CHAPTER-1

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Refrigeration is the process of removing heat from a body at low temperature. A refrigerant is used as a primary working fluid to produce the cooling effect in a refrigeration system. It absorbs heat at low pressure and temperature during evaporation and rejects heat at high pressure and temperature during condensation in a condenser. A refrigeration system consists of a number of components, apparatus, and pipe lines that are coupled in a sequential order to produce the desired cooling effect. There are many refrigeration systems and these are mainly used in heating, ventilation and air conditioning (HVAC) industry for cooling and freezing of products, condensing vapors, maintaining comfort conditions and for cold storage. In almost all refrigeration systems, thermodynamics, fluid mechanics and heat transfer processes are involved and therefore, a thorough understanding of these processes and the systems are extremely important. Coefficient of performance (COP), a dimensionless index, is defined as the figure of merit to estimate the performance of a refrigeration cycle. It is typically defined as the amount of heat removed from the evaporator to the energy input that is required to operate the system. The laws of thermodynamics, specially the first and the second laws are exclusively used in analyzing different refrigeration cycles to evaluate performances of these cycles, not only in terms of COP but also to determine other aspects of thermal performances from the second law point of view. With the improvement of computational and numerical capability, inverse and optimization techniques have also become popular nowadays. Inverse and optimization techniques can be used respectively to (i) identify unknown system characterizing parameters against known design output and (ii) identify the most appropriate system design configuration for maximizing the system performance through determination of optimal parameters.

1.2 Refrigeration systems

Refrigeration is the process of removing heat from a body at low temperature. It is used in HVAC industry for cooling and freezing of products, condensing vapors, maintaining comfort conditions and for cold storage. The following are the basic refrigeration systems that are used in refrigeration and HVAC industry.

(i) Vapour compression refrigeration system (VCRS)

(ii) Vapour absorption refrigeration system (VARS)

(iii) Gas cycle refrigeration system (GCRS)

(iv) Ejector refrigeration systems (ERS)

(v) Thermoelectric refrigeration systems

(vi) Thermo–acoustic refrigeration systems

In these systems, a refrigerant is used to transfer heat from a region of lower temperature to a sink at higher temperature. The refrigerant absorbs the heat because its temperature is lower than the temperature of the source. During system operation, the temperature of the refrigerant increases to a value higher than that of the sink and therefore, it can deliver the heat to the high temperature sink.

1.2.1 Vapour compression refrigeration system (VCRS)

Vapor–compression refrigeration systems are the most commonly used refrigeration systems. A simple schematic of a single stage VCRS is shown in Fig. 1.1. It consists of a compressor, a condenser, an expansion valve and an evaporator. In VCRS, the refrigerant is vaporized and condensed alternately in the evaporator and the condenser respectively. The compressor is used to compress the refrigerant in the vapour phase. The cooling effect is produced in the evaporator.

VCRS has the advantage of high Coefficient of Performance (COP) and large cooling capacity over the other refrigeration systems. Chlorofluorocarbons (CFCs) were used as refrigerants in VCRS earlier. As they have large degree of ozone depletion potential (ODP) and global warming potential (GWP), therefore, CFCs, are being phased out gradually and these are now substituted with HCFCs, HCs and HFCs. Compared to CFCs, HCFCs, HCs and HFCs have relatively less ODP and GWP. Research is being carried out to evaluate performance of VCRS with low ODP/GWP refrigerants having superior thermo physical and heat transfer properties [1–3].

The single stage VCRS is however not suitable for industrial cooling applications such as precipitation hardening of special alloy steel, rapid freezing, storage of blood and frozen food, liquefaction of petroleum vapor and atmospheric gases etc. where it is required to achieve temperature at very low level. This is due to operational difficulty associated with the compressor in compressing refrigerants with extremely large specific volume, low volumetric efficiency and excessive stresses on compressor parts due to higher compressor pressure ratio. In such cases, cascade refrigeration systems are used by combining two or more refrigeration cycles with different refrigerants in series. Say for example, in a two stage system, if the same refrigerant is used in both low and high temperature circuit, then it becomes difficult to achieve low temperature due to property limitation of the individual refrigerants. Therefore, in cascade refrigeration systems, different refrigerants are used in the individual circuits.

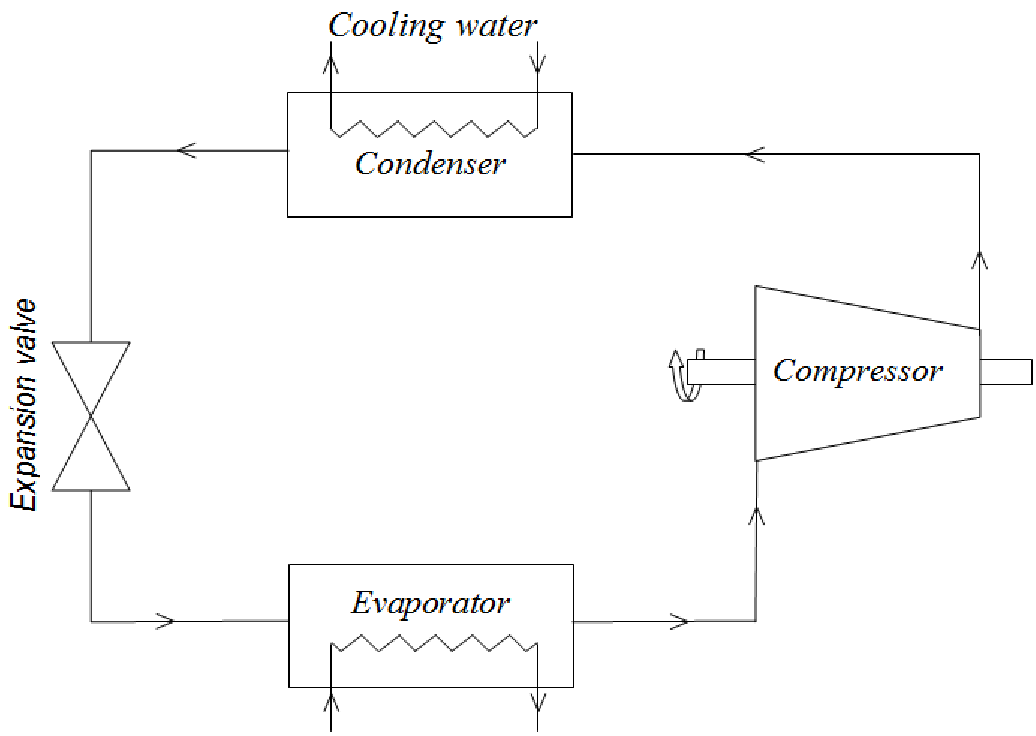

Fig. 1.1: Schematic of a single effect VCRS

Current research trend with cascade refrigeration system is towards the use of natural refrigerant pairs such as NH_3 (R717), CO_2 and the HCs which are naturally available and also have comparatively low GWP and ODP. The CO_2/NH_3 has been widely investigated both theoretically and experimentally [4, 5, 6–10].

1.2.2 Vapour absorption refrigeration system (VARS):

VARS is a heat driven cooling system that is used mainly for meeting cooling and air conditioning demands. Any heat source such as geothermal energy, solar energy, process steam or waste heat from cogeneration plants and even natural gas can be used to operate VARS. VARS has received significant research interest in recent times due to a global trend for rational utilization of energy sources and environment protection from global warming and ozone depletion. Although the COP of VARS is low compared to VCRS, but the advantage is that low grade heat can be used as driving energy source for operating VARS. Most importantly in VARS, the use of CFCs as working fluid can be avoided. It is a highly reliable technology which is quiet in operation and it has a long service life. Today, the absorption technology has improved a lot. Moreover, the rising cost of electricity and the environmental pollution caused by the power generating stations are also some of the factors that are contributing to the growth and popularity of absorption systems as an alternative to the VCRS.

In VARS, the refrigerating effect is produced by using a binary mixture of refrigerant and absorbent and low grade energy (heat) instead of high grade electrical energy as in the VCRS [11]. A commonly used single effect VARS is shown in Fig. 1.2.

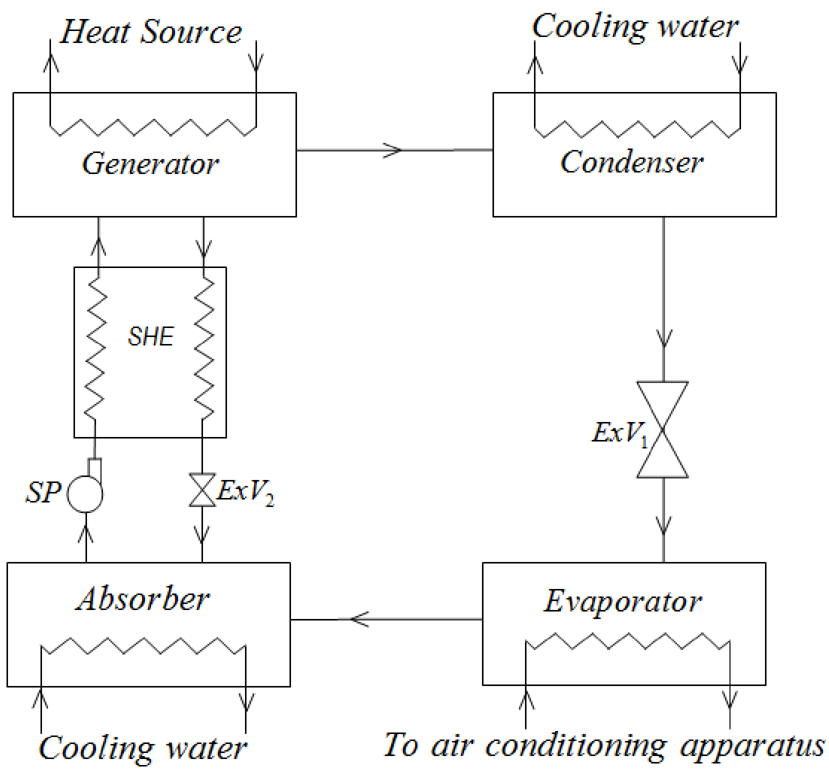

Fig. 1.2: Schematic of a single effect VARS

The system schematic is different from the VCRS in the sense that the compressor of the VCRS is replaced by a complex generator–absorber assembly with a solution pump

(SP), a solution heat exchanger (SHE) and an expansion valve in the assembly. The refrigerant vapour, after it is vapourized in the evaporator, goes to the absorber where it is absorbed by a weak solution of the refrigerant in the solvent. Next, the strong refrigerant solution from the absorber is pumped by the SP to the generator via the SHE where heat is supplied from the source. The function of the SHE is to preheat the strong solution with the heat of the generator off weak solution prior to its entry to the generator. This partially reduces the heat load in the generator. When heat is supplied in the generator, the refrigerant evaporates and the vapour then goes to the condenser where it is condensed to liquid. The expansion valve in the refrigerant side is used to reduce the pressure of the liquid refrigerant from the condenser to the evaporator pressure. The liquid refrigerant then enters the evaporator and this completes the cycle.

Selection of an appropriate combination of refrigerant and absorbent is very crucial in a VARS as it decides the efficiency of the absorption process as well the performance of the overall system. Ammonia–water (NH_3–H_2O) and water–lithium bromide (H_2O–LiBr) are the most widely used refrigerant and absorbent pair in a VARS. The H_2O–LiBr pair is used mainly for air–conditioning and chilling applications over 4°C because of the ice formation problem at low temperature and crystallization of LiBr at moderate concentration. On the other hand, NH_3–H_2O is used for large capacity industrial applications requiring low temperature for process cooling below 0°C. The above schematic shown in Fig. 1.2 will however be slightly different for the NH_3–H_2O VARS. A rectifier needs to be fitted between the generator and the condenser in NH_3–H_2O VARS for separating the water and allowing only NH_3 vapour to go into the condenser.

1.2.3 Absorption/vapor–compression cascade refrigeration system

Cascaded absorption–VCRS is a two stage system (Fig. 1.3). It consists of a single effect VARS in the HTC and a single stage VCRS in the low temperature circuit (LTC). The VARS at the top and VCRS at bottom share a common heat exchanger that acts as the evaporator of the absorption system and the condenser of the compression system. Since the topping cycle is a heat driven VARS, therefore, low temperature can be achieved in the bottoming VCRS without using a conventional two stage cascade VCRS in which two separate VCR systems are combined as HTC and LTC. Through use of such a cascade system, it is possible to save considerable amount of electrical energy. Since the system is obtained by cascading the VARS with the VCRS, it offers the advantages of both the vapour absorption and the vapour compression systems.

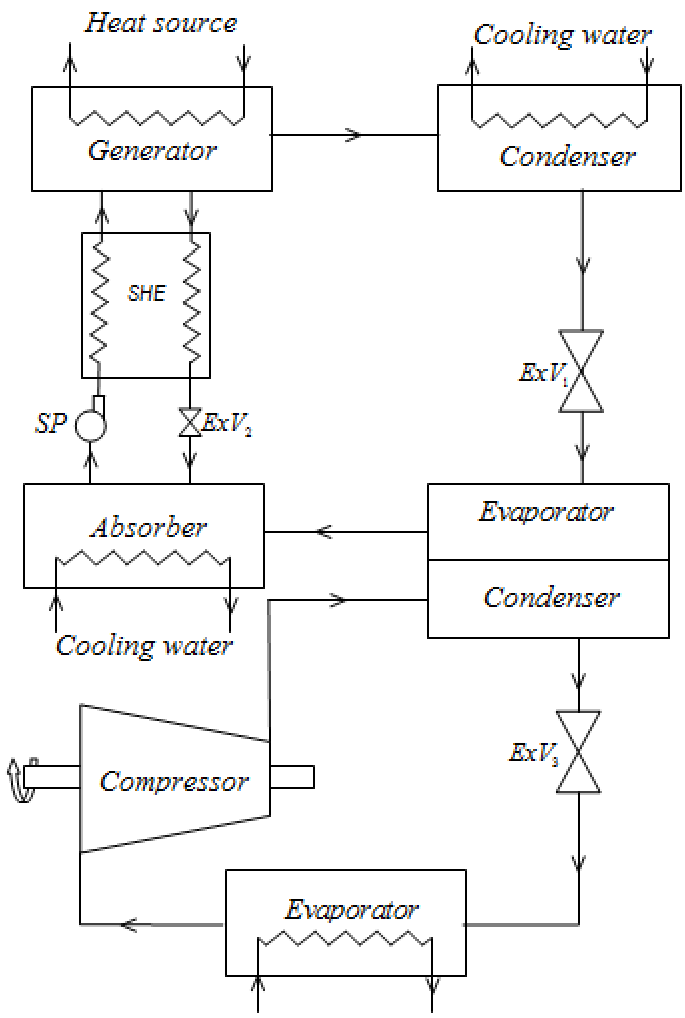

Fig. 1.3: Schematic view of Cascaded absorption–VCRS

Cascaded absorption–VCRS was analyzed for performance evaluation in various studies [12–16]. However, due to presence of the VARS and its bulky components at the top, a cascaded absorption– VCRS becomes complex and bulky, but the overall operating cost of the system reduces because of simultaneous usage of electricity and heat energy for refrigeration [16].

1.2.4 Gas cycle refrigeration system

In a GCRS, either air or some other gas is used as refrigerant and the gas does not undergo any phase change during the cycle. Consequently, all the internal heat transfer processes involved in a GCRS are sensible heat transfer processes. GCRS works on reversed Brayton cycle. It uses simple and lighter components such as axial flow compressor and gas turbine which make them suitable for air craft cabin cooling. The schematic of a simple aircraft refrigeration system is shown in Fig. 1.4. There is also modified system such as Bootstrap system which uses two heat exchangers (air cooler and after cooler), instead of only one air cooler (AC) as in the simple system. COP of these systems are however low compared to VCRS.

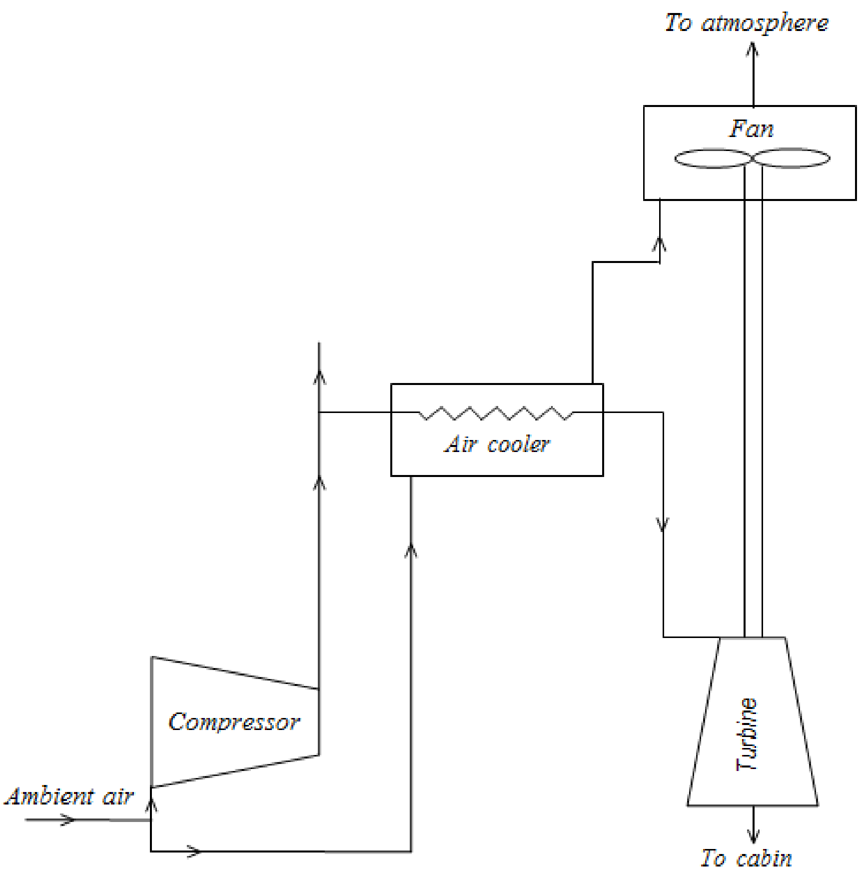

Fig. 1.4: Schematic of a simple aircraft refrigeration cycle

Regeneration can be employed in GCRS for temperature lowering and therefore, often regenerative GCRS is used for liquefaction of gases and cryogenic applications requiring temperature below –100°C. Regenerative cooling is achieved by using a regenerator in the cycle as shown in Fig. 1.5. With the use of the regenerator, the gas temperature can be lowered further before it enters the gas turbine. The temperature decreases further when the gas expands in the turbine and thus, extremely low temperatures can be achieved by using a regenerative GCRS.

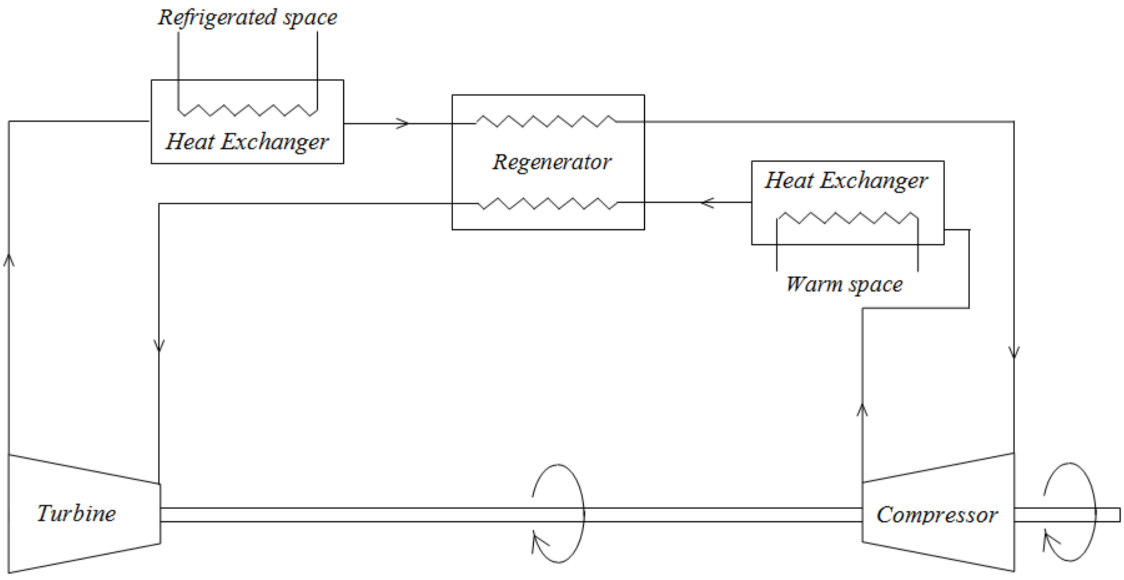

Fig. 1.5: Schematic of a regenerative GCRS

Temperature of such low magnitude is difficult to be achieved using other refrigeration systems. Several complex refrigeration cycles such as Joule–Thomson cycle, Linde–Hampson cycle and Claude cycle are used for liquefaction of gases. Oxygen and nitrogen separation from air, preparation of liquid propellants for rockets, study of material properties at low temperatures are some other specific application areas where GCRSs are used [17].

1.2.5 Ejector based refrigeration systems

Ejector refrigeration or jet pump refrigeration is a heat driven cooling system with an ejector that is used to increase the pressure of the refrigerant vapour [18–20]. The compressor, which is common in VCRS, is not required in an ejector based refrigeration system and hence, the consumption of electrical energy is greatly reduced [18–22]. A basic ejector based refrigeration cycle is shown in Fig. 1.6. It consists of a boiler/generator, an ejector, a condenser and an evaporator. Heat is applied in the boiler/generator to vaporize the high pressure liquid refrigerant. The high pressure vapour (also known as primary fluid) then flows through an ejector where it first accelerates through a nozzle. The pressure of the primary fluid reduces at the nozzle outlet and it helps in inducing the refrigerant vapour from the evaporator side (known as the secondary fluid) to the ejector. The primary and secondary fluids mix in a mixing chamber and then flow through the diffuser section of the ejector where it decelerates and pressure recovery takes place at the diffuser outlet. The vapour mixture then enters the condenser where it is condensed by rejecting heat to a surrounding medium. The condensed liquid refrigerant is divided into two streams, one stream goes via the expansion valve to the evaporator where it evaporates to producing the cooling effect and the other portion is pumped to the boiler/generator. The primary vapour from the generator goes to the ejector and gets mixed with the secondary vapour coming from the evaporator to complete the cycle.

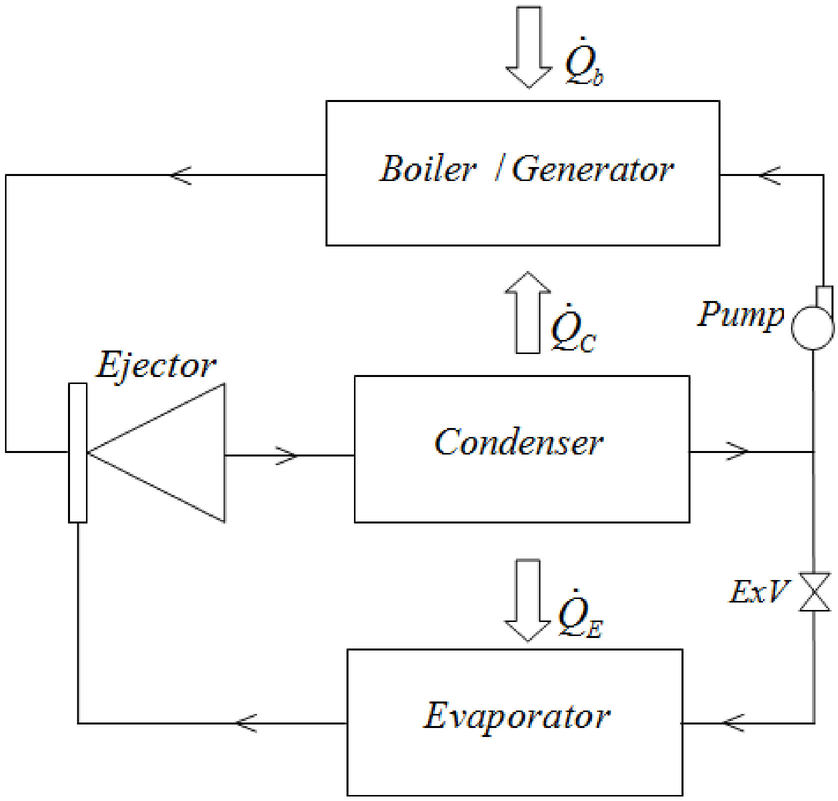

Fig. 1.6: Ejector refrigeration system [21]

Ejector based refrigeration systems are simple in configuration with relatively less number of moving parts, quiet operation with low noise and less vibration. The installation and operating cost is also low. These are developed and available in various capacities for a wide range of applications [23–27], but these systems have very low COP compared to VCRS and VARS. COP however can be improved through modification of the simple ejector cycle. Modified and combined configurations such as ejector cycle with additional jet pump [28], combined ejector–VCRS [21, 29–30], hybrid absorption–ejector refrigeration system [31] and combined vapour compression–absorption–ejector refrigerator systems [32] have been developed and investigated.

The following schematic shown in Fig. 1.7 shows a combined ejector based and VCRS [19–21].

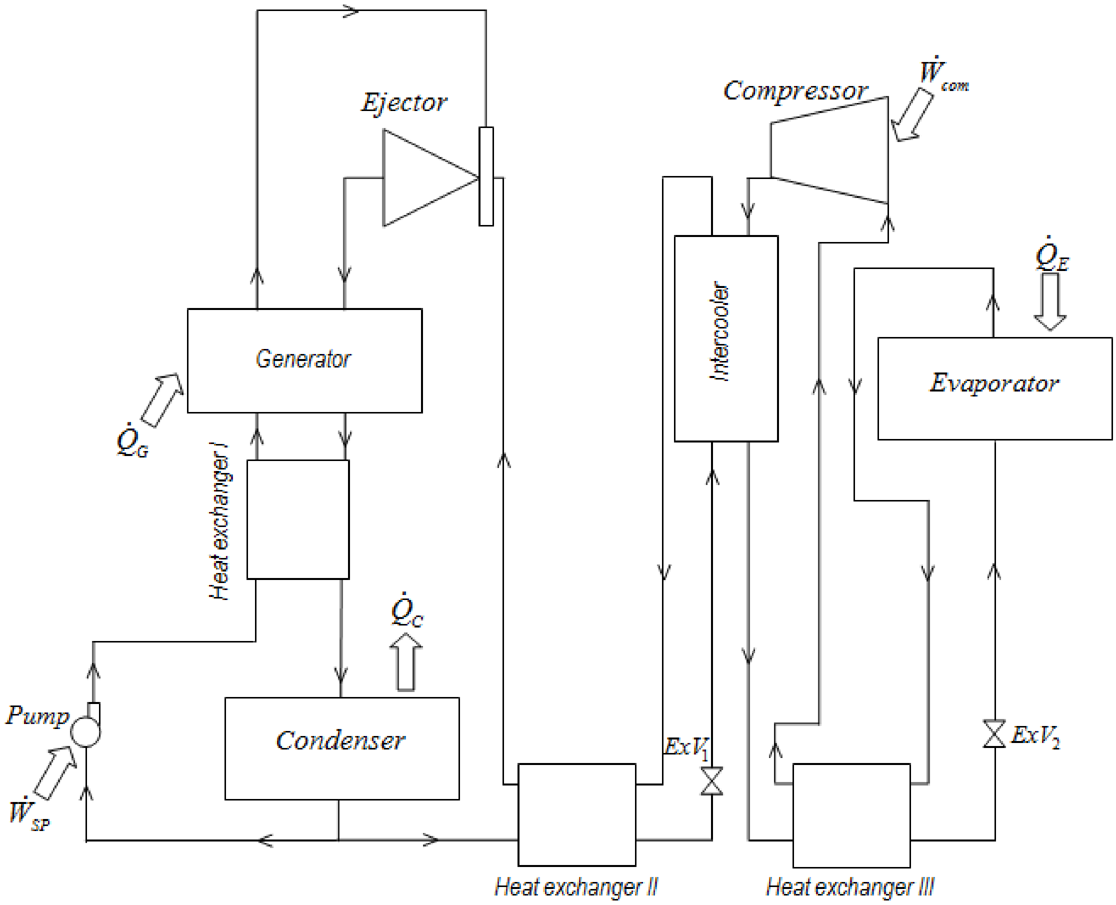

Fig. 1.7: Schematic of a combined ejector based refrigeration cycle and VCRS

It consists of an ejector based refrigeration system which is similar to Fig. 1.6 and a normal conventional VCRS. The two cycles are connected by means of the intercooler which acts as evaporator of the ejector based cycle and condenser of the VCRS. Since, it is a kind of cascade; therefore, two different refrigerants can be used to take advantage of each individual cycle.

An ejector based refrigeration cycle can be combined with a single effect VARS as shown below in Fig. 1.8 [33]. The VARS has its normal components and the VARS generator serves both the purpose of evaporator and condenser of the ejector based cycle.

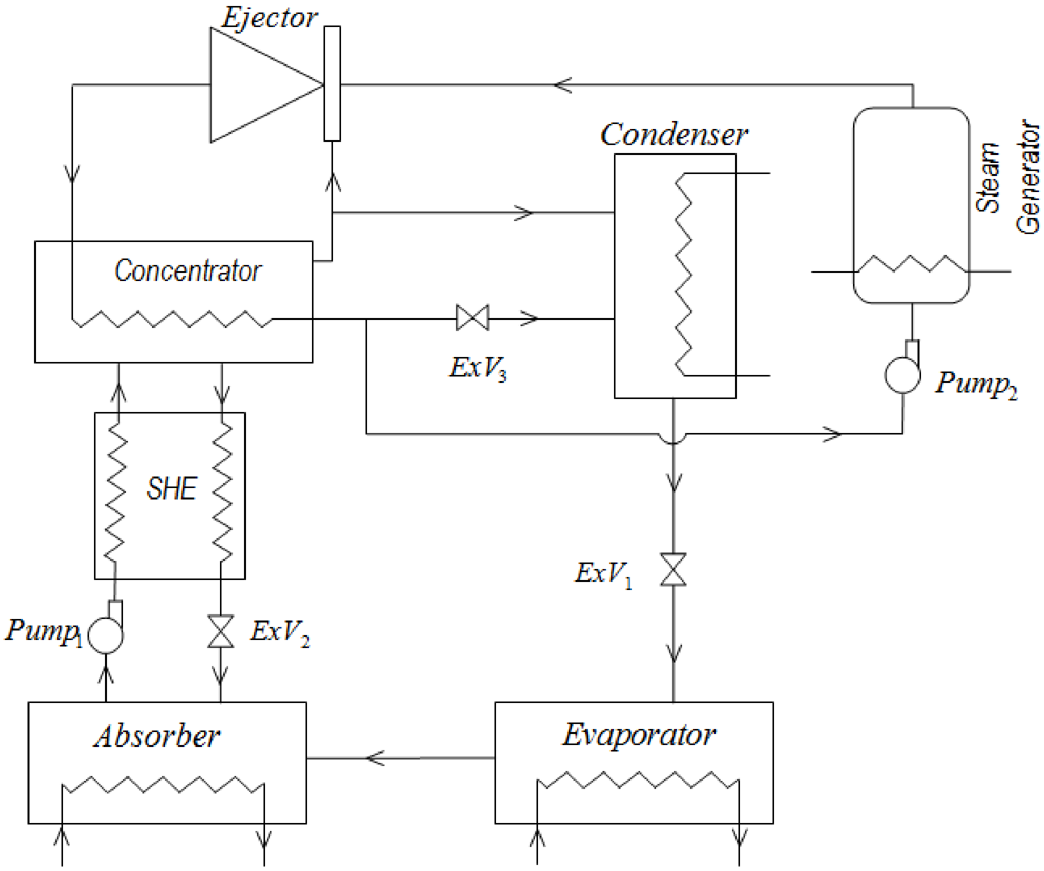

Fig. 1.8: Schematic of the combined ejector–VARS [33]

Significant effort has also been made to develop solar driven ejector based refrigeration systems [30, 34–37].

1.2.6 Thermoelectric refrigeration systems

The working principle of a thermoelectric refrigerator is based on Peltier effect. In a circuit containing two junctions of two dissimilar conductors or semiconductors, heat may be transferred from one junction to the other by applying DC sources such as photovoltaic (PV) cells, fuel cells and car DC electric sources. Semiconductors are better than metals for producing Peltier effect [38] and in a practical thermo–electric refrigerator, N and P type semiconductors are connected in series (Fig. 1.9). The heat from the refrigerated space is transferred through semiconductor elements to the hot–side heat sink which rejects the heat to the environment. The N type material has an excess of electrons, while the P type material has a deficit of electrons. Through an interconnector, when the electrons move from the P type to the N type material, the electrons jump to a level of higher energy state by absorbing heat from the refrigerated space. Again when the electrons flow back from the N type to the P type materials, the energy level of the electrons drop down as the heat energy is released to the heat sink [11].

Thermoelectric refrigerators are simple and reliable. This is the only cooling system where no refrigerant is required to produce the cooling effect. There are also no mechanical moving parts in these systems, hence they are quiet in operation and also compact in size and light in weight. However, these systems involve high cost and provide low energy efficiency. Due to their low COP, they cannot compete with the conventional refrigeration cycles [38]. Therefore, these are used in electronic, medical, automobile, telecommunications and space applications where system cost and energy efficiency are less important than system size, weight, reliability and quiet operation environment [39].

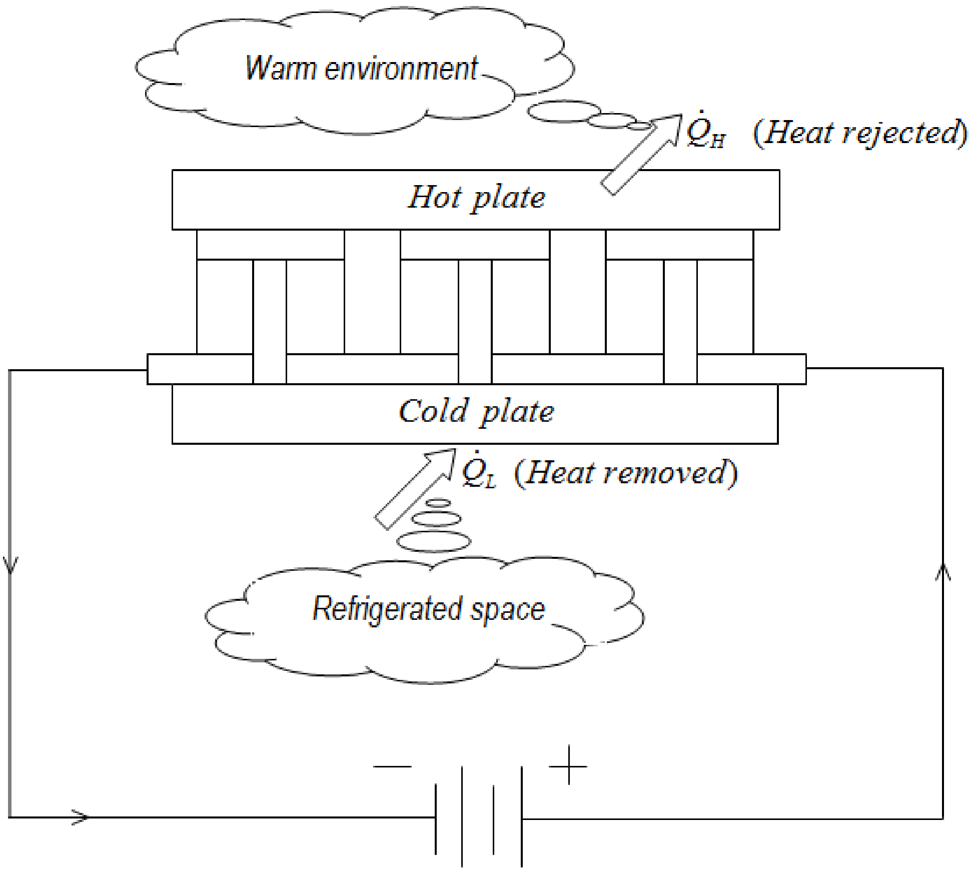

Fig. 1.9: Schematic of a thermoelectric refrigerator

Zhao and Tan [39] made a detail review on the development of thermoelectric cooling with a particular focus on advances in materials, modeling and optimization approaches, and applications. They emphasized on developing thermoelectric materials with improvised efficiencies and improvising on working conditions through cooling system's thermal design and optimization.

1.2.7 Thermoacoustic refrigeration systems

Thermoacoustic refrigeration is another technology where cooling is produced without the use of conventional toxic and harmful refrigerants and no moving parts. It utilizes either standing or travelling acoustic waves through a system of gas/gas mixture in a resonator to produce the cooling effect. The main components of a standing acoustic wave based thermoacoustic refrigerator (Fig. 1.10) are a closed cylinder, an acoustic driver, a porous stack, a hot and a cold heat exchanger. When the acoustic wave is applied through the acoustic driver, it makes the gas resonant which starts oscillating due to repeated compression and expansion of the gas in the resonator caused by the sound pressure of the wave. This oscillation of the gas creates a temperature difference due to which heat transfer occurs between the gas and the stack. Consequently, heat is removed from the cold side and rejected at the hot side of the system. In a travelling acoustic wave based system, the sound pressure is created by a moving piston and the conversion of acoustic power to heat occurs in a regenerator rather than a stack. The regenerator contains a matrix of channels that ensures good thermal contact between the gas and the matrix. The gas moves towards the cold heat exchanger during expansion when the pressure is low and absorbs heat. The gas rejects heat during compression when the gas moves towards the hot heat exchanger due to high pressure.

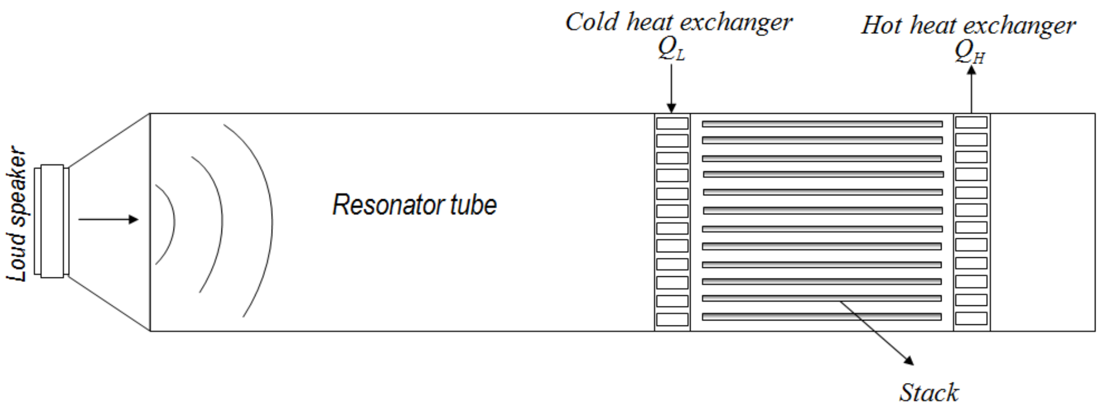

Fig. 1.10: Standing wave based thermoacoustic refrigerator

A number of design concepts and prototypes are developed and proposed in various research studies [40, 41, 42]. This cooling technology has the potential for further development but design improvements are necessary for efficiency improvement.

Therefore, the current research in this field is directed towards development of new design concepts and optimization of multistage systems combined with thermo–acoustic engines and refrigerators [43, 44, 45].

1.2.8 Metal hydride (MH) based refrigeration systems

Metal hydrides are a special type of alloys which can absorb and desorbs hydrogen in a reversible manner. When say an MH alloy–A (MH–A) comes into contact with hydrogen, it absorbs hydrogen by an exothermic reaction and store it as metal hydrides. In the reverse process, another MH alloy–B (MH–B) discharges hydrogen by an endothermic reaction and refrigeration effect is produced. Thus a MH based refrigeration system uses a pair of MH alloys, in which MH–A works at a higher temperature and MH–B at a lower temperature, under their own equilibrium hydrogen pressure. Research is being carried out to with different hydride pairs such as ZrMnFe / $MmNi_{4.5}Al_{0.5}$[46], $MmNi_{4.6}Al_{0.4}$ / $MmNi_{4.6}Fe_{0.4}$ [47], $LaNi_{4.61}Mn_{0.26}Al_{0.13}$ / $La_{0.6}Y_{0.4}Ni_{4.8}Mn_{0.2}$, $LaNi_{5x-y}Mn_xAl_y$ / $La_{1-x}Y_xNi_{5y}$, $LaNi_{4.7}Al_{0.3}$ / $MmNi_{4.15}Fe_{0.85}$, $LaNi_{4.65}Al_{0.35}$ / $MmNi_{4.0}Fe_{1.0}LaNi_{4.7}Al_{0.3}$ / $MmNi_{4.15}Fe_{0.85}$ etc. to evaluate their hydrogen absorption properties [48]. The basic working principle of a MH based refrigeration system is illustrated in Fig. 1.11. The process of regeneration and refrigeration associated with a MH based refrigeration system is explained in brief.

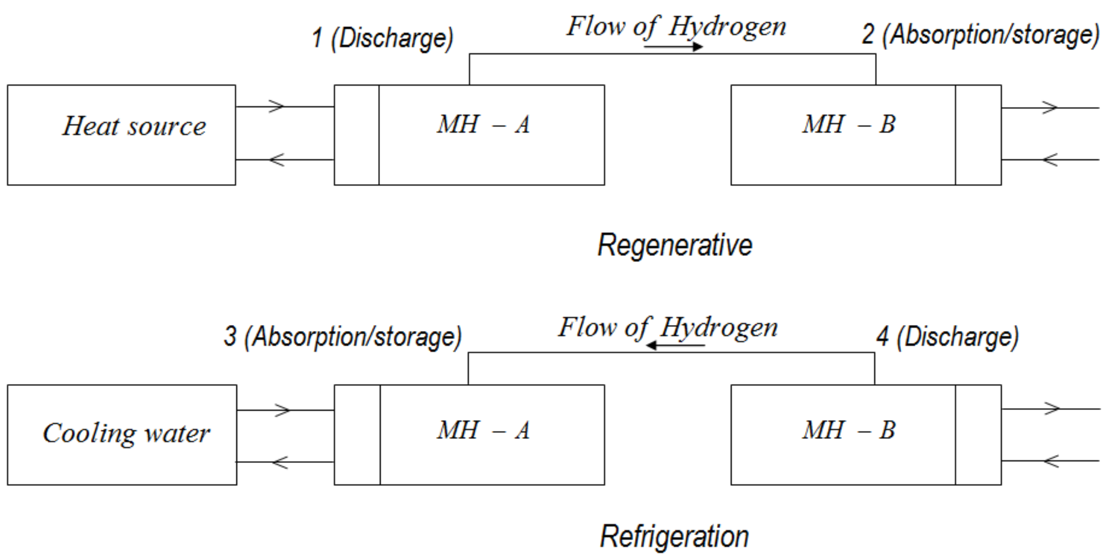

Fig. 1.11: Schematic of Regeneration and refrigeration associated with a MH

(a) Regeneration process

In this process, MH–A is heated and during heating when the temperature raises, the hydrogen pressure of the MH–A increases. MH–A, discharge hydrogen which moves to the MH–B, because the hydrogen pressure in MH–B is low. Next, MH–B absorbs hydrogen and generates heat and therefore, it is cooled and kept at low temperature by circulating a cooling medium in order to prevent rising of hydrogen pressure in MH–B. The MH–B continues to absorb hydrogen and hydrogen is stored as metal hydrides.

(b) Refrigeration process

After all the hydrogen from MH–A is transferred to MH–B, MH–A is cooled by circulating a cooling medium. The hydrogen pressure in MH–A reduces and MH–B starts discharging hydrogen to MH–A. In this process, the temperature decreases and heat is absorbed by MH–B to produce the refrigerating effect.

The advantages with MH based refrigeration systems are that they are compact and environmental friendly. By changing the alloy composition, the pressure–temperature characteristics can be adjusted to suit various heating and cooling requirements [46].

1.3 More on different VARS configurations

Depending on the configurations and the number of generators used, VARSs are classified into different categories. The single effect VARS with a single vapour generator is the most common that has already been explained. Half effect cycle (Fig. 1.12) is also available, however its COP is low; even lower than the COP of single effect VARS [49, 50]. The operating pressure and temperature of the generator are usually low in the half and single effect cycle.

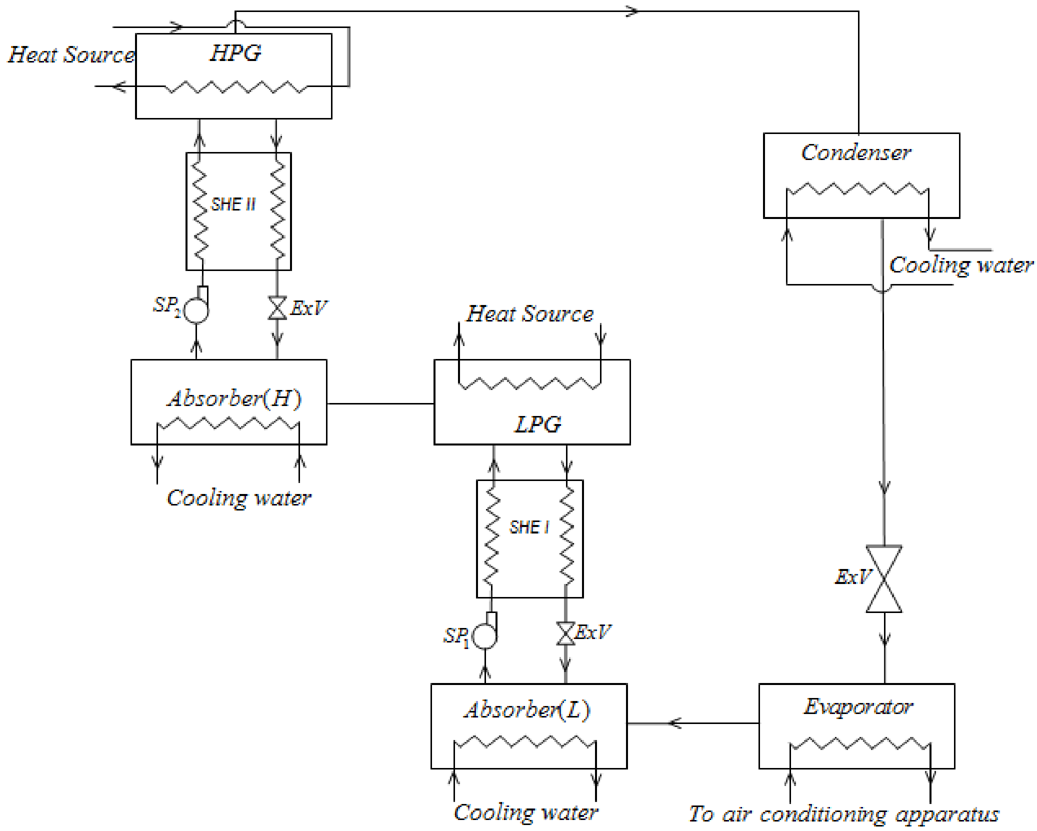

Fig. 1.12: Half effect cycle [49, 50]

The half cycle shown in Fig. 1.3 is also referred as double lift/double stage cycle [49, 51]. It consists of the low and high pressure cycles with one generator, one absorber, one SHE, one SP and one solution expansion valve in each cycle. The superheated water vapour generated in the generator of the low pressure cycle goes to the absorber of the high pressure one where it is absorbed by the dilute salt solution.

Multi effect (double and triple effect) cycles were developed at later stage in order to utilize heat source at higher temperature and also to overcome the problem of low COP associated with half and single effect cycles. The number of generators used in the double and triple effect cycles is more than one. As the name suggests, there are two generators in the double effect and three generators in the triple effect cycle respectively. VARS COP usually increases with the number of effects and thus, the triple effect system gives the highest COP; followed by the double effect and then the single effect. But, the gain in COP with use of every additional effect should actually be sufficiently high to justify the added cost and complexity involved with multi effect systems due to presence of more number of generators and other associated system components. But unfortunately, the highest gain in COP is achieved with the addition of the first effect and the COP gain usually diminishes with addition of every subsequent generator [52]. This is the reason that the double effect systems are preferred more for commercial use in the refrigeration industry [52–54]. However, this may not be only due to higher COP but also due to the fact that a single effect system is not suitable for utilization of heat source with high temperature as it will lead to higher loss of available energy. As such, heat source temperature is also one of the important criteria for selection and use of proper VARS configuration in cooling application.

A double effect VARS configuration can be of series, parallel and reverse parallel flow type depending on how the vapour generation is distributed among the different vapour generators [49, 52, 53, 55]. Details of all the double effect VARS configurations and their differences are explained in the Refs. [49, 52, 53, 55]. The schematics of the double effect series, parallel and reverse parallel flow configurations are shown in Figs. 1.13–1.15.

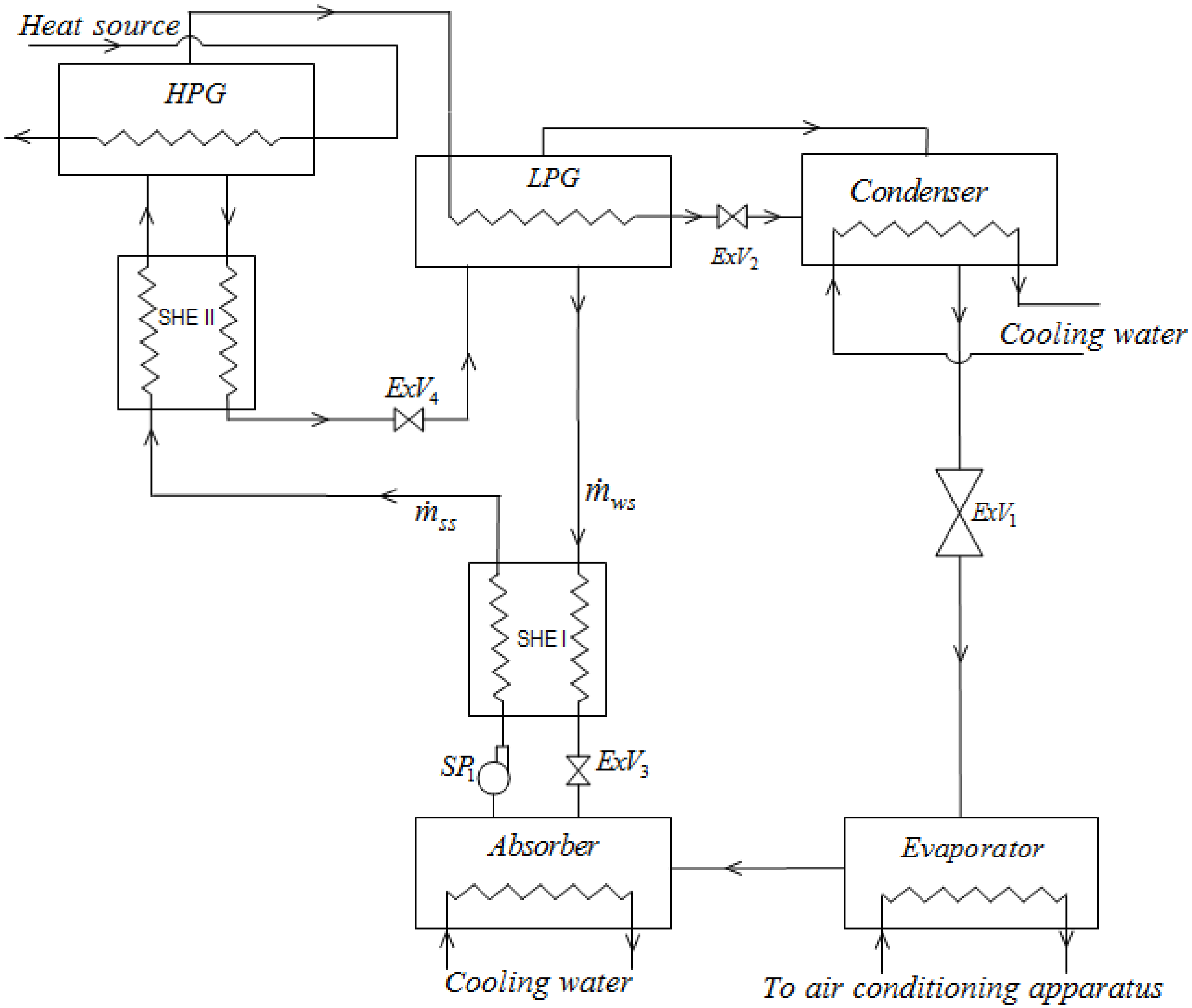

Fig. 1.13: Double effect VARS (series configuration) [52]

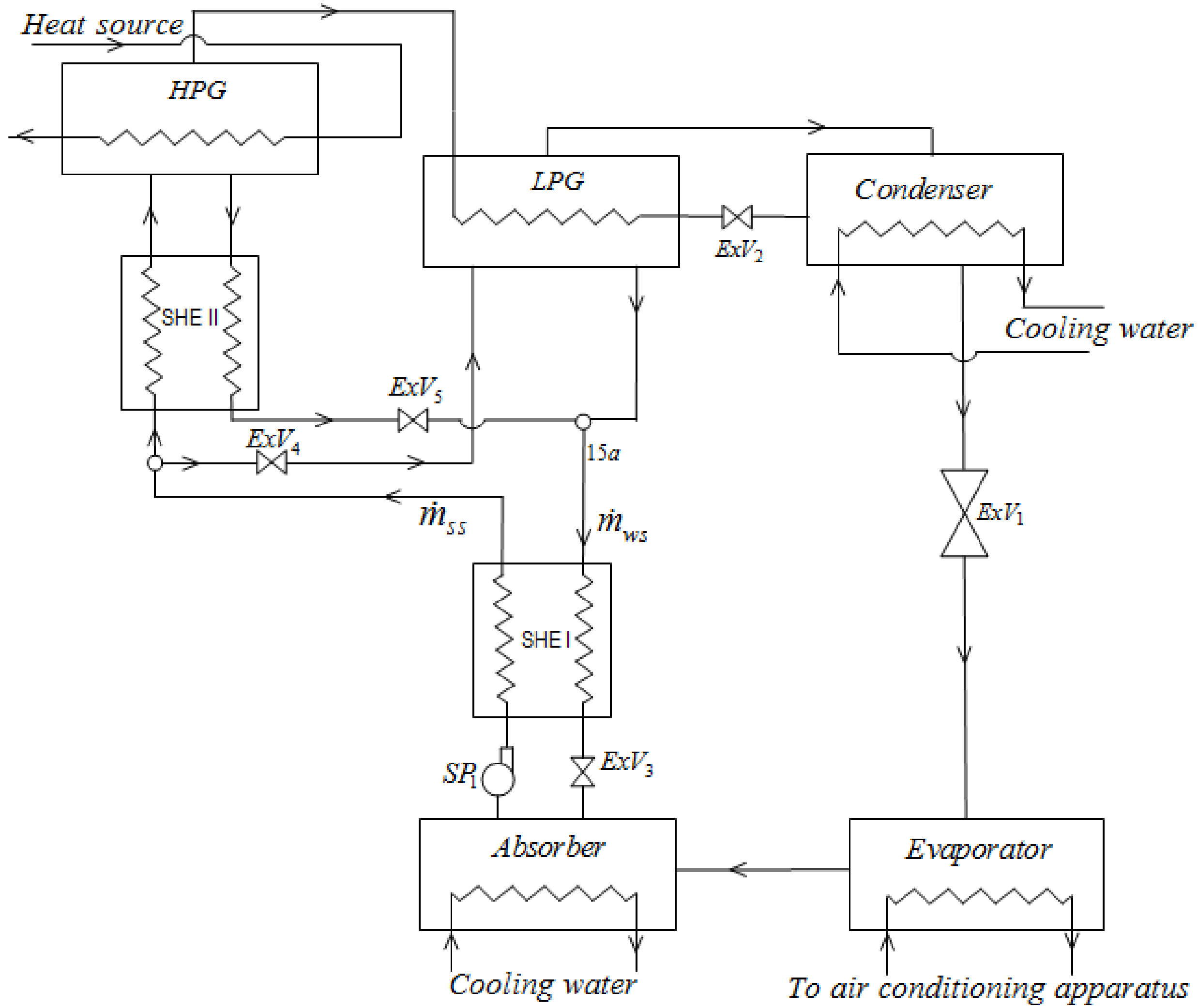

Fig. 1.14: Double effect VARS (parallel configuration) [52]

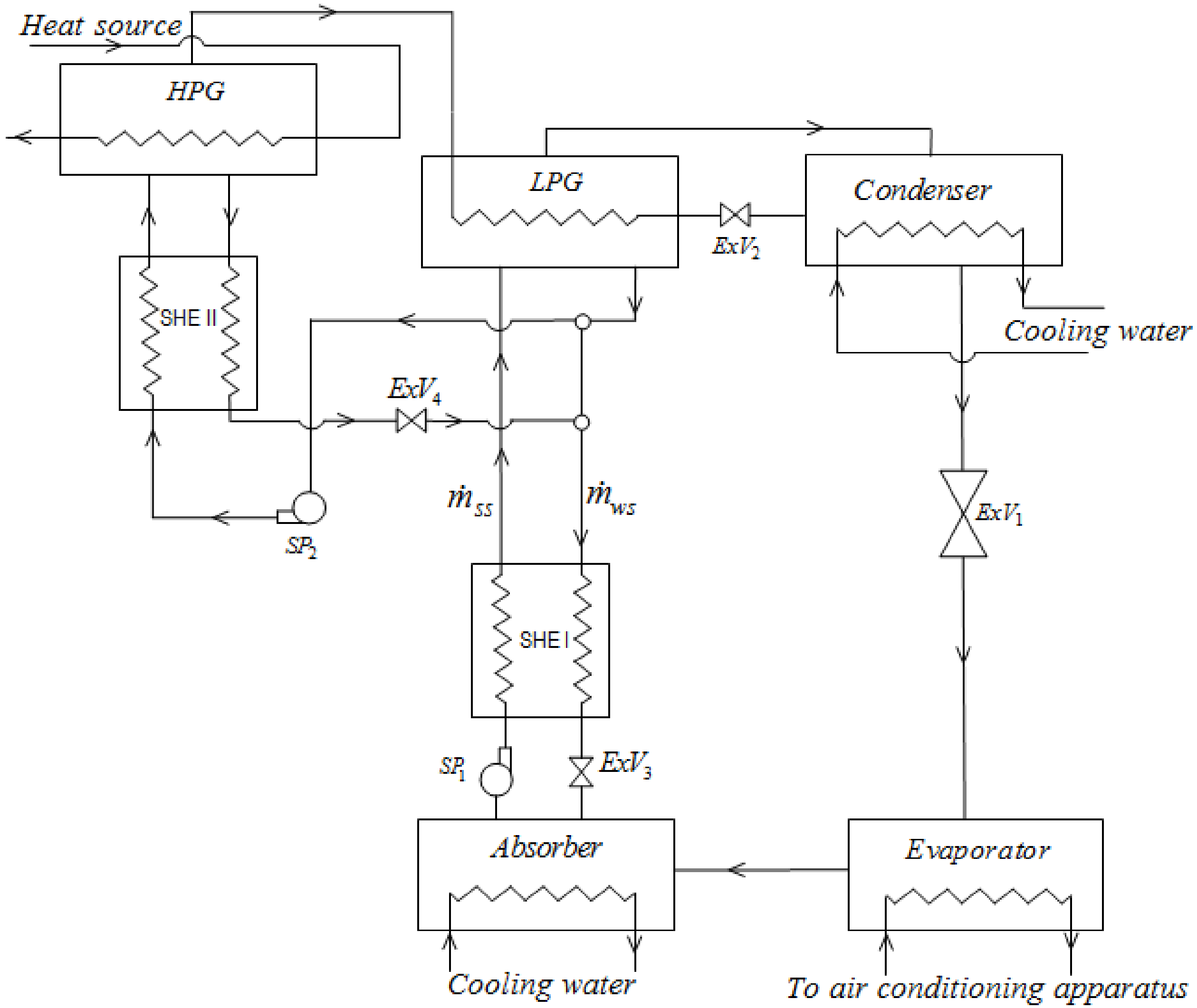

Fig. 1.15: Double effect VARS (reverese parallel configuration) [52]

As can be seen from the above schematic in Fig. 1.13, the strong refrigerant solution from the absorber is pumped directly to the high pressure generator (HPG) in the series flow configuration. In the parallel flow configuration (Fig. 1.14), however, the solution is distributed among the low pressure generator (LPG) and the HPG. In the reverse parallel flow configuration, the solution first goes to the LPG for partial vapour generation. The solution which comes out from the LPG is distributed and a part of this solution is pumped to the HPG while the other part is mixed with the solution flowing back from the HPG before it goes to the absorber again [52].

The double effect series, parallel and reverse parallel configurations which were shown in Ref. [49] were however little different from the ones in Ref. [52]. A careful look at the schematics presented in Figs. 1.13–1.15 reveals that in all the three schematics, the water vapour generated in the HPG is condensed in the LPG and the heat released during condensation is used to produce water vapour in the LPG from the HPG off medium solution. In the double effect cycles shown in Ref. [49], the authors have however referred to a condenser–generator assemblies ‘‘CG’’, where the heat released by the condensing vapor on the hot side of the heat exchanger (condenser) is used for producing vapor in the solution on the cold side (generator). The function of the CG is more or less similar to the LPG and as such, there is not much difference between the schematics in Ref. [49] and Ref. [52], although schematically, they look little different. But the reverse parallel configuration shown in Fig. 1.15 is somewhat different from the reverse flow configuration of Ref. [49]. In the reverse flow one, the solution from the absorber is first pumped to the LPG and after it is partial vaporizes, the remaining solution is then pumped to the HPG.

The triple effect cycles are shown below. Fig. 1.16 shows the triple effect series configuration, the strong solution from the absorber is pumped to the HPG which after water evaporation is returned back to the absorber through the SHE III, expansion valve III (ExV3), the medium pressure generator (MPG), SHE II, ExV2, LPG and SHE I, ExV1. The high pressure water vapour which is produced in the HPG goes to the MPG and after cooling in the MPG, it is mixed with the medium pressure water vapour generated in the MPG. The vapour mixture then goes to the LPG where it is condensed and the latent heat of condensation is used to generate water vapour at low pressure. Both the low pressure water vapour and the condensed liquid water go to the condenser where the water vapour is condensed. The condensed water is sent via the pressure reducing expansion valve (ExV4) to the evaporator where the water is evaporated by supplying heat from a source to produce the cooling effect. The vaporized water then goes to the absorber where the water vapour is absorbed by the lean solution.

In the parallel–flow cycle shown in Fig. 1.17, the absorber leaving strong solution is distributed among the LPG, MPG and HPG. The solution which comes out from the MPG is mixed with the solution that comes back from the HPG via the SHE III. It then enters the SHE II, gets mixed with the LPG leaving solution and passes via the SHE I before finally being sent back to the absorber.

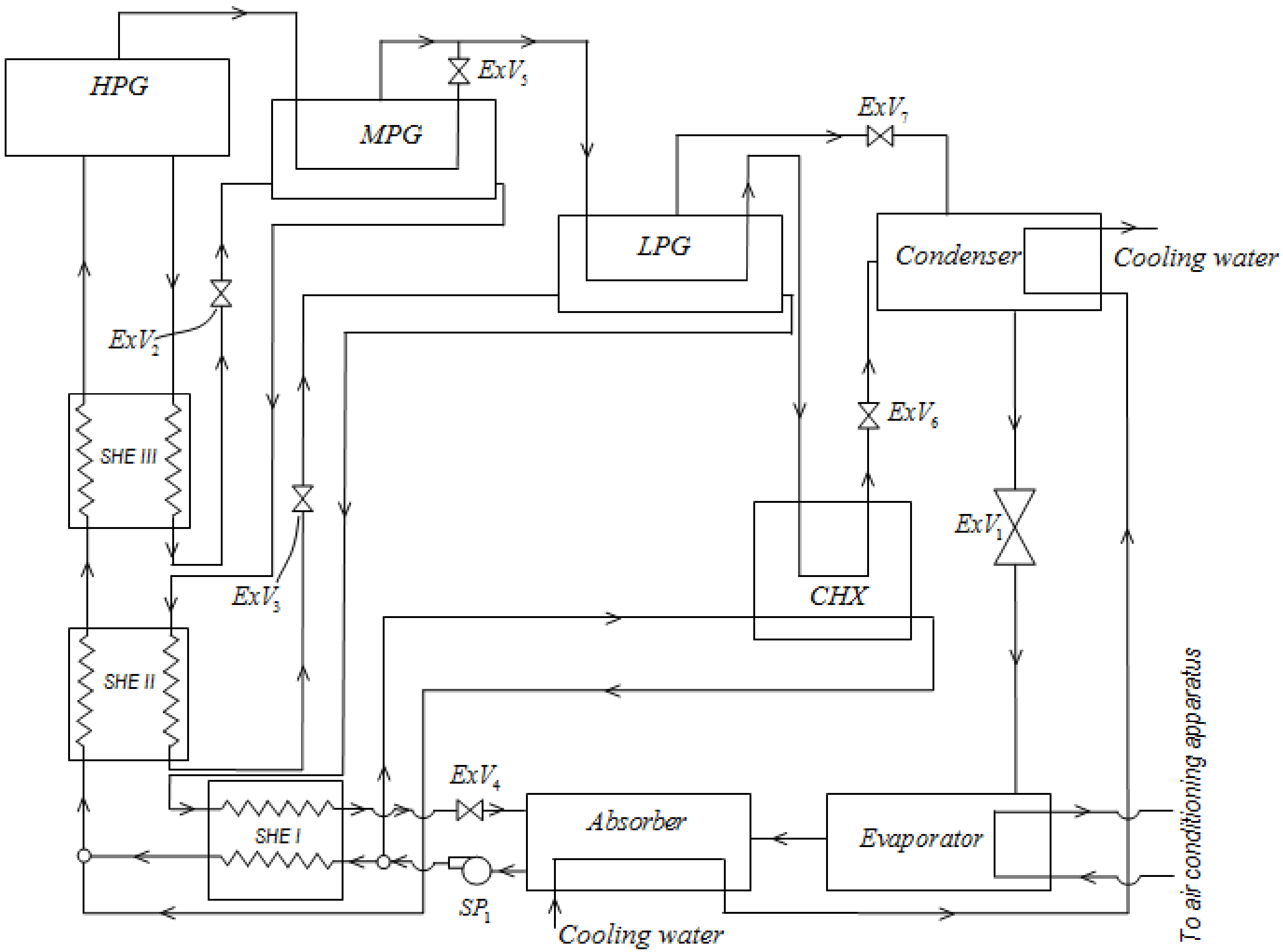

Fig. 1.16: Triple effect VARS (series configuration) [56]

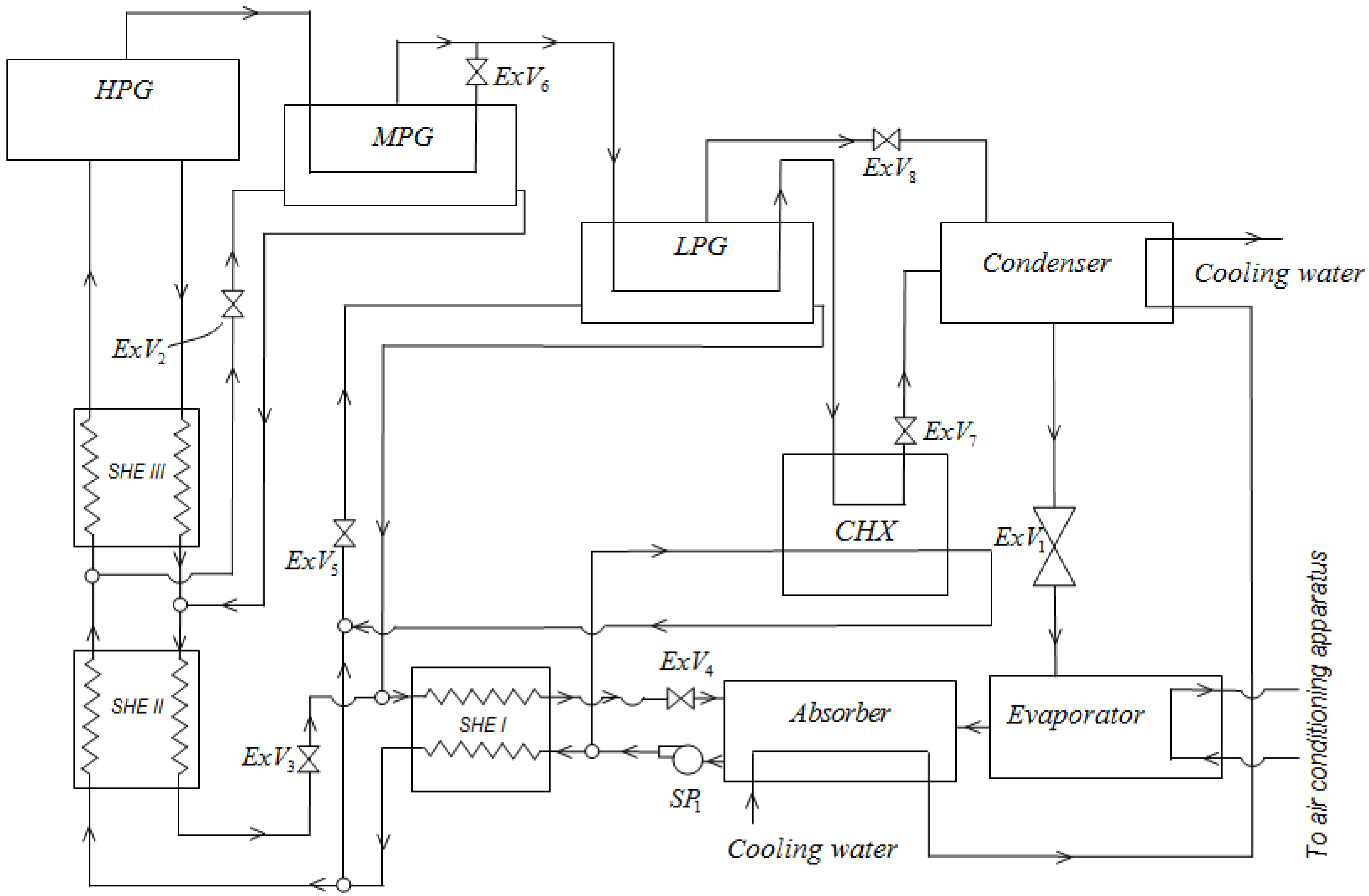

Fig. 1.17: Triple effect VARS (parallel configuration) [56]

In the reverse–flow cycle in Fig. 1.18, the strong solution is first pumped to the LPG, then to the MPG and finally to the HPG. The solution which comes out from the MPG is divided into two streams. One part is pumped to the HPG and the remainder mixes with the solution that comes back from the HPG via the SHE III. The mixed solution then goes to the SHE II and again mixes with one part of the LPG leaving solution (the other part is pumped to the MPG) and finally goes back to the absorber via SHE I and the expansion valve (ExV1).

Kaita [56], in his triple effect cycles also made use of a condensed refrigerant heat exchanger (CHX), where heat of condensed refrigerant leaving the LPG is recovered to preheat a part of the absorber leaving strong solution prior to its entry to the LPG in order to obtain improved performance.

The triple effect series configuration shown in the Ref. [57] is however slightly different in the sense that the HPG off high pressure water vapour is condensed in the MPG and the condensed water is then sent to the condenser via a pressure reducing valve. Similarly, the MPG off water vapour is condensed in the LPG and is passed to the condenser via another pressure reducing valve. This is shown in Fig. 1.19 and in this schematic; the vapour generated in the MPG is not mixed with the HPG off stream like in Fig. 1.18 which is then sent to the LPG and finally to the condenser via CHX. In line with the series flow schematic of Gomri [57], the parallel and reverse parallel configurations can be in the following way as shown in the Fig. 1.20 and Fig. 1.21.

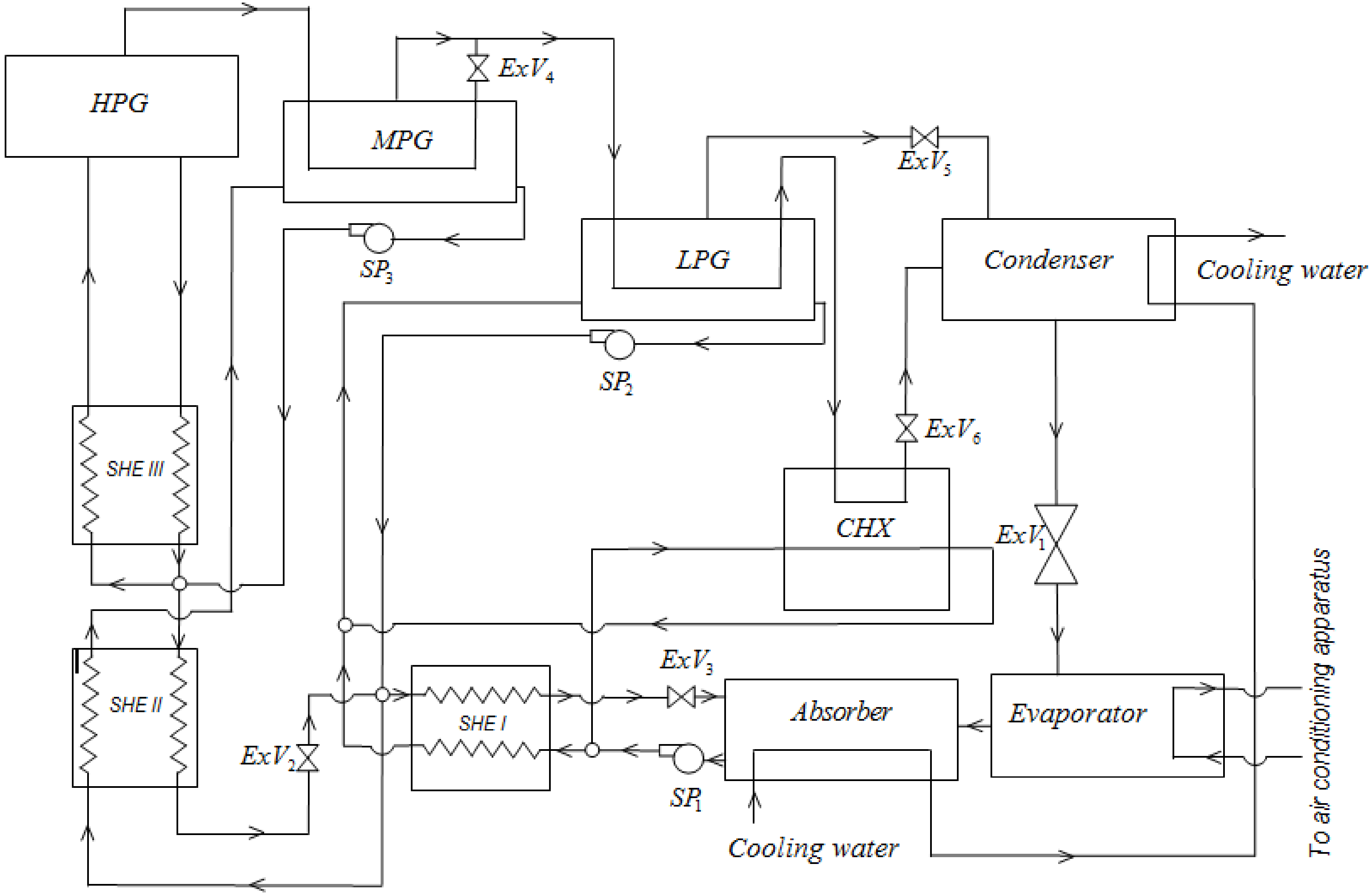

Fig. 1.18: Triple effect VARS (Reverse parallel configuration) [56]

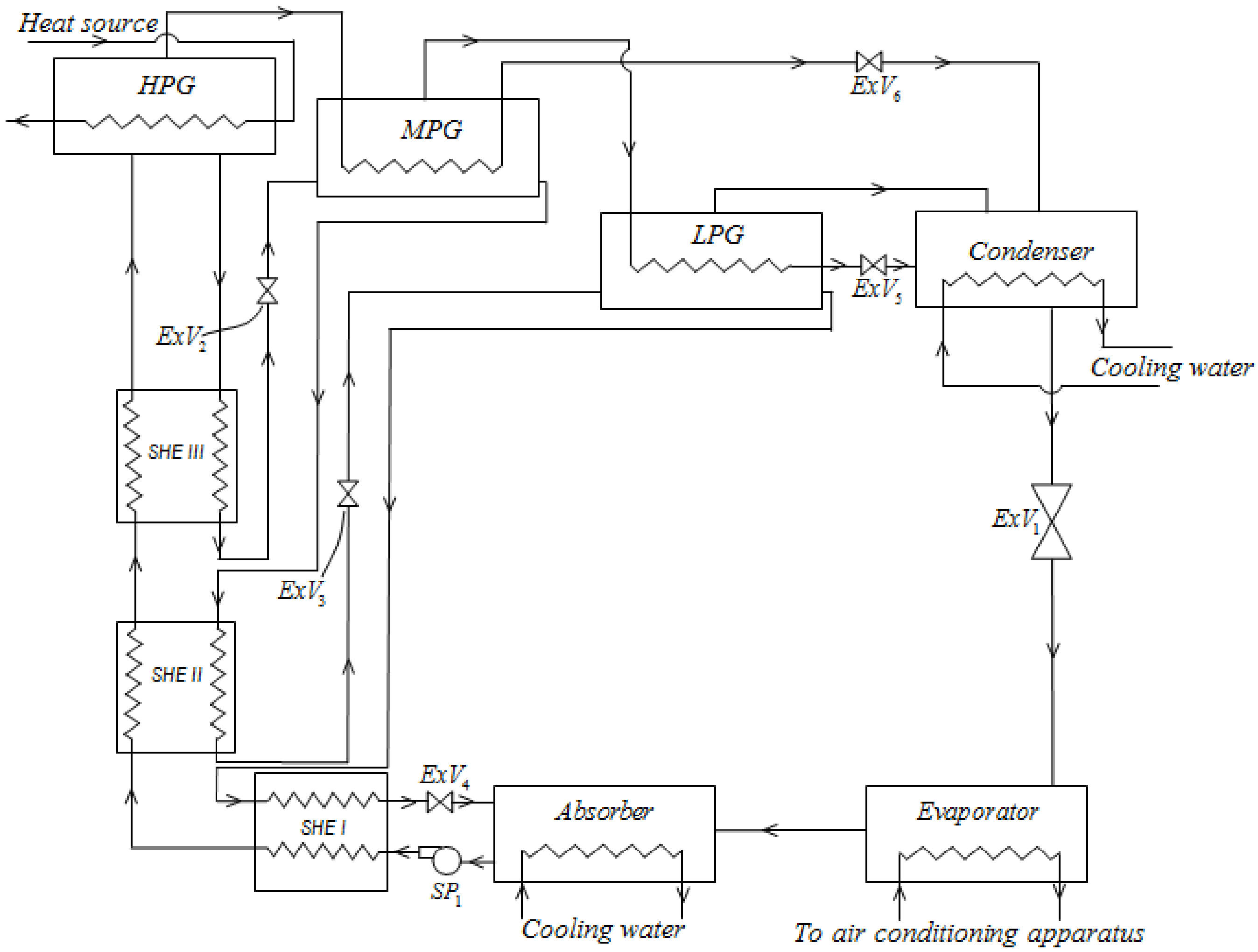

Fig. 1.19: Triple effect VARS (Series configuration) [56]

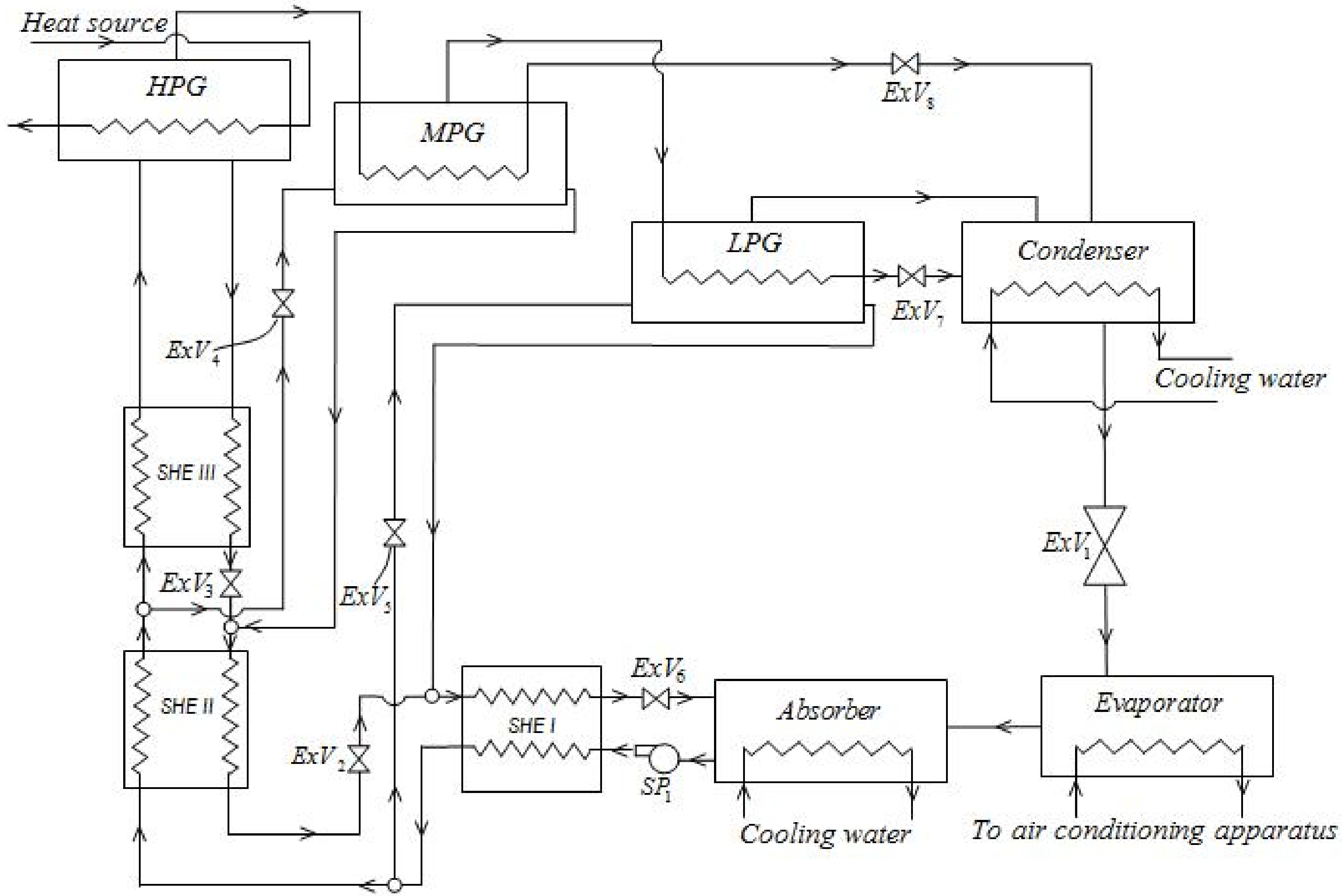

Fig. 1.20: Triple effect VARS (Parallel configuration)

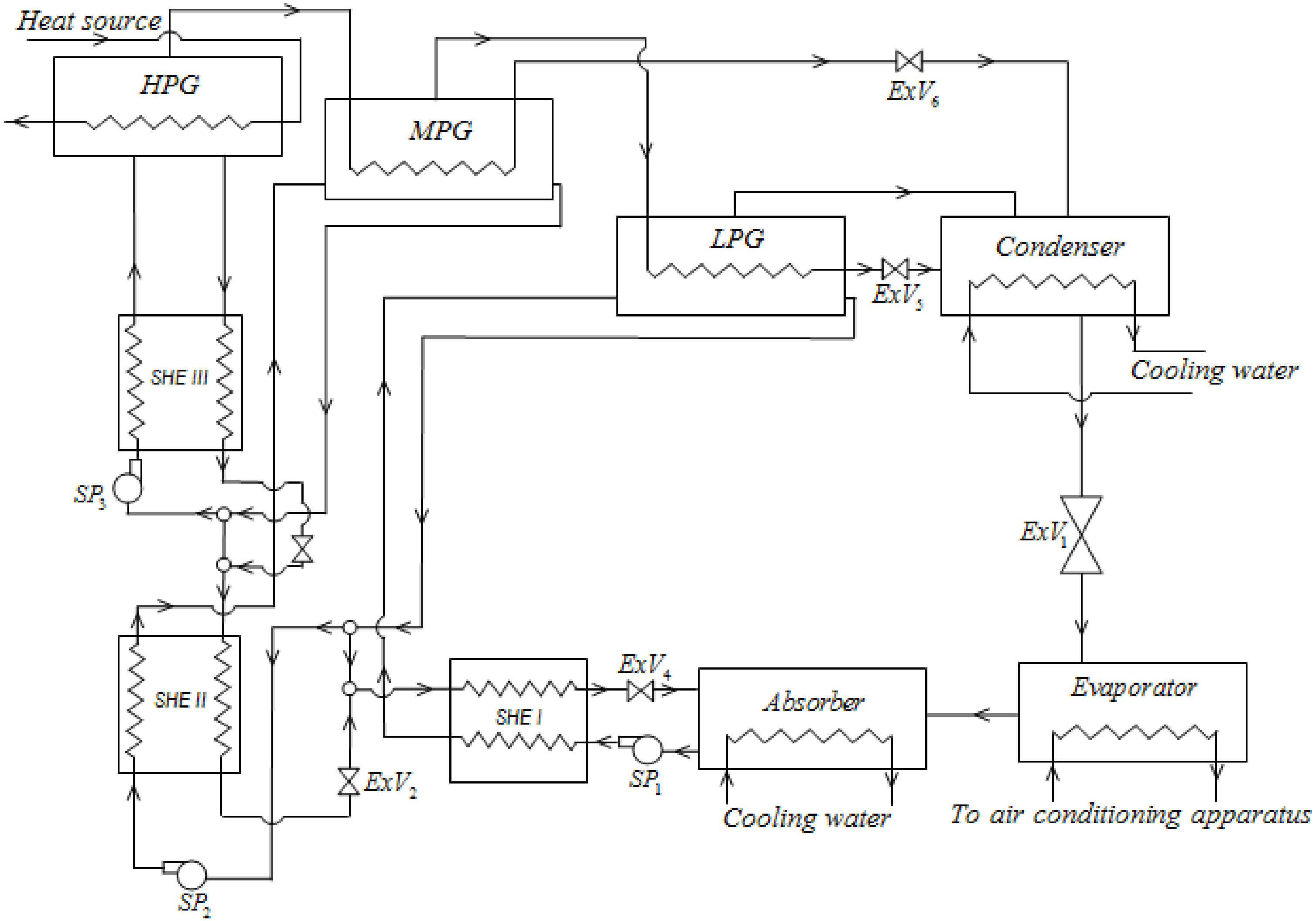

Fig. 1.21: Triple effect VARS (Reverse parallel configuration)

1.4 Some desirable properties of refrigerant/absorbent pair used in VARS

The refrigerant/absorbent pairs which are used in VARS must possess some essential characteristics. Few demanding requirements are listed below [58–59].

(i) Solubility: High solubility between the refrigerant and the absorbent and refrigerant is an important requirement because the refrigerant vapour needs to be absorbed by the absorbent in the absorber. The refrigerant should exhibit negative deviation from Raoult's law at the absorber.

(ii) Volatility: A large difference in the boiling points of refrigerant and absorbent is desired to ensure that only pure refrigerant goes through refrigerant circuit.

(iii) Heat of mixing: The heat of mixing of the refrigerant should be less which however contradicts with the solubility requirement. Hence, a trade–off is required between the two.

(iv) Mobility: The refrigerant/absorbent pair should have low viscosity for high performance. Mobility of the working fluids depends on its viscosity. Low viscosity reduces the pumping power requirement and guarantees higher system performance.

(v) Crystallization: In the refrigerant/absorbent pair, if the absorbent is a salt then it tends to form solid crystals at certain temperature, pressure and salt concentration. In the design limit, the pair must not attain solidification state; this will otherwise block the flow passages and halt the system operation. The VARS operating conditions should be such that the salt concentration in the liquid solution never exceeds the crystallisation limit for continuous operation of the system.

(vi) Vapour pressure: The vapour pressure refrigerant/absorbent pair depends upon the physical nature of the constituting elements. A too high vapour pressure may lead to an unstable system demanding robust structure whereas a high vacuum may cause leakage to the system. A refrigerant/absorbent pair with relatively high vapour pressure is suitable for low temperature operation and vice versa.

(vii) Chemical stability: It is one of the major requirements for thermal equilibrium of any thermally driven system. Chemical bond between absorbent and refrigerant must be stable even in severe physical condition, for smooth and longer operation of the system. Chemical impurities in the working fluids can cause detrimental effect in the system leading to formation of scaling inside equipment wall and unwanted gas etc.

(viii) Heat capacity: A high heat capacity of the refrigerant/absorbent pair is required to maintain low circulation ratio of the working fluid as it ensures high system performance.

(ix) The refrigerant/absorbent pair should be non toxic, non–inflammable and non corrosive to the environment.

1.5 Refrigerant/ absorbent pairs used in VARS

Water is considered to be a good natural refrigerant and eventually most of the VARSs uses water as refrigerant [60]. Marcriss et al. [61] mentioned about 40 various refrigerant compounds and 200 absorbent compounds in his article. Among them, aqua ammonia (NH_3–H_2O) and water–lithium bromide (H_2O–LiBr) refrigerant/absorbent pairs are very common working fluid pairs in VARS [55, 62, 63]. Water–LiBr VARS is suitable for air conditioning application; however, ice formation at low temperature and crystallization of LiBr at moderate concentration are some major problems associated with this system. Aqua ammonia VARS on the other hand is mainly used for large capacity industrial applications where low temperature is required for process cooling. Some additional component such as analyzer, rectifier etc. is required in aqua ammonia VARS and this adds to the system complexity [38]. There is also chance of corrosion taking place if copper pipes are used in the system. Fire risk also increases at high ammonia concentration exceeding 25% (by volume) in air.

The search for newer and better alternative refrigerant–absorbent pairs is on to realize improved system performance. Many other refrigerant/ absorbent pairs had been investigated in various research studies. These include Acetone–zinc bromide [64], water–monomethylamine [65], water–potassium formate [66], water–[lithium bromide + potassium formate] [67], ammonia–lithium nitrate [68], ammonia–sodium thiocyanate [68], methanol–lithium bromide [69], methanol–lithium chloride [70], R134a–dimethyl acetamide [71], water–[lithium bromide + lithium chloride + Zinc chloride] combination [72]; Water–[lithium bromide + potassium formate + sodium formate + potassium acetate and sodium lactate etc. H_2O–LiCl solution pair is another potential candidate with advantage of triple state point (solid, liquid and vapor form), long–term stability in the regeneration process under atmospheric conditions, comparatively less cost and better cycle performance [74–75]. Many experimental studies have been carried out to determine properties of H_2O–LiCl solution pair and mathematical correlations are developed through curve fitting of experimental results [76–78]. In fact when some reasonably accurate empirical correlations are developed by researchers for calculation of thermodynamic properties of such binary mixtures, this is done with the purpose that these might benefit others in carrying out theoretical performance studies of VARS using these binary mixtures as working fluids.

1.6 Exergy analysis and its importance in analyzing thermal systems

Exergy of a system is a composite property linked with the state of the surroundings and it indicates the extent of departure of a system from the equilibrium state. It is the measure of quality and usefulness of energy. It refers to the maximum useful work that can be obtained from a system when it reaches thermal, mechanical and chemical equilibrium with the reference environment.

In most of the cases, the performances of thermal systems are measured in terms of thermal efficiency, specific fuel consumption, coefficient of performance (COP) etc. These are basically first law based performance parameters which are determined through application of energy balance to the system components. The first law based performance parameters don't provide the actual measure of performance as they ignore, in their definitions, the best possible performance of a system under reversible conditions. Thermodynamic analysis based on first law of thermodynamics is called energy analysis. Energy analysis alone is not sufficient to evaluate some features of energy resource utilization as it provides only the quantitative measurement and completely ignores the qualitative aspect of it. It is also not possible, through energy analysis, to have a deeper insight into the system's operation. In order to overcome these limitations of energy analysis and also to evaluate the system performance under reversible conditions, often thermodynamic analysis is carried out on the basis of second law of thermodynamics which is called second law analysis or exergy analysis. Like energy balance in energy analysis, exergy balance is applied in exergy analysis to evaluate exergy destruction (or irreversibility) in various system components and the second law (exergy) efficiency of a thermal system. Any attempt to reduce system irreversibility would result in better performance through a more efficient utilization of energy resources. Therefore, to reduce irreversibility, it is necessary to quantify them for which the exergy analysis is needed [79] and considered more appropriate for performance assessment of thermal systems and energy conversion devices. It is mainly applied to design, evaluate and optimize thermal energy systems to improve their performance. In exergy analysis, the best possible performance of the system under reversible conditions is evaluated through exergy or the second law efficiency. Often energy and exergy analyses are performed together for analyzing thermal systems**.** Combination of energy and exergy analysis is a better approach of performance assessment as it gives a complete depiction of system characteristics [80]. Through exergy analysis it is possible to –

(i) Determine magnitudes, location and causes of irreversibility in a thermal system,

(ii) Analyze the effect of various design, operating and thermodynamic parameters on the exergy destruction,

(iii) Specify the maximum possible performance of thermal systems and identify those aspects of processes that are significant to overall performance, and

(iv) Propose methods for reduction of exergy destruction.

Thus, exergy analysis offers the system designers a plenty of scope for improvement of system operations.

1.7. Inverse analysis

The inverse problem uses the results of a direct/forward problem to estimate the values of some unknown operating or design parameters of a given system under consideration. In inverse analysis, it is assumed that the end results (outputs) are known while the parameters which affect the end results are assumed unknown and estimated. Therefore, it is called inverse of the forward problem. In the direct/forward problem, usually the input parameters are specified which can later be varied to evaluate the effect of these parameters on system output. This is generally known as parametric analysis and through such analysis, certainly it is possible to have a fair idea about the system performance behavior with the changing input parameters. In an inverse analysis however, these parameters are assumed unknown and the unknown parameters are estimated against the known values of the system performance parameters. As such, the inverse problems are more object oriented and it is found that in most of the cases, multiple combinations of parameters satisfy a given objective function/set of objective functions (in this case the outputs). This notable characteristic feature of inverse analysis makes it completely a different approach from that of the direct/forward method. The fact that a multiple set of parameters give the same output can be shown only through inverse analysis which is considered as a major advantage over conventional parametric analysis. Inverse analysis however cannot be performed in isolation. First the forward model is developed and then it is coupled with the inverse model for estimating the unknown parameters. Thus, the combined inverse and the forward model offers lot of flexibility at the designer's hand in selecting the most suitable combination of parameters in satisfying a given set of objective functions. Therefore, the inverse analysis is quite beneficial for obtaining suitable combinations of input parameters which otherwise can't be found out directly from the direct/forward model. Nowadays, inverse techniques are applied for resolving a wide range of engineering problems [81]. Say for example, in heat transfer problems, it can be applied for prediction of unknown boundary temperature or the boundary heat flux against some known temperature at a given location. Inverse analysis is also performed for characterization of the unknown material properties. It is quite an efficient mathematical tool which can be used for estimating operating and design parameters of a given system. The only disadvantage is that they are complex and computationally expensive as they require use of some optimization methods and hence, inverse problems are mathematically more challenging compared to their forward counterpart [82, 83]. Although the inverse analysis is done by using some optimization technique, but it is different from general single/multiple–objective optimization studies where the objective functions are either maximized or minimized as per the optimization requirement. In general single/multiple–objective optimization problems, depending upon the nature of the problem, the objective functions are either maximized or minimized simultaneously and accordingly the optimal decision variables are determined [81]. Say for example, the objective functions in a refrigeration system could be its COP, exergy efficiency and the total system irreversibility. In a general optimization study, the COP, exergy efficiency would require maximization and total system irreversibility would require minimization which is not the case in inverse analysis. Like in the inverse problem, in a general optimization problem, it is not assumed that the objective function values are known a priori, instead they are obtained through optimization of the decision variables. This is how the inverse analysis is different from a general optimization problem. Although it uses an optimization technique but the purpose is different.

1.8 Single/multiple–objective optimization of thermal systems

In many research studies, the performance of thermal systems is maximized by using optimization methods. For this purpose, optimization methods such as nonlinear programming (NLP), simplex search method; conjugate–gradient method, genetic algorithm (GA), differential evolution (DE), particle swarm optimization (PSO) etc. are used to optimize the system performance through determination of optimal parameters. Sometimes, optimization is also done to identify the most efficient/economical system design configuration. To some extent, parametric analysis also helps identifying optimal operating parameters. Although it is possible to have some idea about the optimum system performance from parametric analysis, but certainly, optimization is a better approach as it provides the optimal combinations of operating parameters more precisely than the conventional parametric analysis.

Contrary to single objective optimization where a single objective function is considered, multi–objective optimization is more convenient for optimizing engineering problems dealing with conflicting objectives. In this regard, evolutionary based search algorithms such as GA, DE, PSO are more suitable compared to the conventional gradient based search methods such as NLP, simplex search, conjugate gradient method etc. GA is an evolutionary heuristic search and evolutionary–based search techniques are well known for their ability to deal with nonlinear and complex optimization problems [84]. The primary advantage of evolutionary algorithms over other conventional optimization techniques is that they just require the objective function values, while properties, such as differentiability and continuity are not required [81].

1.9 Motivation and Research Objectives

From the discussion in the preceding sections, it was clear that VARS is one of the most important refrigeration systems used in HVAC industry. The most widely used refrigerant and absorbent pairs in VARS are the NH_3–H_2O and H_2O–LiBr. However, investigations are being done in search for newer refrigerant–absorbent pairs and also to analyze VARS performance with working fluid pairs other than NH_3–H_2O and H_2O–

LiBr [64–67, 70–73]. H_2O–LiCl is also being considered for use in VARS for chilling or air conditioning application. It is relatively cheaper, in fact the cost of H_2O–LiCl is approximately about half of H_2O–LiBr [75]. So, from cost point of view, H_2O–LiCl has the advantage over H_2O–LiBr. Moreover, use of H_2O–LiCl pair in a VARS provides low flow ratio (FR) and better system performance compared to H_2O–LiBr under identical operating conditions [51, 72, 74, 75, 85–89]. Thermodynamic properties of H_2O–LiCl solution pair in the form empirical relations are available [76–78]. H_2O–LiCl has high vapor pressure than H_2O–LiBr and therefore, under the conditions of same vapor pressure and solution concentration, the temperature would be low in case of H_2O–LiCl solution pair. Hence, it is possible to operate a H_2O–LiCl VARS with relatively lower heat source temperature. However, with high heat source temperature, depending on other operating conditions, sometimes, it may pose operational difficulty due to increased risk of crystallization [51].

A few VARS performance analysis has been done using H_2O–LiCl solution pair [85, 86, 72, 51]. These research works are however related to energy analysis which alone is not sufficient to evaluate features of energy resource utilization as it provides only the quantitative measurement and completely ignores the qualitative aspect of it. Energy analysis does not provide the designer the best insight into the system's operation. Exergy analysis based on second law is must if someone desires to evaluate the source of inefficiency and irreversible losses occurring in various system components. Any attempt to reduce system irreversibility would result in better performance through efficient utilization of energy resources. Therefore, to reduce irreversibility, it is necessary to quantify them through exergy analysis as it offers the system designers a plenty of scope for improvement of system operations [17].

Exergy analysis of H_2O–LiBr VARS has been performed in various studies [90–95]. Articles on exergy analysis of NH_3–H_2O systems are also available in the literature [96, 97, 98]. In so far as H_2O–LiCl system is concerned, exergy analysis of H_2O–LiCl VARS is neither available nor it was attempted before to evaluate its exergetic performance analysing the effect of operating temperatures on exergy destruction of individual components or the overall H_2O–LiCl VARS as a whole.

Crystallisation characteristics of a particular aqueous salt solution in a VARS are governed by its component operating temperatures. Therefore, a VARS designer cannot arbitrarily choose operating temperatures for the generator, condenser, evaporator and absorber of the VARS. In articles [76–78, 85], the formulations for thermodynamic properties of H_2O–LiCl solutions were provided for the composition range from pure water to 50% wt. Grover et al. [85] also took the upper limit of solution concentration as 51% and accordingly the operating temperatures were selected. All combinations of operating temperatures don't fulfil this criterion of solution concentration ≤51%. Therefore, selecting the appropriate combination of operating parameters is of paramount importance in absorption refrigeration systems not only from the performance point of view but also to avoid crystallization. This is possible through inverse analysis. Inverse analysis, which uses optimization technique, is an efficient mathematical tool that can used for estimating system's operating parameters against a known system parameter. Unfortunately, inverse analysis, using optimization technique, for estimation of operating parameters of single effect H_2O-LiCl based VARS was never done earlier in any of the previous research studies.

A number of VARS configurations are available starting from half effect to multi–effect (double and triple) systems [49, 52, 53, 55]. Over the years, double effect H_2O–LiBr absorption refrigeration systems have been analyzed in various studies and a good number of research articles on performance analysis of double effect H_2O–LiBr systems [49, 52–53, 55, 99–111] are available in the literature. In so far as H_2O–LiCl is concerned, not much articles are available in the literature and as such, research studies on H_2O–LiCl based VARS performance analysis are limited. Few studies [51, 76–78, 88–102] on single effect H_2O–LiCl VARS are however available and these studies have confirmed better system performance in respect of H_2O–LiCl based VARS compared to H_2O–LiBr under identical operating conditions. Thermodynamic energy and exergy analyses particularly on double effect H_2O–LiCl VARSs are limited [51, 86, 112]. Not many research articles were found related to thermodynamic performance evaluation and comparison of H_2O–LiCl based double effect series, parallel and reverse parallel systems, neither from energetic nor from exergetic point of view. Further, optimization study using evolutionary based optimization technique was not conducted on double effect VARS configurations neither with H_2O–LiCl nor with H_2O–LiBr as solution pairs.

It was felt that a detail thermodynamic modelling and analysis of H_2O–LiCl operated single effect, double effect series, double effect parallel and double effect reverse parallel VARS configurations could be appropriate for the purpose of the current research. Through a detailed parametric analysis, it would be possible to study and understand the effect of operating parameters on the energetic and exergetic performance of all the above system configurations operated with relatively a newer working fluid pair i.e. H_2O–LiCl. Further, a performance comparison of the single and double effect VARS configurations separately with H_2O–LiCl and H_2O–LiBr is possible as it would indicate the details regarding performance of the systems with the two solution pairs. Identification of optimal operating parameters is also crucial for obtaining maximum performance from double effect VARS configurations when their performances are governed by many operating parameters. Particularly with the two different working fluid pairs (H_2O–LiCl and H_2O–LiBr) with different properties, certainly the optimal operating conditions would not be identical. In this regard, research studies on optimization of double effect VARS configurations using evolutionary based search algorithm was not found in the literature neither for the H_2O-LiBr nor for H_2O-LiCl operated systems. Therefore, to address the knowledge gap associated with exergetic and optimal performances of single and double effect H_2O–LiCl VARS configurations and also to enumerate their performance differences with H_2O–LiBr counterparts. This research work considers fulfillment of the following objectives.

(i) To perform energy and exergy based parametric analyses of a single effect H_2O–LiCl operated VARS with operating temperatures estimated through inverse analysis considering the weak solution concentration at absorber exit as objective function.

(ii) To provide performance comparison between single effect H_2O–LiCl and H_2O–LiBr operated VARS under identical operating conditions.

(iii) To analyze the energetic and exergetic performances of H_2O–LiCl operated double effect VARS configurations (series, parallel and reverse parallel) and identify the optimal operating parameters through parametric variation.

(iv) To provide performance comparison between H_2O–LiCl and H_2O–LiBr operated double effect VARS configurations under identical operating conditions.

(v) To optimize the performances of H_2O–LiCl and H_2O–LiBr operated double effect series and parallel VARS configurations using a GA based optimization method and also to provide a comparative assessment between H_2O–LiCl and H_2O–LiBr based systems at the optimized operating conditions.

1.10 Outline of the thesis chapters

The thesis consists of total seven (7) chapters. Introduction is provided in the first chapter where the motivation and research objectives of the current research are highlighted at the end. The remaining six chapters are organized as follows:

• Chapter 2 presents a detailed review of previous studies related to thermodynamic performance analyses and optimization of single and multi effect VARS configurations operated with various refrigerant/absorbent pairs. The scope of the present research work is highlighted at the end.

• Chapter 3 presents the thermodynamic modeling and a detail energy and exergy based parametric analysis of the single effect H_2O–LiCl operated VARS. First the component operating temperatures of the LiCl–H_2O based single effect VARS are estimated using an inverse technique with weak solution concentration at absorber exit as objective function such that it is always within 50% limit. For this, purpose, a DE based optimization algorithm is used for objective function minimization. Total 34 combinational temperatures are obtained and for each of these combinations, the VARS performance results are obtained and presented. Parametric analysis is performed to show system COP, exergy efficiency and total irreversibility variation with component temperatures. The performance comparison between the LiCl–H_2O and LiBr–H_2O systems under identical operating conditions is also provided in this chapter.

• Chapter 4 describes the thermodynamic modeling of double effect series, parallel and reverse parallel VARS configurations. Energy analysis is done in this chapter to evaluate and compare the performances of the three double effect VARS configurations with H_2O–LiCl as the refrigerant/absorbent pair. A parametric analysis shows the performance variation of the three configurations with LPG and HPG temperatures at four cases of fixed evaporator, condenser and absorber temperatures. The parametric analysis is done with lot of maneuvering in the computer simulation programs through (i) simultaneous change in LPG temperature (T_{LPG}) and HPG temperature (T_{HPG}) and (ii) change in T_{HPG} at fixed T_{LPG} to find out the optimal T_{LPG}, T_{HPG} and additionally the distribution ratio in case of the parallel configuration for various cases.

Further, the performances of the double effect H_2O–LiCl systems are compared with their counterparts operated with H_2O–LiBr pair under identical operating conditions. Details regarding performance of double H_2O–LiCl VARS configurations and their operational difference with corresponding double effect H_2O–LiBr VARS configurations are also highlighted in this chapter.

• In Chapter 5, the exergy based parametric analysis is carried out to evaluate and compare the exergetic performances of the H_2O–LiCl operated double effect series, parallel and reverse parallel VARS configurations. The exergy efficiency and total system irreversibility of the double effect VARS configurations are evaluated through parametric variation of component temperatures and distribution ratio in a similar manner as in Chapter 4. Further, the exergetic performances of the H_2O–LiCl based double effect series, parallel and reverse parallel systems are compared with their H_2O–LiBr counterparts under identical operating conditions.

• In Chapter 6, multi–objective optimization of the series and parallel flow type double effect absorption refrigeration systems is presented. An evolutionary based optimization algorithm (GA) is used to find the optimal solutions and the Pareto–optimal fronts. The optimization is done considering COP, exergy efficiency and the total system irreversibility rate as objective functions. The LPG and HPG temperatures are taken as decision variables in the series configuration while for the parallel system; additionally the distribution ratio is also taken as a decision parameter. The optimization is done for four different cases of fixed evaporator, absorber and condenser temperatures. For each case, the optimal decision parameter values are determined for both the H_2O–LiCl and H_2O–LiBr operated double effect series and parallel configurations.

• Chapter 7 summarizes the important observations and conclusions made in this research work. The possible scope of future research in the field of vapour absorption refrigeration systems is highlighted at the end of this chapter.

Bibliography

[1] Dalkilic, A. S. and Wongwises, S. A performance comparison of vapour compression refrigeration system using various alternative refrigerants. *International communication in Heat and Mass Transfer,* 37(9):1340–1349, 2010.

[2] Padilla, M., Revellin, R., and Bonjour, J. Exergy analysis of R404a and R507a refrigerant mixtures in a experimental double–stage vapour compression plant. *Energy Conversion and Management,* 51(11):2195–2201, 2010.

[3] Llopis, R., Torella, E., Cabello, R., and Sanchez, D. Performance evaluation of R413a as replacement of R12 in a domestic refrigeration system. *Applied Energy,* 87(5):1546–1553, 2010.

[4] Lee, T. S., Liu, C. H., and Chen, T. W. Thermodynamic analysis of optimal condensing temperature of cascade–condenser in CO_2/NH_3 cascade refrigeration systems. *International Journal of Refrigeration,* 29(7):1100–1108, 2006.

[5] Getu, H. M. and Bansal, P. K. Thermodynamic analysis of an R744–R717 cascade refrigeration system. *International Journal of Refrigeration,* 31(1):45–54, 2008.

[6] Bingming, W., Huagen, W., Jianfeng, L., and Ziwen, X. Experimental investigation on the performance of NH_3/CO_2 cascade refrigeration system with twin–screw compressor. *International Journal of Refrigeration,* 32(6):1358–1365, 2009.

[7] Dopazo, J. A. and Seara, J. F. Experimental evaluation of a cascade refrigeration system prototype with CO_2 and NH_3 for freezing process applications. *International Journal of Refrigeration,* 34(1):257–267, 2011.

[8] Dopazo, J. A., Seara, J. F., Sieres, J., and Uhia, F. J. Theoretical analysis of a CO_2–NH_3 cascade refrigeration system for cooling applications at low temperatures. *Applied Thermal Engineering* 29(8–9):1577–1583, 2009.

[9] Ming, M., Yu, J., and Wang, X. Performance evaluation and optimal configuration analysis of a CO_2/NH_3 cascade refrigeration system with falling film evaporator–condenser. *Energy Conversion and Management,* 79:224–231,

2014.

[10] Aminyavari, M., Najafi, B., Shirazi, A., and Rinaldi, F. Exergetic, economic and environmental (3E) analyses, and multi objective optimization of a CO_2/NH_3 cascade refrigeration system. *Applied Thermal Engineering,* 65(1–2):42–50, 2014.

[11] Dincer, I. and Kanoglu, M. *Refrigeration systems and applications.* John Wiley and Sons, 2nd Edition, 2010.

[12] Kairouani, L. and Nehdi, E. Cooling performance and energy saving of a compression–absorption refrigeration system assisted by geothermal energy. *Applied Thermal Engineering*, 26(2–3):288–294, 2006.

[13] Fernandez–Seara, J., Sieres, J., and Vazquez, M. Compression–absorption cascade refrigeration system. *Applied Thermal Engineering*, 26(5–6): 502–512, 2006.

[14] Garimella, S., Brown, A. M., and Nagavarapu, A. K. Waste heat driven absorption/vapour–compression cascade refrigeration system for megawatt scale, high–flux, low–temperature cooling. *International Journal of Refrigeration*, 34(8):1776–1785, 2011.

[15] Cimsit, C. and Ozturk, I. T. Analysis of compression absorption cascade refrigeration cycles. *Applied Thermal Engineering*, 40:311–317, 2012.

[16] Jain, V., Kachhwaha, S. S., and Sachdeva, G. Thermodynamic performance analysis of a vapour compression–absorption cascaded refrigeration system. *Energy Conversion and Management*, 75:685–700, 2013.

[17] Cengel, Y.A. and Boles, M.A. *Thermodynamics–An engineering approach.* Fifth ed.: Tata McGraw Hill, New Delhi, 2006.

[18] Decker, L. O. Consider the cold facts about steam–jet vacuum cooling. *Chemical Engineering Progress,* 89(1):74–77, 1993.

[19] Sun, D. W. Variable geometry ejectors and their applications in ejector refrigeration systems. *Energy*, 21(10):919–929, 1996.

[20] Sun, D. W. Thermodynamic design data and optimum design maps for absorption refrigeration systems. *Applied Thermal Engineering*, 17(3):211–221, 1997.

[21] Sun, D. W. Evaluation of a combined ejector–vapour–compression refrigeration system. *International Journal of Energy Research*, 22(4):333–342, 1998.

[22] Chunnanond, K. and Aphornratana, S. Ejectors: applications in refrigeration technology. *Renewable and Sustainable Energy Reviews*, 8(2):129–155, 2004.

[23] Alexis, G. K. Exergy analysis of ejector–refrigeration cycle using water as working fluid. *International Journal of Energy Research*, 29(2): 95–105, 2005.

[24] Yapici, R. and Ersoy, H. K. Performance characteristics of the ejector refrigeration system based on the constant area ejector flow mode. *Energy Conversion and Management*, 46(18–19): 3117–3135, 2005.

[25] Yapici, R. and Yetisen, C. C. Experimental study on ejector refrigeration system powered by low grade heat. *Energy Conversion and Management*, 48(5):1560–1568, 2007.

[26] Alexis, G. K. Performance parameters for the design of a combined refrigeration and electrical power cogeneration system. *International Journal of Refrigeration*, 30(6):1097–1103, 2007.

[27] Sankarlal, T. and Mani, A. Experimental investigation on ejector refrigeration system with ammonia. *Renewable Energy*, 32(8): 1403–1413, 2007.

[28] Yu, J., Chen, H., Ren, Y., and Li, Y. A new ejector refrigeration system with an additional jet pump. *Applied Thermal Engineering*, 26(2–3):312–319, 2006.

[29] Zhu, Y. and Jiang, P. Hybrid vapour compression refrigeration system with an integrated ejector cooling cycle. *International Journal of Refrigeration*, 35(1): 68–78, 2012.

[30] Chaobin, D., Nakamura, Y., and Hihara, E. Study on Ejector–Vapour Compression Hybrid Air Conditioning System Using Solar Energy. In *International Refrigeration and Air Conditioning Conference*, Purdue, July 16–19, 2012.

[31] Jiang, L., Gu, Z., Feng, X., and Li, Y. Study of New Absorption–Ejector Hybrid Refrigeration System. In *International Refrigeration and Air Conditioning Conference*, Purdue, 2002.

[32] Goktun, S. Performance analysis of a heat engine driven combined vapour compression–absorption–ejector refrigerator. *Energy Conversion & Management*, 41(17):1885–1895, 2000.

[33] Eames, I. W. and Wu, S. Experimental proof–of–concept testing of an innovative heat–powered vapour recompression absorption refrigerator cycle. *Applied Thermal Engineering* 20:721–736, 2000.

[34] Sun, D. W. Solar powered combined ejector–vapour compression cycle for air conditioning and refrigeration. *Energy Conversion & Management*, 38(5):479–491, 1997.

[35] Nguyen, V. M., Riffat, S. B., and Doherty, P. S. Development of a solar–powered passive ejector cooling system. *Applied Thermal Engineering*, 21(2):157–168, 2001.

[36] Alexis, G. K. and Karayiannis, E. K. A solar ejector cooling system using refrigerant R134a in the Athens area. *Renewable Energy*, 30(9):1457–1469, 2005.

[37] Abdulateef, J. M., Sopian, K., Alghoul, M. A., and Sulaiman, M. Y. Review on solar–driven ejector refrigeration technologies. *Renewable and Sustainable Energy Reviews*, 13(6–7):1338–1349, 2009.

[38] Talukdar, K. Thermodynamic modelling and analysis of steam turbine based cogeneration systems with single and double effect absorption cooling as bottoming cycles. Ph.D’s thesis, School of Engineering, Tezpur University, October 2017.

[39] Zhao, D. and Tan, G. A review of thermoelectric cooling: Materials, modeling and applications. *Applied Thermal Engineering*, 66:15–24, 2014.

[40] Wetzel, M. and Herman, C. Design optimization of thermoacoustic refrigerators. *International Journal of Refrigeration*, 20(1):3–21, 1997

[41] Piccolo, A. Optimization of thermoacoustic refrigerators using second law

analysis. *Applied Energy*, 103:358–367, 2013.

[42] Nouh, M., Aldraihem, O., and Baz, A. Piezo–driven thermoacoustic refrigerators with dynamic magnifiers, *Applied Acoustics* 83:86–99, 2014.

[43] Saechan, P. and Jaworski, A. J. Numerical Studies of Co–Axial Travelling–Wave Thermoacoustic Cooler Powered by Standing–Wave Thermoacoustic Engine. *Renewable Energy,* 2019. (DOI: 10.1016/j.renene.2019.02.059)

[44] Saechan, P. and Jaworski, A. J. Thermoacoustic cooler to meet medical storage needs of rural communities in developing countries–High pressure system. *Thermal Science and Engineering Progress,* 8:31–46, 2018.

[45] Tartib, L. K. Developing more efficient travelling–wave thermo–acoustic refrigerators: A review. *Sustainable Energy Technologies and Assessments* 31:102–114, 2019.

[46] Gopal, M. R. and Murthy, S. S. Performance of a metal hydride cooling system. *International Journal of Refrigeration*, 18(6):413–420, 1995.

[47] Satheesh, A., Muthukumar, P., and Dewan, A. Computational study of metal hydride cooling system. *International Journal of hydrogen energy,* 34:3164–3172, 2009.

[48] Qin, F., Chen, J., Lu, M., Chen, Z., Zhou, Y., and Yang, K. Development of a metal hydride refrigeration system as an exhaust gas–driven automobile air conditioner. *Renewable Energy* 32:2034–2052, 2007.

[49] Gebreslassie, B. H., Medrano, M., and Boer, D. Exergy analysis of multi–effect water–LiBr absorption systems: from half to triple effect. *Renewable Energy,* 35(8):1773–1782, 2010.

[50] Maryami, R. and Dehghan, A. A. An exergy based comparative study between LiBr/water absorption refrigeration systems from half effect to triple effect. *Applied Thermal Engineering,* 124:103–123, 2017.

[51] She, X., Yin, Y., Xu, M., and Zhang, X. A novel low–grade heat–driven absorption refrigeration system with LiCl–H_2O and LiBr–H_2O working pairs. *International Journal of Refrigeration,* 58:219–234, 2015.

[52] Farshi, L. G., Mahmoudi, S. M. S., Rosen, M. A., and Yari, M. A. Comparative study of the performance characteristics of double-effect absorption refrigeration systems. *International Journal of Energy Research,* 36:182–192, 2012.

[53] Farshi, L. G., Mahmoudi, S. M. S. and Rosen, M. A. Analysis of crystallization risk in double effect absorption refrigeration systems. *Applied Thermal Engineering,* 31(10):1712–1717, 2011.

[54] Talukdar, K. and Gogoi, T. K. Exergy analysis of a combined vapor power cycle and boiler flue gas driven double effect water–LiBr absorption refrigeration system. *Energy Conversion and Management,* 108:468–477, 2016.

[55] Arun, M. B., Maiya, M. P., and Murthy, S. S. Performance comparison of double effect parallel–flow series flow water–lithium bromide absorption systems. *Applied Thermal Engineering*, 21(12):1273–1279, 2001

[56] Kaita, Y. Simulation results of triple–effect absorption cycles. *International Journal of Refrigeration,* 25(7):999–1007, 2002.

[57] Gomri, R. *Thermal Issues in Emerging Technologies,* ThETA 2, Cairo, Egypt, Dec 17–20th, 2008.

[58] *ASHRAE Handbook–Fundamentals*, SI Edition, 2013.

[59] Arora, C. P. *Refrigeration and Air Conditioning.* 3rd Edition, Tata McGraw–Hill Education, India.

[60] Holmberg, P. and Berntsson, T. Alternative working fluids in heat transformers. *ASHRAE Trans*, 96:1582–1589, 1990.

[61] Marcriss, R. A., Gutraj, J. M., and Zawacki, T. S. Absorption fluid data survey: final report on worldwide data. Technical report ORLN/sub/8447989/3, Institute of Gas Technology, U. S Department of Energy, Chicago, 1988.

[62] Pongsid, S., Satha A., and Supachart C. A review of absorption refrigeration technologies. *Renewable and Sustainable Energy Reviews,* 5:343–372, 2001.

[63] Wu, S. and Eames, I. W. Innovations in vapour–absorption cycles. *Applied Energy*, 66:251–266, 2000.

[64] Ajib, S. and Karno, A. Thermo physical properties of acetone–zinc bromide for using in a low temperature driven absorption refrigeration machine. *Heat and Mass Transfer*, 45(1):61–70, 2008.

[65] Pilatowsky, I., Rivera, W., and Romero, R. J. Thermodynamic analysis of monomethylamine–water solutions in a single–stage solar absorption refrigeration cycle at low generator temperatures. *Solar Energy Materials and Solar Cells,* 70(3)**:**287–300, 2001.

[66] Riffat, S. B., James, S. E., and Wong, C. W. Experimental analysis of the absorption and desorption rates of HCOOK/H_2O and LiBr/H_2O. *International Journal of Energy Research*, 22(12):1099–1103, 1998.

[67] Lucas, D., A. Donate, M. Molero, C. Villasenor, J., and Rodriguez, J. F. Performance evaluation and simulation of a new absorbent for an absorption refrigeration system. *International Journal of Refrigeration*, 27(4):324–330, 2004.

[68] Ferreira, C. A. I. Thermodynamic and physical property data equations for ammonia–lithium nitrate and ammonia–sodium thiocyanate solutions. *Solar Energy,* 32(2):231–236, 1984.

[69] Safarov, J. T. The investigation of the (p, r, T) and (ps, rs, Ts) properties of (1–x) CH_3OH + xLiBr for the application in absorption refrigeration machines and heat pumps. *Journal of Chemical Thermodynamics*, 35:1929–1937, 2003.

[70] Safarov, J. T. Study of thermodynamic properties of binary solutions of lithium bromide or lithium chloride with methanol. *Fluid Phase Equilibrium*, 236(1–2):87–95, 2005.

[71] Muthu, V. Saravanan, R., and Renganarayanan, S. Experimental studies on R134a–DMAC hot water based vapour absorption refrigeration systems. *International Journal of Thermal Science,* 47:175–181, 2008.

[72] Saravanan, R. and Maiya, M. P. Thermodynamic comparison of water–based working fluid combinations for a vapour absorption refrigeration system, *Applied Thermal Engineering,* 18(7):553−568, 1998.

[73] Donate, M., Rodriguez, L., Lucas, A. D. and Rodriguez, J. F. Thermodynamic evaluation of new absorbent mixtures of lithium bromide and organic salts for absorption refrigeration machines. *International Journal of Refrigeration*, 29:30–35, 2006.

[74] Olsson, R., Olsson, M., and Kaarebring, J. S. A chemical heat pump. 2000: World Patents Register. WO0037864.

[75] Gommed, K., Grossman, G., and Ziegler, F. Experimental Investigation of a LiCl–Water Open Absorption System for Cooling and Dehumidification. *Journal of Solar Energy Engineering,* 126:710–715, 2004.

[76] Conde, M. R. Properties of aqueous solutions of Lithium and Calcium chlorides: formulations for use in air conditioning equipment design. *International Journal of Thermal Science,* 43:367–382, 2004.

[77] Patek, J. and Klomfar, J. Solid–liquid phase equilibrium in the systems of LiBr–H_2O and LiCl–H_2O. *Fluid Phase Equilibrium,* 250:138–49, 2006.

[78] Patek, J. and Klomfar, J. Thermodynamic properties of the LiCl–H_2O system at vapour–liquid equilibrium from 273 K to 400 K. *International Journal of Refrigeration*, 31:287–303, 2008.

[79] Sarmah, P. Performance analysis of Solid oxide fuel cell (SOFC) integrated combined power cycles and parameter estimation using inverse method. Ph.D's thesis, School of Engineering, Tezpur University, October 2017.

[80] Gogoi, T. K. and Dixit, U. S. Basics and Applications of Thermal Engineering. In: Davim, J. (eds) *Introduction to Mechanical Engineering. Materials Forming, Machining and Tribology,* Springer, Cham, 2018.

[81] Gogoi, T. K. Estimation of operating parameters of a water–LiBr vapour absorption refrigeration system through inverse analysis. *Journal of Energy Resources Technology,* 138:022002, 2015.

[82] Gogoi, T. K. and Das, R. Inverse analysis of an internal reforming solid oxide fuel cell system using simplex search method. *Applied Mathematical Modelling*, 37:6994–7015, 2013.

[83] Gogoi, T. K. and Das, R. A combined cycle plant with air and fuel recuperator for captive power application. Part 2: Inverse analysis and parameter estimation. *Energy Conversion and Management*, 79:778–789, 2014.

[84] Kenneth, A. and Jong, D. Evolutionary computation: a unified approach. Cambridge (MA): MIT Press; 2006.

[85] Grover, G. S., Eisa, M. A. R., and Holland, F. A. Thermodynamic design data for absorption heat pump systems operating on water–lithium chloride–part one, Cool. *Heat Recovery Systems, CHP,* 8:33–41, 1988.

[86] Won, S. H. and Lee, W. Y. Thermodynamic Design Data for Double–Effect Absorption Heat Pump Systems Using Water–Lithium Chloride Cooling. *Heat Recovery System and CHP*, 11(1):41−48, 1991.

[87] Parham, K., Atikol, U., Yari, M., and Agboola, O.P. Evaluation and optimization of single stage absorption chiller using (LiCl+H_2O) as the working pair. *Advances in Mechanical Engineering,* 2013.

[88] Gunhan, T., Ekren, O., Demir, V., Hepbasli, A., Erek, A., and Sahin, A.S. Experimental exergetic performance evaluation of a novel solar assisted LiCl–H_2O absorption cooling system. Energy and Building, 68A:138–146, 2014.

[89] Bellos, E., Tzivanidis, C., and Antonopoulos, K. A. Exergetic and energetic comparison of LiCl–H_2O and LiBr–H_2O working pairs in a solar absorption cooling system. *Energy Conversion and Management,* 123:453–461, 2016.

[90] Sencan, A., Yakut, K. A., and Kalogirou, S. A. Exergy analysis of lithium bromide/water absorption systems. *Renewable Energy*, 30:645–657, 2005.

[91] Lee, F. S. and Sherif, S. A. Thermodynamic analysis of a lithium bromide/water absorption system for cooling and heating applications. *International Journal of Energy Research,* 25:1019−1031, 2001.

[92] Talbi, M. M. and Agnew, B. Exergy analysis: an absorption refrigerator using lithium bromide and water as working fluid. *Applied Thermal Engineering,* 20, 619−630, 2000.

[93] Kilic, M. and Kaynakli, O. Second law–based thermodynamic analysis of water–

lithium bromide absorption refrigeration system. *Energy,* 32:1505−1512, 2007.

[94] Gomri, R. and Hakimi, R. Second law analyses of double effect vapour absorption cooler system. *Energy Conversion and Management,* 49:3343–3348, 2008.

[95] Kaynakli, O., Saka, K., and Kaynakli, K. Energy and exergy analysis of a double effect absorption refrigeration system based on different heat sources. *Energy Conversion and Management,* 106:21–30, 2015

[96] Adewusi, S. A. and Zubair, S. M. Second law based thermodynamic analysis of ammonia–water absorption systems, *Energy Conversion and Management,* 45:2355–2369, 2004.

[97] Aman, J., Ting, D. S. K., and Henshaw, P. Residential solar air conditioning: energy and exergy analyses of an ammonia–water absorption cooling system. *Applied Thermal Engineering,* 62:424–32, 2014.

[98] Yari, M., Zarin, A., and Mahmoudi, S. M. S. Energy and exergy analyses of GAX and GAX hybrid absorption refrigeration cycles. *Renewable Energy,* 36:2011–20, 2011.

[99] Kaushik, S. C. and Chandra, S. Computer modeling and parametric study of a double effect generation absorption refrigeration cycle. *Energy Conversion and Management*, 25:9–14, 1985.

[100] Xu, G. P., Dai, Y. Q., Tou, K. W., and Tso, C. P. Theoretical analysis and optimization of a double–effect series–flow–type absorption chiller. *Applied Thermal Engineering*, 16:975–987, 1996.

[101] Xu, G. P. and Dai, Y. Q. Theoretical analysis and optimization of a double–effect parallel–flow–type absorption chiller. *Applied Thermal Engineering*, 17:157–170, 1997.

[102] Arun, M. B., Maiya, M. P., and Murthy, S.S. Equilibrium low pressure generator temperatures for double–effect series flow absorption refrigeration systems. *Applied Thermal Engineering*, 20:227–242, 2000.

[103] Liu, Y. L. and Wang, R. Z. Performance prediction of solar/gas driving double

effect LiBr–H_2O absorption system. *Renewable Energy*, 29:1677–1695, 2004.

[104] Torrella, E., Sanchez, D., Cabello, R., Larumbe, J. A., and Llopis, R. On–site real–time evaluation of an air–conditioning direct–fired double–effect absorption chiller. *Applied Energy*, 86:968–975, 2009.

[105] Yin, H., Qu, M., and Archer, D. H. Model based experimental performance analysis of a microscale LiBr–H_2O steam–driven double–effect absorption Chiller. *Applied Thermal Engineering*, 30:1741–1750, 2010.

[106] Ravikumar, T. S. Suganthi, L. and Anand, A. S. Exergy analysis of solar assisted double effect absorption refrigeration system. *Renewable Energy*, 14:55–59, 1998.

[107] Khaliq, A. and Kumar, R. Exergy analysis of double effect vapour absorption refrigeration system. *International Journal of Energy Research*, 32:161–174, 2008.

[109] Figueredo, G. R. Bourouis, M., and Coronas, A. Thermodynamic modelling of a two stage absorption chiller driven at two–temperature levels. *Applied Thermal Engineering*, 28:211–217, 2008.

[110] Kaushik, S.C. and Arora, A. Energy and exergy analysis of single effect and series flow double effect water–lithium bromide absorption refrigeration systems. *International Journal of Refrigeration*, 32:1247–1258, 2009.

[111] Gomri, R. Investigation of the potential of application of single effect and multiple effect absorption cooling systems. *Energy Conversion and Management*, 51:1629–1636, 2010.

[112] Bellos, E., Tzivanidis, C., Pavlovic, S., and Stefanovic, V. Thermodynamic investigation of LiCl–H_2O working pair in a double effect absorption chiller driven by parabolic trough collectors. *Thermal Science and Engineering Progress,* 3:75–87, 2017.