12th IEA Heat Pump Conference 2017

Et Thermosiphones which collect heat from surface ground layers for CO_2 refrigeration cycle in cold climates: Effect of artery tube on performance

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Abstract

The utilization of heat pumps in Hokkaido, where temperature is very low in winter, remains as low as 5% so it is required to develop high-efficiency heat pumps used in the region. In this study, thermosiphons with a new shape have been designed and evaluated to collect heat from surface ground layers. Those thermosiphons utilize underground heat as a heat source for evaporators in order to prevent a reduction in heat pump efficiency. In the experiment, the performance of two types of thermosiphons, a 1-m-long corrugated and a smooth tube, has been evaluated. Moreover, the artery tube is inserted inside the thermosiphon to facilitate the recirculation of the working fluid from the condenser. Four types of artery tubes were used, three of which were showering tubes. Each showering tube is blocked on the bottom, with three holes opened on the side wall. As a result of performance evaluation, the smooth tube showed higher performance than the corrugated tube. This is attributable to the fact that the corrugated tube has a lower heat-transfer coefficient than the smooth tube because of a metal net reinforcement covering the tube. And by inserting the three-direction showering tube, the heat flux in the evaporator increased up to at around 9000 W/m^2 and dry out was not observed. Based on the test results, an empirical correlation of heat-transfer coefficients was developed, and then integrated into heat pump simulations. It was found that the use of a three-direction showering tube successfully reduced the number of thermosiphons required.

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Key Words: Underground heat; Thermosiphons; Heat pumps

1. Background and purpose

In Hokkaido, where temperature is very low in winter, the energy used for space heating accounts for about 42% of annual energy consumption per household. In addition, that used for space heating and hot water supply combined exceeds 80% as showed in figure 1[1]. This indicates that huge amounts of energy are consumed for space heating and hot water supply. They use kerosene as a heat source in the region since the price is cheap, and this accounts for 58% in the total energy consumption. In the recent years, owing to increasing awareness of environmental issues and soaring oil prices, heat pump type of hot-water heating begins to spread, but the utilization of heat pumps in the region remains as low as 5% as showed in table 1[2]. This is because low outdoor temperature in cold climate cause various problems such as frost formation in the outdoor units, reduction in heat pump efficiency and so on. For these reasons, the development of high-efficiency heat pumps used in cold climate

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for space heating and hot water supply is considered an effective measure at households for energy saving and CO_2 emission reductions.

In this study, thermosiphons with a new shape have been designed and evaluated to collect heat from surface ground layers. The new thermosiphons utilize underground heat as a heat source for evaporators in order to prevent a reduction in heat pump efficiency. In addition, heat pump simulations have been conducted by incorporating the test results of the thermosiphons and COP2.8 was set as the performance target.

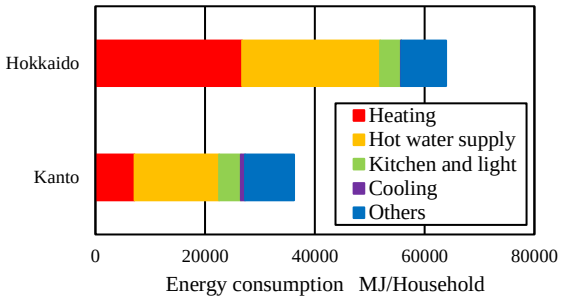


Fig. 1. Regional comparison of energy consumption per household

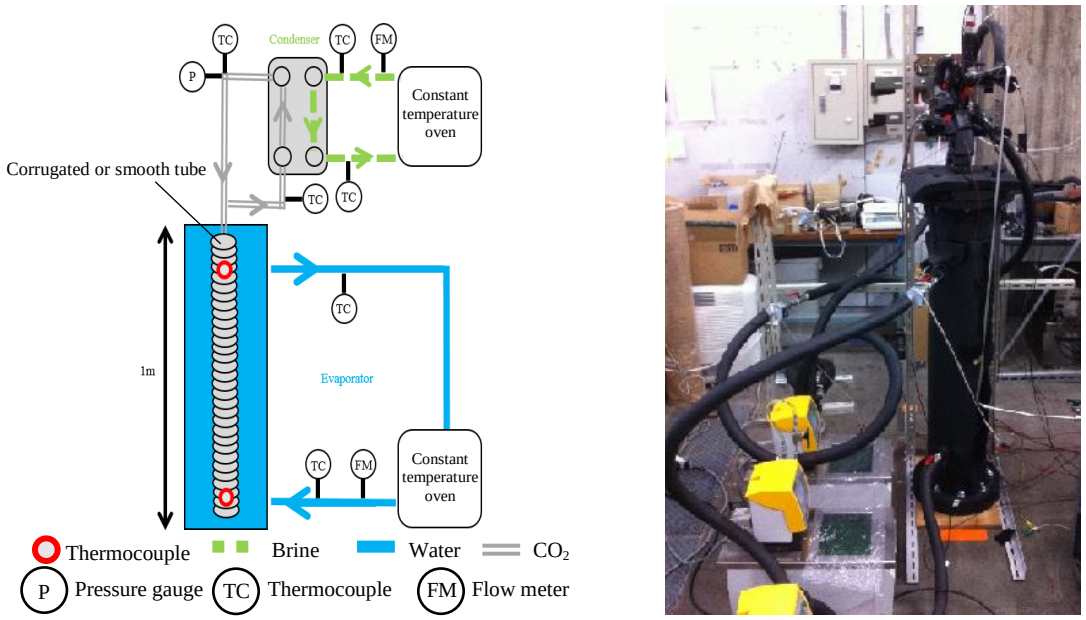
Table 1. Percentage of heat pump installations at new all-electric homes in Hokkaido

Hot water supply		Space heating		Actual sales in FY2010 [households]	Percentage [%]
Heat source	APF	Heat source	SPF		
EcoCute	2.3	Heat pump	2.3	800	5
		Heater on snow melting power	1	1532	10
		Heater on midnight power	1	553	4
Heater	1	Heat pump	2.3	693	4
		Heater on snow melting power	1	8766	57
		Heater on midnight power	1	3172	20
Total				15516	100

2. Experiments

2.1 Experimental test facility

The performance of two types of thermosiphons, a 1-m-long corrugated tube and a 1-m-long smooth tube, have been evaluated. Table 2 and 3 shows the dimensions of each tube. The corrugated tube is designed to increase its heat-transfer area and give it the effect of preventing flooding by having the wall surface trapping the working fluid. The corrugated tube is covered with a stainless steel net reinforcement on the external wall to ensure its pressure resistance. The smooth tube has a smaller heat-transfer area than the corrugated tube, but has a higher heat-transfer coefficient because there is no need to cover the tube with a metal net reinforcement. A plate heat exchanger is set up above a thermosiphon. The whole thermosiphon area works as an evaporator, and the plate heat exchanger works as a condenser. Figure 2 is the outline of experimental apparatus and its photograph. In this test, an artery tube is inserted inside each thermosiphon. The artery tube is inserted inside the thermosiphon to facilitate the recirculation of the working fluid from the condenser. Leading the condensate of working fluid into the artery tube can separate the gaseous flow path from the liquid flow path, which is expected to prevent flooding from incidence. Four types of artery tubes were used in this study, three of which were showering tubes. Each showering tube is blocked on the bottom, with three holes opened on the side wall. The recirculated working fluid is distributed through these holes like showering, which was expected to prevent localized dry out. Three types of artery tubes were prepared with different hole positions: 1) the three holes face in one direction, 2) face alternately in two directions at the intervals of 180 deg., and 3) face in three directions at the intervals of 120 deg. In addition, another artery tube was prepared, whose bottom was not blocked, and which had no holes on the side wall. This artery tube was named “double tube”. Figure 3 shows the shapes of these artery tubes and table 4 shows the dimensions of them. Table 5 shows the experimental conditions.



(a) Outline of experimental apparatus (b) Photograph of experimental apparatus

Fig. 2. Experimental apparatus

Table 2. Dimensions of corrugated tubes

Corrugated tube	Length	mm	900	Wall thickness	mm	0.25
	ID (min. ID)	mm	10	Inner heat-transfer area	m^2	0.0882
	OD	mm	15.0	Volume	m^3	10.5×10^{-5}
	Groove pitch	mm	2.4	Material		SUS316L
	Groove depth	mm	2.25			
Straight portion (edges)	Length	mm	30,70	Wall thickness	mm	1.0
	OD	mm	12.7	Material		SUS304
Total	Length	mm	1000	Volume	m^3	11.4×10^{-5}
	Inner heat-transfer area	m^2	0.0916			

Table 3. Dimensions of smooth tubes

Heat exchanger	Length	mm	930	Wall thickness	mm	1.2
	ID	mm	14.9	Heat-transfer area	m^2	0.0509
	OD	mm	17.3	Volume	m^3	10.5×10^{-5}
	Material		SUS304			
Heat exchanger joint area (edges)	Length	mm	70	Wall thickness	mm	1.2
	OD	mm	12.7	Material		SUS304
	ID	mm	10.3			
Total	Length	mm	1000	Volume	m^3	16.7×10^{-5}
	Heat-transfer area	m^2	0.0537			

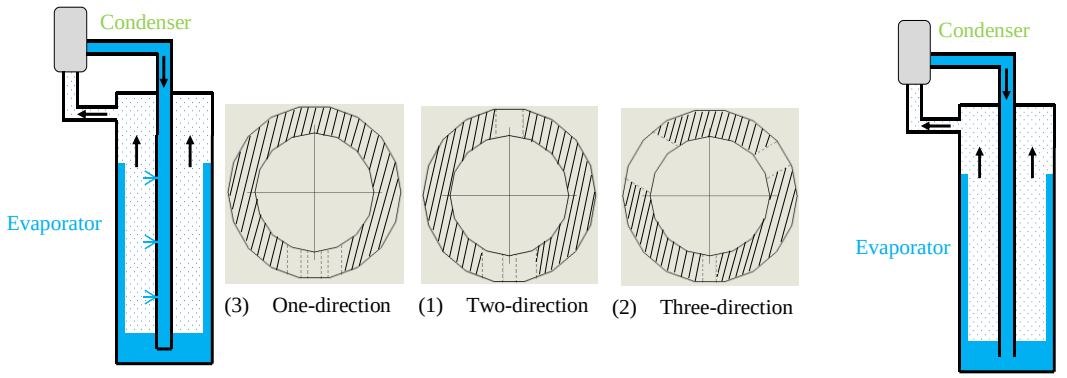
Table 4. Dimensions of artery tubes

Length	mm	1170
ID	mm	4.4
OD	mm	6.4
Hole height, hole diameter	mm	250, 0.5
		570, 1.0
		890, 2.0

Table 5. Experimental conditions

Quantity of CO_2 charged	V	g	43~153
Percentage of CO_2 charged	V+	%	19.1~67.4
Setting temperature of evaporator	T_e	℃	4～22
Setting temperature of condenser	T_c	℃	0

Flow rate of water around evaporator	m_e	L/min	4.0~5.1
Flow rate of brine around condenser	m_c	L/min	2.0~2.4



(a) Showering tube (b) Double tube

Fig. 3. The shapes of artery tubes

2.2 Data reduction

We use formulas written here to evaluate the performance of the thermosiphon.

$$Q_e = m_e c_e \Delta T_e \quad (1)$$

$$Q_c = m_c c_c \Delta T_c \quad (2)$$

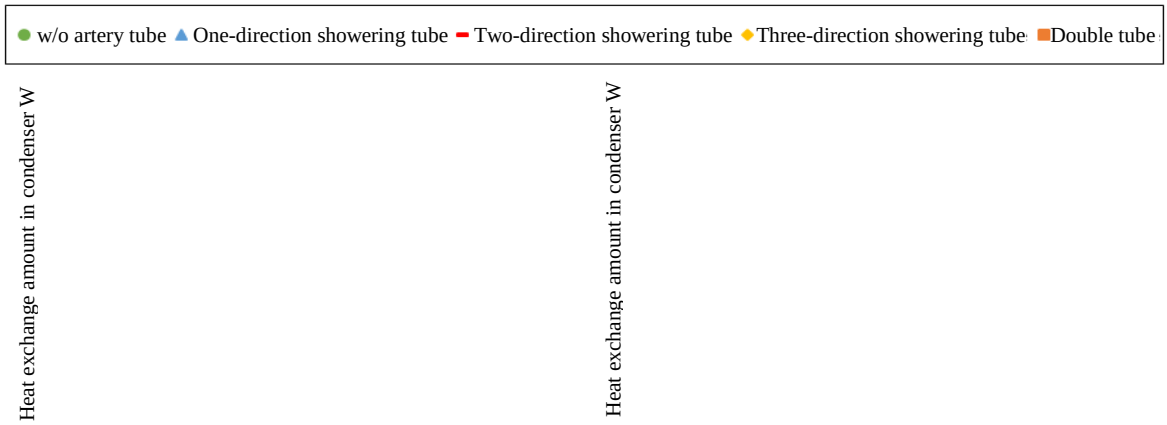
$$q_e = \frac{Q_e}{A_e} \quad (3)$$

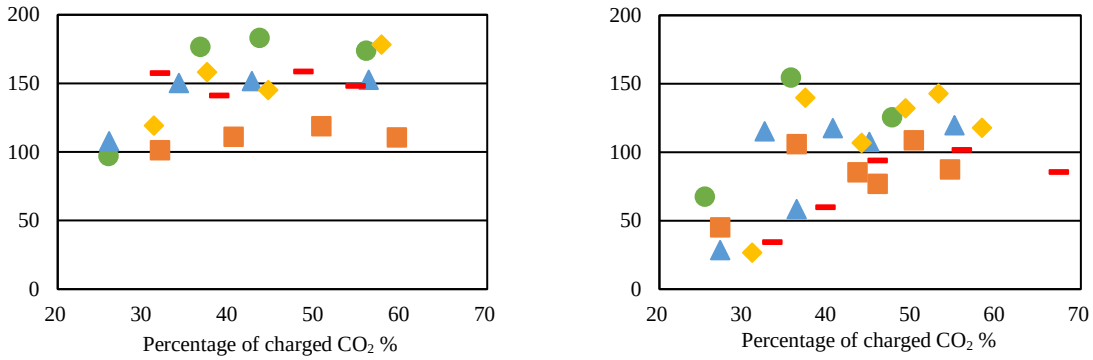
$$h_e = \frac{Q_e}{A_e\left(T_{surf.} - T_{sat.}\right)} \quad (4)$$

Q_e : Heat exchange amount in evaporator	[W]	Q_c : Heat exchange amount in condenser	[W]
$\dot{m}_e$: Mass flow rate of water	[kg/s]	$\dot{m}_c$: Mass flow rate of brine	[kg/s]
c_e : Specific heat of water	[J/kgK]	c_c : Specific heat of brine	[J/kgK]
ΔT_e : Temperature difference of evaporator	[K]	ΔT_c : Temperature difference of condenser	[K]
q_e : Heat flux in evaporator	[W/m^2]	A_e : Heat transfer area of evaporator	[m^2]
h_e : Heat transfer coefficient of evaporator	[W/m^{2}K]	$T_{surf.}$: Heating surface temperature	[K]
$T_{sat.}$: Saturation temperature	[K]		

3. Results and discussion

As a result of performance evaluation showed in figure 4, the smooth tube showed higher performance than the corrugated tube. This is attributable to the fact that the corrugated tube has a lower heat-transfer coefficient than the smooth tube as showed in figure 5. For the purpose of making it flexible, the corrugated tube used in this test is designed to have a very thin wall thickness. The corrugated tube is covered with a metal net reinforcement on the external surface to be able to withstand its internal pressure. It is thought that this construction made the tube's heat-transfer resistance higher than intended.





(a) Smooth tube (b) Corrugated tube

Fig. 4. Quantity of heat exchanged in condenser

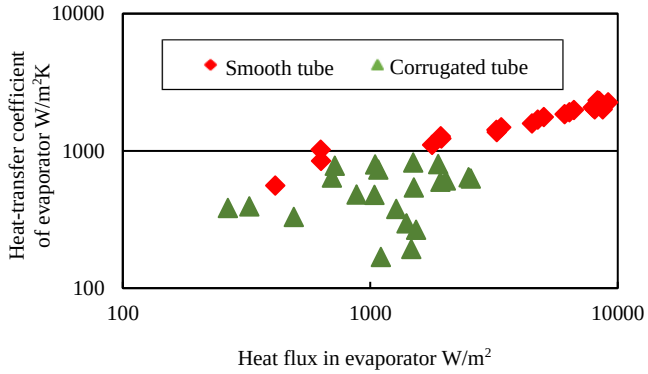
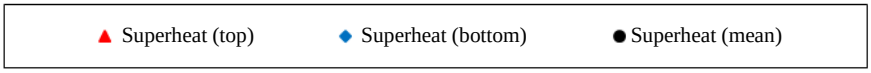
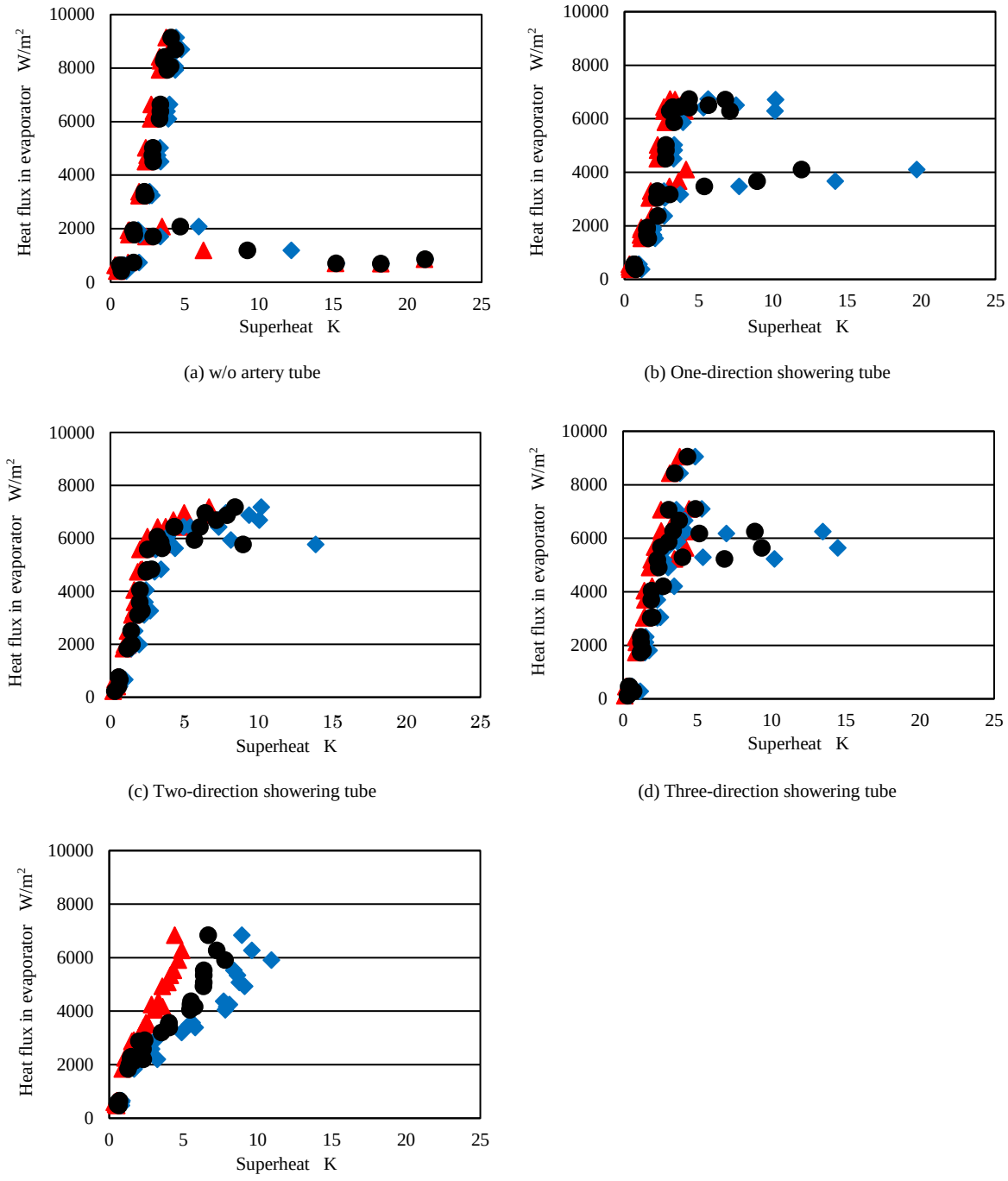


Fig. 5. Heat transfer coefficient of evaporator

To discuss the effect of the artery tubes, figure 6 shows the relationships in smooth tube between the evaporator's heat flux and the degree of superheat on the evaporator wall surface. In the three cases where a one-direction showering tube, a two-direction showering tube, and a double tube were used, the heat flux in the evaporator peaked out at around 7000 W/m^2, which indicates that dry out happened. On the other hand, in the cases where no artery tube and a three-direction showering tube were used, the heat flux in the evaporator increased up to at around 9000 W/m^2. Although dry out started at around 1000 W/m^2 in the case where no artery tube was used, such dry out was not observed in the case where a three-direction showering tube was used. Based on these results, it was found that the use of a three-direction showering tube was most effective when smooth tubes were used.





(a) w/o artery tube

(b) One-direction showering tube

(c) Two-direction showering tube

(d) Three-direction showering tube

(e) Double tube

Fig. 6. Evaporator's heat flux vs. degree of superheat on the evaporator wall surface (smooth tube)

4. Heat pump simulation

Our objective system is CO_2 binary refrigeration cycle. In this cycle, refrigerants exchange heat in a cascade heat exchanger to keep the compression ratio low of the each stage and it makes it possible to obtain high performance with lowering the work load of the compressor. Our simulations have been conducted as shown in Figure 7[3]. By using this, the number of thermosiphons required to achieve the targeted COP2.8 have been calculated. Based on the test results showed above, an empirical correlation of heat-transfer coefficients was developed, and then integrated into heat pump simulations. In this study, the experimental equation was developed based on the experimental data obtained from the smooth tube without an artery tube, and the smooth tube with a three-direction showering tube, both of which showed the highest test performance. The following is the empirical correlation for heat-transfer coefficients of two-phase closed thermosiphons developed by Imura, et al.[4]

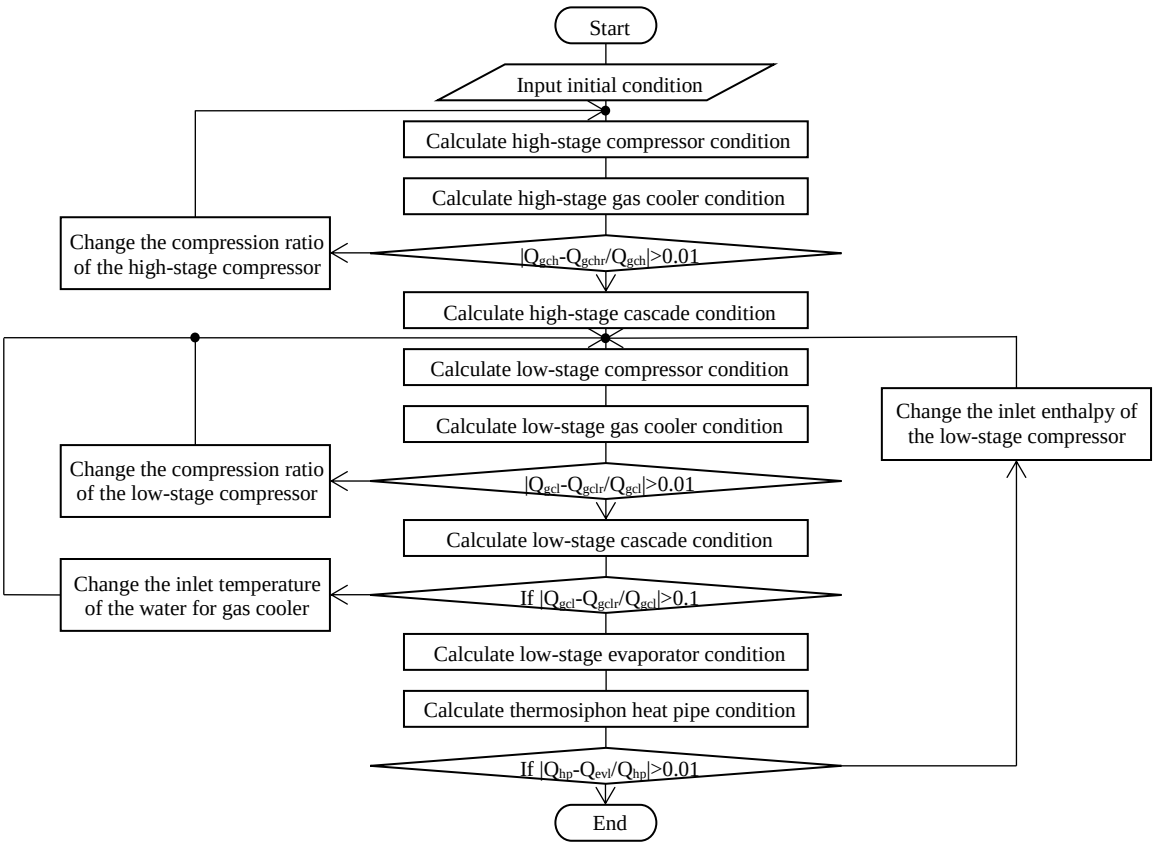


Fig. 7. Flow chart of the heat pump simulations

$$h_e = 0.32\frac{\rho_l^{0.65}\lambda_l^{0.3}C_{p_l}^{0.7}g^{0.2}q_h^{0.4}}{\rho_g^{0.25}L^{0.4}\mu_l^{0.1}}\left(\frac{P_{in}}{P_a}\right)^{0.3} \quad (5)$$

h_e : Heat transfer coefficient	[W/m^{2}K]	ρ_l : Liquid density	[kg/m^3]
ρ_g : Vapor density	[kg/m^3]	C_{p_l} : Liquid specific heat	[J/kgK]
g : Gravitational acceleration	[m/s^2]	L : Vapor latent heat	[J/kg]
μ_l : Liquid viscosity coefficient	[kg/ms]	λ_l : Liquid thermal conductivity	[W/mK]
q_h : Heat flux in evaporator	[W/m^2]	P_{in} : Internal pressure	[Pa]
P_a : Atmospheric pressure	[Pa]		

Where

$$X = \frac{\rho_l^{0.65}\lambda_l^{0.3}C_{p_l}^{0.7}g^{0.2}q_h^{0.4}}{\rho_g^{0.25}L^{0.4}\mu_l^{0.1}} \quad (6)$$

Using the above equations, empirical correlations of heat-transfer coefficients have been developed for the smooth tube without an artery tube, and the smooth tube with a three-direction showering tube as showed in figure 8.

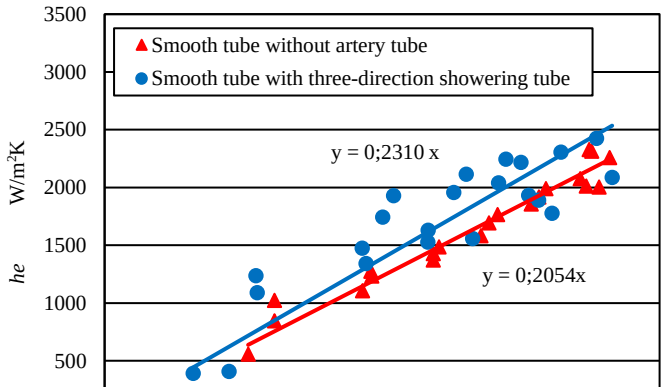


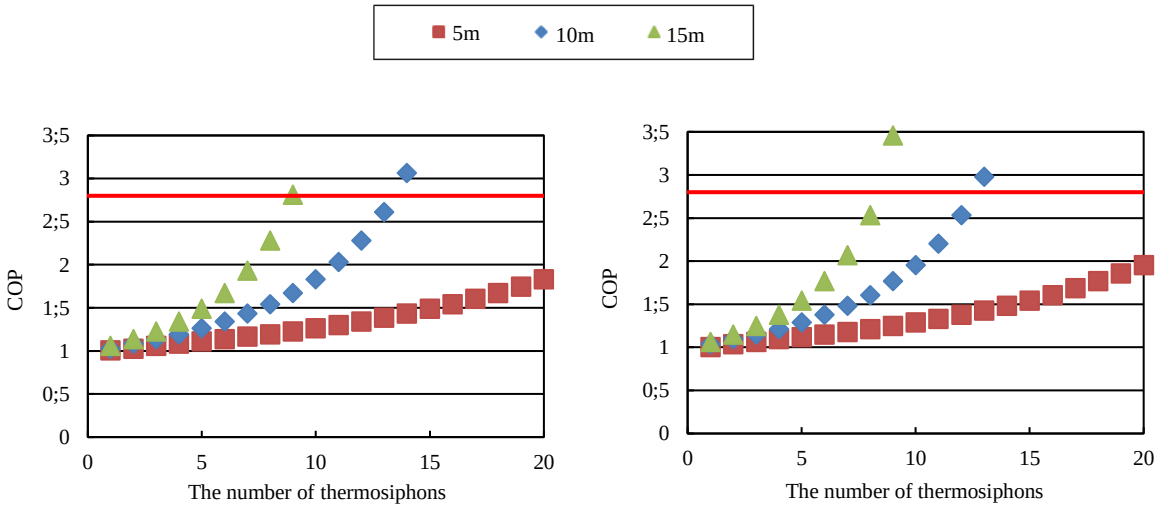
Fig. 8. Experimental equations for heat transfer coefficients

The empirical correlations were developed by altering the slope of the equation developed by Imura, et al. The followings are the empirical correlations for the cases where no artery tube and a three-direction showering tube are used.

$$h_e = 0.2054X\left(\frac{P_{in}}{P_a}\right)^{0.3} \quad (7)$$

$$h_e = 0.2310X\left(\frac{P_{in}}{P_a}\right)^{0.3} \quad (8)$$

Figure 9 shows the results of heat pump simulations for the smooth tube without an artery tube, and the smooth tube with a three-direction showering tube. In the case where no artery tube is used, the number of thermosiphons required to achieve the targeted COP2.8 is 14 for the tube length of 10 m, and 9 for 15 m. Similarly, in the case where a three-direction showering tube is used, the number of thermosiphons required is 13 for the tube length of 10m, and 9 for 15 m. Based on these results, in the case where 10-m-long thermosiphons are used, the use of a three-direction showering tube successfully reduced the number of thermosiphons required (in this connection, note that COP was calculated as APF here).



(a) Smooth tube without artery tube (b) Smooth tube with three-direction showering tube

Fig. 9. The results of the heat pump simulations

5. Conclusions

- When comparing the performance between the smooth tube and the corrugated tube, higher performance was obtained when the smooth tube was used because heat transfer coefficient's lowering have the larger effect on performance than the increase of the heat transfer area. Supposing soil temperature of 10 ℃, the maximum heat exchange amount in condenser was 180 W.
- It was found that the dry out happened from the bottom of both smooth and corrugated thermosiphons by discussing from the superheat on the evaporator.
- Based on the test results, an empirical correlation of heat transfer coefficients was developed.
- By using the developed correlations, COP was calculated. The number of thermosiphons required to achieve the objective COP2.8 is 14 for the tube length of 10m, and 9 for 15m in the case where the smooth tube without artery tube is used. In the case where the smooth tube with three-direction showering tube is used, the number of thermosiphons required is 13 for 10m and 9 for 15m.

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