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Performance Analysis of Sensible and Latent Energy Recovery System for Thermally Controlled Facility

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Key words: Sensible and latent energy recovery system, Energy performance simulation, Thermally controlled facility, TRNSYS(Transient System Simulation) program

Abstract

Simulation was conducted using TRNSYS to evaluate the thermal performance of a facility. This facility has a condensing-type heat exchanger which is able to recover the latent energy for the purpose of reducing the heating energy in winter. The boiler and chiller are selected based on the annual peak loads and controlled to maintain the facility at the set temperature of 14~17℃. Supplied energy by the boiler and recovered energy by the heat exchanger were calculated as a function of number of pass through heat exchanger, kind of fuel and hot water velocity. Simulation results show that about 20% of the total heating load can be recovered by the heat exchanger and the amount of latent heat is increasing with the number of pass. This means that the efficiency of the waste energy recovery system can be increased by using a condensing-type heat exchanger rather than a traditional sensible heat exchanger.

Nomenclature

A_p : heat transfer area of boiler [m^2]
C_p : specific heat [kJ/kg℃]
CAP : heat capacity [kJ/℃]
h : convective heat transfer coefficient [W/m^2℃]
k : thermal conductivity [W/m℃]
LHV : lower heating value [kJ/kg]
$\dot{m}$: mass flow rate [kg/hr]
$\dot{Q}$: heating or cooling load [kJ/hr]
T : temperature [℃]
U : overall heat transfer coefficient [W/m^2℃]
w : mass flow rate [kg/hr]

Greek symbols

ε : emissivity
σ : Stefan-Boltzmann constant
ω : ratio of humidity

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Subscripts

a : air
b : building
e : exhaust gas
f : fuel
g : gas
i : inlet
o : outlet
TCF : thermally controlled facility
w : water

1. Introduction

Since the oxides of sulphur and nitrogen are contained in the industrial exhaust gases of fossil fuel burning industry, cold corrosion could be caused by condensation of water vapor when heat is recovered with a heat exchanger. Thus, in most cases, sensible heat is only recovered while maintaining the exhaust temperature above 130℃ to prevent condensation of moisture. Recently, due to the development of anti-corrosive materials such as titanium and coating techniques, condensing-type heat exchangers have been developed which can withstand the exhaust gas temperature as low as 50℃ such that the efficiency of energy utilization has substantially increased. Additionally, this type of heat exchanger could contribute to reduced environmental pollution by removal of particles in the exhaust gas.

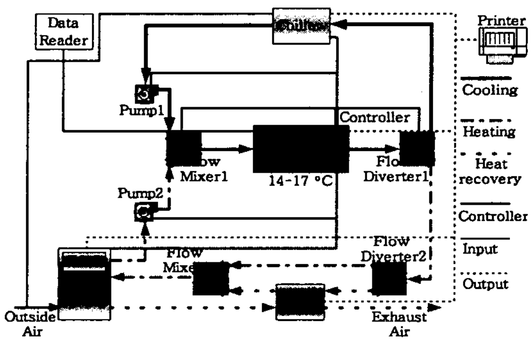

Fig. 1 Schematic of sensible and latent heat recovery system.

In this study, dynamic thermal characteristics of thermally controlled facility (TCF) is simulated with TRNSYS to investigate the efficiency of a waste energy recovery system by using the weather data of Incheon.

2. Energy recovery system

Fig. 1 shows the schematic of a sensible and latent energy recovery system using a condensing-type heat exchanger. The system consists of thermally controlled facility providing the heating and cooling load and several devices such as boiler, cooler, waste heat recovery heat exchanger (HX), distributor and pump. There are also controllers for the boiler, cooler and pump which maintain the system at set conditions.

2.1 Modeling of heating and cooling load

2.1.1 Structure of heating and cooling space

The thermally controlled facility being modeled is an aquafarm which is contained in the

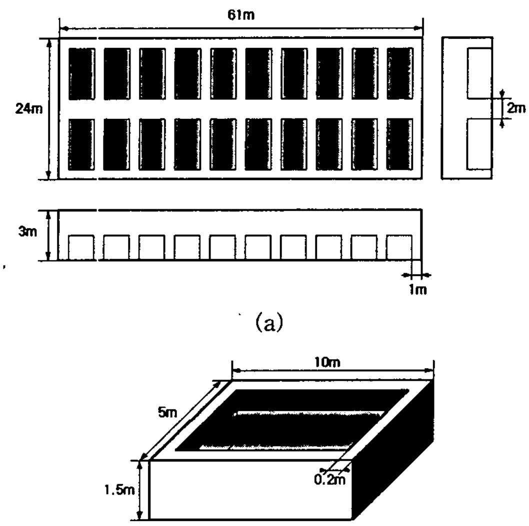

Fig. 2 Dimensions of the TCF model.

outer structure with the dimension of 24×61×3 m as shown in Fig. 2 (a). The aquafarm consists of 2 rows of 10 water tanks each with dimensions shown in Fig. 2 (b). Type 12 of TRNSYS Ver. 14.2 was used to calculate the heating and cooling load of the outer structure containing aquafarm.(1) Type 12 is an analytical module for single zone buildings and can calculate the energy flow including the heating and cooling load.

2.1.2 Overall heat transfer coefficient

In order to calculate the load of the facility, 2 modules of Type 12 were used in series and the heat loss of the aquafarm is considered as the heat gain of the outer building. The temperature of the aquafarm is set at 14℃ in summer and 17℃ in winter and kept constant by controlling the flow rate of hot water or cold water through heat exchanger. Fig. 3 shows the wall structure of the building with a total thickness of 70 mm which consists of 66 mm insulation with 2 mm thick steel plates on both sides. The inside and outside heat transfer coefficient is assumed to be a constant value of 16.95 W/m^2℃. The overall heat transfer coefficient of the wall can be calculated by

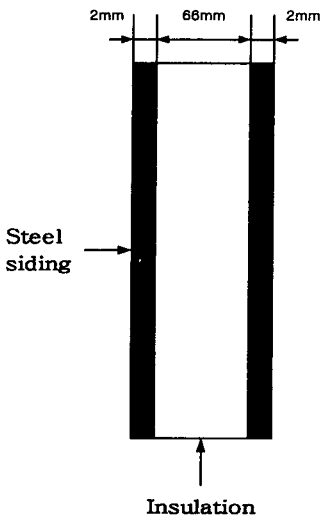

Fig. 3 Structure and dimension of the wall.

Table 1 Thermal capacitance and overall heat transfer coefficient of TCF

		Thermal capacitance (CAP) kJ/℃		Overall heat transfer coefficient (UA) W/℃
Outer building	Panel	61,764	66,965	57,358
	Air	5,200		
Water tank	Wall	167,600	4,067,960	110,800
	Water	35,798		

the Eq. (1) and summarized in Table 1.

$$U=\frac{1}{\frac{1}{h_i}+\frac{\Delta x}{k}+\frac{1}{h_o}}=\frac{1}{R_i+R_s+R_o} \quad (1)$$

The wall of water tank is made of concrete (k=1.731 W/m℃) with a thickness of 200 mm. The overall heat transfer coefficient through the wall, water surface and bottom of the tank can be calculated in a similar way and the value is shown in Table 1.

2.1.3 Thermal capacitance

The dynamic thermal characteristics of the facility are varying according to Eq. (2).

$$CAP\frac{dT_{TCF}}{dt}=UA(T_{amb}-T_w)-\dot{Q} \quad (2a)$$

$$\dot{Q}=\dot{m}C_p(T_{h,i}-T_{h,o})_{heating} \quad (2b)$$

where CAP is the thermal capacitance which is determined as follows. First of all the thermal capacitance of the building can be considered as the sum of walls and inside air capacitance. Similarly, thermal capacitance of each water tank can be calculated by summing that of the concrete walls and water. Table 1 shows the magnitudes of the overall heat transfer coefficient and thermal capacitance of the outer building and water tank. The thermal capacitance of the water tank is the sum of the value for 20 water tanks.

2.1.4 Load by make-up water

Since the water in the aquafarm is polluted by fish excrement, a certain portion of the water is replaced by fresh water. In this study it is assumed that the water in each tank is completely substituted once a day. The temperature of the make-up water in winter is assumed to be 5℃. The make-up water is assumed to mix completely in the tank and the heating load by making up is determined by the temperature difference with the set value.

2.2 Modelling of waste heat recovery HX

2.2.1 Structure of waste heat recovery HX

The heat exchanger used in this system is able to recover the latent heat as well as sensible heat and was modified from Type 52 in TRNSYS so as to treat the number of passes as one of the parameters. Fig. 4 shows the front view of the heat exchanger and side view of 2 passes. As shown in the figure, one pass consists of 5 tubes staggered in 4 rows. The exhaust gas exchanges heat with water tubes in cross flow. The water exiting one pass goes into the next pass. The parameters of the HX are shown in Table 2.

The temperature of exhaust gases decrease as they pass across the tubes until the dew point is reached, when water vapor condenses on the tube surface.[2] By determining the position of condensation, the surface area ratio of dry tube to wet tube can be calculated and the heat transfer can be calculated by the following equations.[3]

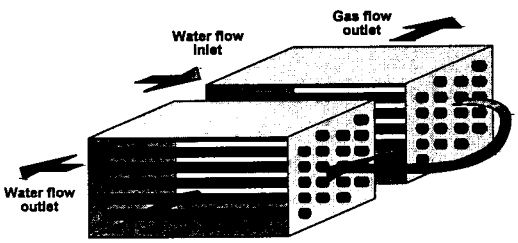

Fig. 4 Arrangement of condensing-type heat exchanger and flow direction of fluid.

Table 2 The parameters of heat exchanger component configuration

Parameter	Value
Number of pass	1~20
Number of tubes	5
Duct height	0.16 m
Duct width	0.32 m
Outside tube diameter	0.019 m
Inside tube diameter	0.018 m
Tube spacing	0.05 m
Thermal conductivity (Titanium)[4]	79 kJ/(hr · m · ℃)

$$\dot{Q} = \dot{m}_w C_{pw}(T_{w,o} - T_{w,i}) \quad (3a)$$

$$\dot{Q}_{lat} = \dot{m}_a(\omega_{a,i} - \omega_{a,o})h_{fg} \quad (3b)$$

$$\dot{Q}_{sens} = \dot{Q} - \dot{Q}_{lat} \quad (3c)$$

where h_{fg} is the latent heat of the gas and $\dot{m}_a$ and $\dot{m}_w$ are mass flow rate of exhaust gas and hot water respectively.

2.2.2 Mass flow rate of exhaust gas and water

The fluid speed in tubes of the heat exchanger is in the range of u_w=0.2~2 m/s and the typical velocity range of exhaust gas is 1~10 m/s.[5] Boiler heat capacity is determined from maximum heating load and the amount of supply air is determined by the maximum supply heat using chemical equilibrium equation. The amount of supply air and exhaust gas is calculated from the excess air ratio. The mass flow rate of hot water is calculated by trial and error such as the exhaust gas temperature at the outlet of heat exchanger could be within 50~60℃. Thus, the average velocity of exhaust gas and hot water is determined from the each mass flow rate by using the average density and section area.

The velocity of hot water is set to 1.3 m/s and that of exhaust gas is set to 5.8 m/s.

When the hot water velocity is considered as a variable, calculation was conducted in the range of u_w=0.2~2 m/s.

2.3 Model of boiler

The chemical equilibrium equation is required to calculate the quantity of water vapor in the exhaust gas which is needed to analyse the thermal efficiency of the heat exchanger. The TRNSYS code does not have a module for the above analysis, therefore a new boiler module was developed to calculate the quantity of water vapor and exhaust gas temperature.(6) The parameters and the values for the module are shown in Table 3.

The heat transfer from the exhaust gas to the boiler water is expressed in Eq. (4).(6)

$$\begin{aligned} Q &= A_p \varepsilon_f \sigma (T_g^4 - T_w^4) \\ &= (w_f LHV - w_g C_p T_e) \end{aligned} \quad (4)$$

Generally $\Delta T_{ge} = T_g - T_e \cong 150$℃, combustion gas temperature (T_g) can be expressed in terms of exhaust gas temperature (T_e) and substituted in Eq. (4) to find the exhaust gas temperature by iteration.

The boiler is controlled by on-off mode depending on the setting temperature. Boiler capacity is determined by the heat transfer from the water tank to the outer building and that from the simple structure to the external environment.(7) In this study liquified natural gas and diesel oil were used as heating fuel and the chemical formula was assumed as CH_4, and $C_{12}H_{26}$, respectively.(8)

Table 3 The parameters of combustion process

Parameter	Value
Number of carbon	1
Number of hydrogen	4
Maximum boiler heat rate	830,000 kJ/hr
Boiler area	15 m^2
Set point water temp	90℃
Air specific heat	1.09 kJ/kg℃
Excess air	100%
Efficiency of boiler	0.9
Water mass flow rate	6,596 kg/hr
Air mass flow rate	568 kg/hr

2.4 Control and maintenance of system

The system control is required to maintain the temperature of the facility and to make the thermal system efficient. In order to keep the temperature of the water tank constant the capacity of the boiler and cooler should meet the maximum heating and cooling load, respectively. It is assumed the quasi-steady state so that the heat loss of the water tank to the inside air is equal to the heat loss of inside air to outdoor air. Thus, Eq. (5) can be written to find the capacity of the boiler or cooler.

$$Q = \frac{(UA)_s \cdot (UA)_{TCF}}{(UA)_s + (UA)_{TCF}} (T_w - T_{amb}) \quad (5)$$

where $(UA)_s$ and $(UA)_{TCF}$ are overall heat transfer coefficient of the outer structure and the facility respectively. The capacity of boiler and cooler estimated by Eq. (5) is as follows.

▷ capacity of boiler : 830,000 kJ/hr
▷ capacity of cooler : 490,000 kJ/hr

The heating or cooling energy is supplied by on-off control function. The operating condition

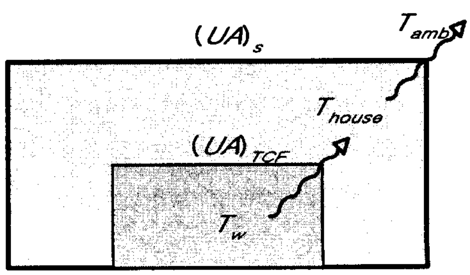

Fig. 5 Heat flow from heating space.

of the boiler, cooler and pump is determined by the temperature of the water tank. In this study, the aquafarm was chosen as a constant temperature facility and the setting temperature is as follows.

▷ operating condition of boiler : $T_w < 14℃$

▷ operating condition of cooler : $T_w > 17℃$

▷ stop condition of boiler and cooler :
$14℃ \le T_w \le 17℃$

3. Results and discussions

3.1 Annual temperature variations

Fig. 6 shows the annual environmental temperature of the Incheon area for 1998 as well as the temperature of inside air and the controlled water tank. As shown in the figure, the water tank temperature is fairly well kept constant at the temperature of 14℃ in winter and 17℃ in summer. If maximum heating capacity is required, water tank temperature may not meet the setting temperature. Fig. 7 shows the degree of deviation from the setting temperature depending on the number of pass through the heat exchanger. It was shown that a heat exchanger with 8 or more passes will enable control within 0.2℃.

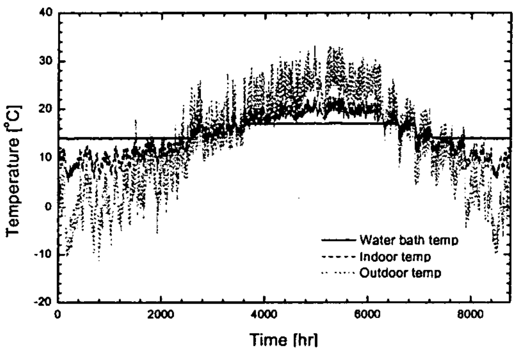

Fig. 6 Outdoor temperature variations of water tank, indoor and outdoor.

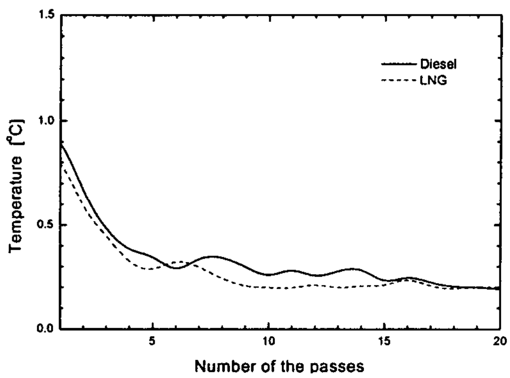

Fig. 7 Deviations from the setting temperature of water tank at the peak heating load.

3.2 Monthly heating and cooling load and recovered energy

Fig. 8 shows the monthly heating and cooling load profile and recovered energy by the heat exchanger with 8 passes using diesel oil. Heating season starts in October and ends in April. Heating and cooling load was highest in January and August, respectively and especially both heating and cooling are required in October. The heating and cooling load are defined as the required thermal energy to maintain the water temperature in the set temperature range. The amount of sensible and latent energy recovered by the heat exchanger is almost 20% of the total heating load. Also, re-

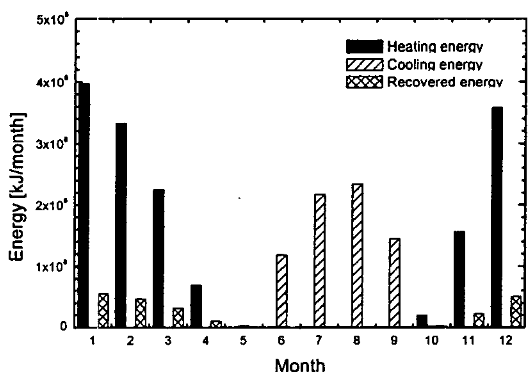

Fig. 8 Monthly energy supply for the facility and recovered energy by heat exchanger.

covered energy is increased in proportion to the heating energy.

3.3 Recovered energy and exhaust gas temperature

Fig. 9 shows the latent, sensible and total recovered energy depending on the number of passes when diesel oil is used. Although the total recovered energy is increased as the number of passes increases, an optimal number of passes should be chosen to balance the rate of energy recovery with the installation cost.

When the ratio of latent heat recovery is compared with that of sensible heat in Fig. 10, it is found that the recovery rate of latent heat is increasing with the number of passes. The condensing-type heat exchanger has about 20% to 40% higher heat recovery than traditional heat exchanger. The temperature of exit gas and hot water passing through the heat exchanger is shown in Fig. 11. The exit temperature of exhaust gas initially decreases with increasing number of pass. On the other hand the increase in hot water temperature at the exit is very small compared to the change in exit gas temperature. Since the mass flow rate and specific heat of hot water are big, hot water temperature does not change remarkably.

3.4 Supplied and recovered energy with fuel

It is shown in Fig. 12 that supplied energy by the boiler is decreasing as the pass number

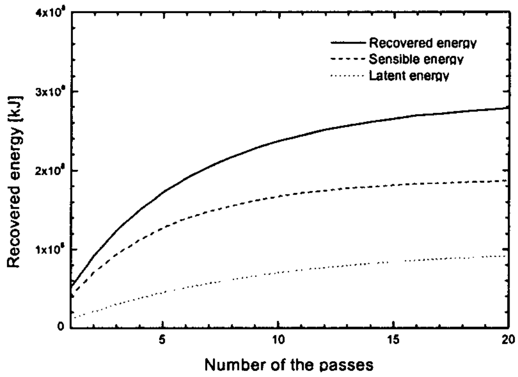

Fig. 9 Recovered total, sensible and latent heat as a function of number of pass.

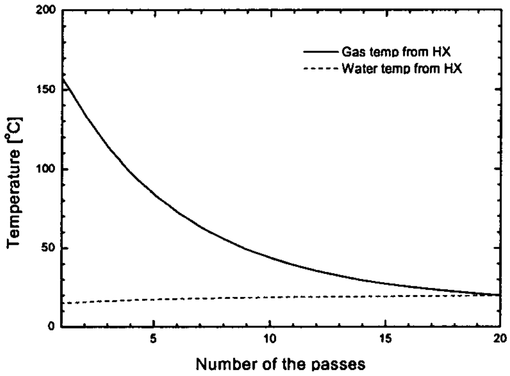

Fig. 11 Temperature of gas and water at HX outlet as a function of number of pass.

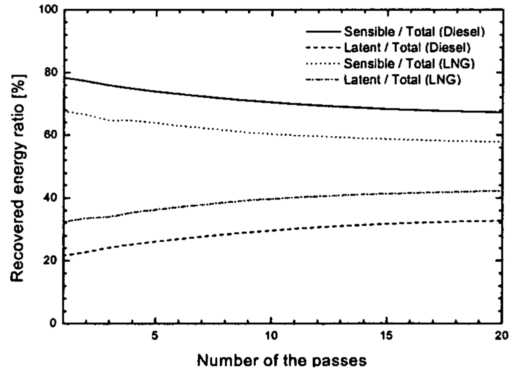

Fig. 10 The ratio of sensible/latent to total heat as a function of number of pass.

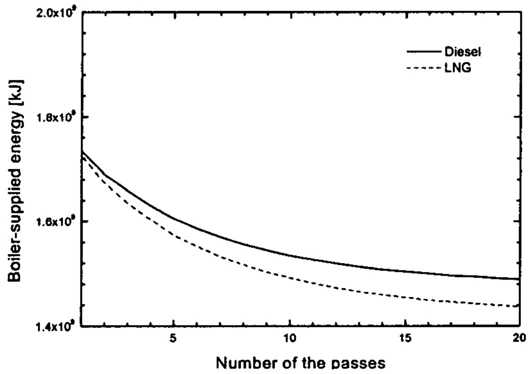

Fig. 12 Boiler-supplied energy as a function of number of pass.

increases. Quantity of recovered energy when LNG is used is larger than that using diesel oil as a fuel. Fig. 13 shows the comparison of recovered sensible and latent energy where sensible heat is similar for both fuels but latent energy using LNG is greater than that using diesel. The reason is that combustion of LNG generates more water vapor than diesel oil with the condition of same calorific value. Fig. 14 shows the ratio of recovered energy to supplied energy as a function of the number of passes through the heat exchanger and indicates that using LNG is more advantageous than diesel in terms of recovered fraction of energy. This means that the efficiency of the waste energy recovery system can be greatly increased by using a condensing-type heat exchanger rather than a traditional sensible heat exchanger.

3.5 Supplied and recovered energy as a function of the flow velocity of hot water

Fig. 15 shows the heat energy supplied by the boiler depending on the hot water velocity with a HX containing 10 passes. The supplied energy rapidly decreases up to a flow velocity of 0.2 m/s after which the supplied energy shows a gradual decrease. This appears to be due to the enhancement of heat transfer by transition of flow regime from laminar to turbulent. With the pipe diameter of 0.018 m, the

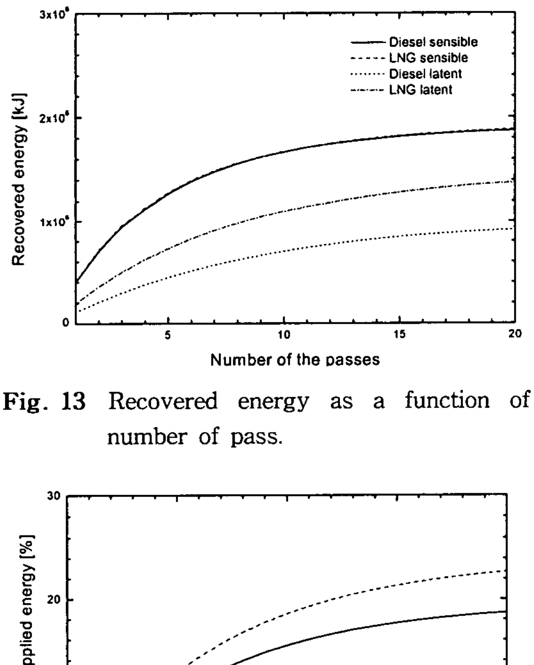

Fig. 13 Recovered energy as a function of number of pass.

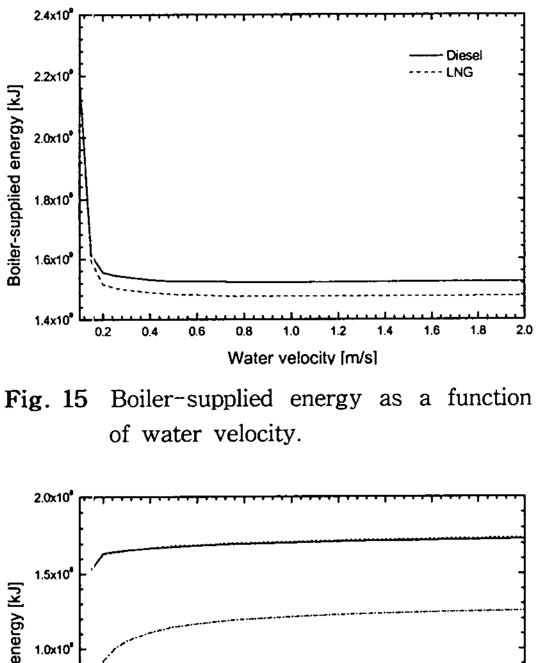

Fig. 15 Boiler-supplied energy as a function of water velocity.

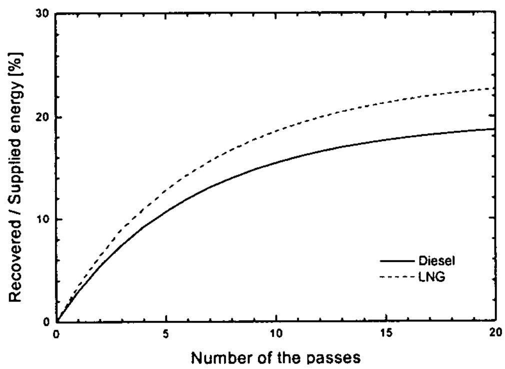

Fig. 14 The ratio of recovered energy to supplied energy as a function of number of pass.

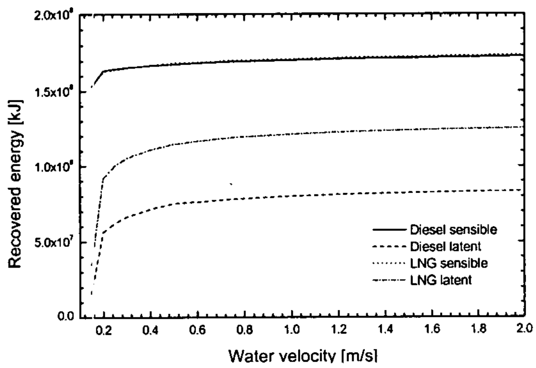

Fig. 16 Recovered energy as a function of water velocity.

flow velocity for transition corresponds to 0.128 m/s if the critical Reynolds number for transition is 2300 and that for the fully developed region corresponds to 0.222 m/s for Re=4000. Increase of flow velocity above 8 m/s does not contribute to the decrease of supplied energy. Supplied energy is more needed when using diesel rather than LNG, but there is not much difference within the low velocity region.

Fig. 16 shows the sensible and latent heat recovery for hot water velocity which increases with the flow velocity. But, the quantity of recovered energy is almost constant above a certain velocity because the heat exchanger cannot take heat any more exceeding the maximum. In the low velocity region, the increased rate of latent energy recovery is greater than that of sensible energy and total energy recovery depends primarily on the latent energy recovery. Once the system is defined, there is an optimal range of flow velocity. It is highly recommended to choose the flow velocity in order to avoid unnecessary power consumption by the pump. In this study, the flow velocity of 1.3 m/s is within acceptable range of flow velocity.

4. Conclusions

A numerical study was conducted to analyse a specific thermally controlled facility by using TRNSYS version 14.2. This facility has a waste energy recovery heat exchanger which is able to recover the latent energy for the purpose of reducing the heating energy. Supplied energy by the boiler and recovered energy by the heat exchanger were calculated for the parameters such as pass number, kind of fuel and hot water velocity. The following conclusions are made:

(1) The temperature of the thermally controlled facility was well kept at the setting temperature even in the change of outdoor temperature by the control of boiler or chiller.

(2) With the increase of pass number, the variation of temperature of system from setting value is decreased at the maximum heating load.

(3) The recovered latent energy using LNG is greater than that using diesel oil as a fuel while the recovered sensible energy is almost same regardless of the fuel.

(4) The ratio of recovered energy to the supplied energy increases with the pass number, but the suitable number of pass should be chosen in consideration of expenses.

(5) Since the heat energy supplied by boiler becomes almost constant once the flow velocity exceeds the certain value, the flow velocity should be chosen in this range.

Acknowledgements

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