International Refrigeration and Air Conditioning Conference
School of Mechanical Engineering

2008

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ABSTRACT

The purpose of the present study is to propose a new refrigeration cycle and clarify the performance characteristics of the cycle. (1) We defined a vapor compression refrigeration cycle based on the Ericsson cycle on a T-s diagram and a p-h diagram. (2) We clarified the practical method employed of achieving isothermal compression and expansion processes through utilization of multi-stage adiabatic compressors and an ordinary expansion valve. (3) Theoretical investigation of the cycle performance using a single stage compressor and an expansion valve demonstrated that COP values become maximum and constant when the dryness fraction is kept in a definite range. (4) The calculation results of performance characteristics of 12 different but typical refrigerants under the conditions of -40 °*C* evaporative temperature and 40 °*C* condensing temperature showed distinct improvements in COP for most of the sampled refrigerants.

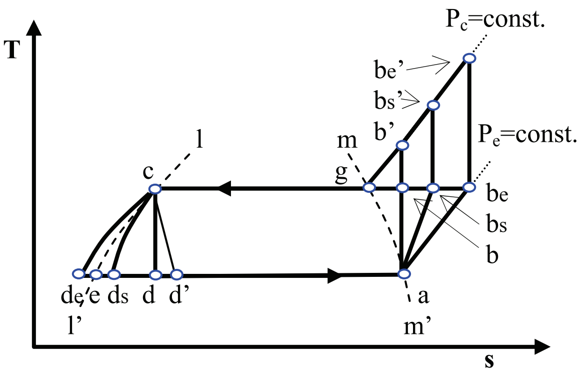
1. INTRODUCTION

Rapid progress has been made in the development of refrigerating and air conditioning equipment in recent years. The technological development and production of energy saving components were accelerated by the two so-called "oil crisis" in 1970s and kept continued up to the mid 1980s. In 1992, the 4th Meeting of the Parties to the Montreal Protocol (MOP4) held in Copenhagen amended and adjusted the Montreal Protocol to call for the complete abolishment of the use of CFC refrigerants by 1995 and HCFC by 2020. Stemming from the amendment of this protocol, complex technological development collaboration between industry, government and academia commenced in Japan. Since then, the development of energy-saving industrial refrigeration facilities and air conditioning systems has made enormous contributions to society (Nagatomo 2005, Nishimura 2005). Except for absorption and adsorption type refrigerators, industrial refrigeration and air conditioning systems have always utilized the vapor compression refrigeration cycle with the Rankine cycle as the basic cycle. On the other hand, however, research and development as to the refrigeration cycle using a new basic cycle have been started recently, together with popularization of CO2 as a natural refrigerant (Meunier *et al.* 2006, Hugenroth *et al.* 2006, Ino *et al.* 2007).

2. THEORETICAL ANALYSIS

2.1 Basic Cycle of Vapor Compression Refrigeration Cycle

Figure 1 shows the vapor compression refrigeration cycle as a practical utilization cycle using three different basic cycles on a T-s diagram. The basic cycle of the conventional vapor compression refrigeration cycle ab'gcd'a is an inverse Rankine's cycle, though the basic cycle thereof would be considered to be an inverse Carnot cycle abcda. Therefore, the inverse Carnot cycle abcda is regarded as the basic cycle of the vapor compression refrigeration cycle. The Stirling cycle and Ericsson cycle would be also regarded as other basic cycles available for practical vapor compression refrigeration. The vapor compression refrigeration cycle $ab_sb_s'gcd_sa$ corresponds to the inverse-Stirling cycle ab_scd_sa, and the cycle $ab_eb_e'gcd_eea$ corresponds to the inverse-Ericsson cycle ab_ecd_ea respectively. The purpose of the present study is to clarify the performance characteristics of a vapor compression refrigeration cycle based on the Ericsson cycle, and to determine the most suitable method for selecting a refrigeration cycle and a refrigerant in



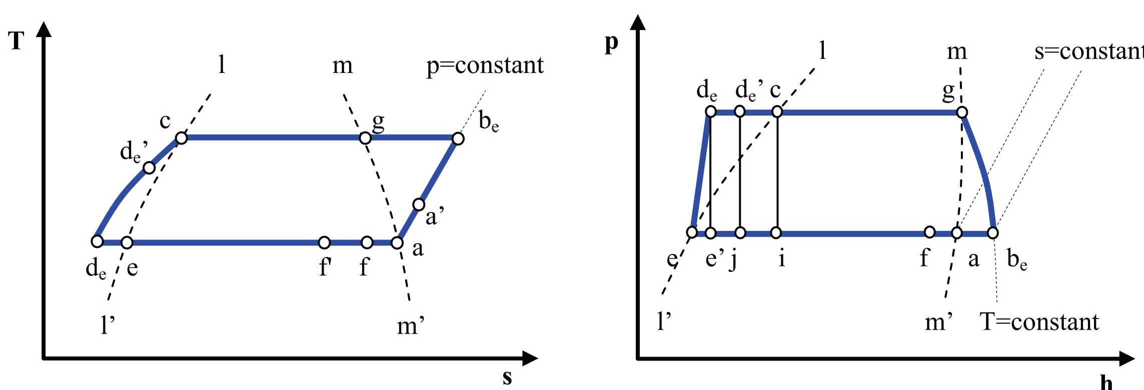
Basic cycle process configurations			
Process	Ericsson	Stirling	Carnot
isobaric regeneration	ab_e		
isochoric regeneration		ab_s	
adiabatic compression			ab
isothermal compression	b_eg	b_sg	bg
condensation	gc	gc	gc
isobaric regeneration	cd_e		
isochoric regeneration		cd_s	
isothermal expansion	d_ee		
adiabatic expansion			cd
evaporation	ea	d_sa	da
Practical compression process			
compression and cooling	$b_eb_e'g$	$b_sb_s'g$	ab'g

Figure 1: Typical basic cycles for vapor compression refrigeration cycle

comparison with the existing conventional refrigeration cycle.

2.2 Irreversibility in Regenerative Heat Transfer Process

The reversibility is discussed here of the three basic cycles shown in Fig.1, which are utilized for the vapor compression refrigeration cycles including phase changes. It is obvious that the inverse-Carnot cycle abcda is reversible, but it is not clear whether the inverse-Ericsson cycle ab_ecd_ea and the inverse-Stirling cycle ab_scd_sa are reversible or not. In Figure 2, the vapor compression refrigeration cycle with the inverse-Ericsson cycle as a basic cycle is shown on the T-s diagram and the p-h diagram. Assuming the reversible cycle ab_ecd_ea, the cycle consists of four basic processes which are ①isobaric regeneration process ab_e for super-heating evaporative vapor, ②isothermal process b_ec of compression b_eg and condensation gc, ③isobaric regeneration process cd_e for sub-cooling condensate liquid and ④isothermal process d_ea of expansion d_ee and evaporation ea. In the Ericsson cycle, both isobaric processes of ab_e and cd_e are required to exchange an equal amount of heat using a counter flow heat exchanger (hereinafter called the "regenerative heat exchanger"). But in this case, the two flows in the heat exchanger will experience different temperature changes due to the different capacitance rates. Therefore, the point d_e' on the process cd_e can be defined as the point that the enthalpy difference of the process cd_e' is equal to that of the process ab_e, satisfying the Equation (1). All the Equations are shown in numbered order at the end of the sub section **2.3**.
In the same manner, the point f on the evaporative process ea can be defined as the point satisfying the Equation (2). Equation (2) is the conditional formula required to establish the inverse-Ericsson cycle ab_egcd_eea. Equation (2) also indicates that the gas inlet point of the regenerative heat exchanger is transferred from a saturated vapor point a to a wet vapor point f on an evaporating line. Next, we investigate the reversibility of the regenerative heat exchanger during the heat transfer process. The heat transfer process satisfying Equation (1) is an irreversible process, because the temperature difference $(T_{de'} - T_a)$ is inevitably formed between a liquid refrigerant and a gas refrigerant at the lowest temperature end section of the regenerative heat exchanger shown in Figure 2(a). On the other hand, the heat transfer process satisfying Equation (2) is also an irreversible process. The temperature difference between the liquid



(a) T-s diagram (b) p-h diagram

Figure 2 : Ericsson vapor compression refrigeration cycle

and the gas of a regenerative heat exchanger is zero at both the lowest temperature side and the highest temperature side, but the temperature difference formed in the intermediate heat transfer range is inevitable, therefore the heat transfer process is irreversible. The entropy generation in the irreversible process of a regenerative heat exchanger will be referred to later in the paper. As is clear from the above discussion, the inverse-Ericsson cycle ab_egcd_eea shown in Figure 2 is an irreversible cycle. The same discussion can be applied for the inverse-Stirling cycle ab_sgcd_sea shown in Fig.1 and thus the inverse-Stirling cycle is also an irreversible cycle.

2.3 Theoretical Analysis of Ericsson Vapor Compression Refrigeration Cycle

We select the Carnot cycle and Ericsson cycle from the three different basic cycles shown in Figure 1, then analyze the performance of a vapor compression refrigeration cycle which utilizes an expansion valve in the expansion process. We define the practical cycle abb'gcd'a as the Carnot vapor compression refrigeration cycle and the practical cycle ab_eb_e'gcd_eea as the Ericsson's vapor compression refrigeration cycle. In the paper, the former practical cycle is termed the conventional cycle and the latter is termed this cycle.

2.3.1 Practical isothermal expansion method and regenerative effect: If the inlet gas status of the regenerative heat exchanger is regarded as the saturated vapor point a in Figure 2, the refrigeration capacity $\boldsymbol{\Phi_{oea}}$ of the cycle is shown by Equation (3). If the inlet gas status of the regenerative heat exchanger is regarded as point f, the refrigeration capacity $\boldsymbol{\Phi_{oef}}$ of the cycle can be represented by Equation (4). Point e' shown in the above Equation (4) and Figure 2(b) indicates the terminal point of the isenthalpic expansion process from the liquid outlet point d_e of the regenerative heat exchanger to the evaporative pressure through the expansion valve. In the same manner, point i and point j are the terminal points of the isenthalpic expansion process from the saturated liquid point c of the conventional cycle and the sub-cooled point $d_{e'}$ of this cycle. Referring to p-h diagrams of refrigerants used under usual operating conditions, the isothermal expansion line d_ee and the isenthalpic expansion line d_ee' with their starting points sub-cooled to the evaporative temperature as shown in Figure 2 almost overlap or become close to each other. Figure 3 shows the relationship between these expansion lines of the cycle on a p-h diagram for ammonia refrigerant. As is clear from this diagram, the isothermal expansion line from the starting point d_e to the evaporative pressure line agrees closely with the isenthalpic expansion line d_ee', also getting close to the isentropic line. These cycle properties are observed when the starting temperature of the expansion process is sub-cooled to the evaporative temperature and the temperature difference between the evaporative temperature and the critical temperature is kept larger than a definite range. These thermodynamic characteristics indicate that there is little functional difference between an isothermal expander and an isenthalpic expansion valve in the Ericsson cycle. This is the reason why an expansion valve can be substituted for an isothermal expander. The refrigeration capacity $\boldsymbol{\Phi_{oc}}$ of the conventional cycle is shown by Equation (5). The refrigeration capacity difference $\boldsymbol{\Delta\Phi_{oce}}$ between the conventional cycle and this cycle is shown in Equation (6) which is induced using Equations (2) through (5). As is clear from Equation (6), the refrigeration capacity increased by this cycle is equivalent to the amount of enthalpy exchanged in the regenerative heat exchanger. The increased effect on refrigerating capacity obtainable by utilization of the regenerative heat exchanger is termed the "regenerative effect" hereafter in the paper. The factors influencing the regenerative effects of actual refrigerants are discussed here based on Equation (6). The mass flow rate $\boldsymbol{q_{mr}}$ of an ideal gas is a function of the compressibility factor $\boldsymbol{pv/RT}$ as is obvious from Equation (15) mentioned

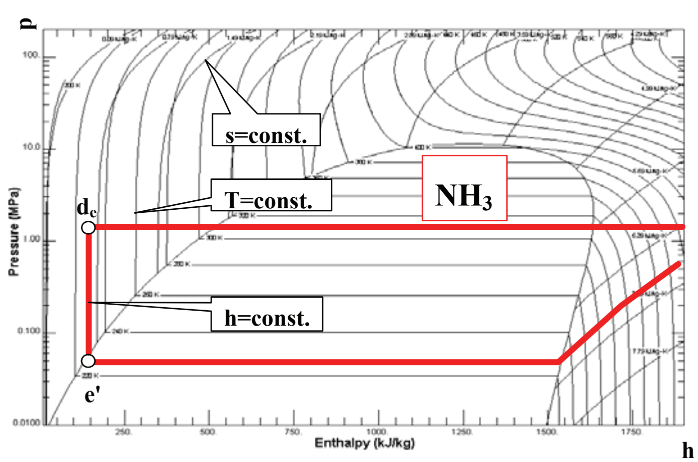


Figure 3: Relationship between contour lines

after. On the other hand, the specific enthalpy difference $(h_{be} - h_a)$ of the ideal gas is a function of isobaric specific heat, that is, a function of the gas constant R and specific heat ratio κ. Therefore the specific enthalpy difference of an actual refrigerant gas can also be regarded as the function of the compressibility factor and the specific heat ratio. Taking the above-mentioned into consideration, the compressibility factor and the specific heat ratio of the actual refrigerants are probably considered as the major factors influencing the regenerative effect in a regenerative heat exchanger.

2.3.2 Influence of dryness fraction on refrigeration capacity: The relationship between the refrigeration capacity and the dryness fraction at the gas inlet point of a regenerative heat exchanger is discussed here. When the gas inlet point is transferred inside and outside of a range fa with the dryness fraction x as shown in Figure 2, the refrigeration capacity can be divided into the following three cases for the dryness fraction x.

Case 1: refrigeration capacity when $x_f \le x \le x_a = 1$

Case 2: refrigeration capacity when $x \le x_f$

Case 3: refrigeration capacity when $x = 1 \,\&\, T_{a'} > T_a$

Case 1 : refrigeration capacity when $x_f \le x \le 1$ Using the earlier Equations (3), (4) and (6), Equations (7) and (8) in relation to refrigeration effects $\Delta h_{oea}, \Delta h_{oef}$ and the refrigeration capacity Φ_{oe} are obtained. From Equation (8), the refrigeration capacity Φ_{oe} becomes constant for the same mass flow rate when the dryness fraction x is in the range $x_f \le x \le x_a = 1$.

Case 2 : Refrigeration capacity when $x \le x_f$ Assuming that the gas inlet of the regenerative heat exchanger is designated at point f' in Figure 2(a), the refrigeration capacity $\Phi_{oef'}$ and the relationship between $\Phi_{oef'}$ and Φ_{oe} can be represented by Equations (9) and (10) respectively. As the dryness fraction x decreases in the range of $x \le x_f$, the refrigeration capacity for the same mass flow rate also decreases from Equation (9).

Case 3 : Refrigeration capacity when $x = 1 \,\&\, T_{a'} > T_a$ Assuming that the gas inlet point of the regenerative heat exchanger is regarded as superheated point $a'(x = 1 \,\&\, T_{a'} > T_a)$ as shown in Figure 2(a), the refrigeration capacity $\Phi_{oea'}, \Phi_{oca'}$ and the relationship between Φ_{oe} and $\Phi_{oea'}$ are indicated by the Equations (11), (12) and (13). Referring to Equation (11), the first right side term Φ_{oc} shows the refrigeration capacity of a conventional refrigeration cycle, the second term $\Phi_{oca'}$ referred to in Equation (12) shows the refrigeration capacity correspondent to the refrigeration effect $(h_{a'} - h_a)$ obtained from superheating the saturated gas, and the third term $q_{mr}(h_{be} - h_{a'})$ shows the refrigeration capacity increased by the regenerative effect. Equation (13) has equality only when the second right side term $\Phi_{oca'}$ in Equations (11) and (12) is perfectly utilized for refrigeration capacity without any loss. The results of the above discussion indicate that the refrigeration capacity Φ_{oe} for the same mass flow rate is maximum and constant when the inlet gas dryness x of the regenerative heat exchanger is in the range of $x_f \le x \le 1$.

2.3.3 Practical isothermal compression method and compression power: In the basic cycle ab$_e$gcd$_e$ea in Figure 1, multi-stage processes of adiabatic compression followed by isobaric inter-cooling can be practically substituted for isothermal compression process b$_e$g. The infinite multi-stage processes are exactly equivalent to the isothermal compression process. Taking the simplification of theory and easy comparison of characteristics with that of the conventional cycle into consideration, however, we discuss the single-stage adiabatic process b$_e$b$_{e'}$g substituted for the isothermal compression process b$_e$g. The theoretical adiabatic compression power $P_{e,ad}$ for ideal gas is calculated using the Equations (14) and (15). Taking the practical usage of compressors into consideration, the performance for the same compressor should be discussed using the same condensing and evaporative temperatures for each different refrigerant. Referring to the adiabatic compression power and refrigeration capacity mentioned in **Case 1** shown in sub-sub section **2.3.2**, the suction temperature of a compressor is always constant and the same as the condensing temperature T_{be}, therefore, the discharge temperature is constant and the specific heat ratio κ of Equation (14) and the refrigerant mass flow rate q_{mr} of Equation (15) are also constant. Consequently, the adiabatic compressor power $P_{e,ad}$ becomes constant and the refrigeration capacity Φ_{oe} becomes constant and maximum under the conditions that the dryness fraction x is in the range $x_f \le x \le 1$ in **case 1**.

2.3.4 COP and entropy generation: The refrigeration capacity and the adiabatic compression power for the same compressor operated under the same condensing and evaporative temperatures have been mentioned in this section. The COP of this cycle can be calculated using those results and the Equation (16). As is clear from the discussions in sub-sub section **2.3.3** and Equation (16), the COP of this cycle is constant when the dryness fraction x at the gas inlet point of the regenerative heat exchanger is kept in the range of $x_f \le x \le 1$. But it is not theoretically clear whether or not this COP is maximum in all the range of $0 \le x \le 1$ including **case 1** and **case 2**. Next, we discuss the amount of Entropy generated in a cycle under the condition of $x_f \le x \le 1$. In the cycle ab_eb_e'gcd_ea, as shown in Figure 1, we need to take into account the next three irreversible processes, which are ①a heat transfer process of a regenerative heat exchanger, ②an isenthalpic expansion process through an expansion valve and ③an isobaric cooling process for gas discharged from the compressor. Taking into consideration the fact that the COP is constant when the dryness fraction x is in the range of $x_f \le x \le 1$, the Equation (17) is derived from the second law of thermodynamics. Where, the entropy generation rates caused by the above process ③ are the same on both hand sides of Equation (17), therefore they are balanced out and not shown in the equation. Equation (17) shows that [(Entropy generation rate caused by the heat transfer process of the regenerative heat exchanger) + (Entropy generation rate caused by the expansion process)] is constant. That is to say, the entropy generation rate of the cycle is constant as long as the gas inlet point of the regenerative heat exchanger is in the range of fa shown in Figure 2.

2.3.5 Differences between Ericsson cycle and conventional cycle: Utilization of a liquid-gas heat exchanger for the conventional refrigeration cycle is widely recognized as one of the simplest and most common means of improving COP(JAR, 1993). Though the system configuration of this cycle is the same as that of the conventional refrigeration cycle with a liquid-gas heat exchanger(JAR 1993), the two cycles are conceptually different in the basic cycle, the working conditions and effectiveness of each component used in the cycle. Major differences between them as well as characteristics are summarized as follows. **(a)** The compression process of the basic Ericsson cycle is theoretically different from that of the conventional cycle. The compression process of the basic Carnot cycle is adiabatic and isothermal, but that of the Ericsson cycle is isothermal, as shown in Figure 1 and sub section 2.2. The multi-stage adiabatic compression processes followed by isobaric cooling processes through intercoolers can therefore be substituted for the isothermal compression process. **(b)** The working conditions and effectiveness of an expansion valve are also significantly different. The expansion process of a conventional cycle is isenthalpic, but that of this cycle can be regarded as isothermal, as mentioned in sub-sub section **2.3.1**. That is why the expansion valve can be substituted for an isothermal expansion process. **(c)** The COP of this cycle is usually larger than that of the conventional cycle, though the former COP does not accord with the latter COP except when the latter starting point of the expansion process reaches point d_e' as shown in Figure 2. But the gas outlet temperature of the liquid-gas heat exchanger, which is equal to the compressor suction temperature, has been designed to be kept lower than the condensing temperature to prevent compressor discharge temperature from rising to an extraordinary level. The liquid outlet temperature of the liquid-gas heat exchanger is therefore kept higher than that of point de'. **(d)** The starting temperature of the compression process in this cycle differs from that of the conventional cycle. The latter starting temperature is designed to be midway between the evaporative temperature and the condensing temperature, while the former starting temperature is always equal to the condensing temperature, which is equal to the gas outlet temperature of the regenerative heat exchanger.

$$(h_{be} - h_a) = (h_c - h_{de'}) \tag{1}$$

$$(h_{be} - h_f) = (h_c - h_{de}) \tag{2}$$

$$\Phi_{oea} = q_{mr} \cdot (h_a - h_j) \tag{3}$$

$$\Phi_{oef} = q_{mr} \cdot (h_f - h_{e'}) \tag{4}$$

$$\Phi_{oc} = q_{mr} \cdot (h_a - h_i) \tag{5}$$

$$\Delta\Phi_{oec} = \Phi_{oea} - \Phi_{oc} = \Phi_{oef} - \Phi_{oc} = q_{mr} \cdot (h_i - h_j) = q_{mr} \cdot (h_{be} - h_a) \tag{6}$$

$$\Delta h_{oea} = (h_a - h_j) = (h_f - h_{e'}) = \Delta h_{oef} \tag{7}$$

$$\Phi_{oe} = \Phi_{oea} = \Phi_{oef} \tag{8}$$

$$\Phi_{oef'} = q_{mr} \cdot (h_{f'} - h_{de}) = q_{mr} \cdot (h_{f'} - h_{e'}) \tag{9}$$

$$\Phi_{oe} > \Phi_{oef'} \tag{10}$$

$$\Phi_{oea'} = \Phi_{oc} + \Phi_{oca'} + q_{mr} \cdot (h_{be} - h_{a'}) \tag{11}$$

$$\Phi_{oca'} = q_{mr} (h_{a'} - h_a) \tag{12}$$

$$\Phi_{oe} \geq \Phi_{oea'} \tag{13}$$

$$P_{e,ad} = \{\kappa/(\kappa-1)\} \cdot p_e \cdot V_e \cdot \{(p_c/p_e)^{(\kappa-1)/\kappa} - 1\} \times 10^{-3} \tag{14}$$

$$p_e \cdot V_e / R = q_{mr} \cdot T_{be} \tag{15}$$

$$COP_e = (\Phi_{oe} \quad or \quad \Phi_{oef'} \quad or \quad \Phi_{oea'}) / P_{e,ad} \tag{16}$$

$$\{(s_{be} - s_a) - (s_c - s_{de'})\} + (s_j - s_{de'}) = \{(s_{be} - s_f) - (s_c - s_{de})\} + (s_{e'} - s_{de}) \tag{17}$$

3. RESUTS AND DISCUSSION

3.1 Method and Results of Calculations

Based on the thermodynamic properties of actual refrigerants, the refrigeration capacity, adiabatic compression power and COP of this cycle have been calculated for the **case 1** and **case 2** mentioned in sub-sub section **2.3.2** using Equations (1) through (16). Since the specific heat ratio κ of an actual refrigerant changes during an adiabatic compression process, the compression power can be calculated using the arithmetic mean value of the specific heat ratio between the compressor suction gas temperature and the compressor discharge gas temperature $80\,^{\circ}C$. The major reason why the compressor discharge gas temperature should be controlled at a constant $80\,^{\circ}C$ is to prevent the discharged gas from overheating through an adiabatic compression process. The discharge gas temperature can easily be controlled at $80\,^{\circ}C$ using an oil injected screw compressor. The results of calculations for the COP_e , the COP increase ratio and the refrigeration increase ratio for 12 different but typical refrigerants (NIST REFPROP Version 7.0) at -40 $^{\circ}C$ evaporative temperature and $40\,^{\circ}C$ condensing temperature are shown in Figures 4 through 6. The COP increase ratio shown as the ordinate axis in Figure 5 means the COP ratio of this cycle with the conventional cycle under the same condensing and evaporative conditions. The axis of abscissa shows the gas outlet temperature $^{\circ}C$ of the regenerative heat exchanger. The ordinate axis of Figure 6 shows the refrigeration capacity increase ratio which also means the refrigeration capacity ratio of this cycle compared with the conventional cycle under the same conditions. The compression power decrease ratio CPR demonstrated in Figure 7 means the compression power ratio of this cycle compared with the conventional cycle under the same conditions. In Figures 4 through 6, the ordinate values at the abscissa value -40 $^{\circ}C$ include the performance values for the conventional refrigeration cycle, while showing that the regenerative heat exchanger exerts no regenerative effect on a cycle. The ordinate values at $40\,^{\circ}C$ show the performances for the Ericsson cycle and the ordinate values intermediate between the abscissa values -40 $^{\circ}C$ and $40\,^{\circ}C$ indicate the performances in the range of $x \leq x_f$ for **case 2**. In case of an ideal gas, the regenerative effect, compression power and COP are a function of the temperature and specific heat ratio. The performance characteristics using an actual refrigerant would therefore also be affected by the temperature and specific heat ratio. To investigate the relationship between performance characteristics and the specific heat ratio under the same evaporative and condensing temperatures, Figure 7 has been prepared. Figure 7 shows the plotted relationships between three characteristic ratios and specific heat ratios of 12 different refrigerants at the gas outlet point of the regenerative heat exchanger. The ordinate axis includes the COP increase ratio $COPR$, the refrigerating capacity increase ratio RCR and the compression power decrease ratio CPR .

3.2 Discussion

The results of discussions regarding characteristics shown in Figures 4 through 7 are summarized as follows: (1) Figure 4 demonstrates that the COP of the cycle is indicated by the total sum of the COP value based on a conventional cycle at the abscissa of -40 $^{\circ}C$ and the COP increment generated by the Ericsson cycle. (2) In Figure 4, the essential reason why the COP of R600a shows a maximum ordinate value at the abscissa of $40\,^{\circ}C$ is attributed to

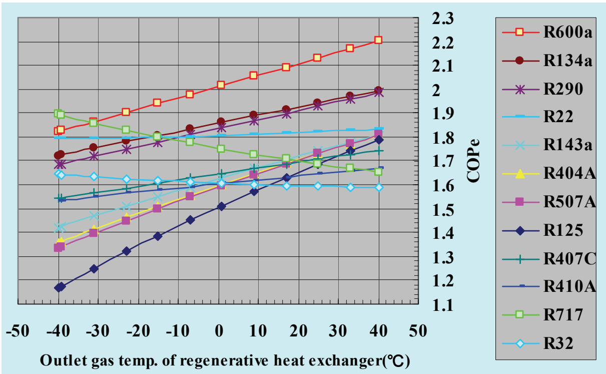


Figure 4: Relationship between COPe and outlet gas temperature at regenerative heat exchanger.

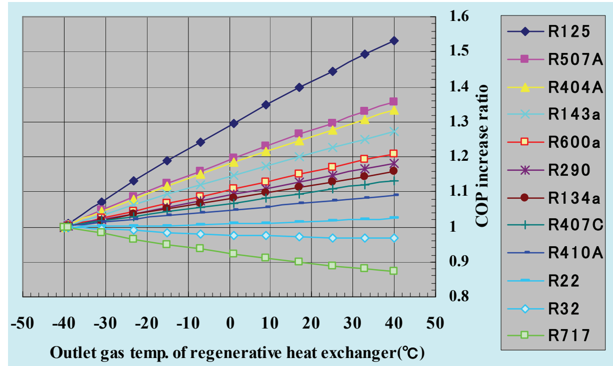


Figure 5: Relationship between COP increase ratio and outlet gas temperature at regenerative heat exchanger.

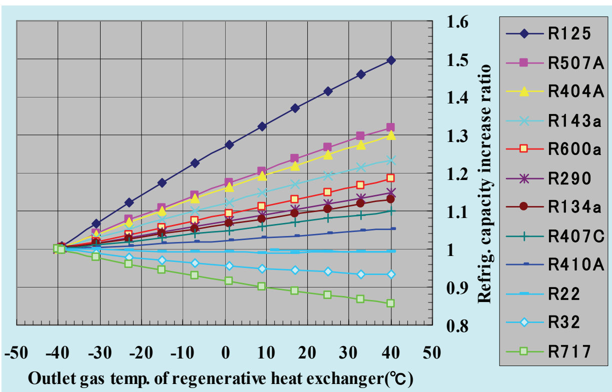


Figure 6: Relationship between RCR and outlet gas temperature at regenerative heat exchanger.

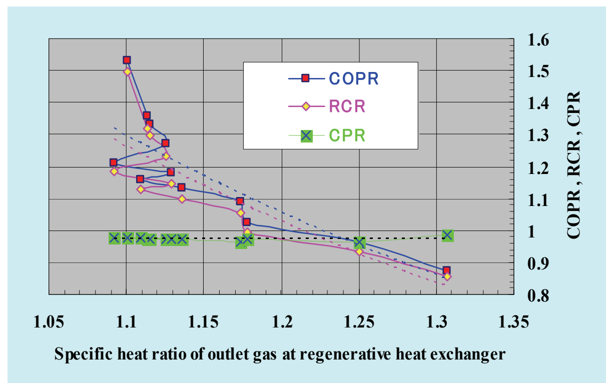


Figure 7: Relationship between characteristic ratios and specific heat ratios of 12 refrigerants.

the second largest ordinate value at the abscissa of $-40^{\circ}C$, while the essential reason why the COP of R717 shows a minimum value at $40^{\circ}C$ results from the negative gradient of the COP curve. (3) The reason why both the ***COPR*** and ***RCR*** shown in Figures 5 and 6 show the same descending order for all the refrigerants results from the small differences between the compression power decrease ratios (***CPR*** s), as shown in Figure 7. (4) The regenerative effect caused by a refrigerant increases as the specific heat ratio approaches 1, though turning to the negative effect when the specific heat ratio exceeds around 1.25, as shown in Figure 7. (5) The first essential factor exerting influence on the COP gradient shown in Figures 4 and 5 is the specific heat ratio, as is clear from the above (4). (6) Judging from the discussion in sub-sub section **2.3.1** and a result of the above (5)., the compressibility factor of a gas refrigerant should be taken into consideration as the secondary essential factor influencing the regenerative effect. (7) It remains to be resolved whether the mutual relationships between refrigerants and their performance characteristics will be further clarified for various types of refrigerants such as HFCs, hydrocarbons, inorganic compounds, organic compounds, etc by carrying out comprehensive research and study. (8) Taking into consideration the fact that the temperatures shown in the abscissa axis of Figures 4 thru 6 correspond to the gas outlet temperatures of a liquid gas heat exchanger used in a conventional cycle, the COP characteristics of the Ericsson cycle are available for estimation of the conventional cycle with a liquid gas heat exchanger. (9) From the results of discussion in the above (8), the maximum COP value attainable by the conventional cycle with a liquid gas heat exchanger is equivalent to the normal COP value of the Ericsson cycle under the same evaporative and condensing temperatures.

4. CONCLUSION

The results of this research and investigation can be summarized as follows: (1) The vapor compression refrigeration cycle based on a theoretical Ericsson cycle was defined on a T-s diagram and a p-h diagram. (2) We proposed the new cycle and clarified the practical method employed of achieving isothermal compression/expansion in the Ericsson cycle through utilization of the multi-stage adiabatic compressors and the usual expansion valve. (3)The calculation results of the cycle using a single stage compressor and an expansion valve demonstrate that the COP values become maximum and constant for the sampled refrigerants other than R32 and R717 when the dryness fraction x is in the range of $x_f \le x \le 1$. (4) From the calculation results of the performance characteristics under the conditions of -40 $^\circ C$ evaporative temperature and 40 $^\circ C$ condensing temperature, the following findings are obtained: (a) The COP of the cycle is indicated by the total sum of the COP based on the conventional cycle and the COP increment by the regenerative effect of the Ericsson cycle. (b) The first essential factor of the regenerative effect in the cycle is the specific heat ratio, while the secondary factor is probably the compressibility factor of the actual refrigerant gas as well as an ideal gas. (c) Both the Ericsson cycle and the conventional cycle with a liquid gas heat exchanger have the same system configuration, however, the two cycles are conceptually different in the basic cycle, the working conditions and effectiveness of each component.

NOMENCLATURE

Symbol	Description	Unit	Symbol	Description	Unit
COP_e	coefficient of performance	—	R	gas constant	(J/kgK)
$COPR$	COP increase ratio	—	RCR	refrig. capacity increase ratio	—
CPR	compression power ratio	—	$s_a, s_b,,$	specific entropy at point a, b, ,	(J/kgK)
$\Delta\Phi_{oec}$	refrig. cap. diff. $(\Phi_{oe} - \Phi_{oc})$	(kW)	$T_a, T_{a'}$	temperature at point a, a’	(K)
Φ_{oc}	refrig. capacity of conv. cycle	(kW)	T_{be}	temperature at point be	(K)
Φ_{oe}	refrig. cap. of Ericsson cycle	(kW)	V_e	theoretical flow rate of comp.	(m^3/s)
$\Phi_{oca'}$	refrig. cap. by superheat aa’	(kW)	x, x_f	dryness, dryness at point f	—
$\Phi_{oea'}$	refrig. capacity at point a'	(kW)		**Subscript**	
Φ_{oef}	refrig. capacity at point f	(kW)	ad	adiabatic	
$\Phi_{oef'}$	refrig. capacity at point f’	(kW)	$a, a', b,$	cycle working point a, a', b,	
$h_a, h_b,,$	specific enthalpy at point a, b,	(J/kgK)	c	conventional or condensing	
κ	specific heat ratio	—	e	Ericsson cycle or evaporative	
$P_{e,ad}$	adiabatic compression power	(kW)	m	mass	
p_e, p_c	evap., condensing. pressure	(Pa)	o	refrigeration capacity	
q_{mr}	mass flow rate	(kg/s)	r	refrigeration	

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