Performance of A Three-Stage Condensation Heat Pump

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Key Words : Multi-stage condensation heat pump, Environmental protection, Energy efficiency, CFC11, HCFC123, HCFC141b

Abstract

In this study, computer simulation programs were developed for single-stage, two-stage, and three-stage condensation heat pumps and their performance with CFC11, HCFC123, HCFC141b was examined under the same external conditions. The results showed that the coefficient of performance(COP) of an optimized 'non-split type' three-stage condensation heat pump is 25-42% higher than that of a conventional single-stage heat pump. The increase in COP, however, differed among the fluids tested. The improvement in COP is largely due to the decrease in average LMTDs in condensers, which results in the decrease in thermodynamic irreversibility in heat exchange process. For the three-stage heat pump, the highest COP is achieved when the total condenser area is evenly distributed among the three condensers. For the two-stage heat pump, however, the optimum distribution of the total condenser area varies with an individual working fluid. For the three-stage system, 'splitting the condenser cooling water' for the use of intermediate and high pressure subcoolers helps increase the COP further. When the individual cooling water entering the intermediate and high pressure subcoolers is roughly 10% of the total condenser cooling water, the maximum COP is achieved showing roughly an 11% increase in COP as compared to that of the 'non-split type' heat pump.

Nomenclature

A : heat transfer area [m²]

Cp : specific heat [kJ/kg℃]

h : enthalpy of refrigerant [kJ/kg]

m : mass flow rate [kg/s]

Q : heat transfer rate [kW]

T : temperature [℃]

TS : temperature of heat transfer fluid [℃]

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U : overall heat transfer coefficient [W/m²℃]

Subscript

C : condenser
E : evaporator
ECALTP : calculated two-phase region of evaporator
ESUP : superheated region of evaporator
ETP : two-phase region of evaporator
HC : high pressure condenser
HCCALTP: calculated two-phase region of high pressure condenser
HCSUB : subcooled region of high pressure condenser
HCSUP : superheated region of high pressure condenser
HCTP : two-phase region of high pressure condenser
IC : intermediate pressure condenser
ICCALTP : calculated two-phase region of intermediate pressure condenser
ICSUB : subcooled region of intermediate pressure condenser
ICSUP : superheated region of intermediate pressure condenser
ICTP : two-phase region of intermediate pressure condenser
LC : low pressure condenser
LCCALTP: calculated two-phase region of low pressure condenser
LCSUB : subcooled region of low pressure condenser
LCSUP : superheated region of low pressure condenser
LCTP : two-phase region of low pressure condenser
RE : refrigerant flowing in the evaporator
RHC : refrigerant flowing in the high pressure condenser
RIC : refrigerant flowing in the intermediate pressure condenser
RLC : refrigerant flowing in the low pressure condenser
WC : water flowing into the condenser
WE : water flowing into the evaporator
WHS : water flowing in the high pressure subcooler
WIS : water flowing in the intermediate pressure subcooler
WLC : water flowing in the low pressure condenser

1. Introduction

Since the advent of the industrial revolution, energy resources have been greatly explored worldwide and these days much concern is expressed for their rapid depletion. Besides this, global warming due to the increase of carbon dioxide from the combustion of fossil fuels has become an important environmental issue requiring a worldwide regulation to prevent a global disaster. For the past few decades, CFCs have been used extensively in various refrigeration and air-conditioning fields due to their excellent properties. But it has been shown that they not only deplete the stratospheric ozone layer but also have an impact on the global warming and hence many nations agreed to regulate the production and consumption of CFCs.[1-3]

At present, the whole world recognizes that energy resources are limited and a reckless use of them would result in a global environmental problem. Hence much effort is expended for the development of highly efficient energy conversion devices. As well known, one way of utilizing energy efficiently is the use of heat pumps.[4-6] For the past decades, Japan has performed various research activities to develop high efficiency energy conversion devices. At present, a large portion of their total energy consumption is for the hot water supply to commercial and residential buildings. And heat pumps have been studied greatly in these applications as a means to reduce the energy consumption. Usually, the temperature of the hot water supplied by conventional heat pumps is in the range of 50-55℃ and much of the heat pump work has been carried out for this temperature range. For the past decade, however, Japanese researchers have examined high temperature/high efficiency heat pumps that can supply hot water in the range of 75-85℃. For this purpose, high temperature/high efficiency heat pumps utilizing multi-stage condensers or economizers have been studied. [(7,8)]

In order to meet the national demand for the conservation of energy and environment, Korea also has actively undertaken various high efficiency heat pump projects for the past years. As part of this on-going research, the performance of a super heat pump with economizers has been studied both experimentally and theoretically.[(9,10)] Little, however, is known about the high temperature/high efficiency heat pumps with multi-stage condensers. Hence in this study, a detailed simulation method for a multi-stage condensation heat pump will be presented, which will be followed by the simulation results. The method as well as the results presented in this paper will enhance our understanding of this kind of a high efficiency heat pump and will be useful to the related industry for the design and manufacture of this kind of system. This, of course, will help alleviate the global environmental problems in the long run.

2. System modeling

The three-stage condensation heat pump examined in this study is one which is capable of raising the temperature of water from 50 to 85℃ in the condensers and at the same time providing the coefficient of performance(COP) of more than 7.0. In order to systematically analyze the factors responsible for the performance increase, single-stage, two-stage, and three-stage condensation heat pumps are simulated separately.

In the simulation of heat pumps, the 'UA' model is often employed in which capabilities of heat exchangers are specified by the product of the overall heat transfer coefficient, U, and the heat transfer area, A. For a multi-stage condensation heat pump, unlike a conventional system, there are more than one condenser. Thus, the performance of the system would vary depending upon the area(or UA) distribution among the condensers e for a given total condenser heat exchange area(or UA). Therefore, in this study the effect of the UA distribution in the condensers will be studied by varying the UAs of the three condensers for a given value of total condenser UA.

In general, there are two types of the multi-stage condensation heat pumps. One is a 'non-split type' in which the total condenser cooling water enters the low, intermediate, and high pressure condensers in succession without having any split at the entrance of the low pressure condenser. Another is a 'split type' in which the condenser cooling water is split at the entrance of the low pressure condenser and the split water enters first the intermediate and high pressure subcoolers and then merges with the main stream. It has been known that the performance of a multi-stage condensation heat pump is improved by having subcoolers with splitting of the condenser cooling water. Therefore, in this study both 'the split type' and 'the non-split type' multi-stage heat pumps will be examined. For the 'split system', the optimum amount of water supplied to the intermediate and high pressure subcoolers for the maximum performance will be determined.

For the multi-stage condensation system, a low pressure refrigerant such as CFC11 has been used so far due to the pressure limitation caused by extremely high water temperature on the high-stage condenser. CFC11, however, is completely phased out in the developed countries and at present HCFC123 and HCFC141b could be used for this application since their vapor pressures are similar to that of CFC11. Hence, in this study the performance of multi-stage condensation heat pumps charged with CFC11, HCFC123, and HCFC141b will be examined.

2.1 Multi-stage condensation heat pump system

In this section, only the three-stage condensation heat pump will be described. Fig.1 shows the schematic diagram of the 'split type' three stage-condensation heat pump while Fig.2 shows the pressure-enthalpy diagram for this system.

The refrigerant flow in the system is as follows: First of all, the refrigerant leaves the evaporator as either saturated or superheated vapor(state 15 or 1) and compressed through a three-stage compression process. In the first stage, a part of the total refrigerant vapor from the evaporator, MRLC, enters the low pressure condenser and leaves the condenser as either saturated or subcooled liquid(state 4 or 5) and finally expands. In the next stage, another part of the refrigerant, MRIC, is compressed and enters the intermediate pressure condenser and leaves the condenser as either saturated or subcooled liquid(state 8 or 9). And this liquid is further subcooled to state 19 via an intermediate pressure subcooler and expands. Finally, the remaining refrigerant, MRHC, is compressed in the next stage and enters the high pressure condenser and leaves the condenser as saturated or subcooled liquid(state 12 or 13). And this liquid is further subcooled to state 17 via another subcooler and expands. The refrigerant streams from the three condensers expand first individually and merge together at the entrance of the evaporator at state 14 to complete the cycle.

For the 'non-split type' system, the total condenser cooling water passes through the three condensers in succession and hence the system is simpler to analyze. It has been known that for the 'split system', roughly 80% of the total condenser cooling water at about 50℃, MWLC, passes through the three condensers in succession as with the 'non-split system' and its temperature increases by about 10℃ for each stage resulting in the exiting hot water temperature of 80℃. Approximately, half of the remaining 20% of the condenser cooling water, MWIS, enters the intermediate pressure subcooler and absorbs heat from the liquid refrigerant and merges with the main cooling water stream from the low pressure condenser and finally enters the intermediate pressure condenser. In the like manner, another half of the remaining cooling water, MWHS, enters the high pressure subcooler directly and absorbs heat from the liquid refrigerant and merges with the cooling water from the intermediate pressure condenser and enters the high pressure condenser.

2.2 System modeling

As mentioned early, the performance of a conventional single-stage heat pump as well as two-stage and three-stage condensation heat pumps is examined individually in this study. For a fair comparison, the evaporator capacity, the evaporator UA, and the condenser UA are held constant for each system. For a two-stage or a three-stage heat pump, the optimum UA distribution for the given total condenser UA was chosen. The heat released through the high, intermediate, low pressure condensers is the sum of the evaporator capacity and the compressor work through the three compressors. The internal energy balance of the refrigerant and the external heat transfer between the refrigerant and water are described elsewhere.(11) For the evaporator and three condensers, the balance equations are similar and hence only the equations for the high pressure condenser will be given as follows:

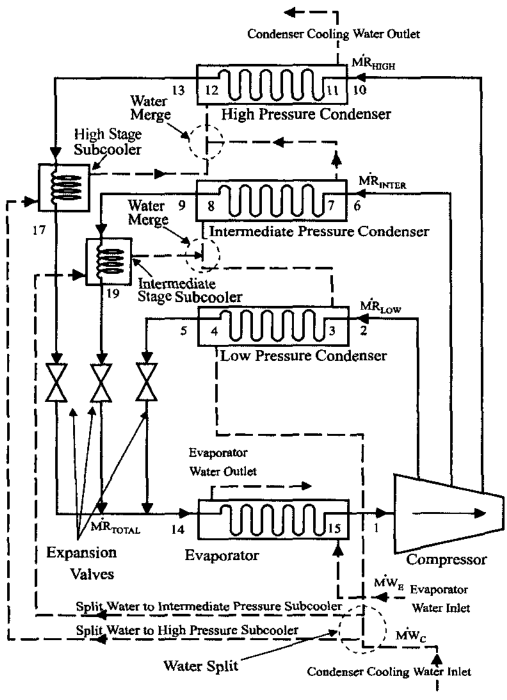


Fig.1 Schematic diagram of a 'split type' three-stage condensation heat pump.

$$F_{HCSUP} = (TS_{10} - TS_{11})/(TS_{10} - TS_{13}) \tag{1}$$

$$F_{HCSUB} = (TS_{12} - TS_{13})/(TS_{10} - TS_{13}) \tag{2}$$

$$F_{HCTP} = 1.0 - F_{HCSUP} - F_{HCSUB} \tag{3}$$

$$LMTD_{HC} = F_{HCSUB} * LMTD_{HCSUB} + F_{HCSUB} * LMTD_{HCSUB} + F_{HCSUP} * LMTD_{HCSUP} \tag{4}$$

$$Q_{HC} = U_{HC} * A_{HC} * LMTD_{HC} \tag{5}$$

where F_{HCSUP} and F_{HCSUB} are the ratios of heat transfer in the superheated and subcooled regions of the high pressure condenser to the total high pressure condenser heat while $LMTD_{HCSUB}$, $LMTD_{HCTP}$, $LMTD_{HCSUP}$ are the log mean temperature difference in the subcooled liquid, two-phase, and superheated vapor regions of the high pressure condenser.

2.3 Numerical methods

Table 1 shows the nineteen balance equations

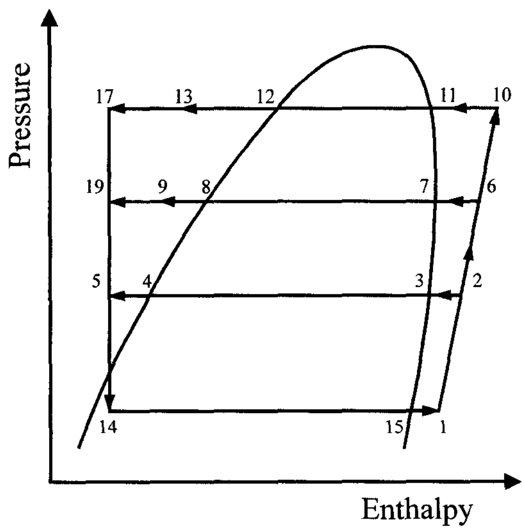


Fig.2 Cycle points of a 'split type' three-stage condensation heat pump.

established for the 'split type' three-stage heat pump.

Since these equations are nonlinearly coupled together, the Newton-Raphson method is utilized for their solution.[11,12,13] Refrigerant properties are calculated by the REFPROP program developed by US National Institute of Standards and Technology.[14]

Table 1 Various balance equations needed for the simulation of a three-stage condensation heat pump

Balance equation	Description
$RE(1) = -A_{ECALTP}+A_{ETP}$	Area balance in evaporator
$RE(2) = -A_{LCTP}+A_{LCCALTP}$	Area balance in low pressure condenser
$RE(3) = -A_{ICTP}+A_{ICCALTP}$	Area balance in intermediate pressure condenser
$RE(4) = -A_{HCTP}+A_{HCCALTP}$	Area balance in high pressure condenser
$RE(5) = -Q_E+M_E*(h_1-h_{14})$	Refrigerant energy balance in evaporator
$RE(6) = -F_{LCSUP}*(h_2-h_5)+(h_2-h_3)$	Definition of F_{LCSUP} in low pressure condenser
$RE(7) = -F_{ICSUP}*(h_6-h_9)+(h_6-h_7)$	Definition of F_{ICSUP} in intermediate pressure condenser.
$RE(8) = -F_{HCSUP}*(h_{10}-h_{13})+(h_{10}-h_{11})$	Definition of F_{HCSUP} in high pressure condenser
$RE(9) = -F_{LCSUB}*(h_2-h_5)+(h_4-h_5)$	Definition of F_{LCSUB} in low pressure condenser
$RE(10) = -F_{ICSUB}*(h_6-h_9)+(h_8-h_9)$	Definition of F_{ICSUB} in intermediate pressure condenser
$RE(11) = -F_{HCSUB}*(h_{10}-h_{13})+(h_{12}-h_{13})$	Definition of F_{HCSUB} in high pressure condenser
$RE(12) = -F_{ESUP}*(h_1-h_{14})+(h_1-h_{15})$	Definition of F_{ESUP} in evaporator
$RE(13) = -M_{RIC}*(h_6-h_9) +(M_{WLC}+M_{WIS})*Cp*(TS_6-TS_9)$	Energy balance between refrigerant and water in intermediate pressure condenser
$RE(14) = -M_{RHC}*(h_{10}-h_{13}) +(M_{WLC}+M_{WIS}+M_{WHS})*Cp*(TS_{10}-TS_{13})$	Energy balance between refrigerant and water in high pressure condenser
$RE(15) = -M_{RIC}*(h_9-h_{19})+(M_{WIS}*Cp*(TS_{18}-TS_{19}))$	Energy balance in intermediate pressure subcooler
$RE(16) = -M_{RHC}*(h_{13}-h_{17})+(M_{WHS}*Cp*(TS_{16}-TS_{17}))$	Energy balance in high pressure subcooler
$RE(17) = \{M_{WLC}*(TS_2-TS_5)+M_{WIS}*(TS_{18}-TS_{19}) +(M_{WLC}+M_{WIS})*(TS_6-TS_9)+M_{WHS}*(TS_{16}-TS_{17}) +(M_{WLC}+M_{WIS}+M_{WHS})*(TS_{10}-TS_{13})\}Cp - Q_C$	Energy balance for all condensers
$RE(18) = -TS_9*(M_{WLC}+M_{WIS}) +(M_{WLC}*TS_2+M_{WIS}*TS_{18})$	Energy balance at the water merging point of intermediate stage(point 18 in Fig.1)
$RE(19) = -TS_{13}*(M_{WLC}+M_{WIS}+M_{WHS}) +((M_{WLC}+M_{WIS})*TS_6+M_{WHS}*TS_{16})$	Energy balance at the water merging point of high stage(point 16 in Fig.1)

($TS_{19} = TS_{17} = TS_5$: Condenser water inlet temperature)

3. Results and discussion

3.1 Verification of the modeling work

A check for the validity of the modeling should be made before any optimization work is carried out using the computer simulation program developed in this study. Therefore, calculated results are compared to the experimental data by Japanese researchers.(8) Table 2 lists the calculated and measured results obtained under the same condition. As one can see easily, the calculated results are very similar to the experimental data.

3.2 Optimized UA distribution in the condensers

Table 2 Comparison between experimental and simulation data

		Experimental results	Simulation results	Deviation(%)
COP		7.6	7.3	3.9
Compressor work(kW)		321	338	5.0
Condenser heat(kW)		2440	2480	1.6
Water temp. at the exit of low pressure condenser(℃)		62.1	61.8	0.5
Water temp. at the exit of intermediate pressure condenser(℃)		73.2	73.7	0.7
Water temp. at the exit of high pressure condenser(℃)		85.1	85.0	0.1
Experimental condition				
Evaporator	Superheat (℃)		3.0	
	Pressure drop(kPa)		10.0	
	UA_E (kW/℃)		479.4	
Condenser	Subcooling(℃)	High	10.0	
		Inter.	7.0	
		Low	11.0	
	Pressure drop(kPa)	High	25.0	
		Inter.	21.0	
		Low	23.0	
	Total UA_C (kW/℃)		326.7	
Compressor efficiency	80.3 %			
Heat source fluid	Water flow rate(kg/s)		102.5	
	Inlet temp.(℃)		50.0	
	Exit temp.(℃)		45.0	
Heat sink fluid	Water flow rate(kg/s)		16.8(*)	
	Inlet temp.(℃)		50.0	

(*) Total mass flow rate is split into 13.4, 1.7, and 1.7kg/s for the low pressure condenser, intermediate and high pressure subcoolers respectively.

Table 3 Combination of UA distribution to the three condensers of the three-stage condensation heat pump

Number	UA in high pressure condenser(%)	UA in inter. pressure condenser(%)	UA in low pressure condenser(%)
1	33.3	33.3	33.4
2	80.0	10.0	10.0
3	70.0	20.0	10.0
4	70.0	10.0	20.0
5	60.0	30.0	10.0
6	60.0	20.0	20.0
7	60.0	10.0	30.0
8	50.0	40.0	10.0
9	50.0	30.0	20.0
10	50.0	20.0	30.0
11	50.0	10.0	40.0
12	40.0	50.0	10.0
13	40.0	40.0	20.0
14	40.0	30.0	30.0
15	40.0	20.0	40.0
16	40.0	10.0	50.0
17	30.0	60.0	10.0
18	30.0	50.0	20.0
19	30.0	40.0	30.0
20	30.0	30.0	40.0
21	30.0	20.0	50.0
22	30.0	10.0	60.0
23	20.0	70.0	10.0
24	20.0	60.0	20.0
25	20.0	50.0	30.0
26	20.0	40.0	40.0
27	20.0	30.0	50.0
28	20.0	20.0	60.0
29	20.0	10.0	70.0

Table 3 shows the UA distribution and Fig.3 shows the COPs calculated for CFC11, HCFC-123, and HCFC141b with the UA combinations in Table 3. As shown in Fig.3, the COPs of the three-stage heat pump vary greatly with the UA distribution. The highest COP is obtained when the total condenser UA is evenly distributed among the three condensers for all three fluids tested.

3.3 Performance improvement with the addition of stages

Table 4 lists various variables calculated for the single-stage, two-stage, and three-stage heat pumps obtained under the same condition's, i.e. the same evaporator capacity, the same evaporator UA, compressors with the same efficiency, and the same total condenser UA. For the two-stage and three-stage systems, results are obtained for the optimum UA distribution in the condensers without splitting the conden-

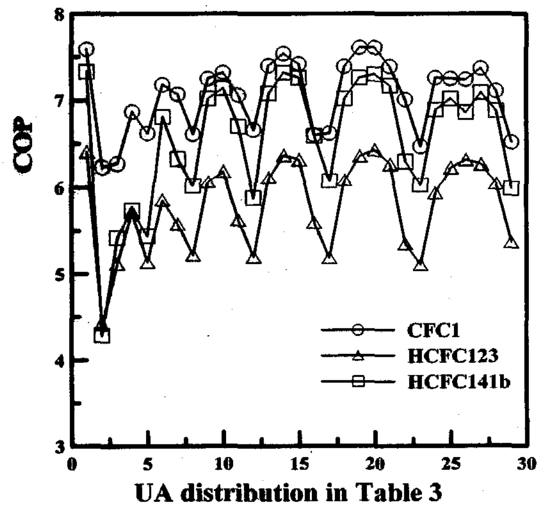


Fig.3 COP of a 'non-split' three-stage condensation heat pump with various UA combinations in three condensers.

Table 4 Optimized results for multi-stage condensation heat pumps

Refrigerant		CFC11			HCFC123			HCFC141b		
Number of stage		1	2	3	1	2	3	1	2	3
Compressor power(kW)		426	369	324	606	438	395	481	366	338
Condenser heat(kW)		2569	2511	2467	2748	2580	2537	2623	2509	2481
Pressure ratio between the evaporator and the low pressure condenser		3.16	2.64	1.27	3.57	2.01	1.32	3.43	2.05	1.32
Pressure ratio between low and intermediate pressure condensers			1.21	1.36		1.74	1.50		1.67	1.37
Pressure ratio between intermediate and high pressure condensers				1.93			1.90			1.97
Refrigerant flow rate(kg/s)		16.0	15.5	15.1	18.5	17.3	16.8	13.1	12.4	12.2
UA distribution (%)	High pressure condenser	100.0	57.0	30.0	100.0	70.0	30.0	100.0	65.0	33.0
	Inter. pressure condenser			40.0			30.0			33.0
	Low pressure condenser		43.0	30.0		30.0	40.0		35.0	34.0
Optimum COP		6.02	6.81	7.60	4.54	5.90	6.43	5.46	6.85	7.34

ser cooling water at the entrance of the low pressure condenser.

The results in Table 4 indicate that for the two-stage condensation heat pump, the optimum UA distribution varies among the refrigerants examined. Therefore, it is important to determine the optimum UA distribution for the two-stage heat pump charged with a certain refrigerant by computer simulation before

manufacture. Fig.4 shows that as the number of stage increases, the system COP increases for all refrigerant resulting in a significant improvement of 25-42%. For CFC11, 13.1% increase in COP was observed for the two-stage system as compared to the single-stage system and an additional 11.6% increase was achieved for the three-stage system resulting in a total increase in COP of 24.7%. For HCFC123, 30.0% and 11.6% increases in COP are achieved for each additional stage while for HCFC141b, 26.0% and 7.0% increases in COP are achieved respectively.

For all systems tested, the COP of CFC11 was the highest among the refrigerants examined, which was followed by HCFC141b and HCFC123. Even though HCFC123 showed the lowest COP, the increase in COP with additional stages was the highest and it also had the lowest compressor discharge temperature.

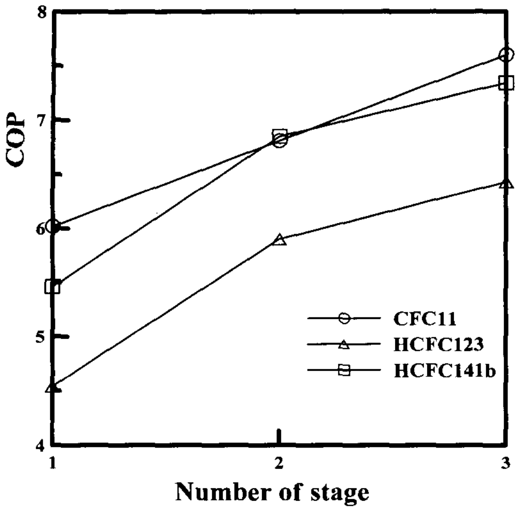


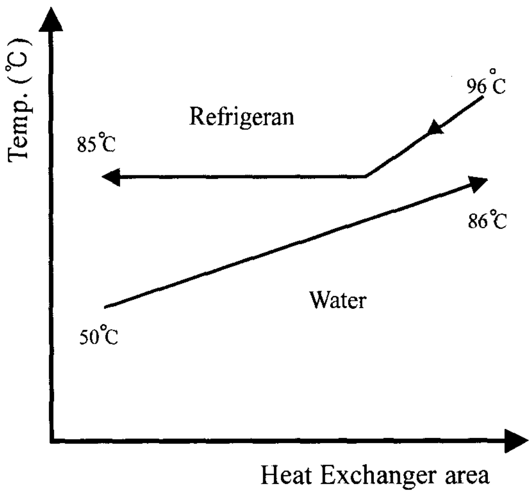
Fig.4 COP of 'non-split type' multi-stage condensation heat pumps.

3.4 Temperature distribution in condenser

Figure 5 shows the temperature distribution in the condenser for the single-stage and three-stage heat pumps using CFC11. For the single-stage system, the temperature difference at the exit of the condenser is quite large resulting in a large average LMTD in the condenser. For the three-stage system, however, 30-40% of the total condenser heat is released in the high pressure condenser and another 30-40% is released in the intermediate pressure condenser. Hence, the temperature difference between the refrigerant and water for each condenser is sm-

Table 5 LMTDs in the condensers for multi-stage condensation heat pumps

System \ LMTD		Average Log Mean Temperature Difference(℃)		
		CFC11	HCFC123	HCFC141b
1-stage system condenser		19.3	18.4	18.0
2-stage system	High pressure condenser	11.0	12.0	13.3
	Low pressure condenser	18.9	11.7	11.7
3-stage system	High pressure condenser	12.7	10.6	11.4
	Inter. pressure condenser	14.3	9.1	11.5
	Low pressure condenser	11.9	9.6	10.9



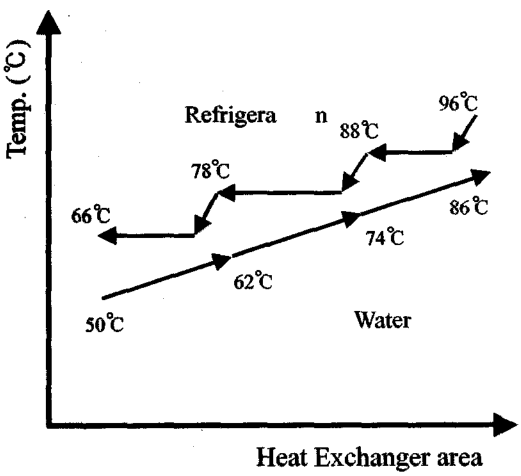


Fig.5 Temperature profiles of refrigerant and water in the condensers of the single-stage and three-stage heat pumps.

aller resulting in a smaller average LMTD in the overall condensation process. Due to the decrease in temperature difference in the heat exchange process, thermodynamic irreversibility decreases resulting in a significant increase in COP.(15)

In order to examine how the average LMTD varies with additional stages, an overall condenser LMTD is estimated from the results obtained for the three-stage system with three fluids. Fig.6 shows the average LMTD values in the condensers and Table 5 gives the calculated values.

As the system changes from a conventional single-stage to a three-stage heat pump, the average LMTD in the condensers decreases by 32%, 47%, and 37% for CFC11, HCFC123, and HCFC141b respectively. Based upon the results, it can be said that for the multi-stage system the temperature difference between the refrigerant and water in the condenser decreases significantly resulting in a decrease in thermodynamic irreversibility and a consequent increase in the system COP. Especially, HCFC123 showed the largest decrease in the average LMTD among the three fluids examined and hence it showed the largest increase in the system COP with the addition of stages as illustrated in Table4.

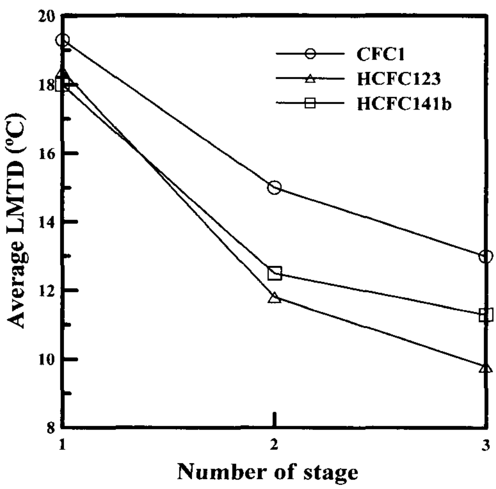


Fig.6 Average LMTDs in the condensersof the multi-stage condensation heat pumps.

3.5 Performance improvement by having subcoolers

To predict the system performance improvement by having subcoolers utilizing the split water at the entrance of the low pressure condenser, a three-stage heat pump with high and intermediate pressure subcoolers was simulated under the same condition. In the simulation, the performance of the subcoolers was specified by heat exchanger effectiveness.

Figures 7 and 8 show the refrigerant mass flow rate and compressor discharge temperature for 'the split type' three-stage system. In the simulation, the amount of the split water entering the high and intermediate pressure

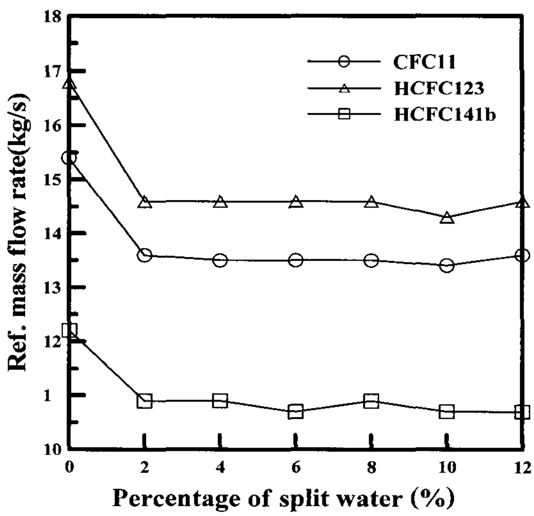


Fig.7 Refrigerant mass flow rate as a function of the amount of split water entering the high and intermediate condensers for the 'split type' three-stage heat pump.

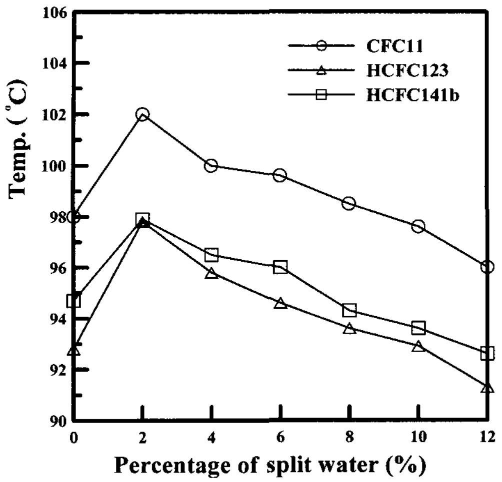


Fig.8 Compressor discharge temperature as a function of the amount of split water entering the high and intermediate condensers for the 'split type' three-stage heat pump.

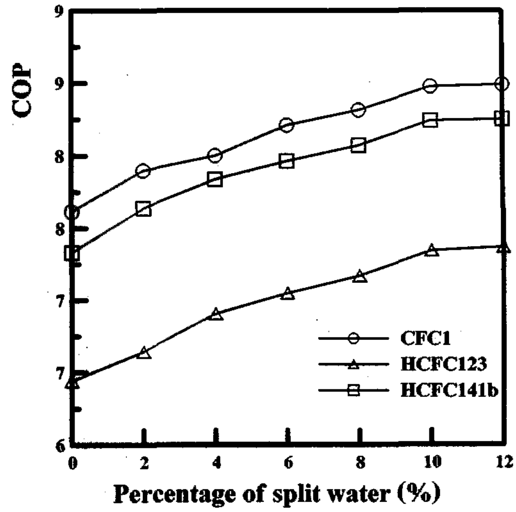


Fig.9 COP as a function of the amount of split water entering the high and intermediate condensers for the 'split type' three-stage heat pump.

subcoolers was assumed to be the same and the abscissas in Figures 7 and 8 are the percentages of the split water entering each subcooler. For all refrigerants examined, the mass flow rate decreases for 'the split type' system and hence the compressor work is decreased resulting in an increase in COP. The evaporator capacity, however, is still the same since the refrigeration effect per unit mass increases due to the subcoolers. Since the refrigerant mass flow rate decreases, the compressor discharge temperature increases by 3-4℃. However, as the amount of the water entering each subcooler increases roughly to 10% of the total condenser cooling water, the discharge temperature becomes quite similar to that without splitting the condenser cooling water.

Figure 9 shows how COP changes as the amount of the split water entering each subcooler increases. For all refrigerants, the system COP increases as the amount of the split water increases. However, when the amount of the split water for each subcooler exceeds roughly 10% of the total cooling water, the COP remains almost constant. From this result, it can be concluded that the highest COP of 'the split type' three-stage heat pump is obtained when the amount of the split water entering each subcooler is roughly 10% of the total condenser cooling water. This finding is also quite comparable with the Japanese experimental results(8) and thus indirectly validates the com-

Table 6 Comparison between 'split' and 'non-split' three-stage condensation heat pumps

Refrigerants	COP of 'non-split' system	COP of 'split' system	Increase in COP(%)
CFC11	7.614	8.474	10.1
HCFC123	6.44	7.343	12.3
HCFC141b	7.33	8.24	11.0

puter simulation model.

Finally, Table 6 shows the increase in COP of 'the split type' three-stage system as compared to 'the non-split type'. The results are obtained when 10% of the total condenser cooling water is assumed to enter individually into the intermediate and high pressure subcoolers for all refrigerants examined. The increase in COP due to the use of subcoolers varies among the refrigerants but the average increase is about 11.0%. Also HCFC123 benefited most through the use of subcoolers, with water split, which was followed by HCFC141b and CFC11.

4. Conclusions

In this study, thermodynamic performance of multi-stage condensation heat pumps was examined by computer simulation for CFC11, HCF C123, and HCFC141b and the following conclusions were obtained.

(1) For all refrigerants examined, the average LMTD in the condensers decrease with additional stages and hence thermodynamic irreversibility in the condensers decreases resulting in a significant increase in COP. For all systems studied, CFC11 showed the highest COP, which was followed by HCFC141b and HCFC123. As the number of stage increases, however, the performance of HCFC123 increased most. For the optimized 'non-split type' three-stage condensation heat pump, the performance was increased 25 to 42% as compared to the conventional single-stage system.

(2) For a given total condenser UA, the COP of a multi-stage condensation heat pump is affected by the UA distribution among the condensers. For a two-stage system, the optimum UA distribution yielding the highest COP was refrigerant dependent. On the other hand, the optimum UA distribution was almost the same for all refrigerants for a three-stage system and the highest COP was obtained when the total condenser UA was evenly distributed among the three condensers.

(3) For three-stage condensation heat pumps, utilizing subcoolers with the splitting of the condenser cooling water helped increase the system COP. When the amount of split water entering individually the high and intermediate pressure subcoolers was roughly 10% of the total condenser cooling water, an additional 11% increase in COP was achieved as compared to 'the non-split type' system.

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