DEVELOPMENT OF HIGH PERFORMANCE WATER-TO-WATER HEAT PUMP FOR GROUND-SOURCE APPLICATION

Yoshiro Shiba, M. Engineer, Development, Zeneral Heatpump Industry Co., Ltd, Nagoya, Japan
Ryozo Ooka, Ph.D., Associate Professor, University of Tokyo Institute of Industrial Science, Tokyo, Japan
Kentaro Sekine, M. Engineer, Technology Center, Taisei Corporation, Tokyo, Japan

ABSTRACT

High performance water-to-water heat pump for ground source application was developed by using plate heat exchanger, new type compressor and the refrigerate cycle with liquid-gas heat exchanger. Firstly, the performance design program that could compute the refrigerate cycle with liquid-gas heat exchanger was created. The simulating program could predict heating/cooling capacity, electric power and COP deciding surface area of heat exchangers, compressor efficiency, compressor displacement, etc. Secondly, the real heat pump was designed by the result of the program, created, and tested on the performance to compare with the simulation. As a result, it was computed that cooling COP of the designed heat pump attained 5.5 using the simulating model, and measured cooling COP of the real heat pump attained the same value.

Key Words: ***water-to-water heat pump, ground source, high performance.***

1 INTRODUCTION

Generally speaking, heat pumps using ground heat source have higher performance than using air heat source. But ground source heat pumps costs extra money of ground source heat exchangers and construction costs compared with air source heat pumps.

Therefore, reducing both running cost and initial cost of ground source heat pump system is needed for promoting. One of the solution methods is developing high performance and low price heat pump for ground source application.

Here, the performance design program developed to grasp the performance using some methods to improve performance and the experiment result of the experimental machine manufactured based on them are introduced.

2 METHODS IMPROVING PERFORMANCE

Following three are examined as methods improving performance of water-to-water heat pump. The basic composition of conventional machine and developing machine are summarized in Table 1 and Table 2 for comparison.

Table 1. Basic composition of conventional water-to-water heat pump

Compressor	Scroll compressor (Compressor A)
Capacity	15H.P.
Heat exchanger	Copper multiplex pipes
Cycle	Single-stage cycle
Refrigerant	R407C

Table 2. Basic composition of developing water-to-water heat pump

Compressor	New style scroll compressor (Compressor C)
Capacity	13H.P.
Heat exchanger	SUS plates (Having more than twice as much heating or cooling surface area as the conventional machine have)
Refrigerating Cycle	Cycle with a liquid-gas heat exchanger
Refrigerant	R407C (the same)

2.1 Adopting a Plate Heat Exchanger

Conventionally, the copper multiplex pipe heat exchanger has been used as a condenser and/or evaporator of water-to-water heat pump. In the developing machine, SUS plate heat exchangers with a twice as many heating or cooling surface area as the conventional heat exchanger was newly adopted in order to increase the surface area. (Fig. 1) In addition, volume successfully became about 1/2 of the conventional heat exchanger.

Fig. 1. Heat exchangers (Left: Conventional copper multiplex pipes, Right: SUS plates adopted newly)

2.2 Adopting a New Style Compressor

Adopting the scrolling compressor (compressor C) manufactured against the conventional scrolling compressor (compressor A) is examined. The manufacture of the compressor C explains that axial bearing load of the new style compressor become lowest possible (to reduce friction loss), and the compressor uses effect of perpendicular force direction with bigger radial bearing play sealing by centrifugal force in order to improve performance of the compressor. Fig. 2 shows appearances of three kinds of compressors including the compressor B and Fig. 3 shows the relation between the isentropic efficiency and the compression ratio shown by the manufacturer of Compressor C.

Although the conventional machine used a 15 H.P. compressor, the developing machine use smaller 13 H.P. compressor because of rise of capacity by adopting the plate heat exchangers with bigger surface area and the cycle with the liquid-gas heat exchanger described below.

Fig. 2. Compressor appearances (A, B, and C from left)

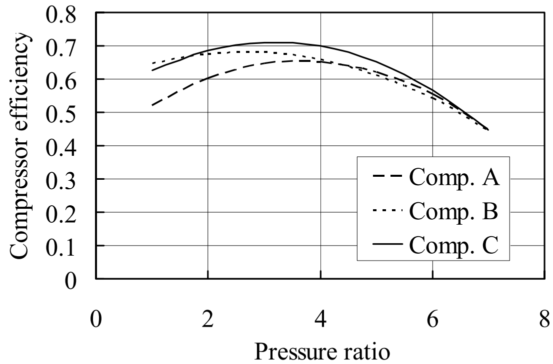


Fig. 3. Relation between compressor efficiency and compression ratio of compressors

2.3 Adopting the Cycle with a Liquid-Gas Heat Exchanger

By changing the basic single-stage cycle (Fig. 4) into a cycle with a liquid-gas heat exchanger (Fig. 5), capacity and COP of the heat pump improve because of raising evaporating temperature. It is because heat transfer is improved by disappear of dry out area and increase of wet area in an evaporator, the thermal gradient in condensation/evaporation process of non-azeotropic refrigerant mixture resemble the thermal gradient of hot/cool water, and the liquid-gas heat exchanger eliminates refrigerant mist. Furthermore, by increasing subcool, the liquid-gas heat exchanger inhibits flash gas which causes capacity depression in front of expansion valve.

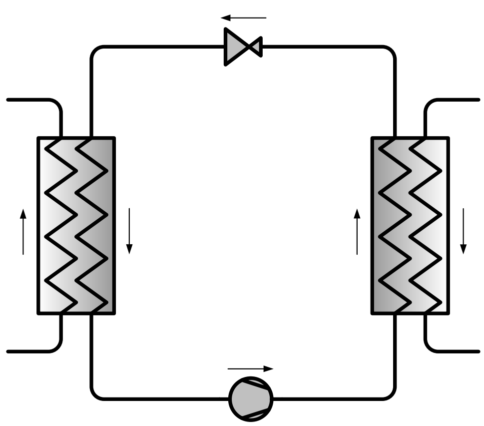


Fig. 4. Single-stage cycle

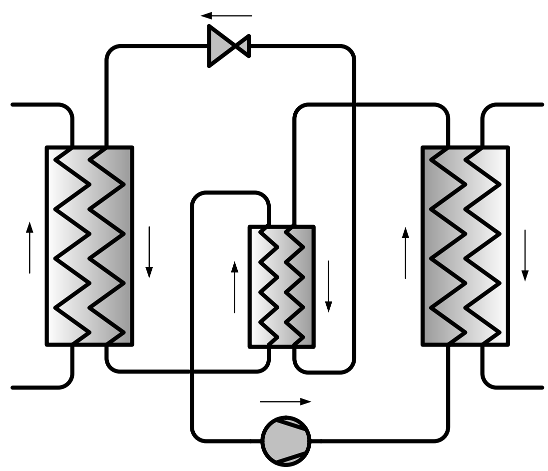


Fig. 5. Cycle with a liquid-gas heat exchanger

3 THE SCHEME OF PERFORMANCE DESIGN PROGRAM

In order to evaluate the performance, it is necessary to calculate various parameters, such as temperature of hot/cool inlet/outlet water, surface area of the heat-exchanger, efficiency of the compressor, electric power frequency, as well as to choice either the single-stage cycle or the cycle with liquid-gas heat exchanger. The performance design program uses spreadsheet software which can calculate performance simply and approximately. The scheme is as follows.

3.1 Variables, Functions, and Fixed Values

The variables used in the program are decided to be only two, the saturated liquid temperature after condensation, and the saturated gas temperature after evaporation.

Using the REFPROP data, polynomial approximations of pressures, densities and specific enthalpies to saturated temperatures and polynomial approximations of temperatures, densities, and specific enthalpies to saturated pressures are created as functions of physical properties.

The polynomial approximation functions of compressor efficiencies to compression ratio are shown in Fig. 3. The other physical properties are used as fixed values in approximation. The variable, function, and the fixed value used in the program are as follows:

1) Variables
  a) saturated gas temperature on the low pressure side: T_{gas}^{L}
  b) saturated liquid temperature on the high pressure side: T_{liq}^{H}
2) Polynomial approximations functions
  a) saturated gas pressure on the low pressure side: $p_{gas}^{L}(T_{gas}^{L})$,
     saturated gas specific enthalpy on the low pressure side: $h_{gas}^{L}(T_{gas}^{L})$,
     saturated gas density on the low pressure side: $\rho_{gas}^{L}\ (T_{gas}^{L})$
  b) saturated liquid pressure on the high pressure side: $p_{liq}^{H}(T_{liq}^{H})$
     saturated liquid specific enthalpy on the high pressure side: $h_{liq}^{H}(T_{liq}^{H})$
  c) saturated gas temperature on the high pressure side: $T_{gas}^{H}(p_{gas}^{H})$
     saturated gas specific enthalpy on the high pressure side: $h_{gas}^{H}(p_{gas}^{H})$
  d) saturated liquid temperature on the low pressure side: $T_{gas}^{L}(p_{gas}^{L})$
     saturated gas specific enthalpy on the high pressure side: $h_{gas}^{L}(p_{gas}^{L})$
  e) compressor efficiencies(of compressor A, B and C) $\eta(p_2 / p_1)$
3) Fixed values of refrigerant characteristics and cycle characteristics
  a) heat capacity ratio: k
  b) pressure drop
   —through gas pipe line on the high pressure side: Δp_{gas}^{H}
   —through condenser on the high pressure side: Δp_{co}^{H}
   —through liquid line on the high pressure side: Δp_{liq}^{H}
   —through liquid-gas heat exchanger on the high pressure side: Δp_{lg}^{H}
   —through evaporator on the low pressure side: Δp_{ev}^{L}
   —through liquid-gas heat exchanger on the low pressure side: Δp_{lg}^{L}
   —through gas pipe line on the low pressure side: Δp_{gas}^{L}
  c) superheat temperature (of gas) t_{sh}
  d) subcool temperature (of liquid)
   —of condenser t_{sc}^{co}
   —of liquid-gas heat exchanger t_{sc}^{lg}
  e) specific isobaric heat capacity
   —of saturated gas at 25°C C_{p}^{gas}
   —of saturated liquid at 25°C C_{p}^{liq}
  f) overall heat transfer
   —of condensation K_{co}

—of subcooling K_{sc}

—of evaporating K_{ev}

—of superheating K_{sh}

g) volumetric efficiency η_v

4) Fixed values used as parameters

a) compressor displacement (dependent on a power frequency) V

b) surface area of condenser or evaporator A

c) temperature of inlet/outlet hot/cool water (omitted here)

3.2 Various Conditions

Temperatures, pressures, and specific enthalpies before and behind compression, condensation, expansion, and evaporation in single-stage cycle, and before and behind compression, condensation, cooling in liquid-gas heat exchanger, expansion, evaporation, and heating in liquid-gas heat exchanger in the cycle with a liquid heat exchanger as well are as follows in approximation.

1) Single-stage cycle

a) After evaporation and before compression

$$T_1 = T^L_{gas} + t_{sh}, \quad p_1 \cong p^L_{gas} - \Delta p^L_{gas}, \quad h_1 \cong h^L_{gas} + C^{gas}_p (T_1 - T^L_{gas}) \tag{1}$$

b) After compression and before condensation

Specific enthalpy h_2^* and temperature T_2^* of superheat gas considered ideal gas are as follows:

$$h_2^* = h_1 + \frac{k}{k-1} p_1 v_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right\}, \quad T_2^* = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} \tag{2}$$

where $v(=1/\rho)$ is specific volume. In fact, isentropic efficiency η is taken into consideration.

$$h_2 = h_1 + \frac{\left(h_2^* - h_1\right)}{\eta}, \quad T_2 \cong T_2^* + \frac{\left(h_2^* - h_2\right)}{c^{gas}_p} \tag{3}$$

The pressure of the condition is as follows from saturated liquid pressure on the high pressure side and pressure drops.

$$p_2 = p^H_{liq} + \Delta p^H_{co} + \Delta p^H_{gas} \tag{4}$$

c) After condensation and before expansion

$$T_3 = T^H_{liq} - t^{co}_{sc}, \quad h_3 \cong h^H_{liq} - c^{liq}_p \left(T^H_{liq} - T_3\right), \quad p_3 \cong p^H_{liq} \tag{5}$$

d) After expansion and before evaporation

$$h_4 \cong h_3, \quad p_4 = p^L_{gas} + \Delta p^L_{ev}, \quad T^L_{liq} = T^L_{liq}(p^L_{liq}), \quad p^L_{liq} = p^L_{gas} + \Delta p^L_{ev},$$

$$T_4 \cong h_4 (T^L_{gas} - T^L_{liq}) / (h^L_{gas} - h^L_{liq}) + T^L_{liq} \tag{6}$$

2) Cycle with liquid-gas heat exchanger

In case of the cycle with liquid heat exchanger, temperatures, pressures, and specific enthalpies before compression, after compression and after condensation are the same expressions as in case of the single-stage cycle. The expressions except them are peculiar in the cycle with liquid-gas heat exchanger.
The notation (') attached to the number shows the cycle with liquid-gas heat exchanger.

a) after heating in liquid-gas heat exchanger and before compression

$$T_{1'} = T_1,\ h_{1'} = h_1,\ p_{1'} = p_1 \qquad \textbf{(7)}$$

b) after compression and before condensation

$$T_{2'} = T_2,\ h_{2'} = h_2,\ p_{2'} = p_2 \qquad \textbf{(8)}$$

c) after condensation and before cooling in liquid-gas heat exchanger

$$T_{3'} = T_3,\ h_{3'} = h_3,\ p_{3'} = p_3 \qquad \textbf{(9)}$$

d) after cooling in liquid-gas heat exchanger and before expansion

$$T_{4'} = T_{3'} - t_{sc}^{lg},\ \ p_{4'} = p_{liq}^{H} - \Delta p_{liq}^{H} - \Delta p_{lg}^{H},\ \ h_{4'} \cong h_{liq}^{H} - c_p^{liq} t_{sc}^{lg} \qquad \textbf{(10)}$$

e) after expansion and before evaporation

$$h_{5'} \cong h_{4'},\ \ p_{5'} = p_{gas}^{L} + \Delta p_{lg}^{L} + \Delta p_{ev}^{L},\ \ T_{liq}^{L} = T_{liq}^{L}(p_{liq}^{L}),\quad p_{liq}^{L} \cong p_{5'},$$

$$T_{5'} \cong h_{5'}(T_{gas}^{L} - T_{liq}^{L})/(h_{gas}^{L} - h_{liq}^{L}) + T_{liq}^{L} \qquad \textbf{(11)}$$

f) after evaporation and before cooling in liquid-gas heat exchanger

$$p_{6'} = p_{gas}^{L} + \Delta p_{lg}^{L}\ ,\ \ h_{6'} \cong h_{1'} - c_p^{gas} t_{sc}^{lg},\ \ T_{6'} \cong h_{6'}(T_{gas}^{L} - T_{liq}^{L})/(h_{gas}^{L} - h_{liq}^{L}) + T_{liq}^{L} \qquad \textbf{(12)}$$

3.3 Overall Heat Transfer Equals Latent and Sensible Heat Fluxes

Overall heat transfer Q^{trans} of heat exchanger in condenser and evaporator equals latent and sensible heat fluxes Q^{flux} of refrigerant, and the formulas are summarized as follow:

$$Q^{trans} = Q^{flux},\ \ Q^{trans} = KA\Delta T_{mean},\ \ Q^{flux} = \rho\eta_v V \Delta h \qquad \textbf{(13)}$$

where ΔT_{mean} is logarithmic mean temperature difference which is decided by temperatures of refrigerant and water before and behind heat exchangers.

3.4 Calculated Performance

The following performance values are calculated by the functions, the fixed values, and the formulas mentioned above.

1) Heating and cooling capacity

$$Q_{heat} = \rho\eta_v V(h_3 - h_2), Q_{cool} = \rho\eta_v V(h_4 - h_1),$$

$$Q'_{heat} = \rho\eta_v V(h_{3'} - h_{2'}), Q'_{cool} = \rho\eta_v V(h_{6'} - h_{5'}) \qquad \textbf{(14)}$$

2) Electric power

$$W = \eta_v \rho V(h_2 - h_1), \quad W' = \eta_v \rho V(h_{2'} - h_{1'}) \tag{15}$$

3) Heating and cooling COP (coefficient of performance)

$$\varepsilon_{heat} = Q_{heat} / W, \varepsilon_{cool} = Q_{cool} / W, \quad \varepsilon'_{heat} = Q'_{heat} / W', \varepsilon'_{cool} = Q'_{cool} / W' \tag{16}$$

The performance data and charts are calculated automatically by changing the temperatures of hot/cool inlet/outlet water.

3.5 Calculation Approach

By using Goal Seek function of spreadsheet software, T_{gas}^{L} and T_{liq}^{sat} are changed by turns so that overall heat transfer of heat exchanger and sensible and latent heat fluxes of refrigerant become equal as shown in (13). If the calculation is successful, it is considered that each value used by calculation is a solution. The calculation is completed, i.e. each value is converged, only if all parameter is input reasonable, but the calculation fails (some value overflows) if only one parameter is input un-reasonable. For example, if compressor capacity and the heat exchanger capacity are unbalance, the calculation is not completed, so it is considered that the refrigerating cycle is unstable and the heat pump with unbalance components cannot operate.

Therefore, the program cannot only calculate performances, but also check capacity balance of heat pump components.

4 CALCULATION RESULTS

The following capacity tables, comparison tables, p-h charts, capacity charts, etc. can be created speedy and simply by the performance design program, and the performances with different components can be compared.

4.1 Basic Performance

The performances calculated about the conventional machine and the developing machine are shown in Table 3 and Table 4, and the performances of not only cooling and heating but ice thermal storage, thaw and hot-water supply were calculated about each machine. Capacity and COP are shown in the tables about both cooling and heating because they are required to calculate pump discharges on the side of heat lord and on the side of heat source, and heat recovery performance.

In each operation mode, it is found that the developing machine has almost same capacity and higher COP compared with the conventional machine. Especially about cooling, it is found that COP of the development machine is 5.5, about 1.6 times as many as of the conventional machine, which COP is 3.5.

Table 3. Performance calculation result of the conventional heat pump

Operation mode	Cool inlet water Temp. [°C]	Cool outlet water temp. [°C]	Hot inlet water temp. [°C]	Hot outlet water temp. [°C]	Cooling capacity [kW]	Heating capacity [kW]	Power [kW]	Cooling COP	Heating COP
Cooling	12	7	25	30	44.0	56.6	12.7	3.5	4.5
Heating / Heat recovery	12	7	40	45	37.4	54.8	17.5	2.1	3.1
Ice thermal storage	-2	-5	20	25	31.3	42.4	11.1	2.8	3.8
Thaw	0	-5	20	25	33.0	42.9	10.0	3.3	4.3
Hot-water supply / Heat recovery	12	7	15	50	39.7	55.3	15.7	2.5	3.5

Table 4. Performance calculation result of the developing heat pump

Operation mode	Cool inlet water temp. [°C]	Cool outlet water temp. [°C]	Hot inlet water temp. [°C]	Hot outlet water temp. [°C]	Cooling capacity [kW]	Heating capacity [kW]	Power [kW]	Cooling COP	Heating COP
Cooling	12	7	25	30	45.6	53.9	8.3	5.5	6.5
Heating / Heat recovery	12	7	40	45	40.0	51.8	11.9	3.4	4.3
Ice thermal storage	-2	-5	20	25	30.7	38.6	8.0	3.8	4.8
Thaw	0	-5	20	25	32.1	39.1	7.1	4.5	5.5
Hot-water supply / Heat recovery	12	7	15	50	41.6	52.4	10.9	3.8	4.8

4.2 Effect of Methods Improving Performance

The effects of each above-mentioned methods improving performance were calculated and shown as follows.

4.2.1 Effect of plate heat exchanger

The calculated performances of the developing machine and a machine changed to only heat exchangers into the conventional heat exchangers were compared. According to the result (Table 5), it is found that COP improves about 28% by adopting plate heat exchangers.

Table 5. Effect of plate heat exchanger

Heat Exchanger	Cooling capacity (kW)	COP	COP ratio
Multiple Pipes (conventional)	41.2	4.28	100%
Plate (developing)	**45.6**	**5.48**	**128%**

Condition : frequency:60Hz; cool water:12→7°C, hot water:25→30°C, cycle with liquid-gas heat exchanger, new type compressor (C)

4.2.2 Effect of new type compressor

In the same way, the calculated performances of the improved developing machine and a machine changed to only a compressor into the conventional compressor A were compared. According to the result (Table 6), it is found that COP improves about 12% by adopting new type compressor C.

Table 6. Effect of new type compressor

Compressor	Cooling capacity(kW)	COP	COP ratio
A(conventional)	45.3	4.89	100%
B	47.8	5.26	108%
C(developing)	**45.6**	**5.48**	**112%**

Condition : frequency:60Hz; cool water:12→7°C, hot water:25→30°C, cycle with liquid-gas heat exchanger, plate heat exchangers

4.2.3 Effect of liquid-gas heat exchanger

In the same way, the calculated performances of the improved developing machine and a machine changed to only a cycle into the conventional single-stage cycle were compared. According to the result (Table 7), it is found that COP improves about 7% by adopting new cycle with liquid-gas heat exchanger.

Table 7. Effect of cycle with liquid-gas heat exchanger

Liquid-gas heat exchanger	Cooling Capacity(kW)	COP	COP ratio
None	42.7	5.12	100%
Equipped	**45.6**	**5.48**	**107%**

Condition : frequency:60Hz; cool water:12→7°C, hot water:25→30°C,
plate heat exchangers, new style compressor (C)

4.3 P-H Diagram

The P-H chart of a refrigerating cycle can be created on the spreadsheet performance design program. For example, the charts in the cooling conditions of the conventional machine and the developing machine are shown in Fig. 6 and Fig. 7. Figure 6 shows the single-stage cycle of the conventional machine and Table 7 shows the cycle with liquid-gas heat exchanger of the developing machine. Figure 7 shows that the specific enthalpy change from 3' to 4' equals that from 6' to 1' by heat exchange in the liquid-gas heat exchanger.
Figure 7 shows that the capacity and COP are improved since the compression ratio become small and the low pressure elevates compared with Fig. 6.

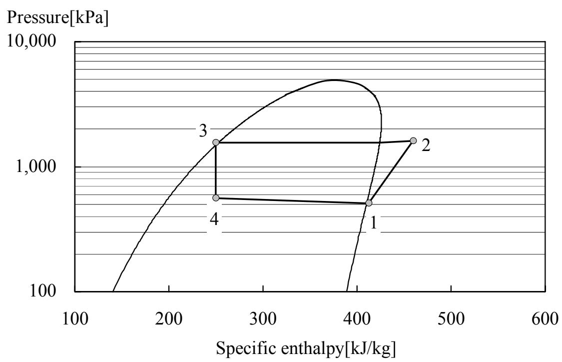


Fig. 6. P-H Diagram(the conventional machine)

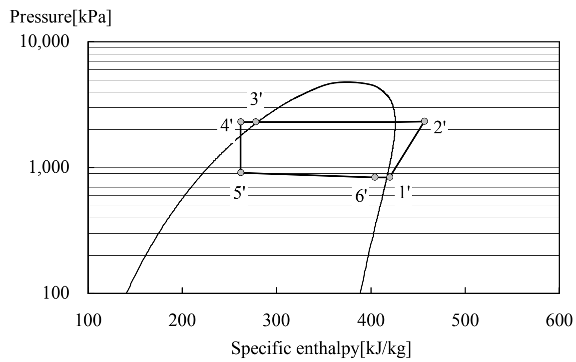


Fig. 7. P-H Diagram (the developing machine)

4.4 Performance Chart

When designing the water-to-water heat pump system for ground source application, temperatures of cool/hot inlet/outlet water are needed to design because the temperatures depend on the purpose, length of ground heat exchanger, soil temperature, etc. Therefore, the capacity diagram and COP chart are needed to decide the temperatures.

Here, the capacity charts (Fig. 8, Fig. 9 and Fig. 10) and COP charts (Fig. 11, Fig. 12) of the developing machine created by the spreadsheet performance design program are shown.
In addition, the computing time of creating all of the capacity charts and the COP charts was about 20 seconds using the computer with Intel Pentium4 2.4 GHz CPU and 512MB memories.

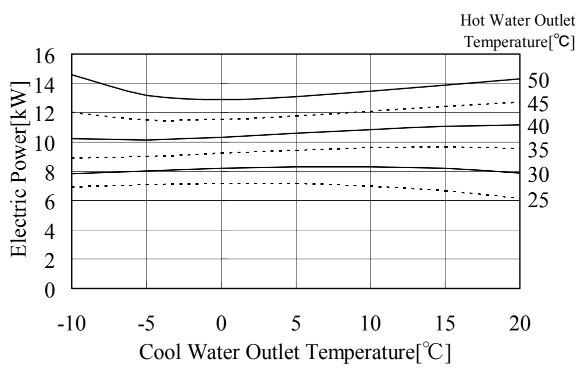
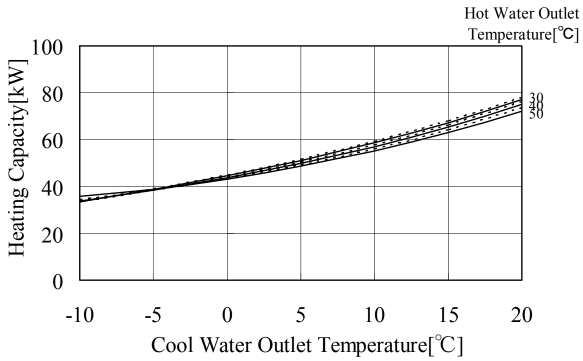


Fig. 8. Power chart (developing machine)



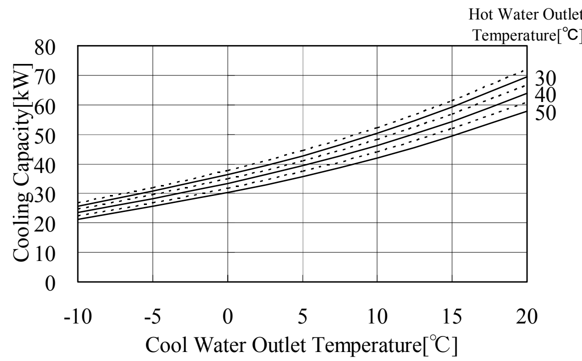
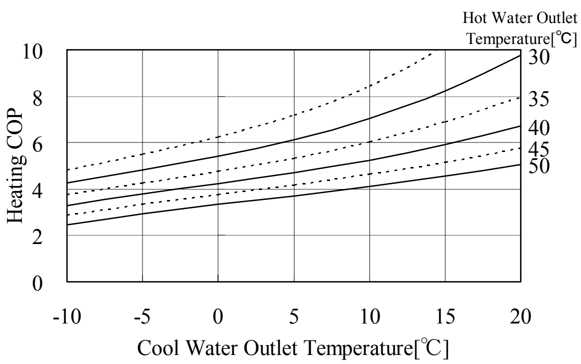


Fig. 9. Heating capacity chart (developing machine) **Fig. 10. Cooing capacity chart (developing machine)**



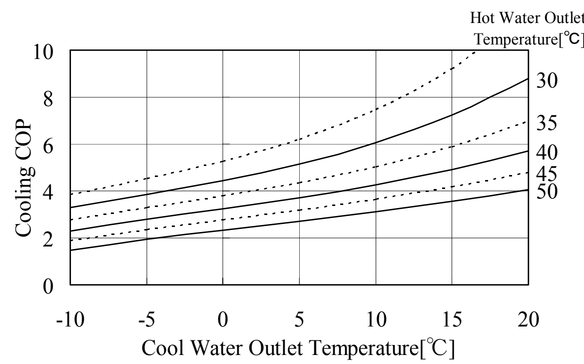


Fig. 11. Heating COP chart(developing machine) **Fig. 12. Cooling COP chart(developing machine)**

5 MANUFACTURE AND TEST

Before manufacture of trial developing machine, the following other methods improving performance were performed besides the three above-mentioned developments improving performance (adoptions of the plate type heat exchangers, the new style compressor, and the liquid-gas heat exchanger).

1) Withdrawal of accumulator

Since liquid-gas heat exchanger can be used as an accumulator in place, there is little liquid back to a compressor even if there is no accumulator in a heat pump with liquid-gas heat exchanger. Moreover, while the pressure loss on the low pressure side is reduced, capacity and COP improve by removing an accumulator, and initial cost of heat pump become cut also. Therefore, it is decided that the accumulator which the conventional machine has is removed in the developing machine

2) Withdrawal of oil separator

The oil separator used as usual is not needed if a heat pump has no oil stay, so decided to be removed. By removing the oil separator, capacity and COP improve since an oil return circuit is removed and the amount of refrigerant circulations increases. Moreover, the compression ratio of a compressor is reduced and electric power decreases by reducing the pressure loss on the high pressure side, and the initial cost becomes cut.

3) Flow counter in evaporator

In a heat pump using a four way valve, refrigerant and water in an evaporator flow parallel normally. Then, by installing two 3-way valves and reversing water flow, refrigerant and water in the evaporator flow counter. By doing so, capacity and COP improve since the logarithmic mean temperature difference of refrigerant and heat-source water becomes small and evaporating pressure raises. The trial machine adopting all of the methods improving performance was manufactured, and when actually run and measured, cooling COP attained 5.5.

6 CONCLUSION

The following developments for improving performance of the water-to-water heat pump for ground source application,

1) Adopting plate heat exchangers
2) Adoption new style compressor
3) Adopting a cycle with a liquid gas heat exchanger

were inquired, and the performance design program for simulating the effects of them was created.
As the result of the performance design program, and the result of measuring trial developing machine, both cooling COP attained 5.5.

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