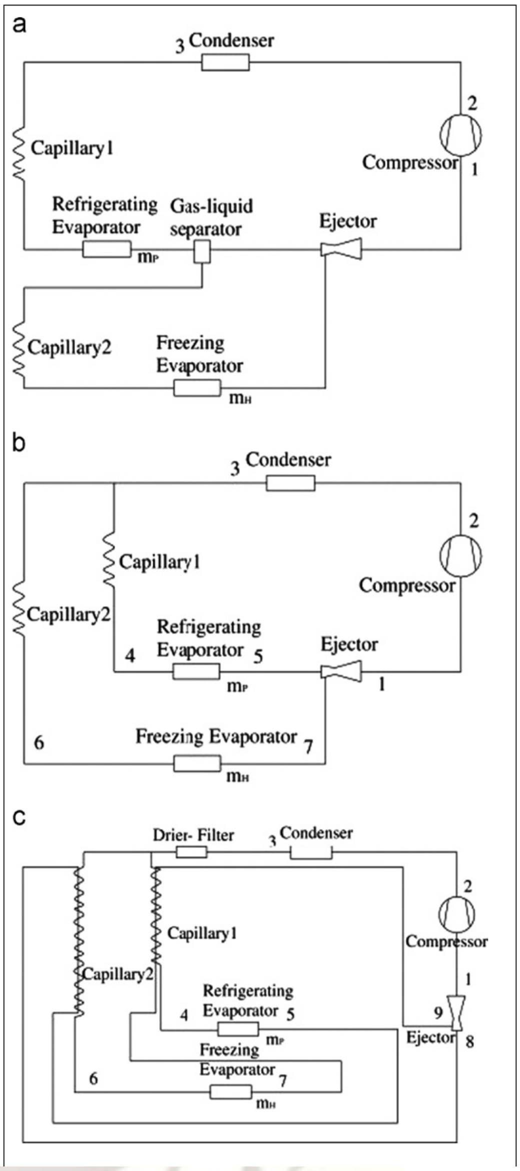


Fig. 9Schematic Diagram of (a) Two-Evaporator Refrigeration Cycle in Series Hybrid System, (b) Two-Evaporator Refrigeration Cycle in Parallel Hybrid System, (c) Two-Evaporator Refrigeration Cycle in Parallel and Crossed-Regenerative Hybrid Systems [88]

The pressure at the connector of the three ejectors and the power consumption were measured, and the performances of the three different connection forms for compressor-jet mixing of the refrigeration cycle were compared. Results showed that for the compression–injection crossed-regenerative hybrid refrigeration cycle system, loss of heat in the throttle processing was decreased effectively by an ejector. Energy consumptions of the first two prototypes were 0.775kWh/day and 0.748 kWh/day, which were higher than the traditional prototype. The power consumption can be reduced to 0.655kWh/day for the third one, which was 7.75% lower than the traditional prototype.Autocascade refrigeration system can use only one compressor to obtain lower refrigerating temperature between -40°c to -180°c. In order to reduce the throttling loss generated by throttling devices, an ejector is introduced to the system to recover the kinetic energy in the expansion process. Yu et al. [28] applied an ejector in autocascade refrigerator with refrigerant mixture of R23/R134a. As shown in Fig. 10, the ejector is set between the evaporative, condenser and the evaporator. Thermo- dynamic analysis showed that the system employed with an ejector had merits in decreasing the pressure ratio of the compressor as well as increasing COP. With condenser outlet temperature of 40°c, evaporator inlet temperature of -40.3°c and the mass fraction of R23 were0.15; the COP was improved by 19.1% over the conventional autocascade refrigeration cycle.

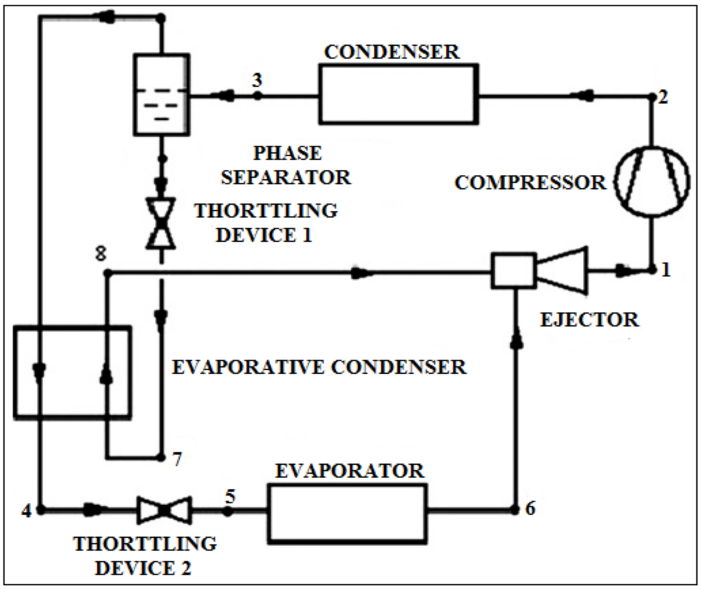
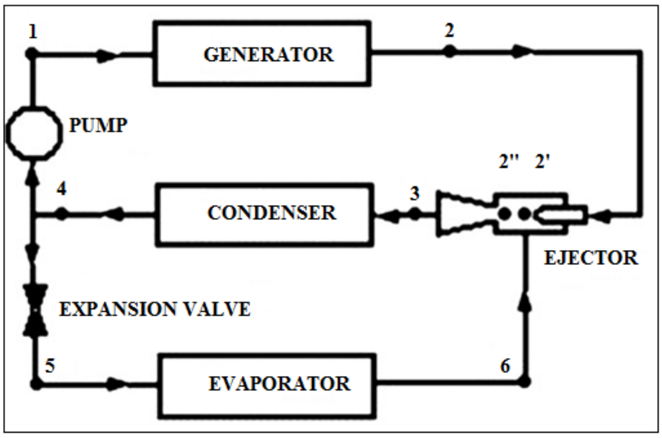


Fig. 10 Schematic Diagram of Autocascade Refrigeration Cycle with an Ejector [28]

4.3 Transcritical Ejector Refrigeration Systems

Different from other ERS whose refrigerants are working in their subcritical cycle, the refrigerant of transcritical ERS (TERS) operates in transcritical process. Characteristic for the process is heat rejection in the supercritical region, introducing a gliding temperature instead of condensation at constant temperature. Compared with an ERS, the TERS has a higher potential in making use of the low-grade thermal energy with gradient temperature due to a better matching to the temperature glide of the refrigerant. Yu et al. [89] carried out a theoretical study of a transcritical ERS (TERS) with R134a as working fluid. The schematic diagram is shown in Fig. 11. The study calculation model for the ejector is the constant-pressure mixing model. The generating temperatureranged from 60 to 100°C with a pressure range of 6–10 MPa.The numerical results indicated that COP of TERS were between0.35 and 0.75, almost double that of conventional ERS, withgenerator temperature at 80°C, evaporator temperature in therange of 10–15°C and the condensing temperature in the range of30–40°C. The authors conclude that the higher working pressurein the TERS resulted in a more compact system. However, no experimental verification is available.

Fig. 11 Schematic Diagram of the Transcritical Ejector Refrigeration System [89]



Similarly, a number of studies [90–93] have concentrated on TERSwith CO2 as refrigerant. Li and Groll [90] investigated theoretically theperformance of transcritical CO2 refrigeration cycle with ejector-expansion device (as shown in Fig. 12). This system incorporated avapour backflow valve to relax the constraints between the entrainment ratio of the ejector and the quality of the ejector outlet stream.The effect of different operating conditions on the relative performance of the ejector expansion transcritical CO2 cycle was alsoinvestigated using assumed values for the entrainment ratio andpressure drop in the receiving section of the ejector. The resultsdemonstrated that the ejector expansion cycle improved the COP bymore than 16% compared to the basic cycle for typical air conditioning applications.

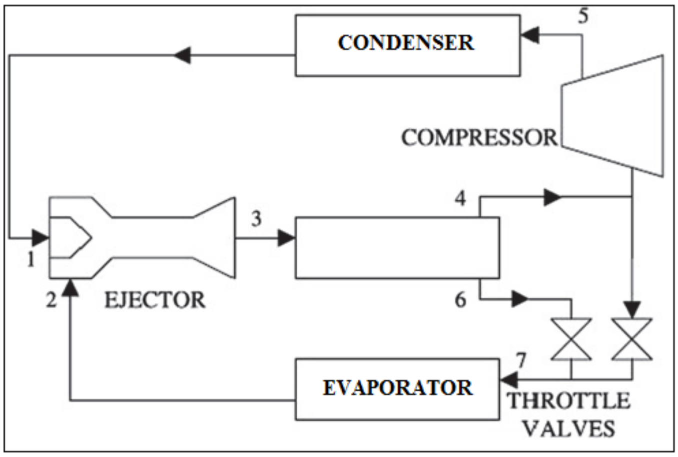


Fig. 12 Schematic Diagram of the New Ejector Expansion Refrigeration Cycle [90]

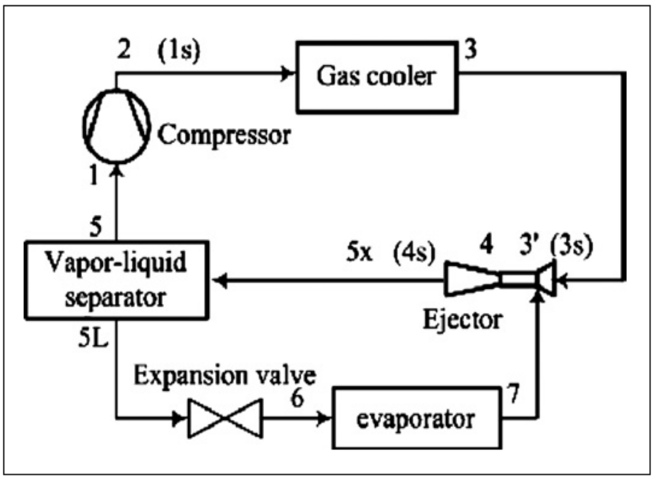


Fig. 13 Schematic Diagram of the Ejector Expansion System [91]

Deng et al. [91] presented a theoretical analysis of a transcritical CO2 ejector expansion refrigeration cycle (shown in Fig. 13),which uses an ejector as the main expansion device instead of anexpansion valve. The results indicated that the ejector entrainment ratio significantly influence the refrigeration effect with anoptimum ratio giving the ideal system performance. It was foundthat for the working conditions described in their paper, theejector improved the maximum COP to 18.6% compared tothe internal heat exchanger system and 22% compared to theconventional system.Yari and Sirousazar [94] investigated a new transcritical CO_2refrigeration cycle (TRCC) with an ejector, internal heat exchangerand intercooler (shown in Fig. 14). This cycle utilized the internalheat exchanger and intercooler to enhance its performancesignificantly. It was found that, the new ejector expansion TRCCimproved the maximum COP and second law efficiency up to 26%compared to conventional ejector-expansion TRCC.

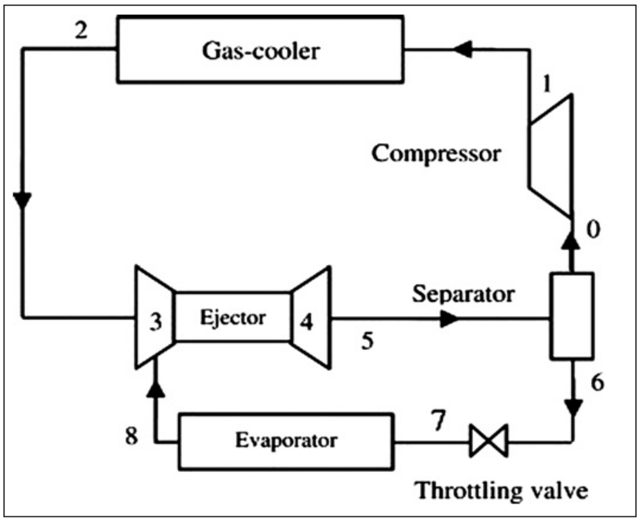


Fig. 14 Schematic Diagram of Transcritical Carbon Dioxide Cycle with Ejector [94]

4.4 SOLAR-DRIVEN EJECTOR REFRIGERATION SYSTEM

Because of the ability of harnessing solar energy, the solar-driven ERS is less energy demanding and more environmentalfriendly in comparison with conventional vapour compressionrefrigeration system. However, due to the intermittent feature ofsolar energy, the unstable heat gains from solar sources inherently affect the operation of solar-driven ERS. Thus, thermalstorage system integrated with solar-driven ERS is becoming ahot research topic.

4.4.1 Conventional Solar-Driven Ejector Refrigeration System

Conventional solar-driven ERS has been widely studied duringpast decade. Heat from the solar collector is carried by theintermediate medium and transferred to the refrigerant by theheat exchanger. The heat transfer mediums should have theboiling point higher than the possible temperature in the system,low viscosity and good heat transfer properties. Water with acorrosion inhibitor additive and transforming oil are recommended for operating temperature below and above 100°C,respectively. However, since water will freeze below 0°C, Vargaet al. [43] found that the system working at very low evaporatortemperature was not suitable for using water as refrigerant.Theoretical analysis of a solar-driven ERS in the Mediterraneanwas carried out by Varga et al. [43]. Based on a simplified 1-Dmodel, the authors studied both the refrigeration and solarcollector cycles for a 5 kW cooling capacity. The results indicatedthat, in order to achieve acceptable COP, generator temperatureshould not fall below 90°C and solar collector output temperature of about 100°C would be required. For higher condensertemperatures (>35°C) and lower evaporator temperature(<10°C), the solar collector area required for 5 kW cooling loadwas larger than 50 m2.R134a was proposed as a refrigerant for a solar-driven ejectorsystem by Alexis and Karayiannis et al. [22]. It was found that COPof ejector cooling system varied from 0.035 to 0.199 for generatortemperatures ranging from 82 to 92°C, condenser temperaturesranging from 32 to 40°C and evaporator temperatures rangingfrom-10 to 0°C.Ersoy et al. [17] conducted a numerical investigation on theperformance of a solar-driven ejector cooling system using R114under Turkish climatic conditions. When generator, condenser,and evaporator temperatures were taken at 85°C, 30°C and 12°C,respectively, the maximum overall COP and the cooling capacityobtained were as 0.197 and 178.26 W/m2.

4.2.2 Solar-Driven Ejector Refrigeration System with ThermalStorage System

During some adverse weather conditions, the cooling capacityprovided by available solar energy cannot be essentially matchedwith the cooling demand. Taken this into consideration, energystorage technology was applied in solar-driven ERS. Two kinds ofthermal storage are considered: hot storage-high temperatureenergy from the solar collectors, and cold storage-low temperature energy from the evaporator.Guo and Shen [95] numerically investigated a solar-driven airconditioning system with hot storage for office buildings. Withgenerator temperature of 85°C, evaporator temperature of 8°Cand condenser temperature varying with ambient temperature,the average COP and the average solar fraction of the system were0.48 and 0.82, respectively. It was concluded that the systemcould save approximately 75% of the electricity used for conventional air conditioning under Shanghai's climatic conditions.In contrast, Pridasawas and Lundqvist [39] reported that thesize of the hot storage tank did not improve significantly theperformance of the system. Hence, cold storage, with the help ofPhase changing materials, cold water or ice storage, was recommended by Bejan et al. [96]. Moreover, using computer simulations, Diaconu et al. [97] analyzed a solar-assisted ejector coolingsystem with cold storage (as shown in Fig. 15) over one year inAlgeria. Compared to that without cold storage, the annual energyremoval of the system with cold storage achieved higher values.

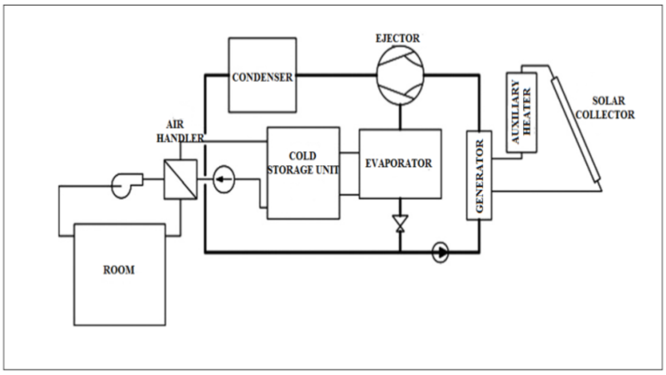


Fig. 15 Schematic Diagram of Solar-Assisted Ejector Refrigerator with Cold Storage [97]

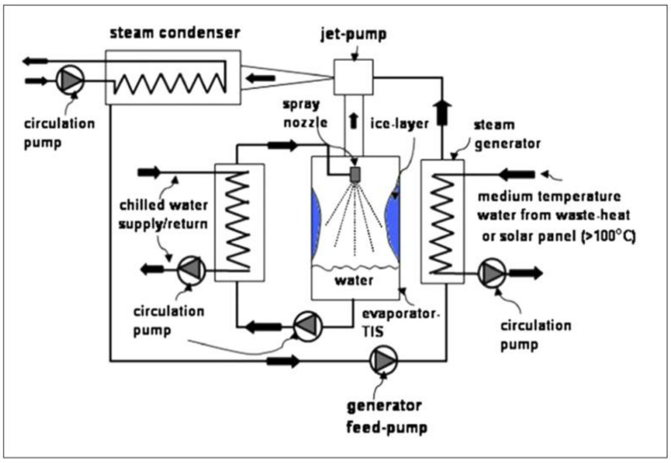


Fig. 16 Schematic Diagram of Ejector Refrigeration System with Thermal Ice Storage [100]

In order to provide better compliance with varying ambientconditions, a variable geometry ejector with cold storage wasinvestigated by Dennis et al. [98]. The annual cooling simulation
results concluded that a variable geometry ejector was able toincrease yield by 8–13% compared to a fixed geometry ejector.The modelling further showed that the solar collector area may bedecreased if a cold storage was used.Worall et al. [99] and Eames et al. [100]carried an experimental investigation of a novel ejector refrigeration cycle withthermal ice storage system (as shown in Fig. 16). Ice was formedin the evaporator vessel under normal operation and acted as acoolth storage medium. The low evaporator temperature resultedin a relatively low COP of 0.162 during experiments. The authorsargued that such system powered by solar energy would help tostore the coolth to level out the off-peak conditions.

5. APPLICATION OF EJECTOR REFRIGERATION SYSTEM COMBINED WITH OTHER SYSTEMS

5.1 Combined Ejector-Absorption Refrigeration System

Absorption system can also make use of low-grade heat sources, such as solar energy, waste or exhaust heat. However, because of its complex configuration and low COP, it is less competitive than the conventional vapour compression system. Applying ejector to the conventional absorption systems is one of the remarkable alternatives. The appropriate installation configuration can help to improve the system performance almost similar to multi-effect absorption cycle machine. Moreover, due to the simplicity of the combined ejector-absorption refrigeration machine, its capital investment cost is comparatively low com- pared to other conventional high performance absorption cycle systems.

Recently, Sozen et al. [46] proposed a solar-driven ejector- absorption system (shown in Fig. 17) operated with aqua- ammonia under the climatic condition of Turkey. Ejector was located at the absorber inlet, which helped the pressure recovery from the evaporator. According to results obtained in this study, using the ejector, the COP was improved by about 20%. For 8–9 months (March–October) of the year, the collector surface area of $4m^2$ was sufficient for different applications of refrigeration all over Turkey.

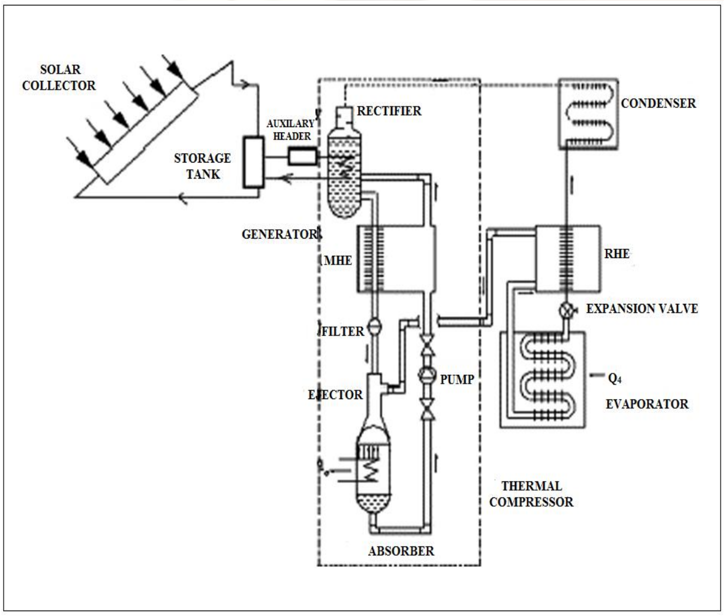


Fig. 17 Schematic Diagram of a Novel Ejector-Absorption Combined Refrigeration Cycle [46]

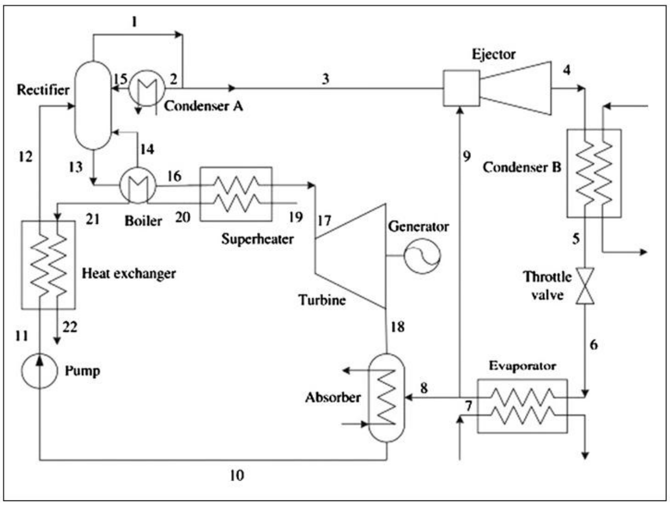


Fig. 18 Schematic Diagram of the Combined Power and Ejector-Absorption Refrigeration Cycle [47]

Wang et al. [47] presented a combined power and ejector- absorption refrigeration cycle with aqua-ammonia as working fluids. This system (shown in Fig. 18) combined the Rankine cycle with ejector-absorption refrigeration cycle, and could produce both power output and refrigeration output simultaneously. This combined cycle introduces an ejector between the rectifier and the condenser, and provides a performance improvement without greatly increasing the complexity of the system. The comparisons of the parametric results between a similar combined system without ejector [101] and this system showed that refrigeration output increased from 149 kW to 250 kW at evaporator temperature of -8°C and generator temperature of 87°C.In order to make sufficient use of high-grade heat with a simple structure refrigeration system, Hong et al. [102] proposed a novel ejector-absorption combined refrigeration cycle (shown in Fig. 19).

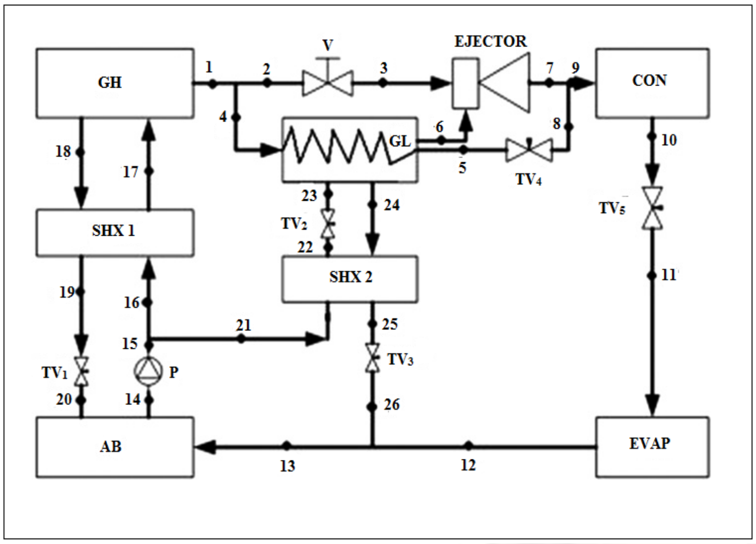


Fig. 19 Schematic Diagram of a Novel Ejector-Absorption Combined Refrigeration Cycle [102] (Ab, Absorber; Con, Condenser; Evap, Evaporator; Gh, High-Pressure Generator; Gl, Low-Pressure Generator; P, Pump; Shx, Heat Exchanger; V, Valve).

When the temperature of the heat source is high enough, the cycle would work as a double-effect cycle. Two generators were used in the cycle, so that the pressure of the high-pressure generator and that of the low-pressure generator could be optimized to get maximum COP at any given working condition. The simulation results showed that system COP was 30% higher than that of the conventional single-effect absorption refrigeration cycle.

However, no experimental validation was available. Theoretical and experimental study of solar-ejector absorption refrigeration system (shown in Fig. 20) was conducted by Abdulateef et al. [103]. The effects of the operating conditions on the COP and the cooling capacity of the system were investigated. A mathematical model was developed for design and performance evaluation of the ERS. A wide range of compression, expansion and entrainment ratios, especially those used in industrial applications were covered in the mathematical model. With the aim of overcoming the intermittency of adsorption refrigeration, Li et al. [104] presented a novel combined cycle solar-powered adsorption–ejection refrigeration system using Zeolite 13X-water as the pair. The cycle consisted of two sub- systems (as shown in Fig. 21): ejector sub-system to provide refrigeration during the day and an adsorption sub-system which refrigerates at night-time. Detailed system description can be found in [104].

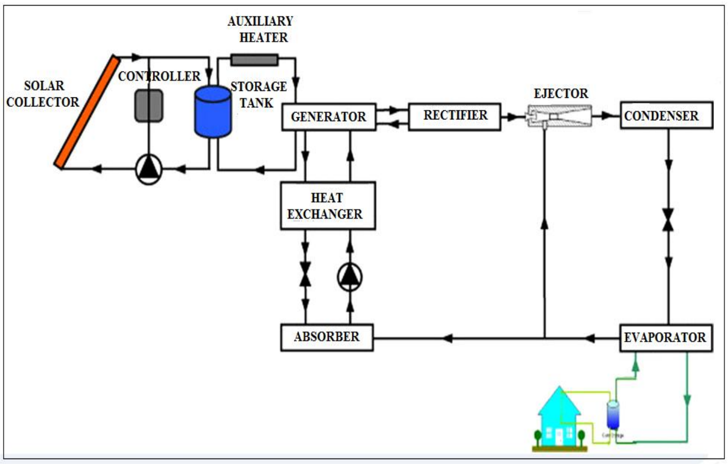


Fig. 20 Schematic Diagram of a Novel Ejector-Absorption Combined Refrigeration Cycle [46]

It was demonstrated that the COP of the ejection sub-system improved when the temperature of the adsorbent increased or when the pressure decreased. A COP of 0.4 was achieved with 9°C evaporating temperature, 40°C condensing temperature, 120°C regenerating temperature and 200°C desorbing temperature. It was further concluded that by increasing the temperature or reducing the pressure within the adsorbent bed, the COPs of the ejection sub-system could be improved slightly.

5.2 Ejector Refrigeration System with Compressor or Vapour Compression System

Since the system performance of ERS is determined by ejector entrainment ratio $\lambda$and operating conditions, one way to enhance the performance is to increase the secondary flow pressure. In1990, Sokolov and Hearshgal [105], introduced new configurations of efficient uses of the mechanical power in order to enhance the secondary pressure without disturbing the refrigeration temperature, which are: (1) the booster assisted ejector cycle and (2) the hybrid vapour compression-jet cycle. Their simulated results showed that the compression enhanced ejector could significantly improve system performance.

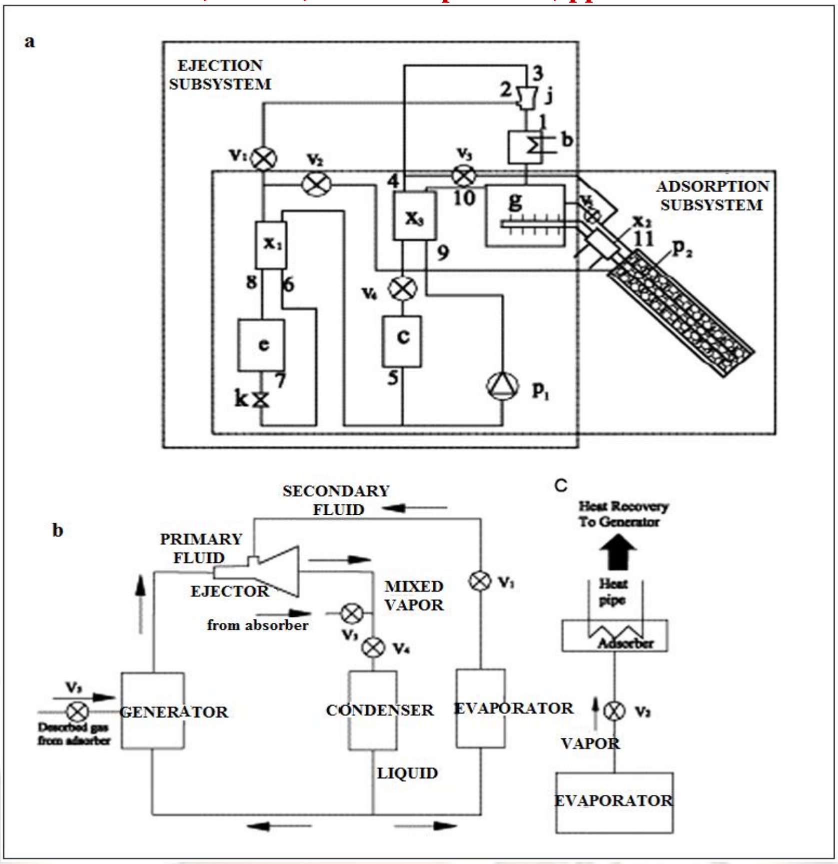


Fig. 21 Schematic Diagram of a Solar-Powered Adsorption–Ejection Refrigeration System [104]
(a) System Layout, (b) Ejector Refrigeration System During Daytime and (c) Adsorption Refrigeration System At Night (A, Absorber; B, Auxiliary Heater; C, Condenser; E, Evaporator; G, Generator; J, Ejector; K, Expansion Valve; P1, Pump; P2, Heat Pipe; X, Heat Exchangers; V, Valves)

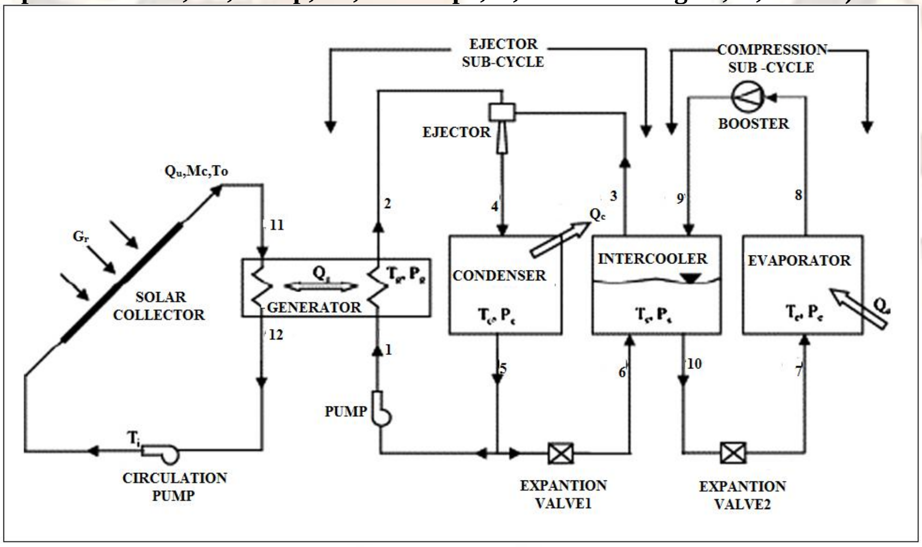


Fig.22 Schematic Diagram of a Solar-Powered ERS [106]

Recently, Sokolov et al. [106] improved their system by using a booster and intercooler in a solar-powered ERS (as shown in Fig. 22). The system consisted of a conventional compression andejector sub-cycles with an intercooler as an interface betweenthem. The intercooler is a heat and mass exchanger throughwhich the two sub-cycles interact. Heat absorbed in the evaporator is boosted up to the intercooler pressure and temperature bythe compression cycle. The elevated suction pressure from theintercooler to the ejector results in a higher mass flow rate withwhich the ejector operates. The ejector sub-cycle further raisesthe heat from the evaporator to the condenser's pressure andtemperature. The system operated at 4°C evaporator temperatureand 50°C condenser temperature, with cooling capacity of3.5 kW. The overall system COP could reach up to 0.5. The grouplater revised their work [107] by substituting the refrigerant ofR114 with R142b. The results indicated that R142b providedhigher efficiency than the one operating with R114.

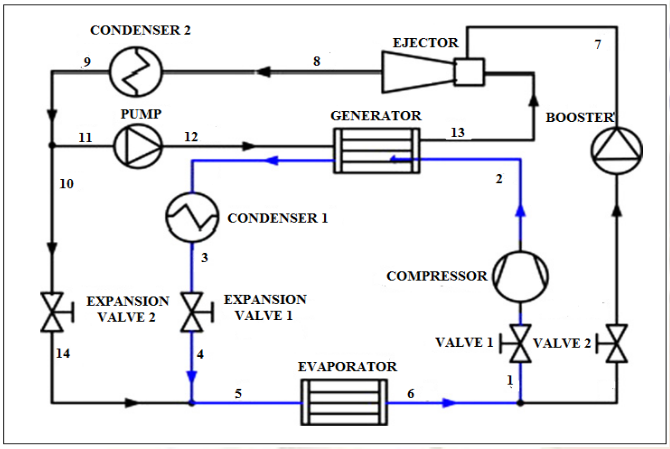


Fig. 23 Schematic Diagram of a Refrigeration System with the Integrated Ejector [38]

A similar system configurations was presented by Hernandezet al. [23] with R134a and R142b as working fluids. The theoretical analysis demonstrated that the optimum COP of 0.48 couldbe achieved at condenser temperature of 30°C and generatortemperature of 85°C, with R134a as working fluid. The authorsfurther indicated that when higher condenser temperature wasimposed, the system with R142b would perform better.Vidal et al. [24] implemented a computer simulation on a solarassisted combined ejector-compression system. The mechanicalcompression cycle and the thermal driven ejector cycle wereperformed with two different refrigerants, R134a and R141brespectively. The final optimum results showed that an intercooler temperature of 19°C resulting in a solar fraction of thesystem of 82% and a COP of the combined ejector cycle of 0.89.A 10.5 kW cooling capacity was achieved with the flat platecollector area of 105 m2. Zhu et al. [38] proposed a hybrid vapour compression refrigeration system which combined with an ejector cooling cycle (as shown in Fig. 23). The ejector cooling cycle was driven bywaste heat from the condenser in the vapour compressionrefrigeration cycle. The additional cooling capacity from theejector cycle in directly input to the evaporator of the vapourcompression refrigeration cycle. Simulation results showed thatCOP increased by 5.5% with R152a and 8.8% with R22 comparedwith the basic system. However, no experimental results wereavailable to validate these.

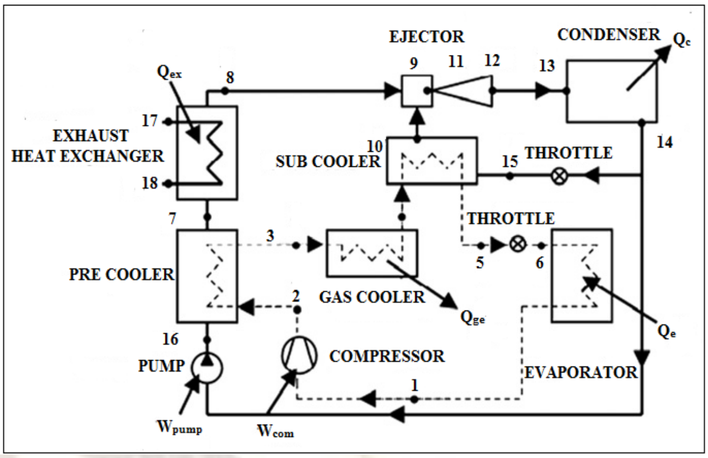


Fig. 24 Schematic Diagram of Hybrid CO2 Ejector and Vapour Compression System [108]

Worall et al. [93,108] proposed a similar hybrid CO2 ejectorand vapour compression system as shown in Fig. 24. The ejectorrefrigeration system was proposed to extract heat from theexhaust of an independent diesel engine and sub-cool the CO2vapour compression system. The modelling results showed that atan evaporator temperature of -15°C, an ambient temperature of35°C and a generator temperature of 120°C, COP could beincreased from 1.0 to 2.27 as sub-cooling increased from 0 to 20°C. At the same time, the compressor work could be reduced by 24% at 20°C sub-cooling. The group [109] later carried outpreliminary experimental investigations on the ejector cycle.

5.3 Combined Power and Ejector Refrigeration System

Recently, many combined power and refrigeration cycles havebeen proposed to make better use of low grade heat sources.Zhang and Lior [110] discussed the combined power and refrigeration cycles with both parallel and series-connected configurations. The cycle has large refrigeration capacity. However, itoperated at temperatures about 450 1C, which is incompatiblewith low temperature heat sources such as solar thermal andwaste heat.Wang et al. [111] proposed a combined power and refrigeration cycle as shown in Fig. 25, which combined the Rankine cyclewith the ERS by adding an extraction turbine between heatrecovery vapour generator (HRVG) and ejector. This combinedcycle could produce both power output and refrigeration outputsimultaneously. The HRVG is a device in which high pressure andtemperature vapour is generated by absorbing heat from sourcessuch as solar thermal, geothermal and waste heat.

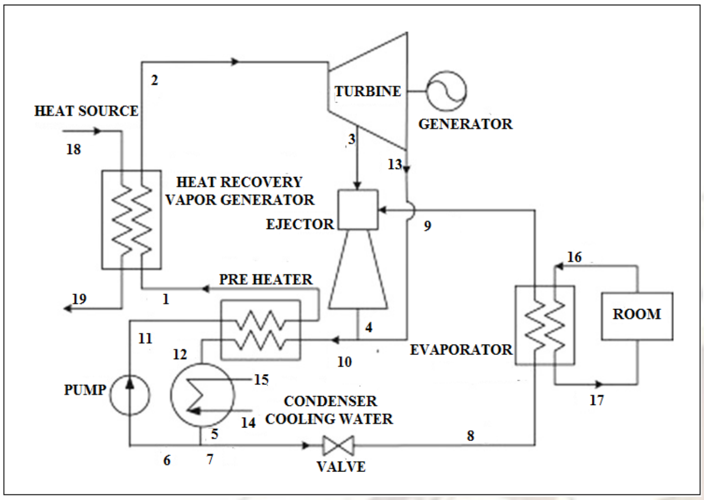


Fig. 25 Schematic Diagram of Combined Power and ERS [111]

The parametric analysis results concluded that the amounts of exergy destruction in the HRVG, ejector and turbine accounted for a large percentage. The author suggested several methods to improve system efficiency including increasing the area of heat transfer and the coefficient of heat transfer in the HRVG, optimization design parameters in the ejector and turbine. Similarly, Alexis et al. [112] studied a combined power and ejector cooling cycle (Fig. 26) in which extracted steam from the turbine in Rankine power cycle was used to heat the working fluid in an independent steam ejector refrigeration cycle.

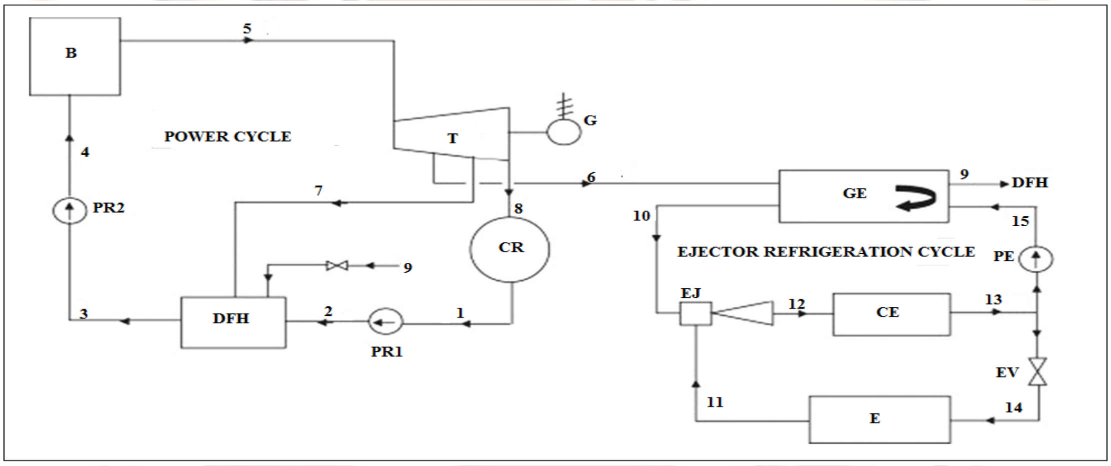


Fig. 26Schematic View of Combined Refrigeration and Electrical Power Cogeneration System [112] (B, Boiler; T, Turbine; G, Generator; Cr, Main Condenser; Pr1, Condensate Pump; Dfh, Deaerating Feed Water Heater, Pr2; Feed Water Pump; Ge, Heat Generator; Ej, Ejector; Ce, Condenser; E, Evaporator;

Pe, Pump; Ev, Expansion Valve).

Rankine cycle and steam ejector refrigeration cycle produced electrical powerand refrigeration capacity, respectively. Computer modellingresults showed that when the ratio between electrical powerand heat transfer rate was varied between 0.1 and 0.4, the ratiobetween electrical power and refrigeration capacity was variedbetween 0.23 and 0.92.A combined power and ERS with R245fa as working fluid waspresented by Zheng et al. [113]. Simulation results showed that athermal efficiency of 34.1%, an effective efficiency of 18.7% and anexergy efficiency of 56.8% could be obtained at a generatortemperature of 122 1C, a condensing temperature of 25 1C andan evaporating temperature of 7 1C. It was also noted that whilethe generator temperature increased the fluid inlet pressure of the ejector increased.

Exergy analysis of combined power and ejector refrigeration cycle presented by Wang [101] showed that the largest exergy destruction occurred in the heat recovery vapor generator (HRVG) followed by the ejector and turbine. In order to recover some of the thermal energy from the turbine exhaust, Khaliq [35] combined a Libr-H2O absorption system with power and ejector refrigeration system using R141b as refrigerant. The results of first and second law investigation showed that the proposed congeneration cycle yielded better thermal and exergy efficiencies than the cycle without absorption system.

However, no experimental results were available. Godefroy et al. [114] designed a small CHP-ejector trigeneration system which combined heat and power (CHP) to drive an ejector cooling cycle. In the system (shown in Fig. 27) consisted of a CHP unit and an ejector cooling cycle. The ejector cooling cycle was driven by heat from the CHP unit supplied through a flat-plate heat exchanger to bring the refrigerant to its vapour state. The design had been tested and validated by a model based on the real fluid properties. The results showed that this system offered an overall efficiency around 50% and would have an almost neutral effect on overall emissions.

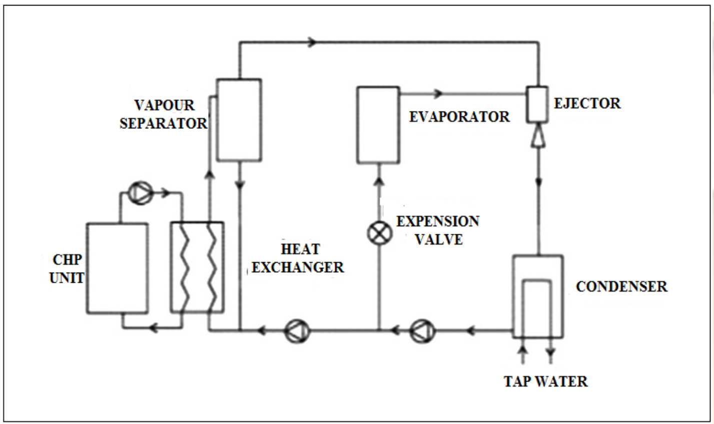


Fig. 27Schematic Diagram of a CHP-Ejector System [114]

5.4 Ground Coupled Steam Ejector Heat Pump

Ground coupled heat pump (GCHP) is being used for heatingand cooling residential and commercial buildings by exchangingheat with the ground as the thermal source or sink. However theinitial investment is higher than that for the air source heatpumps due to the costs of ground loop pipes, wells, channels andcirculation pumps. Ejector systems, with its advantage of longoperating lifetime, high reliability, low maintenance cost, are onealternative to reduce the initial cost of GCHP. Sanaye et al. [115] investigated a GCHP (shown in Fig. 28) which included two mainsections of closed vertical ground heat exchanger and steamejector heat pump. Thermal and economic simulation and optimization of the system, optimum design of ejector main cross section and investigation of the effects of weather, soil type, and system capacity on system performance were carried out in this research.

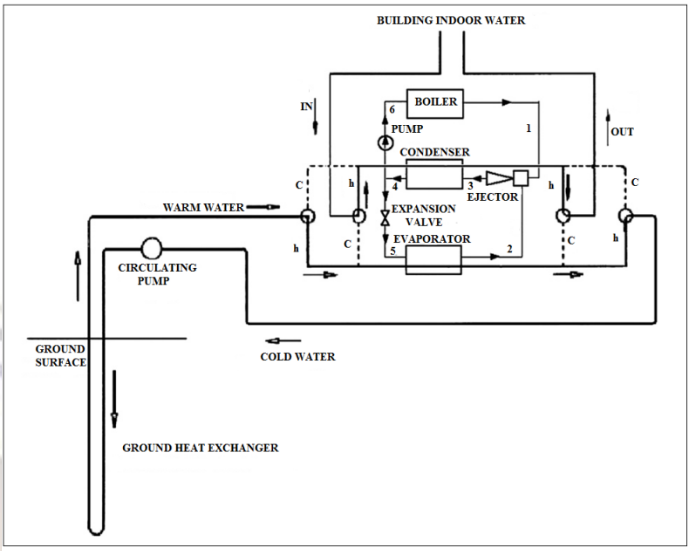


Fig. 28Schematic Diagram of Ground Coupled Steam Ejector Heat Pump System [115]

Simulation results were validated with experimentaldata from the literature. The authors concluded that the system had the smallest mean total annual cost value and maximum overall COP in temperature climates in comparison with cold andtropical climates.

Conclusions

Studies in ejector systems that have been carried out over the past decade involved system modeling, design fundamentals, refrigerants selection and system optimization. The research and development was broad based and productive, concentrating on performance enhancement methodology and feasibility of combining ERS with other systems. This paper presents not only a basic background and principles for ejector design, but also the recent improvement in ejector refrigeration technologies.

The following conclusions can be drawn from the reviewed works that have been carried out in ejector refrigeration system: (1) Attempts have been made on the investigations of proper mathematical models that may help to optimize design parameters. Taking into consideration of friction losses and irreversibilites, some researchers have carried out computer simulations on the improvement of constant-area model and constant-pressure model. A number of researchers have concentrated on the studies of two-phase flow and specific characteristics of working fluids. CFD has been identified as a suitable tool for the turbulence models of the mixing process which can better simulate and optimize the geometry of ejector. Although these simulated results were claimed to become more accurate than others, very few of them were experimentally verified and approved. (2) Different configurations of ejectors with various geometric features were proposed and tested numerically and experimentally. Area ratio and nozzle exit position were the most widely investigated parameters. It can be concluded that the optimal area ratio and NXP have varied for the different operating conditions. A spindle, which can adjust primary nozzle position, could be implemented to provide both flexible area ratio and NXP. (3) Since the ejector refrigeration systems suffer from relatively low COP, a number of studies have focused on system performance enhancement. Operation of ERS without a pump has been declared to considerably reduce the mechanical energy consumption. In contrast, ERS with an additional pump could help to increase the entrainment ratio and COP. In order to cope with variations of working conditions, multi-components ERS are parametrically studied. On the other hand, transcritical ERS is proposed, which provides higher potential in utilizing low-grade heat. The remarkable COP improvements from combined ejector and other types of refrigeration systems (vapor compression, absorption system, etc.) are reported by many research groups. However, most of those studied are limited to numerical analysis, with few experimental results available.

With the concept of energy conservation and environment protection, the utilization of low grade energy, especially solar energy with ERS has been widely studied during past decade. The major technical problem of solar-driven ejector refrigeration system is that the system is strongly reliant on ambient conditions, like the solar radiation, air temperature, cooling water temperature, wind speed and other transient factors. Thus the combination of energy storage in the solar-driven ERS remains to be the research topic in this field of technology.

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