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IMPROVED VAPOUR COMPRESSION REFRIGERATION CYCLES: LITERATURE REVIEW AND THEIR APPLICATION TO HEAT PUMPS

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ABSTRACT

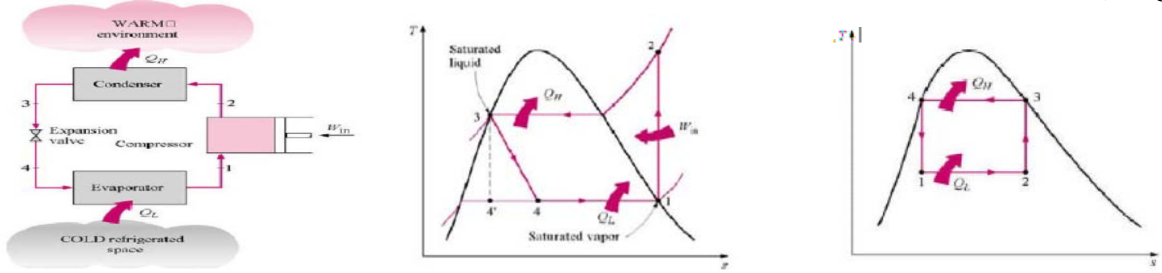
Hot water production as well as space heating using heat pumps is a proven energy efficient heating method. Heat pumps offer the advantages of reducing energy consumption, improving heating performance and reducing the negative effects on the environment compared with other heating methods. The further improvement of performance of vapour compression refrigeration cycles acting as heat pumps has been targeted by several researchers so that heat pumps will be able to achieve wider penetration into the building heating market. The major objective of this paper is to provide a comprehensive understanding of both simple and complex refrigeration cycles and to review the current status achieved in the performance of improved vapour compression refrigeration cycles. Suggestions for further improvement of performance of heat pumps refrigeration cycles and their components are also made.

1. INTRODUCTION

Energy consumption in buildings has become an important aspect on a global scale. Energy costs and environmental concerns have made energy optimisation a crucial issue for buildings. Therein energy efficiency is a prime mover in reducing global warming emissions. In respect to this, new technologies to conserve energy, to use energy effectively, to use alternative energy sources and to reduce the energy running costs of buildings such as solar energy, geothermal energy, wind energy etc. are under continuous development. The energy consumption of households in the UK is a major factor in the current environmental and strategic supply of energy with some 80% of domestic energy associated with space and hot water heating being provided by electrical radiators or boilers fired with fossil-fuel. The rapid escalation in energy costs, the issues of security of supply, the emission of polluting substances as well as global climate change, have all made heating methods in their current forms unsustainable at present and in the future. Therefore to overcome these problems, alternative heating solutions must be studied which focus on the reduction of energy consumption and the improvement of heating performance while reducing adverse effects on the environment. One such heating method is the use of heat pumps. Heat pumps are devices which have an increased efficiency when compared to other traditional heating methods and have the possibility of meeting the requirements of both an environmental and economic sustainable future with regard to domestic energy use. However in order to achieve this, the heat pump must become more flexible in its installation and operation. Thus a critical appraisal of the systems capable of meeting a more flexible installation in a retrofit application is necessary i.e. systems capable of using inexpensive heat sources (e.g. air) and delivering to existing higher temperature heating (hydronic) systems. This paper reviews theoretical and experimental studies of improved vapour compression refrigeration cycles capable of meeting this more flexible agenda.

2. THEORETICAL BACKGOUND

Vapour compression heat pumps are refrigeration systems whose operational cycle is based on the reversed Rankine cycle, requiring work input to accomplish their objective of transferring heat from a lower temperature source to a higher temperature sink as shown in Figure 1a. Their operational mode is virtually identical to that of refrigerators, being only different in its purpose. The low pressure liquid refrigerant is boiled by absorbing heat via an evaporator at the heat source. It is then compressed and passed through a condenser at the heat sink, where it is condensed and heat is released to the heat sink. The high pressure liquid refrigerant is then passed through the expansion device and returns to the heat source heat exchanger. To accomplish this, work is done on the working fluid through the use of a



(a) (b)

Figure 1: Vapour compression heat pumps with (a) Rankine and (b) Carnot cycles

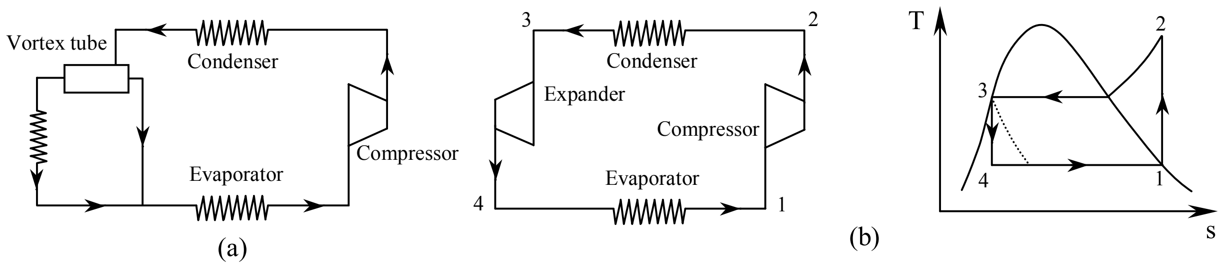


Figure 2: Cycles with (a) vortex tube and (b) with expander

compressor. From a thermodynamics point of view, the performance of a heat pump is evaluated by its heating capacity and coefficient of performance COP_h. The efficiency of the heat pump can also be defined in relation to that of the ideal heat pump operating within a Carnot cycle: $\xi = COP_h/COP_{cartnot}$.

3. IMPROVED SINGLE-STAGE CYCLES

It is known from theory that an ideal reversible cycle having the highest COP is the Carnot cycle which includes two isothermal and two adiabatic processes (Figure 1b). The expansion in most practical real vapour compression cycles occurring in an expansion valve is an irreversible throttling process. This causes a reduced cooling capacity and an increased work requirement in comparison with the Carnot cycle. Based on the principle of the temperature splitting phenomenon of a Ranque Hilsch vortex tube in which a stream of gas divides itself into a hot and cold flow, (identified as a natural heat pump mechanism) Hopper and Ambrose (1972) have recommended an improved expansion process with a replacement of the expansion valve by a Ranque Hilsch vortex tube as shown in Figure 2a. By this, the liquid refrigerant is throttled to the evaporator pressure while the vapour refrigerant is separated with hot and cold flows. The cold vapour along with the liquid is lead to the evaporator while the hot vapour continues rejecting its heat before reaching the evaporator. This leads to a lower quality of refrigerant, mostly at a point very close to state 1 in the Carnot cycle, entering the evaporator. The authors tested the cycle with thirteen refrigerants and showed different improved performances. However their experimental results did not elaborate on how great the improvement was. Ahlborm *et al.* (1998) produced a theoretical analysis explaining the establishment of a secondary circulation in the vortex tube. They demonstrated that a vortex tube satisfies three required processes employed in heat pumps: (i) the working fluid moves heat continuously between a high pressure and a low pressure region, (ii) the compressed working fluid is hotter than the surrounding medium to give off heat and (iii) the expanded working fluid is colder than the surroundings to absorb heat. Although the potential benefits of vortex tubes have been realized, its application to heat pumps has attracted little interest from the engineering community.

Power recovery from the expansion process could also be a means of improving the cycle's performance. Figure 2b shows a heat pump cycle with the expander as a throttling device. The expanders not only perform the isentropic throttling function ideally but also recover the work of expansion. Driver and Davidson (1996) developed a unique positive displacement compressor/expander applied to the heat pump and found a performance improvement of over 25% for R134a compared with the cycle using the conventional expansion devices. They also showed that the higher the compressor efficiency, the higher the potential performance improvement. Tamura *et al.* (1997) tested a high temperature heat pump with a screw compressor and expander separately, while Henderson *et al.* (2000) used a rotary vane compressor/expander as a replacement for both the expansion valve and the single compressor in a heat pump. Their analytical and experimental results all confirmed improvements in performance when using the

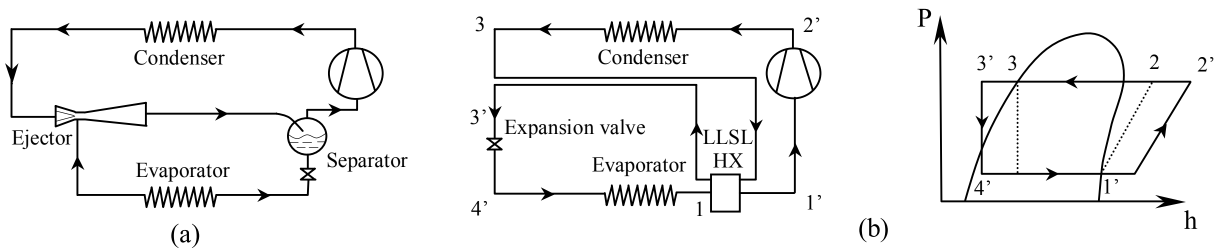


Figure 3: Cycles with (a) ejector and (b) liquid-line/suction-line heat exchanger

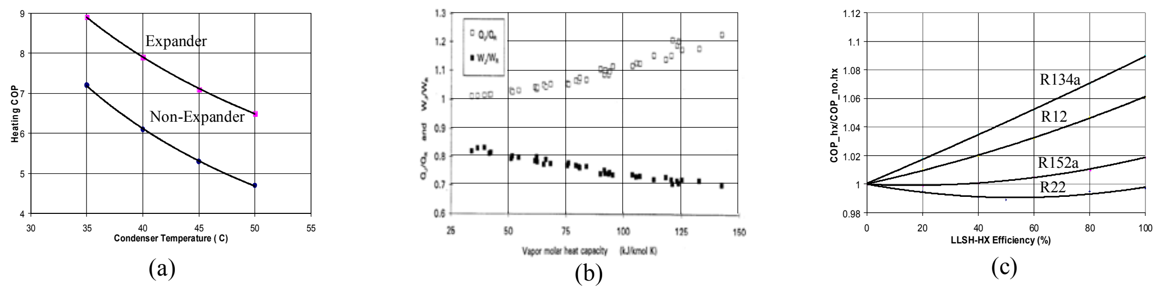


Figure 4: Cycle's performances with (a) expander, (b) ejector and (c) LLSL-HX (Henderson, 2000; Domaski, 1994)

expander as a throttling device (Figure 4a). However the expanders are expensive and are liable to suffer damage because of working with two-phase flow.

Another option for COP improvement is the use of an ejector as an expansion valve in the vapour compression cycle (Figure 3a). As the ejector uses the kinetic energy of the flash gas to increase suction pressure at the compressor inlet, so the ejector expansion cycle also reduces the irreversibilities in the expansion process. This result in reduced compression work and a larger quality difference in the evaporator compared to the conventional cycle. Kornhauser (1990) showed COP improvements of up to 13%, 21%, 20%, 17%, 24%, 22%, 30% and 12% over the conventional cycles at the same working conditions for R11, R12, R22, R113, R114, R500, R502 and R717, respectively. Harrell *et al.* (1995) found a range of improved COP from 3.9% to 7.6% for R134a. Similarly Domanski (1995) studied the ejector expansion cycle with 38 different refrigerants and found improvements in both the reduced work and increased capacity (Figure 4b). He also showed that the COP improvement of the ejector expansion cycle is sensitive to the ejector's efficiency. Menegay and Kornhauser (1996) upgraded further the COP by up to 9% for a heat pump with R12 by modifying the ejector's configuration with the addition of a bubble flow tube fitted upstream of the motive nozzle. Further benefits such as reduced pressure ratio and discharge temperature etc. have also been seen in the ejector expansion cycle (Disawas and Wongwises, 2005).

4. SUBCOOLING CYCLES

The basic heat pump cycle normally has a low degree of subcooling upstream of the expansion valve. This causes a high quality of refrigerant entering the evaporator. As a result of employing the liquid-line/suction-line heat exchanger (LLSL-HX) to the basic cycle (Figure 3b), the subcooling upgrades significantly. Moreover it helps to protect the compressor from two-phase flow if using the refrigerant flow after the evaporator for the subcooling. One of the most pertinent and clearly theoretical analyses of this cycle was reported by Domanski *et al.* (1994). The benefits of LLSL-HX installation depends on a combination of operating conditions and fluid properties such as heat capacity, latent heat and coefficient of thermal expansion with heat capacity being the most influential property. They claimed that for high heat capacity fluids having a low COP in the basic cycle, their performance improves with the LLSL-HX cycle significantly and can exceed the COP of the best performing fluids in the basic cycle. But with fluids of low heat capacity, the benefit from the installation of the LLSL-HX is not significant, or even degrades performance in comparison with the basic cycle, especially for R22 as shown in Figure 4c. Wood and Meyer (1999) tested the heat exchanger accumulator as a LLSL-HX in liquid overfeeding operation conditions. Their test results showed increases of 7.5% in the COP and 6.5% in capacity with reductions in cycling losses and power consumption. A further review of publications on the LLSL-HX cycle was published by Klein *et al.* (2000).

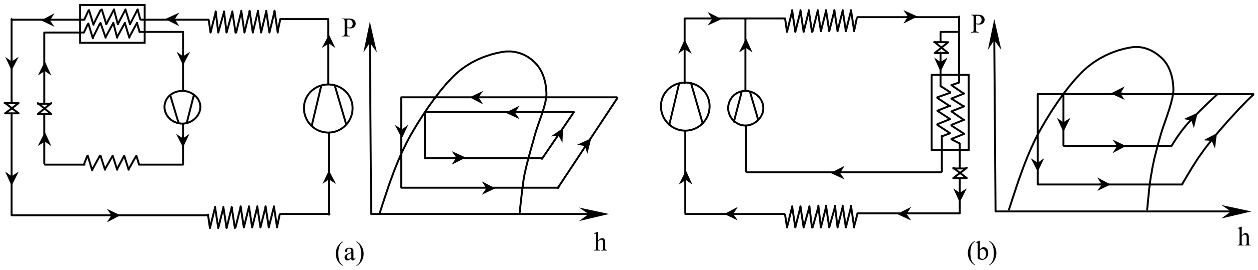


Figure 5: Dedicated (a) and integrated (b) mechanical subcooling cycles

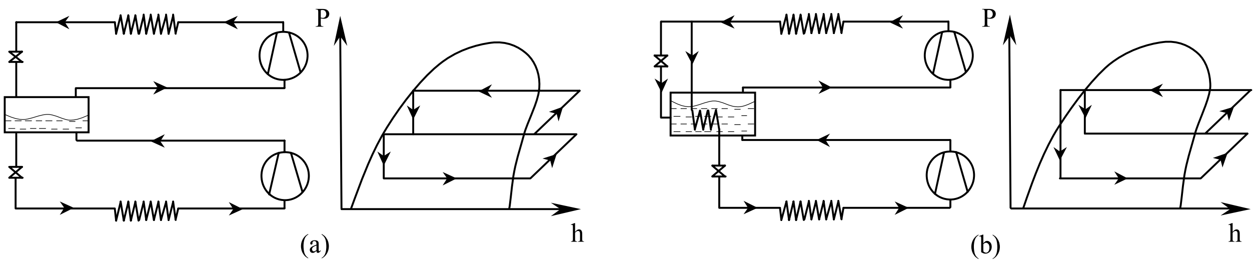


Figure 6: Two-stage cycles with (a) flash tank and (b) shell-and-coil economizers

The authors recommended a new dimensionless quantity to investigate the LLSL-HX effect on the performance with both mass flow rate and no mass flow corrections. It was found that the LLSL-HX is most efficient for refrigerantshaving a relatively small value of that dimensionless quantity and at high temperature lift. Desai *et al.* (1991) installed a secondary subcooler positioned between the first LLSL-HX and the main evaporator which is like a sub-evaporator in parallel with the main evaporator to provide further increased subcooling. This scheme is very beneficial and essential for the efficient use of a binary nonazeotropic refrigerant mixture.

Subcooling can also be gained by adding a mechanical subcooling loop to a basic cycle. The mechanical subcooling cycles include a main cycle and a subcooler cycle which are coupled together as shown in Figure 5. The dedicated mechanical subcooling cycle comprises of two condensers, one for the main cycle and one for the subcooler cycle, whereas the integrated mechanical subcooling cycle consists of only one condenser serving both the main cycle and the subcooler cycle. Couvillion *et al.* (1988) presented a detailed computational model to predict the overall performance of a dedicated mechanical subcooling cycle with several different refrigerant combinations. Thornton *et al.* (1994) and Khan and Zubair (2000a) evaluated this type of cycle's overall performance by using thermodynamic models based on the temperature and refrigerant's property dependences. They all showed an improvement of up to 80% in the COP and 170% in the capacity under certain operating conditions. These improvements mostly depend on the subcooler temperature. For the integrated mechanical subcooling cycle, improved performance was shown in Khan and Zubair's (2000b) study. Component irreversibilities also affect the mechanical subcooling cycle's performance. Based on irreversible analysis using both the first and second laws of thermodynamics, Zubair *et al.* (1996) pointed out a performance improvement of about 24% over the basic cycle.

5. IMPROVED TWO-STAGES CYCLES WITH ECONOMISER

When the temperature lift increases, heat pumps operating with a single-stage cycle become increasingly inefficient. This is particularly true for conventional air-source heat pumps operating in colder climates. These disadvantages may be overcome by operating the heat pump with a two-stage compression/expansion. Figure 6a shows a two-stage cycle utilizing a flash tank economizer. The flash tank has three functions: to separate the liquid and vapour phases, to desuperheat the discharge gas from the low pressure compressor and to cool the liquid from the condenser to the saturated temperature corresponding to the intermediate pressure. The system therefore has less power consumption and higher capacity due to the compression of flash gas only from the intermediate pressure which is higher than evaporating pressure and a lower quality of refrigerant entering the evaporator. Its exergetic efficiencies were 26% and 24% higher than these of the conventional single-stage cycle for R12 and R134a, respectively (Mastrullo and Mazzei, 1987, Zubair *et al.* 1996). Although the addition of extra components such as flash tank, expansion valves

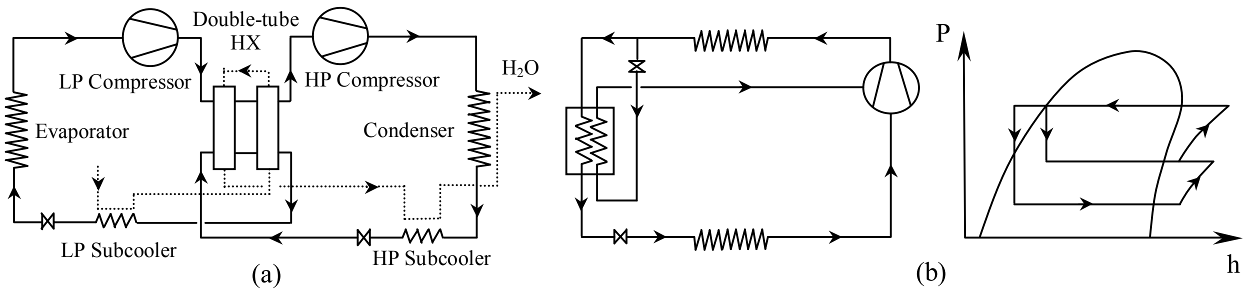


Figure 7: TCCH cycle proposed by Hasegawa (a) and EVI cycle (b)

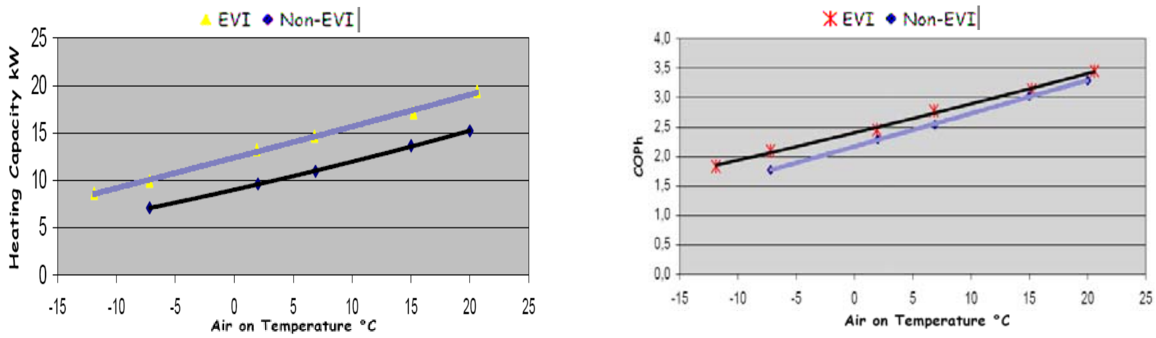


Figure 8: Performance of the EVI cycle at the delivering water temperature of 60°C (Hewitt *et al.,* 2006)

etc. causes additional sources of irreversibility, the total irreversible loss of the overall cycle is still lower than that of the single-stage cycle (Rossi *et al.* 1988, Nikolaidis and Probert, 1998). Khan and Zubair's (1998) thermodynamic models also showed better performance of the two-stage cycle with flash tank than the single-stage cycle. However its performance depends strictly on the inter-stage pressure and temperature (Prasad 1981-82, Gupta and Prasad 1984, Zubair *et al.* 1995, Tiedeman *et al.* 2003).

Because the refrigerant pressure in the flash tank economizer is reduced to the intermediate pressure, the liquid entering the evaporator is not really subcooled liquid. Furthermore the second expansion valve may have control difficulties due to the low pressure differential. An alternative cycle with the shell-and-coil economiser as shown in Figure 6b would overcome the above disadvantages. The refrigerant flow in this scheme is split into two portions: the larger portion of flow works between the condensing and evaporating pressures as in a single-stage cycle, while the smaller portion of flow only works between the condensing and intermediate pressures. From a thermodynamic point of view, this cycle has slightly lower performance due to the impossibility of the liquid intercooling as in the flash tank economizer but it still has exergetic efficiency of about 36.3% higher than the 29.3% of the conventional single-stage cycle (Mastrullo and Mazzei 1987, Rossi *et al.* 1988).

Hasegawa *et al.* (1996) developed a two-stage compression and cascade heat pump (TCCH) cycle with R12 for water heating that is basically similar to the conventional two-stage cycle but differs only in the structure of the intermediate component and the heated water line (Figure 7a). They used two identical vertical shell-and-tube heat exchangers in series connection to create an intermediate heat exchanger between high pressure and low pressure stages. The purpose of this intermediate heat exchanger is to preheat water and to subcool the liquid refrigerant as well as to separate the refrigerant vapour-liquid phases. By this modification, their TCCH cycle can produce hot water up to 65°C with a high COP of 3.73.

Refrigerant injected compressors either twin-screw or scroll types linked with a heat exchanger to establish alternative improved cycles have been employed by some researchers (Figure 7b). The so-called economized vapour injected (EVI) cycle used scroll compressors with vapour refrigerant injections offers many advantages such as the use of a small compressor operating at two stages, higher efficiency with less power, increased compressor lifespan due to lower discharge temperature, easier modulation of capacity etc. Based on its benefits, Ma *et al.* (2003) tested an EVI cycle heat pump for R22 with water as the condenser coolant and glycol solution as the cooling medium. An increased capacity of about 8.6% and improved performance of about 6% were found in their system working at a condensing temperature of 45^0C and evaporating temperature of -15^0C. Hewitt *et al.* (2006) recently designed and

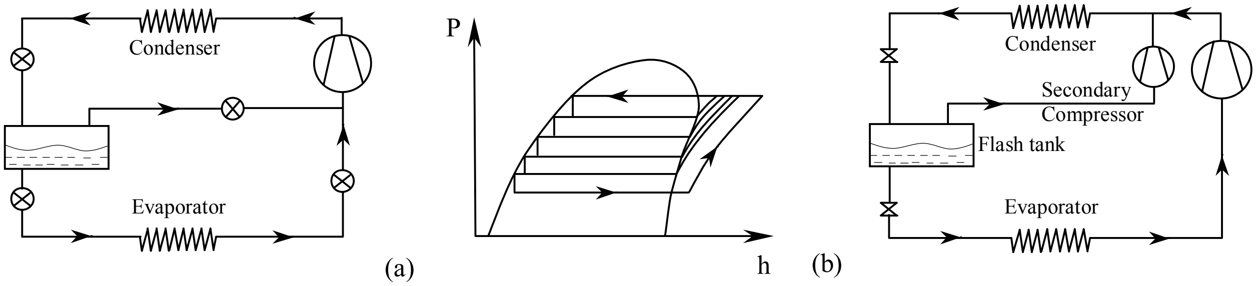


Figure 9: Multi-stage cycles proposed by (a) Granryd and (b) Hewitt

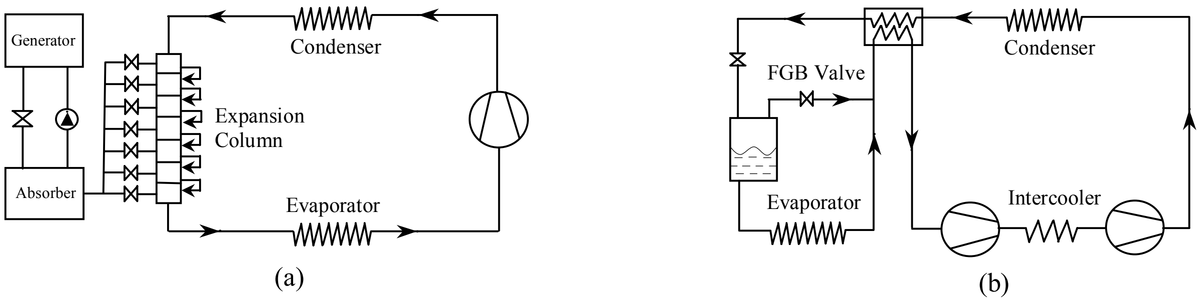


Figure 10: Cycles proposed by (a) Gryzagoridis and (b) Elbel

evaluated an air-source heat pump based on a scroll compressor operating with EVI cycle for hot water production. The results of their experiments shown in Figure 8 indicated that an averaged rise in capacity of 32.9% and COP of 8.52% for R407C were achieved at the delivering water temperature of 60°C. The further evaluations of benefits offered by the EVI cycle for heat pump/refrigeration applications were presented by Beeton and Pham (2003) and Taras (2005). They also reported that a typical performance augmentation by up to 10% could be expected.

6. IMPROVED MULTI-STAGE CYCLES

As previoulsy mentioned, the throttling device is one of the main causes of irreversible loss in cycle efficiency due to a portion of the refrigerant flashing to vapour during the expansion process. Other way to eliminate these throttling losses is to change the single-stage expansion to a multistage expansion where flash gas is removed after each throttling stage. This is achievable with a series of sequent smaller expansions and compressions to reduce the refrigerant pressure to the required pressure of evaporator. Granryd (1975) proposed such a cycle and applied the above multi-compressors effect, with only one compressor and one expansion vessel to make a series of compressions/expansions by periodically switching the main compressor to the expansion vessel (Figure 9a). His computational analyses for R12, R22 and R502 indicated that this cycle offered an increased theoretical capacity without much change in compressor driving power. Szargut *et al.* (1998) continued research on the Granryd cycle and found increased benefit with a saving of the power up to 20% for R22. Hewitt *et al.* (1991) researched a practical Granryd cycle applied to a heat pump by experimental methods, generating a 10% improvement in capacity and performance. Further improvements were not seen because the compressor in the Granryd cycle reduced pressure in the expansion vessel too rapidly to overcome the lack of nucleate boiling sites causing flash gas. Finally, the authors proposed the addition of a smaller secondary compressor performing only the cooling task of the liquid refrigerant in the expansion vessel as illustrated in Figure 9b. Although their modified practical Granryd cycle showed a positive increase in capacity, there was no improvement in performance due to the power requirement of the additional compressor.

Gryzagoridis and Browne (2001) developed a hybrid multi-expansion cycle by adding a small auxiliary absorption system and an expansion column into the vapour compression cycle as seen in Figure 10a. The absorption part would absorb the flash gas from each expansion stage and return it to the condenser using standard absorption techniques. With this hybrid multi-expansion cycle, the capacity increased by up to 10%, while the improvement in the performance was not significant. However if the auxiliary absorption system is operated by waste heat sources, the performance of this hybrid cycle would be higher than that of the conventional cycle. A recent research on the flash gas created after the expansion process was published by Elbel and Hrnjak (2004). Based on the Flash Gas

Bypass (FGB) concept, the authors suggested an idea to bypass refrigerant vapour after expansion around the evaporator. Their system is shown in Figure 10b. A flash tank was installed downstream of the expansion valve which is connected to the compressor suction through a FGB valve. By adjusting this valve, some fractions of refrigerant vapour were removed and the evaporator was fed with only pure liquid. This method brought improvements in the capacity and performance of up to 9% and 7% over that of the conventional cycle at the same operating conditions, respectively. The cycle's performance is very sensitive to the opening of the FGB valve, the authors however did not provide any control strategies.

7. CONCLUSION AND RECOMMENDATIONS

In this review, many alternative vapour compression cycles applied to heat pumps have been identified and presented. These vary from simple to complex cycles as well as from single-stage to multi-stage cycles, all emphasising any improved performance achieved. In general, the alternative cycles all offer several benefits to vapour compression heat pumps such as reduced losses, increased performance, decreased energy consumption etc. when compared to the conventional basic vapour compression heat pumps at the same capacity. Along with improved performance, the complexity of the cycles and the cost of systems increase due to the increased number of stages and the addition of new components. Currently, the economized cycle using a vapour injected compressor seems to be the most efficient for heat pumps for buildings both in its performance and compactness due to the operation of a two-stage cycle with only a single compressor and the versatility of modulating its capacity from a single-stage cycle to a two-stage cycle. Another option is to combine a vapour injected compressor with a flash tank to establish a further improved economized cycle because of the lower cost of the flash tank compared to that of a heat exchanger. Some enhanced techniques to promote more flash boiling of the liquid refrigerant in the flash tank are required in order to minimise the flash gas entering the evaporator. The use of dual-effect or multi-cylinder compressors in economized cycles is also another possible way to improve the cycle's performance. The application of multi-stage cascaded vapour compression cycles for heat pump also has high potential.

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