Performance Evaluation of a Water-to-Water Heat Pump with a Floor Panel Heating System

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The performance of heat pump is severely degraded as the temperature of the heat source is decreased. For air source heat pumps, this results in a serious mismatch of heat pump output and space heating demand. Although the outdoor temperature is below 0℃, the use of a water-to-water heat pump with a floor panel heating system can operate for extended periods of time without frost-defrost cycling and therefore at a high seasonal efficiency. This paper focuses on the performance of a water-to-water heat pump that uses well water as a heat source and a floor panel heating system as a sink.

Field tests of a water-to-water heat pump combined with the floor panel heating system in a heavily insulated miniature test house were conducted to obtain its cyclic performance as well as the seasonal performance. The test house, which had a floor area of 27.5 m^2, consisted of three rooms that were equipped with panel heating coils under floor surface. The heating capacity of the water-to-water heat pump was 3.95 kW with double tube heat exchangers. The averaged COP of the water-to-water heat pump system was measured for every cycle throughout the intermediate season and analyzed as a function of outdoor temperature. The cyclic performance such as part load factor, cyclic operation performance, and degradation coefficient degraded as the outdoor temperature increased, because of relatively increasing off-cycle duration in the cycle as well as lengthening of the cycle duration.

Key Words: Water-to-Water Heat Pump, Floor Panel Heating System, Cyclic Performance

Nomenclature

A : Area
C_D : Degradation coefficient
C_p : Specific heat
$E(t_j)$: Power input for compressor and pump for j^{th} bin
$\dot{m}$: Mass flow rate
n_j : Number of hour for jth bin
No : Ventilation number per hour
$\dot{Q}$: Heat transfer rate
Q_{inf} : Infiltration heat loss
Q_{tl} : Conduction heat transfer through enclosure
t_c : Temperature of attic
t_{in} : Indoor temperature
t_j : Temperature of jth bin
t_{on} : Compressor working time
t_{out} : Outdoor temperature
U : Overall heat transfer coefficient
V : Volume of the heating space
$\dot{W}$: Power input
τ : Period of cycle
Δt : Time interval

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1. Introduction

An air-to-air heat pump has advantages of a safe and lasting useful heat source, with no restriction in installation space and low initial and operating costs. However, it has the disadvantage of the low efficiency dropping dramatically by the heating capacity and the frost of outdoor unit during the heating mode when the outdoor temperature is low (Miller, 1985). Because of this, it is difficult to use the heat pump practically for the most part of Korea where the temperature is almost always below 0℃ throughout winter except the southern regions of Korea. A better heating efficiency can be achieved with the water-to-water heat pump using well water than the air-to-air heat pump because the temperature of well water is higher than that of air in winter. Moreover, well water is abundant and also has no intensive seasonal variation, maintaining a nearly constant temperature throughout the year.

Most of the previous study on the cyclic performance of a heat pump was focused on the air source heat pump systems (Goldschmidt, et al., 1980). Miller (1985) investigated the cyclic performance of an air-to-air heat pump as a function of the on-time ratio and outdoor temperature. Bittle and Goldschmidt (1985) also did an analysis of the laboratory test data and estimated the heating seasonal performance factor using the temperature bin method. Murphy and Goldschmidt (1979) performed field tests for a 3-ton air conditioner and predicted the degradation coefficient. The possibility of the application of a water source heat pump was investigated by several researchers (Mei, 1983; Brown, et al., 1988; Kauffeld, et al., 1990). Brown and Hess (1988) monitored the performance of groundwater heat pump system for office building applications.

In this paper, a heating system consisting of a water-to-water heat pump was combined with a floor panel heating system because the floor heating method coincides with the traditional heating method used in Korea. This system needs no indoor heating equipment that allows efficient space use, and requires almost no repair once the system has been installed. The variations of the operating characteristics and the seasonal performance factors of the water-to-water heat pump due to the changes of outdoor temperature were found by field experiments throughout the intermediate season.

2. Experimental Apparatus and Procedure

2.1 Test facility

The test apparatus consisted of the test house, water-to-water heat pump and the panel heating system. The test house, which had a floor area of 27.5 m^2, consisted of three rooms which were equipped with a panel heating coil under the floor surface. The wall of the test house was well insulated with a overall heat transfer coefficient of 0. 988 kJ/(hr-m^2℃). Figure 1 shows the layout of the panel heating system. The floor coils were arranged in a parallel configuration for each room. The heated water from the condenser passed through the panel heating coil in a closed loop water system. Figure 2 shows the location of the heating coil in the floor and the points of temperature measurement. Regularly spaced heating coils with a pitch of 20 cm were buried 5 cm

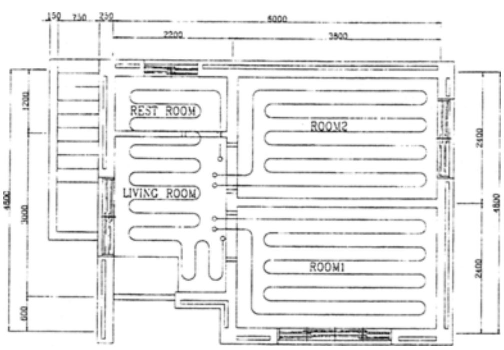

Fig. 1 Schematic of the panel heating system.

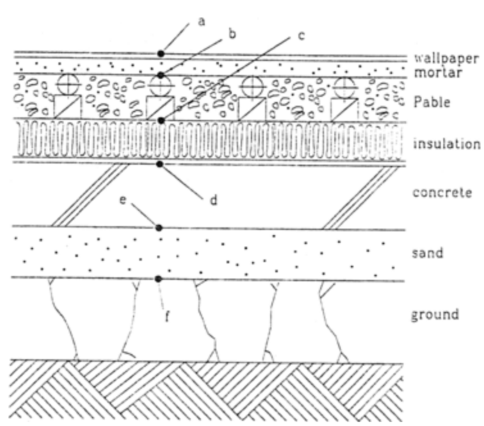

Fig. 2 Cross section view of the floor.

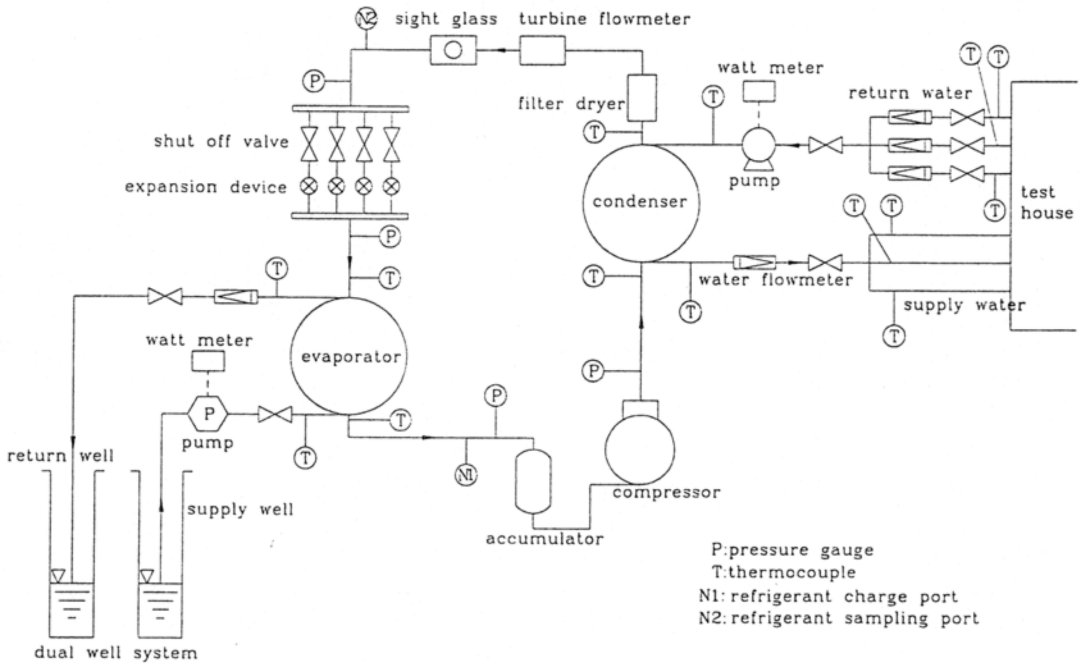

Fig. 3 Schematic diagram of the water-to-water heat pump system.

below the floor surface. The floor slab was separated from the ground by a layer of insulation, concrete, and sand.

The water-to-water heat pump had a nominal capacity of 3.95 kW. It was installed at the basement of the test house. The test heat pump included a hermetically sealed rotary compressor, two double tube heat exchangers (condenser and evaporator), dual well system, and pumps for water circulation. Figure 3 shows the schematic diagram of the test water-to-water heat pump system. The helically coiled double tube heat exchanger was used in both the condenser and evaporator. The refrigerant and water passed through the outer and inner tube, respectively, in a counter flow heat exchanging process. The capillary tube and the thermostatic expansion valve(TXV) were connected in parallel to control the refrigerant flow in the system.

Temperatures of the test house were measured at the following location using thermocouples: both inner and outer wall surface, roof, attic, and room air. Temperatures below the floor surface were also measured at five points from the floor surface to the ground.

The refrigerant circuit was instrumented with thermocouple probes, pressure transducers, watt transducers, and flow meters. Figure 3 shows the location of the basic instrumentation for the test heat pump system. Temperatures and pressures were measured by thermocouples and pressure transducers, respectively, at the entrance and the exit of both condenser and evaporator. The pressure transducers were calibrated with the dead weight tester and the estimated accuracy was 0.5% of the full scale. The temperature was measured using T-type thermocouples which had an accuracy of 0.28℃. The refrigerant flow rate was measured by a turbine flow meter which was calibrated in the flow loop and had an estimated accuracy of 0.5% of the full scale. Flow rates of the heat source and heated water were measured by a turbine flow meter. The power consumption through the compressor was measured using a watt transducer with an instrumentation accuracy of 0.25%. The power input of the water pump was measured using the watt hour meter which had an accuracy of 0.5% of the full scale.

2.2 Experimental procedure

The water flow rate of the heat source and heated water flowing through the heating coils were fixed at 5.1 kg/min and 17.5 kg/min, respectively. The heated water flowing through the coil was selected as 5.45 kg/min and 6.6 kg/min for bedrooms and living room, respectively. The temperature of the heat source varied from 12 to 15℃ during the test period. The refrigerant charge was set in the heating mode with a returned water temperature of 35℃ and source water temperature of 15℃. The unit was charged until the degree of subcooling leaving the condenser reached 5℃. The optimum charge amount for the current system was 1.02 kg.

The designed room temperature was set to 22℃ with a outdoor design temperature of -15℃, and the surface temperature of the floor was maintained at 25℃ using an installed temperature controller. The system including the compressor and pumps was started at the surface temperature of 24℃ and shut off at the temperature of 26℃.

Temperatures and pressures at the eight locations of the system, outdoor temperature, refrigerant flow rate, water flow rate, and power input to the compressor and pump were monitored using the data acquisition system at a 30 second interval. The average value of COP and cyclic performance of the water-to-water heat pump were calculated based on the measured data.

3. Theory and Correlation

Based on the experimental results, the following performance factors for a heat pump were calculated: coefficient of performance (COP), heating load factor (HLF), work load factor (WLF), part load factor (PLF), degradation coefficient (CD), and heating seasonal performance factor ($HSPF$). This section briefly describes the definition of the factors considered in the present paper.

The averaged coefficient of performance for the heat pump was evaluated as a function of outdoor temperature bin:

$$COP_{net}=\frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \tag{1}$$

$$COP_{gross}=\frac{\dot{Q}_{cond}}{\dot{W}_{comp}+\dot{W}_{pump}} \tag{2}$$

where,

$$\dot{Q}_{cond}=\dot{m}C_{pw}\Delta t \tag{3}$$

$\dot{Q}_{cond}$ is the heat rejected through the condenser, and $\dot{W}_{comp}$, $\dot{W}_{pump}$ are work inputs into the compressor and pump, respectively.

Heating load factor (HLF) was defined as the ratio of the heating capacity for the cyclic operation of the compressor to the heating capacity for the steady state operation during a cyclic time of τ.

$$HLF=\frac{Q_{cyc}}{\dot{Q}_{ss}\tau}=\frac{\int_0^{ton}\dot{Q}(t)\,dt}{\dot{Q}_{ss}\tau}=\frac{\dot{Q}_{cyc}t_{on}}{\dot{Q}_{ss}\tau} \tag{4}$$

where, $Q_{cyc}=\dot{Q}_{cyc}t_{on}$ and $\dot{Q}_{cyc}$ is the average heating capacity during the compressor operation (t_{on} period). The term of $\dot{Q}_{ss}$ denotes the steady state heating capacity.

Work load factor (WLF) is the ratio of power input for cyclic operation to that of the steady state operation during a cyclic time of τ.

$$WLF=\frac{W_{cyc}}{\dot{W}_{ss}\tau}=\frac{\int_0^{ton}\dot{W}(t)\,dt}{\dot{W}_{ss}\tau}=\frac{\dot{W}_{cyc}t_{on}}{\dot{W}_{ss}\tau} \tag{5}$$

where, $Wcyc=\dot{W}_{cyc}t_{on}$ and $\dot{W}_{cyc}$ is the average work input to compressor during operation (t_{on} period). The term of $\dot{W}_{ss}$ denotes the steady state work input to the compressor. Average work input used during the cyclic operation is higher than that used during the steady state operation.

Part load factor (PLF) is the ratio of COP for the cyclic operation (COP_{cyc}) to that of the steady operation (COP_{ss}) (Goldschmidt, et al, 1980).

$$PLF=\frac{COP_{cyc}}{COP_{ss}} \tag{6}$$

The following relationship among HLF, WLF, and PLF can be derived from Eqs. (4) ~ (6):

$$PLF=\frac{HLF}{WLF} \tag{7}$$

The reduction of heating capacity due to cyclic operation can be expressed as (1-HLF). The degradation of the system performance, (1-

PLF), can be increased as the (1-HLF) increases. Thus, the degradation coefficient (C_D) can be defined as:

$$C_D=\frac{1-PLF}{1-HLF} \tag{8}$$

Heating seasonal performance factor($HSPF$) is the ratio of the total heating output of a heat pump during its normal heating period to the total electric power input during the same period over a season(ANSI/ASHRAE, 1983, 1984). The temperature bin method was used to determine the $HSPF$ due to the high dependence of $HSPF$ on the climatic region. The procedure to determine $HSPF$ includes the calculation of the following for each temperature bin: building heat load(BL), a heat pump load factor(LF), part load factor(PLF), supplementary resistance heat term(RH), a degradation coefficient(C_D) and heating load factor(HLF) (Bittle and Goldschmidt, 1985).

$$HSPF=\frac{\sum_{j=1}^{k} n_j\cdot BL(t_j)}{\frac{\sum_{j=1}^{k} n_j\cdot LF(t_j)\cdot E(t_j)}{PLF(LF)}+\sum_{j=1}^{k} RH(t_j)} \tag{9}$$

where,

$$BL(t_j)=Q_{tl}+Q_{\mathrm{Inf}} \tag{10}$$

$$Q_{tl}=UA(t_{in}-t_{out}) \qquad Q_{\mathrm{Inf}}=\rho_a V\cdot No\cdot C_{pa}(t_c-t_{out}) \tag{11}$$

$$LF(t_j)=\begin{cases}\frac{BL(t_j)}{Q_c(t_j)} & \text{for } Q_c(t_j)>BL(t_j)\\ 1 & \text{for } Q_c(t_j)<BL(t_j)\end{cases} \tag{12}$$

$$PLF(LF)=1-C_D(1-LF(t_j)) \tag{13}$$

$$RH(t_j)=[(BL(t_j)-Q(t_j)\cdot(HLF)(t_j)]\cdot n_j \tag{14}$$

4. Results and Discussions

Field tests for a water-to-water heat were performed during the intermediate season from March to June. The tests include total of 401 cycles and 2,402 operating hours. Figure 4 shows the percent of occurrence hours for each outdoor temperature bin which was varied from -5.6℃ to 27.8℃. The peak of the percent of occurrence hours was at 22.2℃, which was higher than that of the winter season.

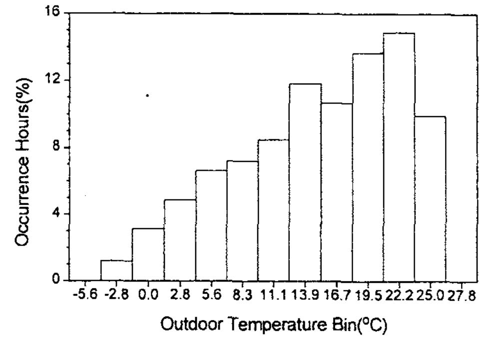

Fig. 4 Occurrence hours as a function of outdoor temperature bin during the test period from March to June.

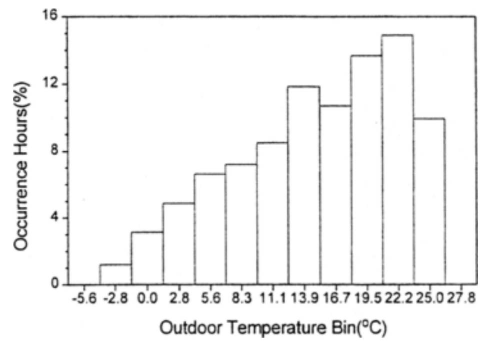

Fig. 5 On-time ratios as a function of outdoor temperature bin during the test period from March to June.

Figure 5 shows the on-time ratio (the ratio of t_{on} to τ during a cyclic operation) as a function of outdoor temperature bins. Generally, the on-time ratio for the intermediate season was lower than 0.6 that was relatively low compared to the value for the winter season due to the decrease of the heating load. Because the heating load was decreased with the increase of outdoor temperature, the compressor operating time and on-time ratio were decreased as the number of cycle was increased.

Figure 6 represents the averaged COP of the water-to-water heat pump as a function of outdoor temperature bins. Generally, as the outdoor temperature increased with a constant indoor temperature, the heating demand increased and heating capacity of an air source heat pump decreased. Thus, there was a serious mismatch in

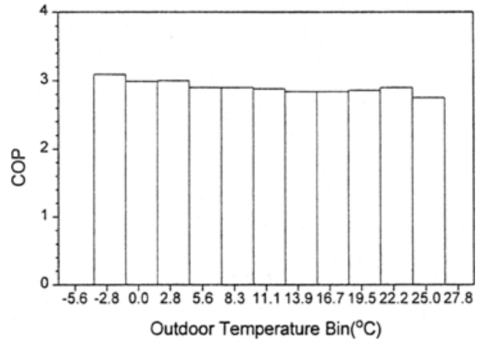

Fig. 6 Averaged COP as a function of outdoor temperature bin during the test period from March to June.

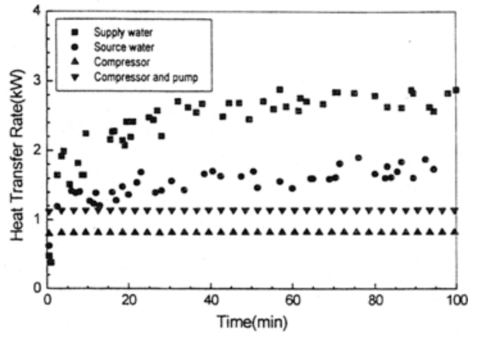

Fig. 7 Variation of heat transfer during the start up of a cycle.

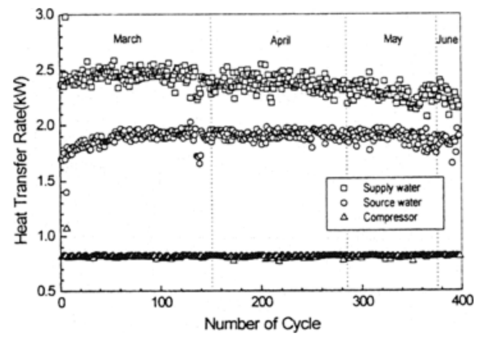

Fig. 8 Variation of heat transfer rate and power input during the test period from March to June.

heat pump output and space heating demand. These often require higher capacity of supplementary heating equipment. The degradation of COP became severe under the frosting conditions of the outdoor coil at low outdoor temperature. For a water-to-water heat pump the averaged COP and heating capacity were maintained relatively constant with the decrease of outdoor temperature by using well water as a heat source. Since the well water had no intensive seasonal temperature variation and does not drop below 0℃, the effects of outdoor temperature on the COP and capacity were reduced and the frosting problems in the outdoor coil at the cold region were prevented. This would suggest that a water-to-water-heat pump could be applied to the cold climate region without having a severe degradation of the seasonal performance and heating capacity.

For the cyclic operation of the heat pump, as the on-time ratio increased up to 0.3, the averaged COP increased. However, the COP was maintained fairly constant with the increase on-time ratio beyond 0.3. During the test period, both the number of cycle per month and on-time ratio were decreased with the increase of outdoor temperature. Therefore, the averaged COP slightly reduced as the outdoor temperature bin increased.

Figure 7 shows the variation in heat transfer during the start up of the system. During the start up period, power input to the compressor and pump reached the steady value within a minute, however, the heating capacity of supply water increased slowly and reached the steady value after 35 minutes from the initial start of a cycle. This heat loss during the start up affects the cyclic response of the water-to-water heat pump. Due to the cyclic loss during start up period, the averaged COP was considerably lower than that of the steady state value.

Figure 8 shows the variation in heat transfer of the supply and source water. The flow rate of the source water was fairly constant during the test period even though the outdoor temperature was varied. The slight increase in heat transfer of the source water in March was due to the increase of ground water temperature. The heat transfer of the supply water was held relatively constant throughout the intermediate season. The slight decrease in heat transfer of supply water with the increase of number of cycle might be due to the increase of outdoor temperature which results in a reduction of heat loss through the envelope and

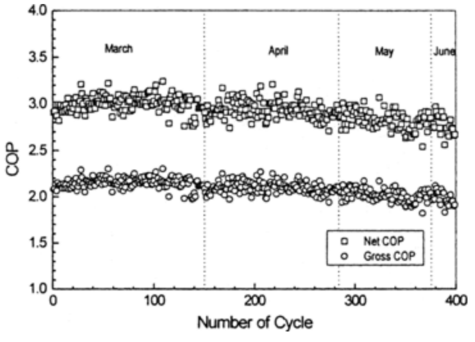

Fig. 9 Variation of the averaged COP during the test period from March to June.

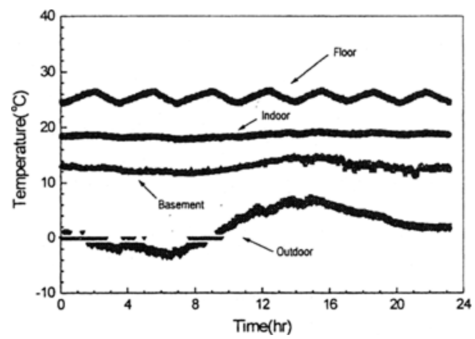

Fig. 10 Variation of floor and indoor temperature during the cyclic operation.

ground. This will decrease the required amount of heat and averaged heat transfer rate as well for a cycle.

Figure 9 represents the variation of COP during the test period. As the number of cycle was increased, the averaged heating capacity slightly reduced and the power input for a cycle also slightly decreased. However, the decrease of heating capacity was larger than that of the power input, which results in a slight decrease of COP with the increase of the number of cycle. The decrease in averaged COP might also come from the cyclic effects of the system operation.

Figure 10 shows the variation of the floor and indoor temperature during a cyclic operation. The floor temperature was maintained at ±1℃ temperature swing from the setting temperature of 25℃. Generally, the floor and indoor temperature were maintained well at the setting temperature regardless of outdoor temperature variation.

Figure 11 shows the relationships among the

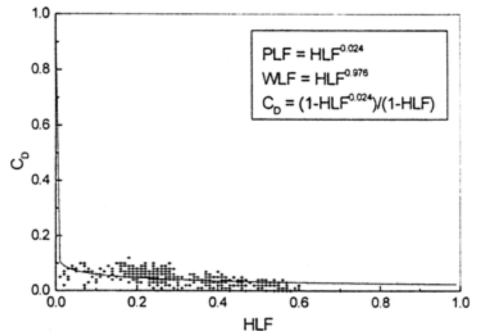

Fig. 11 Variation of degradation coefficient as a function of heat load factor.

factors for the cyclic operation of the tested heat pump. The following correlations were obtained by curvefitting of the experimental data in terms of the nondimensional factors for a cyclic operation:

$$WLF = HLF^{0.976} \tag{15}$$

$$PLF = HLF^{0.024} \tag{16}$$

$$C_D = \frac{1-PLF}{1-HLF} = \frac{1-HLF^{0.024}}{1-HLF} \tag{17}$$

The heating seasonal performance factor was calculated utilizing the bin method with power input and heating capacity for each temperature bin. The estimated *HSPF* for a water-to-water heat pump during the intermediate season from March to June was 2.07.

5. Conclusions

Performance of the water-to-water heat pump system with the floor panel heating system in the miniature test house was measured during the intermediate season. The conclusions obtained are summarized as follows.

The degradation of COP of the water-to-water heat pump with the decrease of outdoor temperature was less than that of the air-to-air heat pump because the temperature of the heat source for the water-to-water heat pump was maintained relatively constant. The cyclic performance deteriorated as the outdoor temperature increased due to the increase of off-cycle duration as well as the increase of a cycle period. The cyclic performance of the tested heat pump throughout the intermediate season (from March to June) can be expressed by the following correlations:

$$WLF = HLF^{0.976}$$

$$PLF = HLF^{0.024}$$

$$C_D = \frac{1-PLF}{1-HLF} = \frac{1-HLF^{0.024}}{1-HLF}$$

In addition, the *HSPF* for the tested heat pump throughout the intermediate season was calculated based on the bin method, and the estimated value was 2.07.

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