

A REVIEW ON SUB-COOLING IN VAPOR COMPRESSION REFRIGERATION CYCLE FOR ENERGY SAVING

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Article history

Received

22 March 2019

Received in revised form

27 June 2019

Accepted

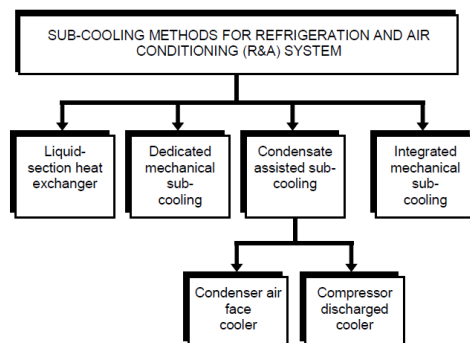
2 July 2019

Published online

26 August 2019

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Graphical abstract



Abstract

Vapor compression refrigeration cycle (VCRC) is widely used in refrigeration and air conditioning (R&A) systems. Sub-cooling is used to improve the coefficient of performance (COP) of the R&A system by enhancing the cooling capacity. This paper presents various sub-cooling methods, which have been established and applied to enhance the performance of the VCRC. In a simple cycle of VCRC, the exit of the condenser is at saturated liquid line. Further cooling of the exit condenser to the sub-cooled region can result in an increase in the cooling capacity due to low vapor quality refrigerant entering the evaporator. As a result, the refrigerant absorbs more heat in the evaporator. The lower the quality of the refrigerant entering the evaporator, the higher the cooling capacity that is produced by the evaporator. This cooling capacity improvement results in an increase in the COP. In the present study, four sub-cooling methods are reviewed, which are liquid-suction heat exchanger, dedicated mechanical sub-cooling, integrated mechanical sub-cooling and condensate assisted sub-cooling. The advantages and drawbacks of each method, as well as future research direction in this research domain were discussed in detail.

Keywords: Sub-cooling, COP, refrigerator, air conditioner, energy-saving

Abstrak

Kitar penyejukan mampatan wap (VCRC) digunakan dengan meluas dalam sistem penyejukan dan penyamanan udara (R&A). Sub-dingin pula digunakan untuk memperbaiki pekali prestasi (COP) sistem R&A dengan meningkatkan kapasiti penyejukan. Artikel ini membincangkan pelbagai kaedah sub-dingin yang telah terbukti dan digunakan di dalam memperbaiki prestasi VCRC. Dalam VCRC yang biasa, keluaran pemelwap berada pada garisan cecair tepu. Penyejukan lanjutan pada keluaran pemelwap ke kawasan sub-dingin boleh memberikan tambahan kapasiti penyejukan disebabkan oleh pecahan kekerangan bahan pendingin yang rendah memasuki penyejat. Kesannya, bahan pendingin menyerap lebih banyak haba di penyejat. Semakin rendah pecahan kekerangan bahan pendingin memasuki penyejat, semakin tinggi kapasiti penyejukan yang dihasilkan oleh penyejat. Peningkatan kapasiti penyejukan ini memberi kesan kepada peningkatan COP. Dalam kajian ini,

empat kaedah sub-dingin dibincangkan iaitu penukar haba sedutan-cecair, sub-dingin mekanikal khusus, sub-dingin mekanikal bersepadu dan sub-dingin berbantu air terpelwap. Kelebihan dan kekurangan bagi setiap kaedah, dan juga arah kajian pada masa depan di dalam domain penyelidikan ini juga dibincangkan dengan terperinci.

Kata kunci: Sub-dingin, COP, peti sejuk, penyamanan udara, penjimatan-tenaga

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1.0 INTRODUCTION

The demand for refrigerators/freezers and air conditioners will increase due to the increment of population and economic growth. A part from that, energy savings in the refrigeration and air conditioning (R&A) systems are important, given the increasingly expensive and limited energy resources. In the commercial buildings, the air conditioning systems consume approximately 50% of the total energy [1-2] whereas in the supermarket, refrigeration systems consume approximately 50-60% of the total energy consumption [3]. As a result, enhancement on performance of the R&A systems can generate a significant energy savings.

There are many methods of energy savings in the R&A systems, from simple to complex methods. Sukri *et al.* [4] discussed that performance of typical R&A systems, i.e. automotive air conditioning system that can be increased through component optimization and/or efficient operational management and control method. Figure 1 shows that sub-cooling control as one of the methods that can be utilized under efficient operational management and control method. It is one of the simple method that can be used to increase the cooling capacity (CC) of R&A systems. By increasing the CC improves the COP of the systems. In simple cycle of vapor compression refrigeration cycle (VCRC), the exit of the condenser is at saturated liquid line. If the condenser exit is further cooled to the liquid region, it causes an increase in the CC due to low quality refrigerant entering the evaporator, and consequently, the refrigerant absorbs more heat in the evaporator [5-9].

There are four methods of sub-cooling, i.e., liquid-suction heat exchanger (LSHX) sub-cooling, dedicated mechanical sub-cooling (DMS), integrated mechanical sub-cooling (IMS) and condensate assisted sub-cooling (CAS) [9-18]. In the LSHX sub-cooling, a heat exchanger is applied for the purpose of transferring heat from the exit of the condenser (liquid line) to the suction of the compressor (vapor line). In the simple VCRC, the exit of the condenser temperature is referred to condensing temperature and the suction temperature is referred to the evaporating temperature. Heat flows from the liquid line to the suction causes the refrigerant temperature in the exit of the condenser decreases, whereas the

refrigerant temperature in the suction increases. As a result, the use of LSHX is always followed by superheating in the suction of the compressor and sub-cooling in the exit condenser. The increase in superheating on the compressor suction causes an increase in the compressor discharge temperature. The increase in the compressor discharge temperature tends to be followed by the increment of input power. As a result, sub-cooling using LSHX does not always increase the COP [10-11, 19] but also depends on the type of refrigerant used.

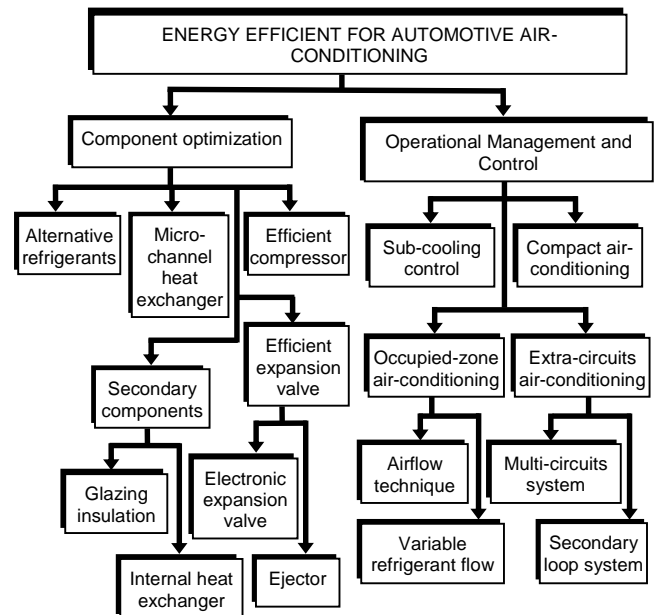


Figure 1 An overview of energy efficiency methods in automotive air-conditioning system [4]

The DMS utilizes an external VCRC, which has smaller capacity than the main system. The external VCRC is used for cooling the refrigerant at the condenser exit. The decrease in refrigerant temperature in the condenser exit decreases the quality of the refrigerant entering the evaporator, hence, it increases the CC. The higher the CC, the higher the COP of the system. Unlike the LSHX sub-cooling, the DMS is not always followed by superheating in the compressor suction

and consequently there is no increase in the compressor discharge temperature [10-16].

Similar to the DMS, the IMS also utilizes an external VCRC but the difference is in the use of the condenser. In the DMS, there are two condensers, where each condenser is used by each system, i.e., main and sub-cooler systems [14-15]. But, in the IMS, there is only one condenser which is used by main and sub-cooler systems.

Meanwhile, CAS utilizes condensate water from evaporator coils as condenser air face cooler or compressor discharged cooler. The potential for condensate water to absorb heat from air face condenser and discharged compressor is high due to significant temperature difference between condensate water temperature and these condenser air face and compressor discharge temperatures [17-18]. In this method, the condensate water will absorb some amount of heat and as a result, there will be sub-cooling on the condenser outlet. Again, the sub-cooling on the R&A system will increase the CC. An increase in CC will reduce the power consumption of the air conditioner for the same or lower system input power.

Therefore, the present study aims to review the literature on theoretical thermodynamics analysis, research advancement, and opportunity for future direction and development of these four methods of sub-cooling in R&A system. The review begins by documenting the theoretical thermodynamic analysis of typical VCRC as a foundation on the establishment of sub-cooling method. As shown in Figure 2, this review is then organized systematically into four methods of sub-cooling to improve the performance of R&A system, known as liquid-suction heat exchanger (LSHX), dedicated mechanical sub-cooling (DMS), integrated mechanical sub-cooling (IMS) and condensate assisted sub-cooling (CAS). The next part of this article reviews the advantages of sub-cooling method as compared to other methods. Then, the final part of this article reviews the direction and development of sub-cooling method in the future

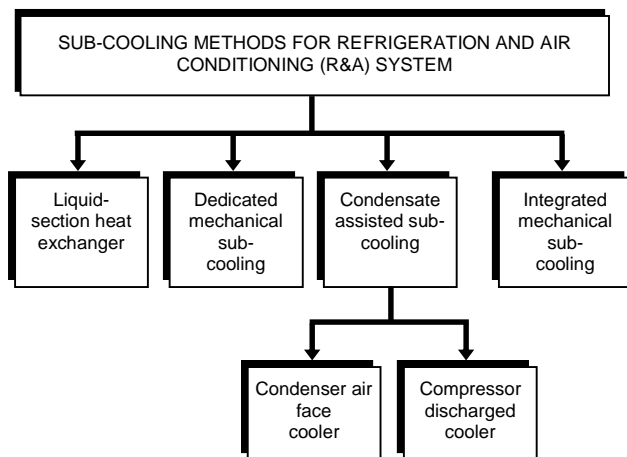


Figure 2 A generalized outline of sub-cooling methods for refrigeration and air conditioning systems

2.0 THERMODYNAMIC ANALYSIS OF VCRC

Evans was the first person who acclaimed the invention of the vapor-compression refrigeration machine in 1805. His concept was modified by Jacob Perkins in 1834. An American professor, namely Alexander C. Twining patented a vapor-compression refrigerator and established a plant that was capable of producing a ton of ice per day [20] in 1853. The numbers and the name of major components of VCRC have not been changed since Alexander C. Twining issued the patent. Nowadays, most air conditioners and refrigerators/freezers that are available in the market use the VCRC.

The VCRC is made of four major components, viz. compressor, condenser, expansion device and evaporator. The name of the components represents their functions. The compressor is used to compress the vapor refrigerant, thereby, increasing the pressure of the refrigerant from low pressure (evaporator) to high pressure (condenser) and in the process, the temperature is also increased. The condenser is utilized to condense the high pressure refrigerant. Because of the condensation of refrigerant, the heat is released to ambient. The quantity of heat which is released by condenser is the amount of heat absorbed by evaporator plus energy consumed by compressor. In order to expand the refrigerant, i.e., to reduce the high pressure refrigerant from the condenser to a low pressure in the evaporator, an expansion device is utilized. Furthermore, an evaporator is used to absorb the heat load from the air conditioned space. The evaporation of refrigerant inside the evaporator needs heat and as a result, the heat load from air conditioned space is absorbed by the evaporator. The vapor refrigerant from evaporator, which is at low pressure enters the suction of the compressor and the cycle is repeated continuously.

The working principle of VCRC is based on the reversed Rankine cycle as shown in Figure 3. The figure shows that to pump heat (q_L) from a lower temperature medium (evaporator) requires input work (W_{in}). The total heat from a low temperature medium plus the input work is equal to heat rejected to the high temperature medium (ambient air) through condenser (q_H) [21]. Unlike the reversed Rankine cycle, a reversed Carnot cycle can use wet-vapor as working fluid. The T - s diagram of the cycle is shown in Figure 4.

The processes of VCRC occur at two constant pressures (evaporation and condensation), i.e., one at constant enthalpy (expansion) and the other at constant entropy (compression). Therefore, it is more convenient to present the VCRC on a P - h (pressure-enthalpy) diagram as shown in Figure 5.

Based on Figure 5, the COP of simple VCRC is called actual COP (COP_{act}), and defined in Eq. (1), where

$$COP_{act} = \frac{q_{evap}}{w_{comp}} = \frac{h_1 - h_4}{h_2 - h_1} \quad (1)$$

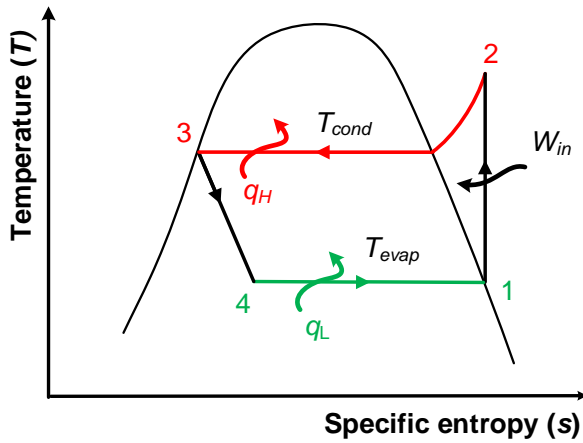


Figure 3 Reversed Rankine cycle in T-s diagram [20]

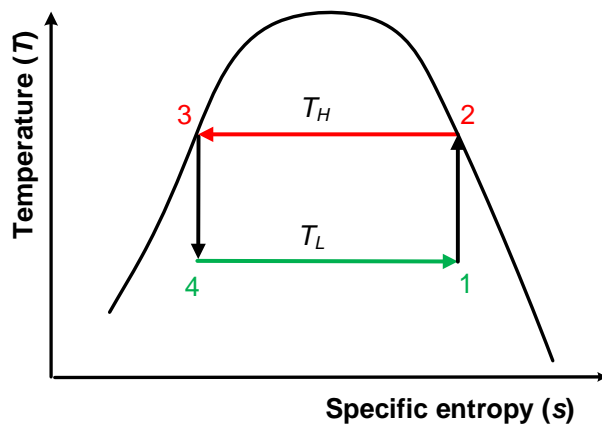


Figure 4 T-s diagram of reversed Carnot cycle with wet-vapor as working fluid [20]

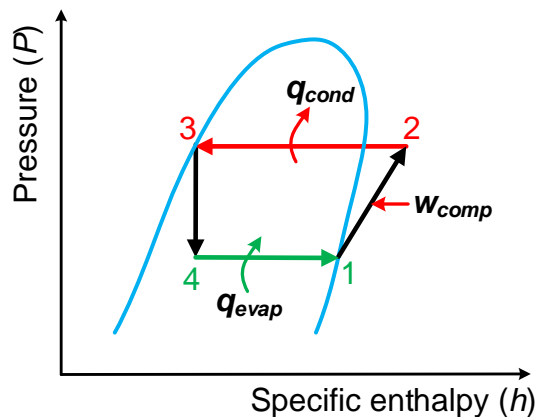


Figure 5 Simple vapor compression refrigeration cycle in P-h diagram [20]

The ratio between COP_{act} and Reversed Carnot cycle (COP_{Carnot}) by using an ideal gas as working fluid is defined as refrigeration efficiency (η_R) as shown in Eq. (2), where

$$\eta_R = \frac{COP_{act}}{COP_{Carnot}} \quad (2)$$

The COP_{Carnot} is defined by Eq. (3) [21], where

$$COP_{Carnot} = \frac{T_L}{T_H - T_L} \quad (3)$$

Hence, the refrigeration efficiency in Eq. (2) becomes:

$$\eta_R = \frac{COP_{act}(T_H - T_L)}{T_L} \quad (4)$$

According to Eq. (4), it can be concluded that for the same condenser (T_H) and evaporator (T_L) temperatures, refrigerators/freezers or air conditioners that have higher COP_{act} indicates that it is more efficient. Usually, the manufacturers have a self-acclaimed COP_{act} for their machines. Due to limited energy resources, researchers and manufacturers of air conditioners and/or refrigerator compete in order to improve the performance of the system. The sub-cooling, which increases the q_{evap} (CC) potentially improves the COP_{act} and refrigeration efficiency of freezers or air conditioners as according to Eq. (1). Therefore, sub-cooling is one of the best and simplest methods in achieving a better energy-efficient R&A systems.

3.0 THE METHODS OF SUB-COOLING

There are four main methods of sub-cooling that can be used to enhance the CC, hence, increase the COP and subsequently improve the performance of the R&A systems. These methods are discussed in detail in the following subsections.

3.1 Liquid-Suction Heat Exchanger Sub-Cooling

This method uses a heat exchanger to transfer heat energy at the condenser exit to the suction inlet. Figure 6 shows the schematic and P-h diagrams of LSHX in the VCRC. The figure shows that the use of sub-cooler (heat exchanger) increases the suction temperature from point 1 to 1', and this phenomenon is called superheating. The figure also shows that the sub-cooler decreases the exit condenser temperature, from point 3 to 3', and this is called sub-cooling. The superheating helps in preventing the liquid refrigerant from entering the suction compressor. The liquid phase refrigerant entering the suction causes problem in the compressor bearings and rings, or valve failure in some extreme cases.

Thermodynamics analysis of the LSHX in Figure 6 is based on Eqs. (5) - (13). The degree of sub-cooling ($T_{cond} - T_{3'}$) and the degree of superheating ($T_{3'} - T_{evap}$) are not the same due to the difference in specific heats of the vapor and liquid phases. An energy balance analysis in the sub-cooler using the effectiveness-NTU method can be utilized to predict the outlet temperatures of the hot refrigerant leaving sub-cooler ($T_{3'}$) and cold refrigerant leaving sub-cooler ($T_{1'}$). By assuming the sub-cooler in Figure 6(a) is well insulated,

the actual heat transfer rate in the sub-cooler (\dot{Q}_{act}) is given in Eq. (5), where

$$\dot{Q}_{act} = \varepsilon \cdot \dot{Q}_{max} = C_c(T_{1'} - T_1) = C_h(T_3 - T_{3'}) \quad (5)$$

where ε is heat transfer effectiveness, C_c and C_h are heat capacity rates of refrigerant at cold and hot conditions respectively. \dot{Q}_{max} is the maximum possible heat transfer rate and can be given as in Eq. (6), where

$$\dot{Q}_{max} = C_{min}(T_3 - T_1) \quad (6)$$

where C_{min} is the smaller of heat capacity rate of C_c and C_h . For double pipe sub-cooler, the effectiveness relations, both for parallel ($\varepsilon_{parallel}$) and counter-flow ($\varepsilon_{counter}$) are given in Eqs. (7) – (8) respectively, where [22]

$$\varepsilon_{parallel} = \frac{1 - \exp[-NTU(1+c)]}{1+c} \quad (7)$$

$$\varepsilon_{counter} = \frac{1 - \exp[-NTU(1-c)]}{1-c \cdot \exp[-NTU(1-c)]} \quad (8)$$

where number of transfers unit, $NTU = UA_s/C_{min}$, $c = C_{min}/C_{max} = (\dot{m}c_p)_{min}/(\dot{m}c_p)_{max}$, A_s is the contact surface area between hot and cold refrigerants and U is overall heat transfer coefficient. If thin pipe is used, surface area in contact with outer refrigerant (A_o) is almost the same as the surface area in contact with inner refrigerant (A_i) where $A_o \approx A_i$. Therefore, thermal resistance in the wall of the pipe (R_{wall}) is negligible since $R_{wall} \approx 0$. Therefore, U can be given as

$$\frac{1}{U} = \frac{1}{H_i} + \frac{1}{H_o} \quad (9)$$

where H_i and H_o are convection heat transfer coefficients for inner and outer refrigerants. Figure 7 shows the relations between ε and NTU for parallel and counter-flow double pipe sub-coolers in a diagrammatic form.

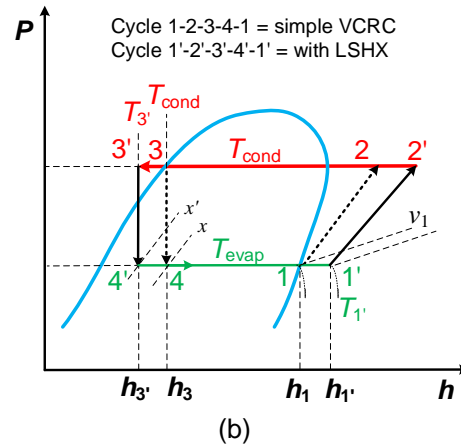
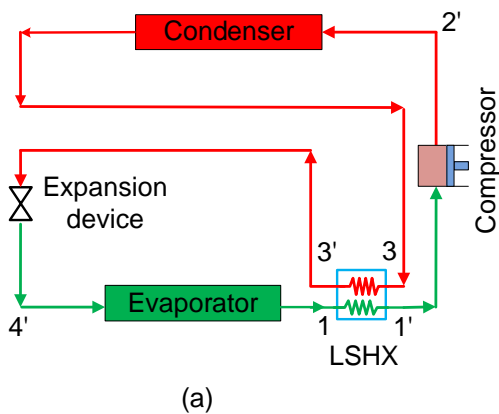


Figure 6 Liquid-suction heat exchanger in the VCRC. (a) Schematic diagram; (b) P-h diagram at steady state condition

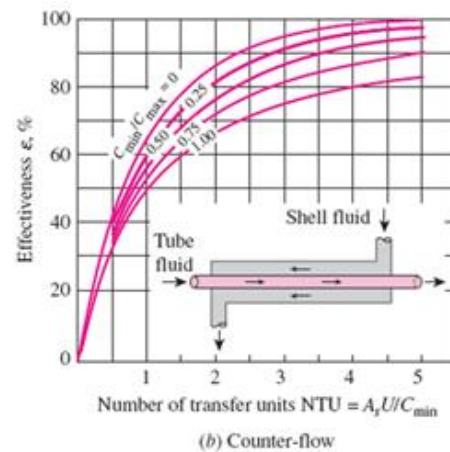
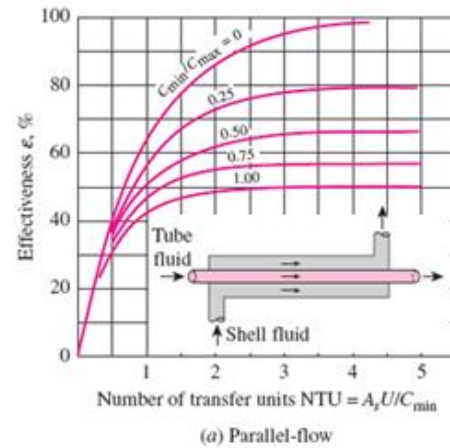


Figure 7 Relations between ε and NTU for double pipe sub-cooler. (a) Parallel-flow; (b) Counter-flow [22]

Meanwhile, by knowing that the hot and cold refrigerants are from the same VCRC, and due to continuity equation, then the mass flow rate of hot refrigerant is equal to the mass flow rate of cold refrigerant. As a result, Eq. (5) can be written as

$$q_{sub-cooling} = q_{superheated} = h_{1'} - h_1 = h_3 - h_{3'} \quad (10)$$

According to Figure 6(b) that shows the comparison between the effect of using LSHX to simple VCRC system, the cooling capacity per unit mass of refrigerant (cc_{LSHX}/cc_{evap}) and compressor work per unit mass of refrigerant (w_{LSHX}/w_{comp}) can be expressed as in Eqs. (11) - (12).

$$\frac{cc_{LSHX}}{cc_{evap}} = \frac{h_1 - h_{4'}}{h_1 - h_4} \quad (11)$$

$$\frac{w_{LSHX}}{w_{comp}} = \frac{h_2' - h_1'}{h_2 - h_1} \quad (12)$$

Then, the COP with the effect of using LSHX (COP_{LSHX}) is

$$COP_{LSHX} = \frac{(h_1 - h_4) + (h_4 - h_{4'})}{h_2' - h_1'} \quad (13)$$

By considering the COP with LSHX sub-cooling, both numerator and denominator of Eq. (13) increase. The net effect whether positive, negative or zero depends on the working fluid and operating temperatures (evaporating and condensing temperatures). As a result, many researchers have carried out investigations to determine the influence of LSHX sub-cooling on COP by using various working fluids.

Domanski *et al.* [23] numerically investigated the performance of 38 different VCRC refrigerants that use LSHX. They reported that refrigerants with low vapor molar heat capacity did not improve the COP as compared to basic cycle (without sub-cooling) when LSHX was applied. They also reported that by using refrigerants with very low heat capacity, the COP of the system decreased when the LSHX was applied.

Klein *et al.* [24] introduced a relative capacity change index (RCI) formula to calculate the CC improvement of using LSHX. The formula is given by Eq. (14),

$$RCI = \frac{CC_{LSHX} - CC_{simple VCRC}}{CC_{simple VCRC}} \cdot 100\% \quad (14)$$

where

CC_{LSHX} = cooling capacity rate with LSHX
 $CC_{simple VCRC}$ = cooling capacity rate for simple VCRC (without LSHX)

Figure 8 shows the correlation between RCI and the effectiveness of heat exchanger, where the effectiveness can be defined as the ratio of actual to maximum possible heat transfer in the heat exchanger. It can be described as in Eq. (15) [24].

$$\varepsilon = \frac{T_1' - T_1}{T_3 - T_1} \quad (15)$$

Figure 8 shows that by applying LSHX in VCRC with eleven refrigerants, three refrigerants decrease their

CC, namely R22, R32 and R717; and the other eight refrigerants increase their CC.

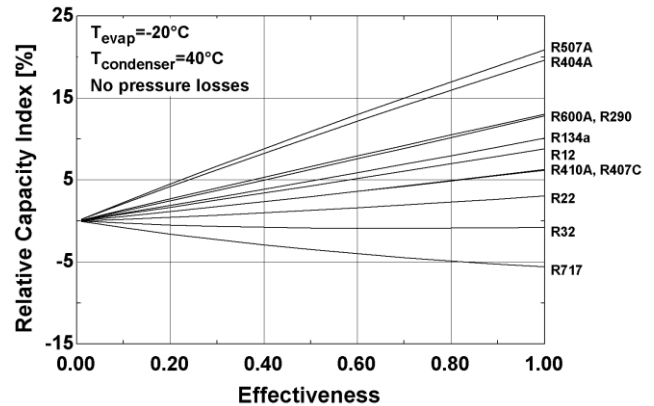


Figure 8 RCI versus effectiveness of heat exchanger [24]

Navarro-Esbri *et al.* [25] experimentally investigated the effect of sub-cooling using LSHX in VCRC by using R22, R134a and R407C as working fluids. Besides studying the influence of sub-cooling on the performance, they also investigated the impact of mass flow rate. Experimental results showed that the mass flow rate reduction occurred for R22 and R134a refrigerants when LSHX was applied. However, although the mass flow rate decreased, the COP did not decrease because the increment of cooling capacity by using LSHX was slightly higher than the mass flow rate. Different results were exhibited by R407C, in which the mass flow rate and cooling capacity increased. As a result, the COP improvement of R407C was the highest for the compression ratio below 5.

Mastrullo *et al.* [19] numerically investigated the advantage of applying LSHX in VCRC. They investigated 19 ozone friendly refrigerants in their study and varied the evaporating and condensing temperatures. They introduced a simple chart that allows the estimation of the effectiveness of using LSHX in the VCRC for various working fluids and specified operating conditions, as shown in Figure 9. The use of Figure 9 is as follows: for example, for a system using R22 as working fluid for evaporating and condensing temperatures of -20°C and 45°C, respectively, the point from these conditions is above the curve of R22. As a result, the use of LSHX improves the system performance. However, if the point from the two conditions (evaporating and condensing temperatures) is below the curve, the benefit of using LSHX disappears. They also found that the advantage of LSHX depends on the combination and operating conditions. The vapor heat capacity is the most influential property on the performance improvement of VCRC.

Potter and Hrnjak [6] carried out a numerical and experimental investigation to study the effect of sub-cooling in VCRC. In the numerical investigation, they used four refrigerants, i.e. R134a, R410A, R1234yf and

R717 in an air conditioner with outdoor and indoor temperatures of 35°C and 27°C, respectively. The numerical results showed that because R1234yf had a lower latent heat of vaporization, this refrigerant had the highest COP improvement as compared to R410A, R134a and R717. Meanwhile, in their experiment of using an automotive air conditioner that originally operated with R134a as working fluid, they reported that for a given operating condition, the COP improvement of R1234yf was higher than R134a, namely up to 18% and 9% for R1234yf and R134a, respectively

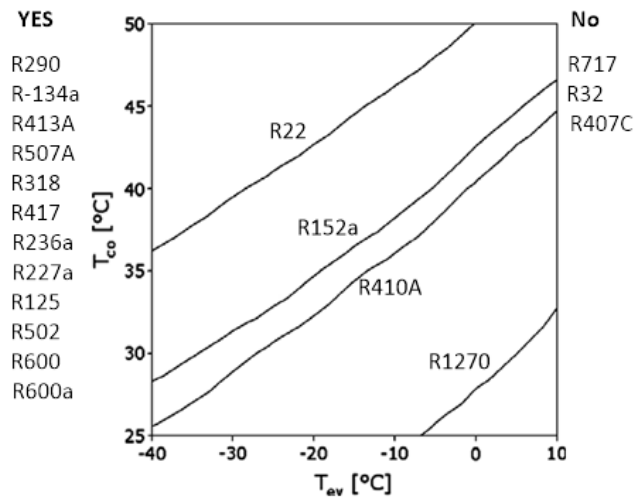


Figure 9 A simple map of comparison of COP with and without LSHX for different operating conditions, for the maximum effectiveness [19]

Later, Potter and Hrnjak [7] carried out an extension experimental investigation to study the effect of sub-cooling on various parameters in the air conditioning system using R134a and R1234yf. The parameters of the air conditioner being studied included isentropic efficiency, compression ratio, condenser pressure drop, suction line pressure drop, saturation temperatures at the evaporator exit and compressor inlet and COP. The experimental results showed that the presence of LSHX in the air conditioner increased the COP up to 18% and 9% for R1234yf and R134a, respectively. The results also showed that there were changes in some parameters due to the presence of sub-cooler using LSHX, as shown in Table 1. The table shows that the use of LSHX sub-cooler is always followed by the increase of CC, compressor work and superheating. Because of COP is the ratio of CC to compressor power, as a result the increase in COP depends on their increments. Most working fluid generate COP improvement of the system when the LSHX sub-cooling is applied, and only few refrigerants (R717, R32 and R407C) showed COP reduction when using LSHX sub-cooling, as shown in Figure 9.

Table 1 Effect of sub-cooler in the VCRC

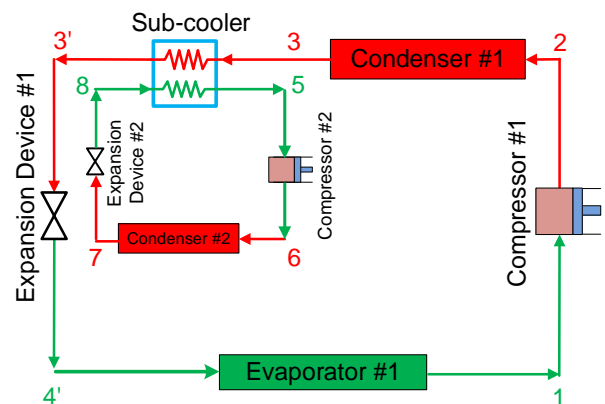
Quantity	LSHX
Cooling capacity	Increases
Condensing pressure	Constant*
Mass flow rate	Vary*
Compressor work	Increases
COP	Increases**
Superheating	Increases

*Depending on the degree of sub-cooling.

**Depending on the working fluid used.

3.2 Dedicated Mechanical Sub-Cooling

The sub-cooling using LSHX can be applied for small up to large capacity freezers and air conditioners, whereas the DMS is usually applied for medium to large capacity. As explained in the previous section, LSHX sub-cooling cools down the refrigerant at condenser exit using the suction line. Whereas, the DMS utilizes a small VCRC to decrease temperature of the condenser exit. Figure 10 shows the schematic and P-h diagrams of DMS. The figure comprises of two cycles of VCRC, namely main and additional (sub-cooler) cycles. The capacity of the main is larger than that of sub-cooler cycle. The refrigerant used in both cycles can either be the same or different. The evaporator temperature of the main cycle must be higher than that of the sub-cooler cycles. The evaporator temperature of main cycle is designed based on the product temperature. Also, the evaporator temperature is designed a few degree under the condenser temperature of the main system. Furthermore, the condenser temperature of main cycle does not always higher than the sub-cooler cycle. If the main and sub-cooler cycles are placed in the same area, both temperatures will be the same. However, because the sub-cooler cycle is usually placed indoor, as a result, the condenser temperature of the additional system is lower than the main cycle.



(a)

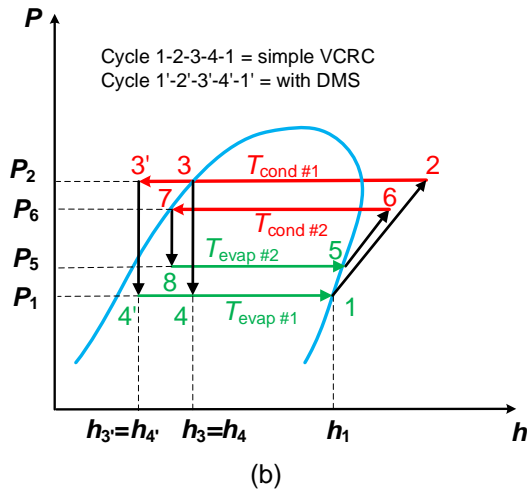


Figure 10 Dedicated mechanical sub-cooling in the VCRC. (a) Schematic diagram; (b) *P-h* diagram at steady state condition

According to *P-h* diagram as shown in Figure 10(b), the cooling capacity per unit mass of refrigerant for simple VCRC (cc_{evap}) and cooling capacity per unit mass of refrigerant with dedicated sub-cooling (cc_{DMS}) can be expressed by Eqs. (16) and (17), respectively.

$$cc_{evap} = h_1 - h_4 \tag{16}$$

$$cc_{DMS} = h_1 - h_{4'} \tag{17}$$

It can be seen from Figure 10 that the CC with DMS ($h_1 - h_{4'}$) is higher than CC without DMS ($h_1 - h_4$). Meanwhile, the compressor work per unit mass of refrigerant for simple VCRC (w_{comp}) and with DMS (w_{DMS}) are determined using Eqs. (18) and (19), respectively.

$$w_{comp} = h_2 - h_1 \tag{18}$$

$$w_{DMS} = (h_2 - h_1) + (h_6 - h_5) \tag{19}$$

Eqs. (18) and (19) show that the compressor work per unit mass of refrigerant with dedicated sub-cooling is higher than without sub-cooling. It is because, DMS requires an additional compressor work, that is, $(h_6 - h_5)$. Therefore, according to Figure 10(b), the COP without DMS or COP for simple VCRC (COP_{act}) and COP with DMS (COP_{DMS}) are:

$$COP_{act} = \frac{h_1 - h_4}{h_2 - h_1} \tag{20}$$

$$COP_{DMS} = \frac{h_1 - h_{4'}}{(h_2 - h_1) + (h_6 - h_5)} \tag{21}$$

According to Eq. (21), to obtain optimum COP_{DMS} , the numerator ($h_1 - h_{4'}$) must be as high as possible, whereas the denominator ($h_6 - h_5$) must be as low as possible. However, to increase the degree of the sub-cooling (point 3 to point 3'), the compressor work of

the sub-cooler cycle must increase as well. Therefore, a trade-off between the degree of the sub-cooling in the main cycle and the amount of the compressor work in the sub-cooler cycle must be performed. As explained in this subsection, in the sub-cooler using LSHX, the degree of the sub-cooling is limited to the discharge temperature of the compressor. Whereas, in the DMS, the limitation is on the compressor work of the sub-cooler cycle. In other words, the higher the degree of the sub-cooling, the higher the discharge temperature of the compressor and the higher the compressor work in LSHX and DMS. To determine the optimum sub-cooling in order to obtain the maximum COP in the DMS, various studies had been carried out by researchers.

Couvillion et al. [26] carried out numerical study using a computational model to estimate the performance of the DMS with several different refrigerant combinations in main and sub-cooler cycles. The model also investigated the effect of degree variation of the sub-cooling in main cycle to the performance of the system. Their result showed that by applying DMS in the VCRC could increase the cooling capacity and COP by 20 to 170% and 6 to 80%, respectively over a simple cycle of VCRC.

Thermodynamic modeling based on refrigerant property of DMS was investigated by Thornton et al. [15]. In their model, they utilized two cycles of VCRC with an ambient temperature of 26.7°C (80°F) for the main and sub-cooler, a main cycle and sub-cooler evaporator temperatures of -28.9°C (-20°F) and -1.1°C (30°F), respectively. Their model was used to obtain the optimum temperature of the sub-cooling, which resulted in maximum COP improvement. The effect of heat exchanger parameters to the performance was also investigated in their model. The results showed that the optimum temperature of the sub-cooling was strongly dependent on the condensing and evaporating temperatures. However, the heat exchanger parameters were weakly dependent on the optimum sub-cooling, the condensing and evaporating temperatures. It depended on the design of the heat exchanger.

Qureshi and Zubair [8] carried out numerical investigation on the effect of DMS on the performances of VCRC. In the first simulation, they utilized R134 as working fluid in the main cycle and R134a, R407C and R410A as sub-cooler cycle, also R717 as main cycle and R134a, R407C and R410A as sub-cooler cycle. They reported that the DMS was more suitable to cycle using R134a than R717 as refrigerant in the main cycle. In addition, they also introduced two dimensionless parameters, namely sub-cooler saturation temperature (θ) and heat exchanger performance (f). By applying these dimensionless parameters, the optimum sub-cooling using DMS could be determined. Because DMS needed an additional sub-cooler, as a result, it caused additional cost. Nevertheless, they concluded that the payback period would depend on the size of the systems. Their report also showed that for air-cooled refrigeration and air conditioning system with

capacity above 100 kW, the payback was expected to be less than 3 years.

An experiment on a split-type air conditioner with CC of 1.5 ton using DMS was carried out by Qureshi *et al.* [9]. Their experiments used R22 and R12 as working fluids in the main and sub-cooler cycles, respectively. Their results showed that the CC increased by approximately 0.5 kW when the main cycle was sub-cooled by 5-8°C. They also reported that the power consumption in the air conditioner increased with the increase in the sub-cooling using DMS. However, the COP always increased, because the increase in cooling capacity was higher than the power consumption. An average COP improvement using DMS was up to 21%. They also reported the effect of DMS on the main system, as shown in Table 2. The table shows that the use of DMS sub-cooler is always followed by the increase of CC and the decrease of main condensing pressure and superheat at main compressor. It means that the use of DMS improves the system performance.

Table 2 Effect of DMS sub-cooler in the main system

Quantity	Effect
Cooling capacity	Increases
Main condensing pressure (and temperature)	Decreases
Superheat temperature at main compressor	Decreases
Evaporator pressure (and temperature)	Increases*

* Only for retrofit applications.

Yang and Zhang [3] introduced the use of DMS in supermarket refrigeration system without an additional sub-cooler for sub-cooling the main cycle. Typically, the supermarket refrigeration system had two different temperature systems, namely for medium temperature (above 0°C) for preservation of fresh foods and beverages, and low temperature (below 0°C) for frozen products. The medium and low temperature systems usually had evaporating temperature of -15°C to 5°C and -30°C to -40°C, respectively. As a result, there was a potential for medium temperature cycle to provide sub-cooling for low temperature cycle. The schematic diagram of their concept in the supermarket refrigeration system is shown in Figure 11. They reported that the maximum energy saving in their numerical investigation in a supermarket refrigeration system using the low temperature as mechanical sub-cooling with R404A or R134a as working fluid were 27% and 20%, respectively.

3.3 Integrated Mechanical Sub-Cooling

Similar to DMS, the integrated mechanical sub-cooling (IMS) is usually applied on medium to large capacity of refrigeration systems. This method uses a small VCRC as a sub-cooler. The evaporator of the sub-cooler is coupled to the exit of the main cycle condenser and it uses the main cycle condenser as its

condenser. The schematic and *P-h* diagram of the IMS are shown in Figure 12.

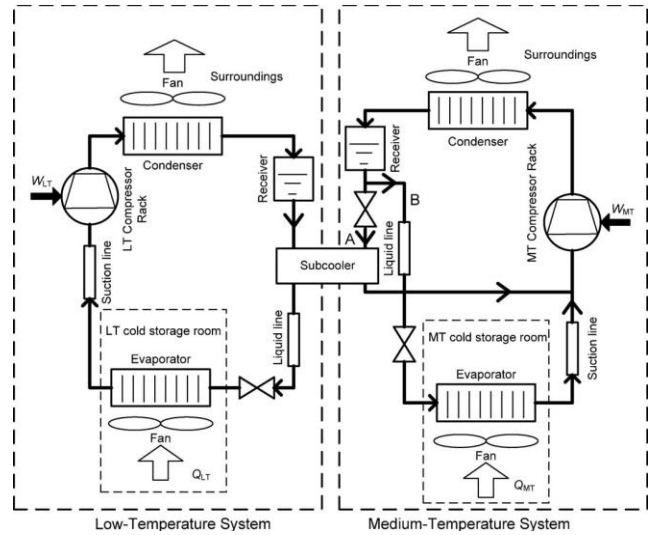


Figure 11 Schematic diagram of mechanical sub-cooling of low temperature using medium temperature system [3]

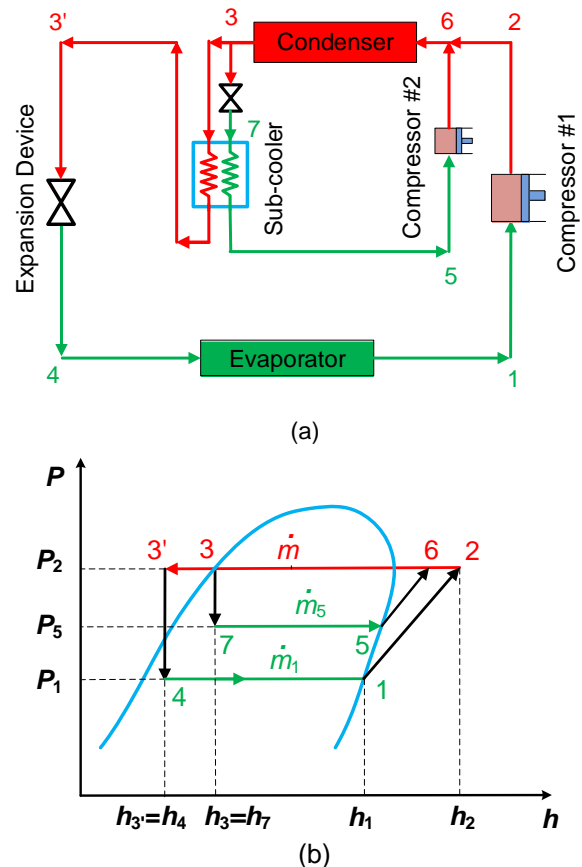


Figure 12 Integrated mechanical sub-cooling in the VCRC. (a) Schematic diagram; (b) *P-h* diagram at steady state condition

By referring to *P-h* diagram as shown in Figure 12(b), the CC of the sub-cooler cycle is equal to the degree of sub-cooling and can be given as in Eq. (22):

$$\dot{m}_5(h_5 - h_7) = \dot{m}_1(h_3 - h_3') \quad (22)$$

where,

$$\dot{m} = \dot{m}_5 + \dot{m}_1 \quad (23)$$

As a result, according to Figure 12(b), the COP with IMS (COP_{IMS}) is,

$$COP_{IMS} = \frac{h_1 - h_4}{\dot{m}_5(h_6 - h_5) + \dot{m}_1(h_2 - h_1)} \quad (24)$$

To obtain maximum COP_{IMS} , the numerator (the CC) must be as high as possible, whereas the denominator has to be as low as possible (the compressor work). In the IMS, to increase the degree of the sub-cooling (point 3 to point 3'), the compressor work of the sub-cooler cycle must be increased; however the increase in compressor work is limited by the condenser capacity. The higher the compressor work, the bigger the condenser capacity. Since this method uses one condenser for two refrigeration systems, the degree of sub-cooling using IMS is determined by the capacity availability in the condenser.

Zubair et al. [14] carried out numerical investigation of IMS using R134a. They presented a graph of normalized COP (improvement) versus sub-cooler saturation temperature for the evaporator and the sub-cooling temperatures of -30°C and 3°C , respectively. The graph shows that the COP improvement increases with an increase in the condenser temperature. There are four curves in their graph, for four different condenser temperatures, namely 40 , 50 , 60 and 70°C , respectively. The COP improvement increases significantly when the condenser temperature is increased, especially when the sub-cooler saturation temperature increases from 0.3 to 0.6 . In addition, the COP improvement increases as the difference between the condensing and evaporating temperatures increases.

Khan and Zubair [27] carried out investigation on the IMS using R134a as working fluid. They developed a model to study the system performance for various conditions. The model could also be utilized to predict an optimum distribution of the total heat exchanger areas between the evaporator and condenser. Based on the numerical investigation for various conditions, they reported that a maximum COP improvement was 7.5% .

Performance characteristic based on thermodynamic standpoint due to fouling in the IMS had been investigated by Qureshi and Zubair [28]. They utilized two sets of refrigerants; the first set consisted of R134a, R410A and R407C and the second consisted of R717, R 404A and R290. Based on the first law standpoint, they reported that COP of R134a and R717 always showed better performance unless when only the evaporator was being fouled for both first and second sets of refrigerants. The efficiency of R134a and R717 also performed the best in all cases studied based on the second-law standpoint. Their numerical study also showed that the fouling in the evaporator

had larger effect on CC. Whereas, the fouling in the condenser degraded the compressor power consumption of the sub-cooler and also on the COP of the system.

Numerical investigation on cost optimization of heat exchanger inventory for IMS was carried out by Qureshi and Zubair [11]. Several non-dimensional parameters were introduced in their model. They reported that for a constant heat rate in the condenser of the main cycle, there was no minima cost function with respect to the absolute temperature ratio and the average sub-cooling absolute temperature ratio. Furthermore, for constant compressor work, ϕ_1 (low-side Carnot absolute temperature ratio for main cycle) and ϕ_2 (low-side Carnot absolute temperature ratio for sub-cooler) provided a theoretical minimum when the function of F_a (non-dimensional cost ratio) was negative. In addition, they found that the cost optimization of the IMS could be generated from the derivatives of the DSM.

3.4 Condensate Assisted Sub-Cooling

Technically, the air that passes through an evaporator coils with temperature of 25°C and relative humidity of 50% has a dew point temperature of 13.9°C . Meanwhile, the evaporating temperature of typical air conditioning systems is about 5°C [17]. As a result, the water vapor will be condensed in the evaporator surfaces because of the condensate temperature is lower than the dew point temperature of the air. Since it is produced by the evaporator, the condensate water temperature is slightly higher than 5°C . In practice, this water is not utilized and discharged to ambient air.

On the other hand, the condenser air face temperature and temperature of compressor discharge are typically about 30°C and 60°C , respectively [17]. It means that the condensate temperature is much lower than the condenser air face temperature and compressor discharge temperature. As a result, by using and additional components, i.e. water pump and sub-cooler as in Figure 13, the condensate water has significant ability to receive heat load either from hot compressed refrigerant or hot air at the inlet condenser. Therefore, it can be utilized to decrease the condenser air face temperature and compressor discharge temperature. The decrease in the condenser air face temperature and compressor discharge temperature can lead to the improved performance of the air conditioning system [29-31].

Figure 14 shows the direct effect of condensate water as a cooler that causes a decrease in the discharge temperature from point 2 to 2'. It shows that the temperature of point 2' is lower than point 2, which resulting a significant reduction in the power consumption of the compressor. Since, $T_{2'}$ is lower than T_2 , with the same size of condenser coil and conditions of air at the inlet of condenser, sub-cooling condition of point 3' is achievable, which later tends to produce

higher cooling capacity per unit mass of refrigerant ($h_{1'} - h_{4'}$) as compared to cooling capacity per unit mass of refrigerant without condensate water ($h_1 - h_4$).

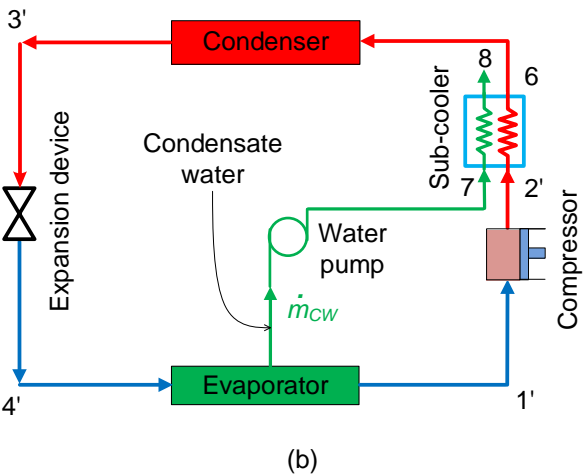
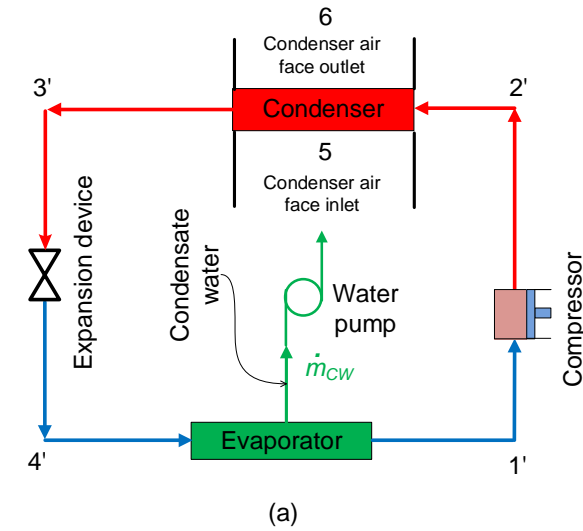


Figure 13 Schematic diagram of condensate assisted sub-cooling in VCRC. (a) Mode of condenser air face cooler; (b) Mode of compressor discharge cooler

In Figure 13(a), with cross-flow (single-pass) condenser, one fluid (refrigerant) mixed and other (air face condenser) unmixed, the ϵ of the cross-flow and single-pass condenser are given as [22]

$$\epsilon_{cross} = \frac{1}{c} (1 - \exp\{-c[1 - \exp(-NTU)]\}) \quad (25)$$

or

$$\epsilon_{cross} = 1 - \exp\left\{-\frac{1}{c}[1 - \exp(-c \cdot NTU)]\right\} \quad (26)$$

where Eq. (25) is for the condition where C_{max} mixed and C_{min} unmixed, and Eq. (26) is for the condition where C_{min} mixed and C_{max} unmixed. Figure 15 shows in a diagrammatic form the relations between ϵ and NTU for cross-flow condenser with one fluid (refrigerant) mixed and other (air face condenser) unmixed.

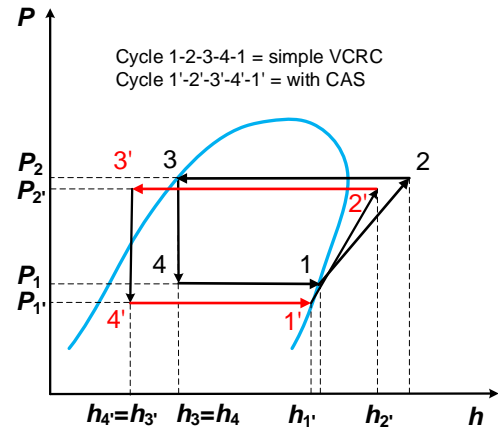


Figure 14 P-h diagram of condensate assisted sub-cooling

According to P-h diagram as shown in Figure 14, the cooling capacity per unit mass of refrigerant with condensate assisted sub-cooling (cc_{CAS}) can be expressed by Eq. (27) where,

$$cc_{CAS} = h_{1'} - h_{4'} \quad (27)$$

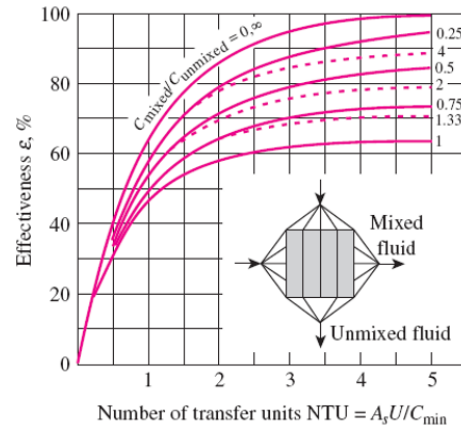


Figure 15 Relations between ϵ and NTU for cross-flow condenser with with one fluid mixed and other unmixed [22]

If condensate water pump is used in CAS system, the input power per unit mass of the system with CAS (e_{CAS}) is the summation of compressor work per unit mass of refrigerant with condensate assisted sub-cooling (w_{CAS}) plus condensate water pump input power per unit mass of condensate water (w_{cwp}). It can be determined by using Eq. (28) as below.

$$e_{CAS} = w_{CAS} + w_{cwp} = h_{2'} - h_{1'} + \frac{W_{wp}}{\dot{m}_{cw}} \quad (28)$$

where W_{wp} is condensate water pump input power in Watt, and \dot{m}_{cw} is mass flow rate of condensate water. Relatively, the water pump input power is very small as compared to compressor input power, thus the portion of W_{wp}/\dot{m}_{cw} in Eq. (26) can be ignored. Then, Eq. (29) becomes

$$e_{CAS} = w_{CAS} = h_{2'} - h_{1'} \quad (29)$$

The COP with condensate assisted sub-cooling (COP_{CAS}) is then defined as,

$$COP_{CAS} = \frac{h_{1'} - h_{4'}}{h_{2'} - h_{1'}} \quad (30)$$

According to Eq. (30), to obtain optimum COP_{CAS} , the numerator ($h_{1'} - h_{4'}$) must be as high as possible, whereas the denominator ($h_{2'} - h_{1'}$) must be as low as possible. However, to increase the degree of the sub-cooling from point 3 to 3' for higher ($h_{1'} - h_{4'}$), the point 1' tends to move on the left side of point 1 (decreasing h). It causes additional compressor work of ($h_1 - h_{1'}$) as compared to system without CAS or simple VCRC system. However, this increment is upset by the decrease of point 2 to point 2', which causes saving in compressor work of ($h_2 - h_{2'}$). In short, a dominant decrease of point 2' from 2, as opposed to a decline of point 1 to 1', as well as a prominent decrease of point 3' from 3, rather than a decline in point 1 to 1', lead to the improvement of COP_{CAS} .

The use of condensate water to lower condenser air face temperature for AC performance improvement has been investigated by several researchers [29-32]. In general, the authors agreed that the use of condensate assisted sub-cooling method provides huge advantage when the condensate water is abundant and water pump input power is relatively small as compared to compressor input power.

Sawant et al. [29] employed condensate water as an evaporative cooling to precool the condenser air face temperature before passing through the condenser coils. Their experimental results showed a decrease in power consumption of the air conditioner up to 10%. Another experimental investigation about the use of condensate water as an evaporative cooling to lower the air temperature of condenser was carried out by Sawan et al. [30]. The experiment was performed on a split type air conditioning unit in June and August and the results showed a significant power consumption reduction of about 5 % and 4.5%, respectively. Meanwhile, Tissot et al. [31] showed experimentally that the use of water spray to lower air temperature upstream of condenser could improved the COP of air conditioner by 28.9%. The latest experimental investigation was carried out by Ibrahim et al. [32]. An air conditioning unit with CC of 1.5 tonnes refrigeration was utilized in their experiment. They reported that the power consumption of compressor decreased by 6.1%, whereas the COP increased by 21.4%.

Recently, instead of using condensate water as an evaporative condenser air face cooler before entering the condenser coils, Sumeru et al. [17-18] used condensate water as a discharge compressor cooler. As evaporative condenser air face cooler, the increase in air conditioner performance due to compressor discharge cooler was also caused by sub-cooling on the condenser outlet.

Sumeru et al. [17] carried out experimental investigation on the performance of a 2.5 kW, split-type residential air conditioner using condensate as a compressor discharge cooler. The utilization of condensate as a discharge cooler decreased the compressor discharge pressure and consequently, reduced the compressor power consumption by 6.3%. In addition, the effect of the decrease in the compressor discharge pressure resulted in the decrement in the temperature of condenser outlet by 2.2°C. The decrease in temperature of condenser outlet led to the increment in the degree of sub-cooling and increased in the CC. More recently, Sumeru et al. [18] conducted an experimental investigation of sub-cooling effects due to compressor discharge cooler on COP of a 0.75 kW residential air conditioning using R410A as refrigerant. In their experiment, the indoor and outdoor temperatures were controlled at 24°C and 32°C. The results showed that the use of condensate water reduced the refrigerant temperature in the condenser outlet and compressor discharge temperature by 2.7°C and 7.6%, respectively. Reduction in refrigerant temperature at condenser outlet led to increase in CC, while reduction in compressor discharge temperature resulted in the decrease in power consumption of the air conditioning system by 5.9%. Consequently, the COP improved by 16.4%.

4.0 THE ADVANTAGES OF SUB-COOLING VERSUS OTHER METHODS

As mentioned in the previous section, sub-cooling is one of the methods for improving the performance of the VCRC. Among the sub-coolers, LSHX and DMS are more viable to be applied in the existing systems. Besides, they are easier to install and both methods also show an increase of COP improvement higher than IMS. On the other hand, authors believe that CAS is very promising method for energy-efficient R&A systems in the future. By using this method, secondary system of condensate water works at low pressure, thus making the system less in complexity. In addition, the cold condensate water is free and abundant.

Another method that can be used for COP improvement is an inverter. Inverter is used to regulate the compressor rotational speed. In the VCRC, the compressor consumes 85% of total power input [33-34], as a result controlling the compressor rotation can produce significant energy savings. Inverter has been applied in air conditioning system since the last two decades. By using an inverter, the speed of the compressor rotation depends on the cooling demand in the air conditioned space. The higher the cooling load, the higher the compressor rotation. All researches reported that the use of inverter in the VCRC yields COP improvement [35-44]. Based on the literature surveys, the highest COP improvement using an inverter is 31.0% with refrigerant of R407C [33].

Besides using sub-cooler and inverter, there are still two alternative methods that can be applied to enhance the COP of the VCRC, i.e. ejector as an expansion device and nano-particle in the compressor lubricant or in refrigerant. However, these methods have not yet been applied in the refrigerator or air conditioners that are available in the market. These methods are intensively investigated by researchers in the last few decades till-date. Ejector as an expansion device reduces energy losses during throttling process [45-51]. Theoretically, throttling process uses conventional expansion device, such as capillarity tube and thermal expansion device, is isenthalpic (constant enthalpy), and energy is lost during expansion. An ejector as an expansion device can recover the energy loss during throttling process and as a result, the process becomes isentropic (constant entropy). A comprehensive review of ejector as an expansion device was reported by Sumeru *et al.* [48].

Also, in the past few decades, research in nanotechnology has led to the emergence of new fluids. These fluids enhance heat transfer by the suspension of nano-scale (1-100 nm) materials inside the base fluids. These fluids are called nano-fluids. There are two ways in which nano-fluids can be applied in R&A systems, i.e., in refrigerants and in lubricants. The comprehensive review paper of nano-fluids as refrigerant and lubricant oil in refrigeration system was presented by Saidur *et al.* [52]. In general, there are three main advantages of nano-particles inside refrigerants and lubricants. It enhances the solubility between refrigerant and lubricant, increases the thermal conductivity and heat thermal characteristic of the refrigerants and decreases the friction coefficient and wear rate [54-55].

Table 3 shows the comparison of the highest COP improvement among those four sub-cooling methods as compared to other methods in the VCRC that have been reported by previous researchers. By comparing to other methods such as an inverter, an ejector as expansion device and nano-particles, the sub-cooling generates moderate COP improvement.

Table 3 Effect of DMS sub-cooler in the main system

Methods	Authors	Refrigerant	COP Improvement (%)	Remarks
LSHX	Pottker [7]	R1234yf	18	Experiment
DMS	Yang [3]	R404A	27	Numerical analysis
IMS	Khan [27]	R134a	7.5	Numerical analysis
CAS	Tissot <i>et al.</i> [31]	R410a	28.9	Experiment
Inverter	Cecchinato [33]	R407C	31.0	Experiment
Ejector as expansion device	Sumeru <i>et al.</i> [50]	R290	23.46	Numerical analysis
Nano-particles	Saidur <i>et al.</i> [52]	R141b	17	Experiment

5.0 FUTURE DIRECTION AND DEVELOPMENT OF SUB-COOLING METHOD

Among the four sub-cooling methods, only LSHX sub-cooling method that has been commercially utilized in the R&A systems. In transportation sector, the LSHX is already being used in the air conditioning system of Toyota Sienna [56]. Meanwhile in building sector, Danfos Group already commercialized tube-in-tube LSHX (Figure 16) for refrigeration plant, applicable for various types of refrigerant i.e. R134a, R22, R290, R404A, etc. [57].



Figure 16 Commercial tube-in-tube LSHX produced by Danfoss Group [57]

Technically, the DMS, IMS and CAS methods can be applied on new or existing R&A systems. However, in terms of techniques, implementation of CAS on the R&A systems have good prospects due to their simple installation. Besides being simple in installation, the CAS method also generates significant performance improvement as compared to other sub-cooling methods as shown in Table 3. Research on CAS method can be considered as new as compared to other three sub-cooling methods. Research review by authors indicated that research activities on CAS method just started in 2011 and 2018 for mode of condenser air face cooler and mode of compressor discharge cooler, respectively. Besides these two modes, it is also possible for condensate water to act as condenser discharge cooler as in Figure 17. However, result related to this mode/conceptual idea is yet to be reported in open literature. Due to these reasons, authors believe that more research related on CAS method will be conducted and published in near future.

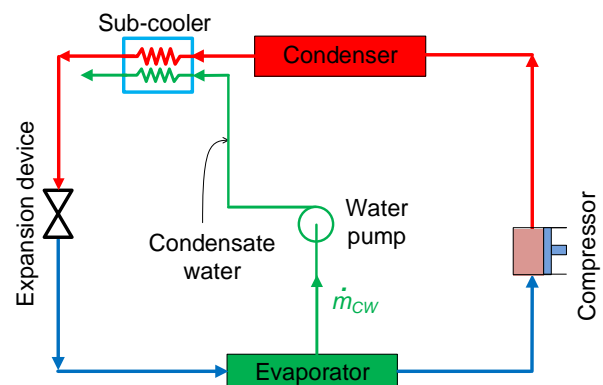


Figure 17 Conceptual idea of condensate assisted sub-cooling in VCRC with mode of condenser discharge cooler

On the other aspect, authors agreed that the IMS method is very complicated for existing R&A systems. It is because, the condenser in existing R&A systems already has fixed specific capacity. As a result, installation of sub-cooler especially using IMS method on existing R&A systems requires complex analysis and consideration. In general, these complex analysis and consideration are related to the determination of optimum capacity of the sub-cooler in order to obtain an optimal improvement performance. These have led to the current research trend, which only focusing on pure experimental study related to energy and performance analyses of the R&A system equipped with sub-cooling method.

The complexity of the analysis and consideration related to determination on optimum capacity of the sub-cooler to obtain an optimal improvement performance of any sub-cooling method gives a great challenge and open up research opportunities in the future. Since the benefit of having DMS, IMS and CAS in R&A systems was proven experimentally by previous researchers, the authors expected that future research and development in these research domain will focus on thermal and energy modelling to predict the optimum capacity of sub-cooler and optimal performance improvement of the R&A systems.

Meanwhile, by adding sub-cooler system to R&A systems can affect the overall equilibrium of the R&A systems. It is therefore expected that studies on the effects of the sub-cooling method on the overall system equilibrium will be carried out in the future. Besides that, it becomes great interest to analyze and quantify how good the proposed sub-cooling method for the R&A systems. Therefore, it is also expected that studies on the exergy analysis will be carried out in the future.

In addition to an increase concern on environmental pollution and cost around the world, the authors also expected that combination of the two or the three sub-cooling methods that possibly produce a great improvement in the COP, and other types of sub-cooler/heat exchanger especially compact sub-coolers/heat exchanger, working with high performance environmentally friendly refrigerants as well as with nano-particles as catalyst for better heat transfer performance will be investigated in the future. Although compact heat exchanger is difficult to be manufactured, which leading to high manufacturing cost as compared to conventional heat exchanger, this disadvantage can be traded off with its lower raw material cost (use less raw material due to compactness) and energy saving during its lifespan operation. In vehicle operation, the reduction in raw material significantly reduces the total weight of the vehicle, which later improves the fuel efficiency of the vehicle itself. However, further detail of life cycle cost analysis is also expected, especially to prove the effectiveness of compact sub-cooler as compared to conventional sub-cooler.

6.0 CONCLUSIONS

A literature review and evaluation of the sub-cooling methods in vapor compression refrigeration cycle for energy saving have been conducted. Key features of each method is presented to identify their merits in improving the efficiency of the R&A system. The review is divided into four main methods, LSHX, DMS, IMS and CAS.

So far, previous researches indicated that among these four sub-coolers, CAS yielded the highest improvement in COP of about 28.9%, followed by DMS (27%), LSHX (18%) and IMS (7.5%). By comparing to other sub-cooling methods, authors agreed that the cold condensate water in CAS system dominantly increased the CC, as opposed to a possible rise in total system input power. As a result, it led to the highest improvement of COP. The huge ability of cold condensate water in CAS to improve the COP of the VCRC was due to high temperature difference between cold condensate water (slightly above 5°C) and condenser air face condenser (of about 30°C), as well as compressor discharge temperature (of about 60°C) of typical air-conditioner used in South-East countries like Indonesia and Malaysia. The huge temperature difference led to high degree of sub-cooling, and consequently produced higher CC. However, for high CC of above 100 kW, the DMS system was more reasonable. It was because, although DMS required an additional sub-cooler to sub-cool the condenser outlet, previous research reported that the payback period for DMS was expected to be much less than 3 years as compared to the VCRC using condenser air face cooler. In general, Table 4 summarized the comparison between these four sub-cooling methods.

Table 4 Comparison between four sub-cooling methods

Criteria	Liquid-section heat exchanger	Dedicated mechanical sub-cooling	Integrated mechanical sub-cooling	Condensate assisted sub-cooling
Required significant additional energy input	No	Yes, for secondary compressor	Yes, for secondary compressor	Yes, for condensate water pump*
System complexity	Low	High	High	Medium
Technological status	Already commercialized	Research stage	Research stage	Research stage
Superheating status	Produces super-heating	No super-heating	No super-heating	No super-heating
COP improvement potential	Low	Medium	Medium	High

*The input energy required for water pump is relatively small as compared to secondary compressor of other methods.

Among the four sub-cooling methods, the LSHX sub-cooling method has been successfully commercialized. The rest of the methods described in this paper are either under ongoing research or in the conceptualization or research stage. In conclusion, by comparing to the other methods, sub-cooling technologies give moderate COP improvement, which is higher than ejector as expansion device and nano-particles but lower than inverter. However, it is expected that the combination of two or the three sub-cooling methods and/or combination with other method, such as compact sub-cooler/heat exchanger, using high performance environmentally friendly refrigerants, and adding nano-particles to refrigerant can give a better improvement in COP. Therefore, research towards this direction is expected in future.

Acknowledgements

This study was funded by the Minister of Research, Technology and Higher Education Indonesia, under research university grant: Penelitian Berbasis Kompetensi: Fund No. 237.2/PL1.R7/LT/2018. The authors were also grateful to Politeknik Negeri Bandung and Universiti Teknikal Malaysia Melaka for the facilities and management supports.

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