CHILLER MODELS FOR HVACSIM$^+$ (J)

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ABSTRACT: Dynamic simulation models to evaluate various performances and dynamic properties of chillers are presented in the present paper. These models are developed for precise energy evaluations that are very important to pursue effective energy savings. The best method to evaluate the energy saving is using dynamic simulation of the buildings. As a dynamic building simulator, HVACSIM$^+$ that was originally developed by NBS, present NIST USDOC is useful. The software has been rearranged and several models were added in Japan to establish HVACSIM$^+$ (J). For the simulation program, an absorption chiller model, reciprocal chiller and turbo-chiller model are developed. The model is described with differential equations of heat balances and material balances. Since the equations simulate physical phenomenon in the chiller, the model simulates even start-up responses and medium load characteristics that are not usually given by manufacturers. The models are examined through comparison with operational data of the chillers.

1. INTRODUCTION

It is a very important issue to decrease earth warming gas, in particular, CO2 emission amount in order to prevent climate changes. Saving energy is the best way to decrease CO2 emission. In the building related energy consumption, air conditioning uses about 50% of total energy consumption. There exist various methods to save building related energy, such as energy efficient machine introductions, building thermal insulation improvements and introduction of energy management system. In any case, precise energy evaluations are very important to pursue effective energy savings. The best method to evaluate the energy saving is using dynamic simulation of the buildings, because flexible conditions can be considered. As a dynamic building simulator, HVACSIM$^+$[1)] that was originally developed by NBS, present NIST USDOC is useful. The software has been rearranged and several component models were added in Japan to establish HVACSIM$^+$ (J)[2)] for the sake of Japanese users.

In the present paper, an absorption chiller model and turbo-chiller model for HVACSIM$^+$ (J) are introduced. The models are described by differential equations of heat balances and material balances. Since the equations simulate physical phenomenon in the chiller, the model simulates even start-up responses and medium load characteristics which are not usually given by manufacturers. The models are examined through comparison with operational data of the chillers. The differences are to be discussed in the paper.

2. CHILLER MODELS

2.1 Absorption Chiller

The model develped is 2-statge absorption chiller (Fig.1). In the chiller, LiBr solution cycle and water cycle works.

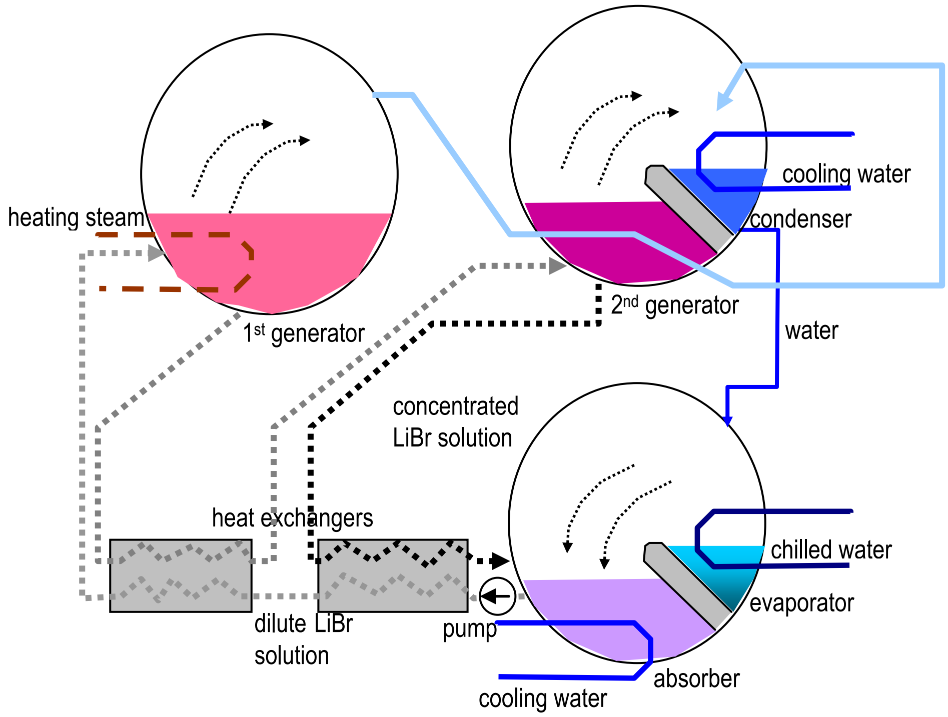


Fig.1 an absorption chiller

In each vessel, the liquid works as follows[3)].

(1)In the evaporator, water evaporates in low-pressure environment and takes heat from chilled water in heat exchange pipe.

(2)In the absorber, LiBr solution absorbs vapor coming from the evaporator. The absorption yields the condensation heat and dilutes the solution. So, the absorber is cooled by cooling water, and its solution is pumped up to the generator through the heat exchanger to condense the solution.

(3)In the 1st and 2nd generators, the solution is heated by steam or hot water so that it boils to remove a part of water. Then the solution become concentrated and return to the absorber.

(4)In the condenser, the vapor from the generator is condensed to liquid water, when the latent heat is released. The condenser is cooled by cooling water.

Fig.2 shows the refrigerant and solvent solution cycle on the Duhring diagram.

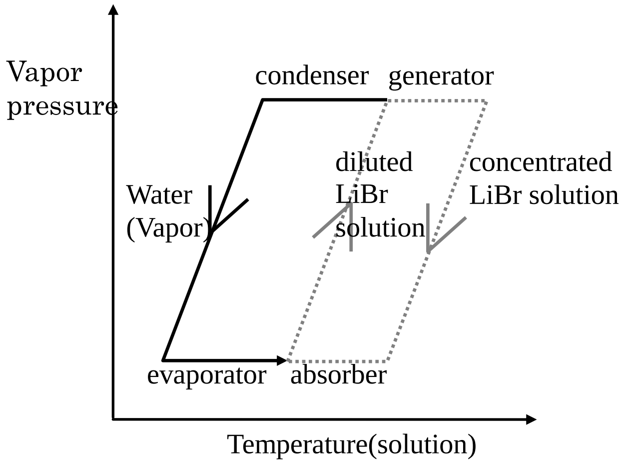


Fig.2 the refrigerant and solvent solution cycle

The model consists of the equations which express the above mentioned phenomena. The equations contain the mass balances, energy balances, and equations for equilibrium states of LiBr/H_2O solution as follows.

In the absorber:

(Heat balance equation)

$$(C_a^{LB}W_a + C_{Va})\frac{dT_a}{dt} + C_a^{LB}T_a\frac{dW_a}{dt} = (r_a + C_W T_V)G_a - \alpha_a(T_a - T_{cool}) + C_r^{LB}G_{dw}T_{ch} - C_a^{LB}G_{up}T_a \tag{1}$$

(Water mass balance equations)

$$\frac{dW_{wa}}{dt} = G_a - (1 - Conc_a)G_{up} + (1 - Conc_r)G_{dw} \tag{2}$$

$$G_a = f(Conc_a, T_v, T_a) \tag{3}$$

The function $f(Conc, T1, T2)$ means the amount of the vapor flow that is described by LiBr solution concentration and the temperature of the absorber and the evaporator.

(Solution mass balance equations)

$$\frac{dW_{LBa}}{dt} = Conc_r G_{dw} - Conc_a G_{up} \tag{4}$$

$$Conc_a = \frac{W_{LBa}}{W_a} = \frac{W_{LBa}}{W_{LBa} + W_{wa}} \tag{5}$$

(Condensation heat)

$$r_a = \frac{q_w}{\mu_w}\frac{T_a^2}{T_v^2}\Delta T_a \tag{6}$$

In the above equation, μ_w implies molecular weight and ΔT_a means the pure water temperature change such that the change yield the same vapor pressure change as one degree change of the LiBr solution temperature yields in the absorber, which is read out from the Duhring diagram.

In generators (2-stage absorption chiller has two generators and the model has 2 sets of following equations) :

(Heat balance equation)

$$(C_r^{LB}W_r + C_{Vr})\frac{dT_r}{dt} + C_r^{LB}T_r\frac{dW_r}{dt} = -(r_r + C_W T_r)G_r + \alpha_r(T_{vpr} - T_r) + C_a^{LB}G_{up}T_{th} - C_r^{LB}G_{dw}T_r \tag{7}$$

(Water mass balance equations)

$$\frac{dW_{wr}}{dt} = -G_r + (1 - Conc_a)G_{up} - (1 - Conc_r)G_{dw} \tag{8}$$

$$G_r = f(Conc_r, T_c, T_r) \tag{9}$$

(Solution mass balance equations)

$$\frac{dW_{LBr}}{dt} = Conc_a G_{up} - Conc_r G_{dw} \tag{10}$$

$$Conc_r = \frac{W_{LBr}}{W_r} = \frac{W_{LBr}}{W_{LBr} + W_{wr}} \tag{11}$$

(Evaporation heat)

$$r_r = \frac{q_w}{\mu_w}\frac{T_r^2}{T_c^2}\Delta T_r \tag{12}$$

Where, ΔT_r is the pure water temperature change such that the change yields the same vapor pressure change as a degree change of the LiBr solution temperature yields in the generator. The equation is the same as the equation (6).

In the condenser and evaporator, as the heat balance and water mass balance equations like equation (7) and (8) hold, equations are omitted. Furthermore, the above heat balance equations use heat transfer coefficients α, presently developed simulator adopts the heat transfer efficiency η defined as follows in order to avoid simulation instabilities due to large heat transfer coefficients.

$$\eta = \frac{T_{out} - T_{in}}{T_{vessel} - T_{in}} \tag{13}$$

In the above equation, T_{in}, T_{out} means the output and input temperature of cooling water or steam and T_{vessel} implies the mean temperature of solution and vessel such as an absorber.

The model is equipped with chilled water temperature controller, by adjusting steam supply. The control algorithm is described as follows.

$$G_{vap} = Kp(T_{clt} - T_{set}) + \frac{Kp}{Ti}\int(T_{clt} - T_{set})dt \tag{14}$$

Finally, the mass transfer between the solvent solution and water vapor is modeled as diffusion process in gas phase and water solutions. The main motive force is the concentration differences in the solution surfaces that are in equilibrium states to the solution or the water in the vessels. In case of the absorber, flow rate is described as follows,

$$G_a = -S_a D_a \frac{w_{sa} - w_{sv}}{d_a} \tag{15}$$

Where, S_a means the absorber cross section, D_a implies diffusion coefficient, w_{sa}, w_{sv} are vapor equilibrium densities to the absorber solution and the evaporator water. d_a is the absorber diameter.

The above mentioned equations are programmed in FORTRAN language as the type function of HVACSIM$^+$(J). Followings are an example of simulation results and its conditions.

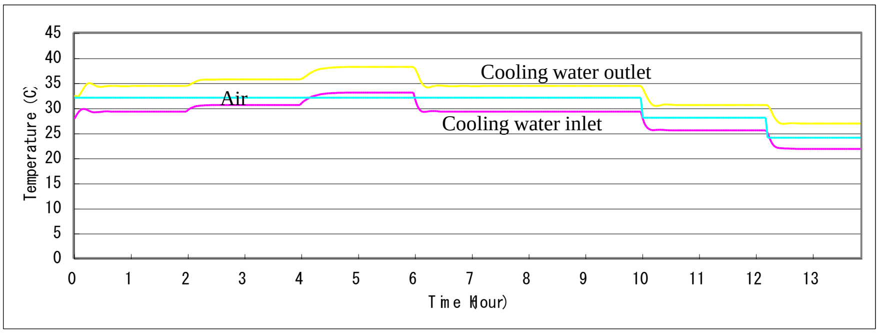
Table 1 Simulation conditions

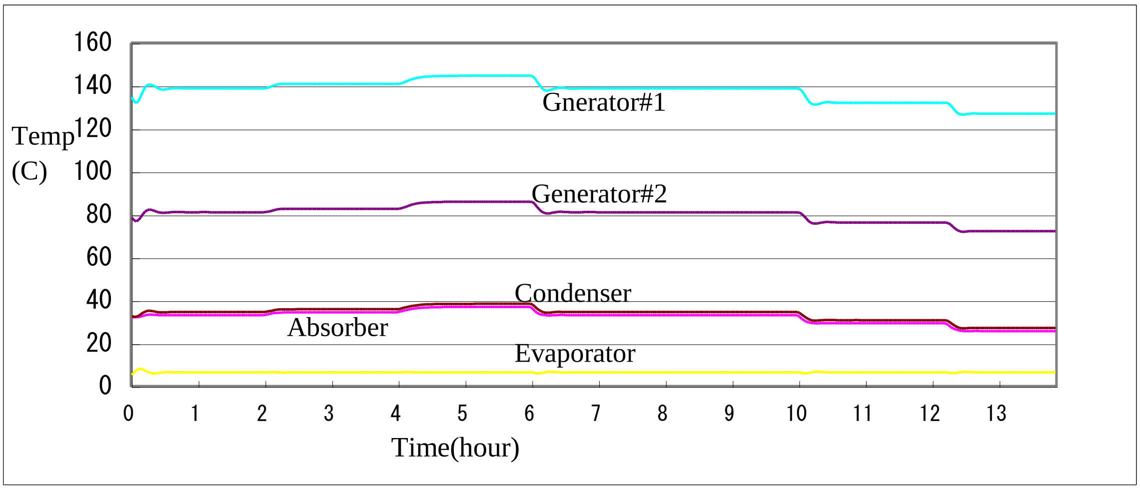
Parameters	*Values*
Cooling water flow rate	19.2kg/sec
Cooling water temperature	32°C
Chilled water flow rate(return)	10.3kg/sec
Chilled water temperature(return)	14°C
vapor temperature	180°C
vapor pressure	800kPa

Table 2 Chiller parameters

Parameters	*Values*
Heat exchange coefficient for absorber	0.9
... forevaporator	0.9
... for condenser	0.9
... for generator#1	0.79
... for generator#2	0.8
... for heat exch.	0.95
Heat capacity for absorber	486kJ/°C
... for evaporator	486kJ/°C
... for generator#1	221kJ/°C
... for generator#2	221kJ/°C
... for condenser	221kJ/°C
Cross section for absorber	5.41m^2
Absorber radius	1m
Cross section for generator#1	0.172 m^2
Generator radius#1	1m
Cross section for generator#2	0.172 m^2
Generator radius#2	1m
Time lag for solution heat exchanger	0.5min
Solution circulation flow rate	10kg/sec
Proportional gain for temp. controller	0.05
Integral gain for temp. controller	300sec
Chilled water temp. set value	7°C
Low temp. protection	6°C

The Chiller is 300kW 2-stage chiller. These parameters are assumed for the chiller. Then the following simulation results are obtained. In this case, cooling water temperature changes from 29°C to 33°C, then it goes down from 29°C to 22°C. With this condition change, chiller temperature changes and COP also changes.





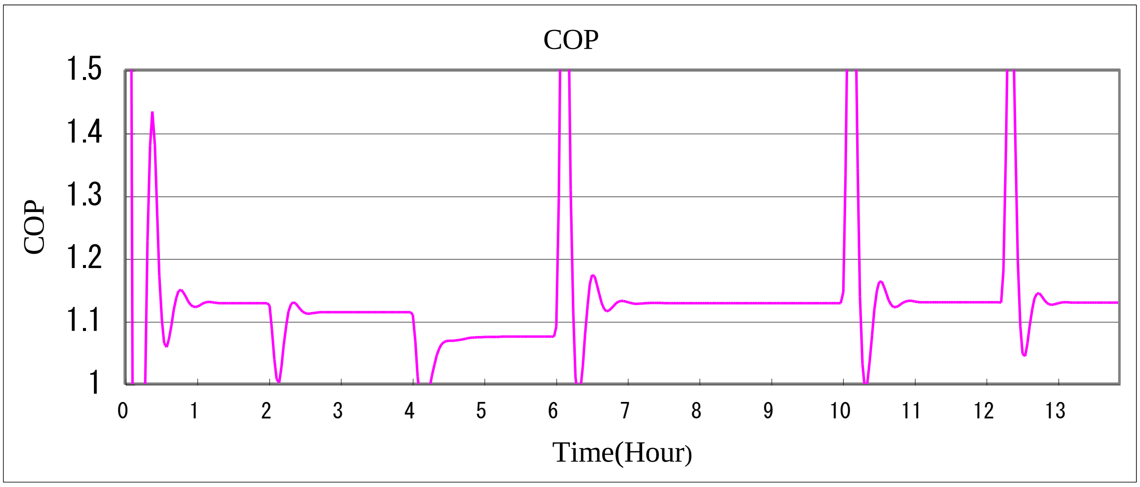


Fig.3 An example of simulation results

The characteristics of the chiller performance obtained by the simulations are described as follows.

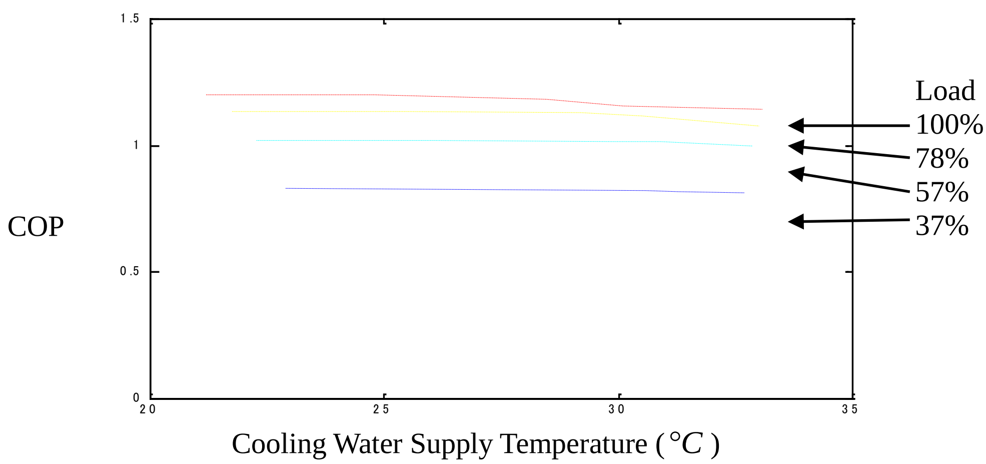


Fig.4 Chiller performance characteristics by simulations

2.2 Reciprocal Chiller

Reciprocal chillers are mainly used in many small scale buildings, small stores and housings. It is considered to play an important role to save nation's energy consumption in Japan, because the number of the chiller is very big.

The chiller cycle consists of a compressor, condenser and evaporator (Fig.5). The compressor pressurizes the refrigerant from the evaporator and sends out to the condenser. In the condenser, the pressurized hot refrigerant gas is cooled to liquid state, transferring its heat to water or air. In case of warming, the heat is used for air-conditioning. Then pressurized liquid refrigerant blows into the evaporator through an expansion valve. Boiling under low pressure, refrigerant temperature goes down and it deprives heat from surrounding water or air. In case of cooling, the heat is used for cooling.

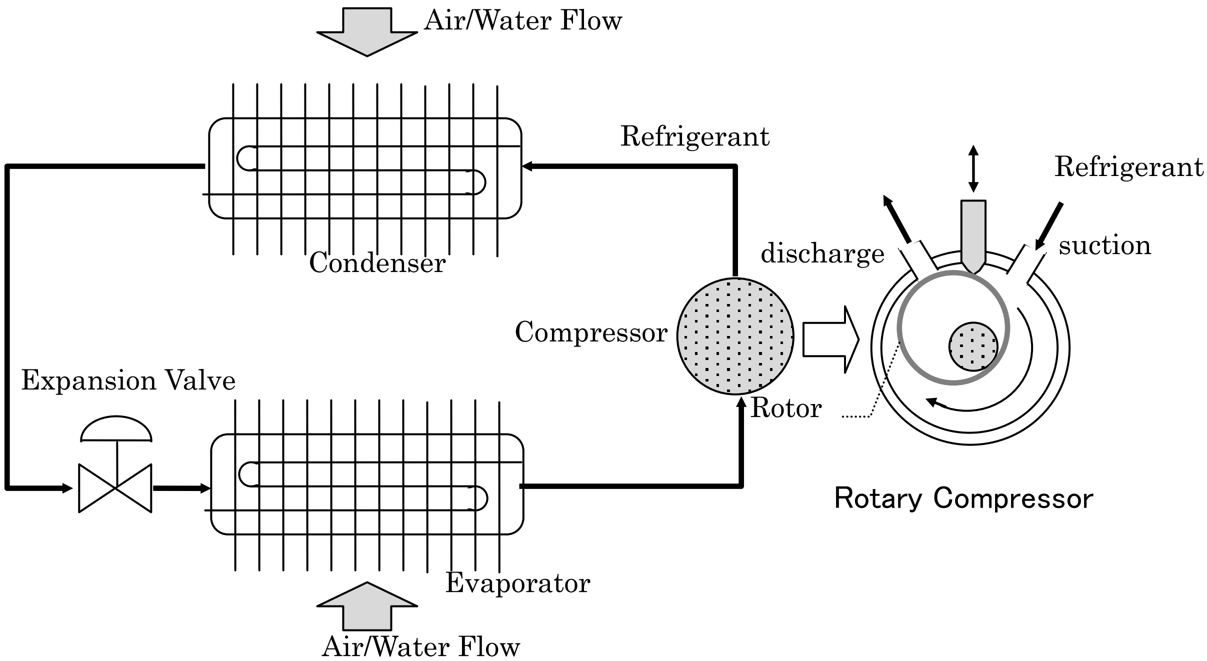


Fig. 5 Refrigeration cycle

2.2.1 Compressor Model

Let P_s refrigerant suction pressure, T_s temperature, h_s enthalpy and P_{cond} condenser pressure be given by other models. The compressor model calculates suction flow rate G_s, discharge flow rate G_d and compressor temperature T_{comp}. The purpose of the model is to calculate energy consumption in the chiller. Then only temperature dynamics are considered. The other mechanisms are described by static equations[4)].

The suction flow rate G_s :

$$G_s = (\eta_v V_{comp}(1/v_s(P_s,T_s) - vo/v_d(P_d,T_d)))N \tag{16}$$

where, $v_s(P_s,T_s)$ is specific volume for suction refrigerant, $v_d(P_d,T_d)$ is specific volume for

discharge refrigerant, η_v is volume efficiency, vo is dead space, and N is rotational speed.
Discharge pressure P_d :

$$P_d = P_{cond} + \Delta P \tag{17}$$

where, ΔP is pressure drop at the compressor outlet.

Compressor power W_{comp} and heat generation Q_{comp} :

$$W_{comp} = {(h_{isen} - h_s) \times G_d} \Big/ {\eta_{mec} \cdot \eta_{mo}}, \tag{18}$$

where, h_{isen} is isentropic enthalpy assuming that the entropy of the refrigerant doesn't change during compression process, η_{mo} is mortor efficiency and η_{mec} is mechanical efficiency, and G_d is discharge flow rate and equal to suction flow rate G_s. Heat generation in the compressor is calculated as the following.

$$Q_{comp} = W_{comp} - h_d \cdot G_d + h_s \cdot G_s \tag{19}$$

Then, the temperature dynamics of compressor inner wall and compressor bulk is described as the followings.

$$C_{comp1} \frac{dT_{in}}{dt} = Q_{comp} - \beta(T_{in} - T_{comp}) \tag{20}$$

$$C_{comp2} \frac{dT_{comp}}{dt} = \beta(T_{in} - T_{comp}) - \beta_2(T_{comp} - T_{out}) \tag{21}$$

where, β and β_2 are heat transfer coefficients between inner wall and the bulk, and between the bulk and surrounding air, C_{comp1} and C_{comp2} are the heat capacity for the compressor of the part, and T_{in} is inner wall temperature and T_{out} is outside air temperature.

2.2.2 Condenser and Evaporator Model

Both components are heat exchangers. In the condenser, refrigerant changes from gas, 2-phase to liquid phase. In the evaporator, refrigerant phase changes from liquid and gas mixed flow to gas flow. Then, model equations are similar to each other. In the following, only condenser equations are described. Because heat exchange is mainly done through 2-phase region, it is very important to predict refrigerant distribution in the heat exchanger. So the model consists of refrigerant distribution model and heat mass balance equations[4)].

(1) Refrigerant distribution model
The 3 node model that consists of gas phase, liquid-gas mixed phase and liquid phase region is adopted to describe continuous condensation phenomena in the tube, in order to minimize calculation complexity for practical engineering work. Furthermore, no pressure drop in the tube is considered for simplicity. The average void ratio is calculated by Zivi's equation[5)] that is derived from minimum entropy principle.
(a) The lengths for gas, liquid gas mixed and liquid phase r_g, r_2, r_l :

$$r_g = \frac{(heat\ amount\ to\ condense\ of\ compressor\ discharge\ refrigerant)}{(heat\ transfer\ rate\ of\ the\ heat\ exchanger)}$$

$$= \frac{G_d \cdot C_r \cdot (T_d - T_{cond})}{\alpha_{cond} \cdot S_{cond} \cdot (T_{tube} - T_{air})}, \tag{22}$$

where, C_r is specific heat capacity for refrigerant, T_{cond} is the condenser 2-phase region temperature, α_{cond} is heat transfer coefficient of the condenser, S_{cond} is heat transfer area of the tube, T_{tube} is the tube temperature and T_{air} is cooling air.

The r_2 and r_l are calculated using average void ratio by Zivi's equation.

$$S_c \cdot r_2 \cdot (1-\alpha_c) \cdot \rho_{lc} + S_c \cdot (l_c - r_2 - r_g) \cdot \rho_{lc} = W^l_{cond} \tag{23}$$

where, S_c is the cross section of the tube, α_c is the average void ratio, ρ_{lc} is average refrigerant density of the condenser, W^l_{cond} is liquid refrigerant weight in the condenser and l_c means the condenser tube length. The quality of the condenser output refrigerant x_c is expressed as the following.

$$x_c = 1 - \frac{W^l_{cond}}{W_{critic}} \tag{24}$$

where W_{critic} means maximum refrigerant inventory in condenser tube when the condenser tube length of $r_2 + r_l$ is fully filled with the refrigerant.

(2) Heat and mass balance model

Heat and mass balance in the condenser is expressed by the following equation.

$$\frac{d}{dt}(W^g_{cond}h^g_{cond} + W^l_{cond}h^l_{cond}) = h_d G_d - ((1-xc)h^l_{cond} + xc \cdot h^g_{cond})G_{cond} - \alpha_{cond}S_{cond}r_2(T_{cond} - T_{ctube}) \tag{25}$$

where left hand terms means the derivative of refrigerant energy in the tube, and right hand terms express the energy balance of the condenser. In the sub-cool region, the tube temperature changes along its location x.

$$G_d \cdot C_p \frac{dT}{dx} = -\alpha_l \cdot S_{cond}(T - T^l_{ctube}) \tag{26}$$

The Mass balance equation is described as the following.

$$\frac{d}{dt}(W^g_{cond} + W^l_{cond}) = G_d - G_{cond} \tag{27}$$

Finally, the condenser tube energy balance is expressed as the next equation.

$$C_{ctube}\frac{dT_{ctube}}{dt} = \alpha_{cond}S_{cond}(T_{cond} - T_{ctube}) - Q_{air} \tag{28}$$

where Q_{air} means heat transfer rate from the tube to the air or the water.

In the expansion valve, it is assumed that enthalpy of the refrigerant does not change. In the simulator, empirical valve lift-flow rate-pressure drop relation is set.

2.2.3 Simulation Example

The above models are combined and programmed in FORTRAN as a type of HVACSIM$^+$(J). The following figures show a simulation example when compressor speed changes from 28Hz to 48Hz at 45min. The chiller is for a housing use. In the figure, "…exp" means experimental data. The simulation results and experimental results show good correspondences after some parameter adjusting.

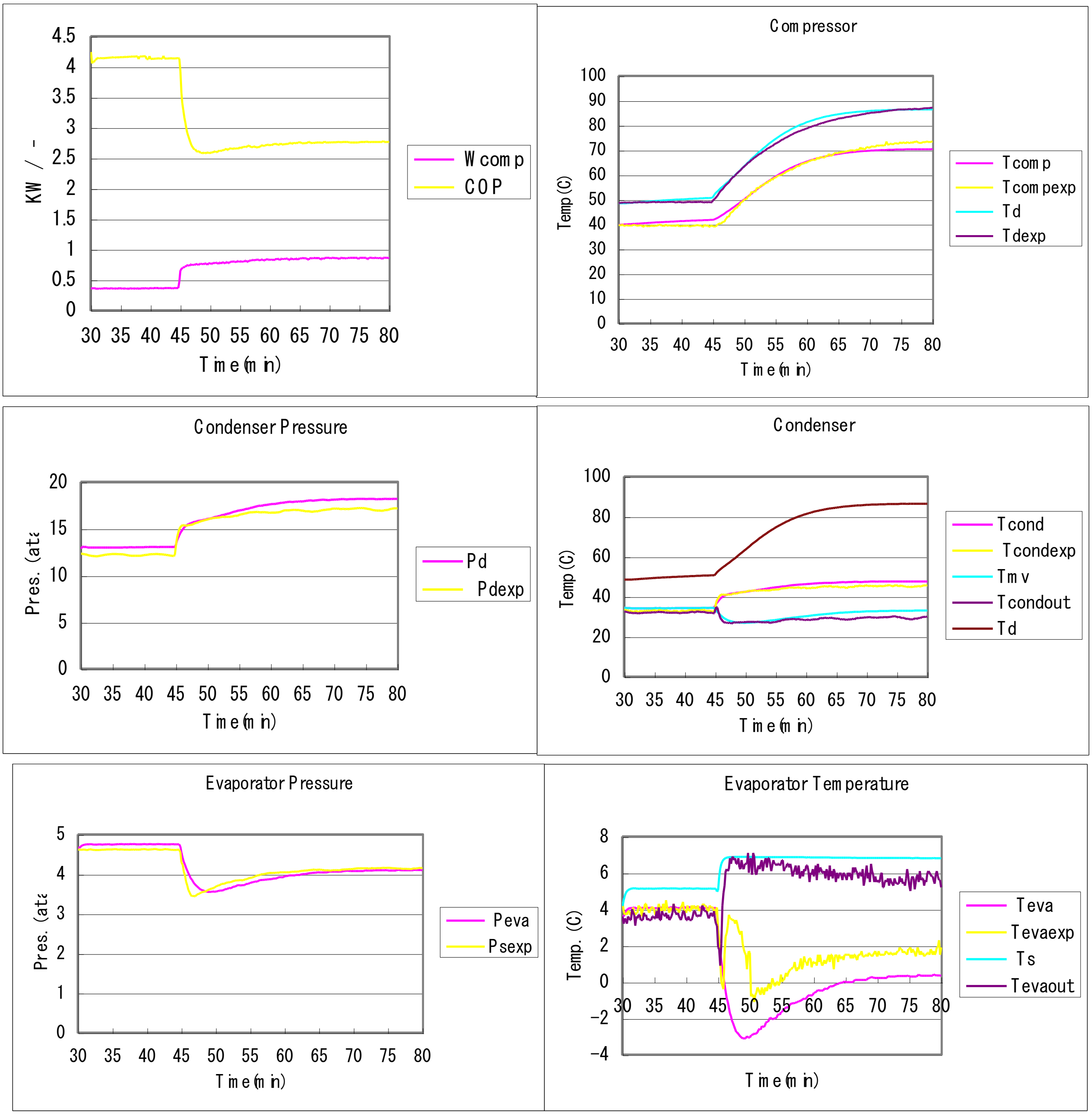


Fig.6 A simulation example for reciprocal chiller

2.3 Turbo Chiller Model

The models of evaporator, condenser and capillary tube for the turbo chillers are the same as the reciprocal compressor type chillers mentioned above. Only difference exists in characteristics of the compressor. The characteristics of the turbo-compressor are described by following well-known equations.

$$Gs' = TurboG(\frac{Pd}{Ps}, n_c, \theta) \quad \text{(Normalized flow rate)}$$

$$\eta = TurboE(\frac{Pd}{Ps}, n_c, \theta) \quad \text{(Efficiency)}$$

where, $n_c = \sqrt{T_0 / T_1} \cdot n$ and $Gs' = \frac{P_0}{P_1}\sqrt{\frac{T_1}{T_0}} \cdot Gs$.

An example of the characteristics is depicted as shown in Fig.7

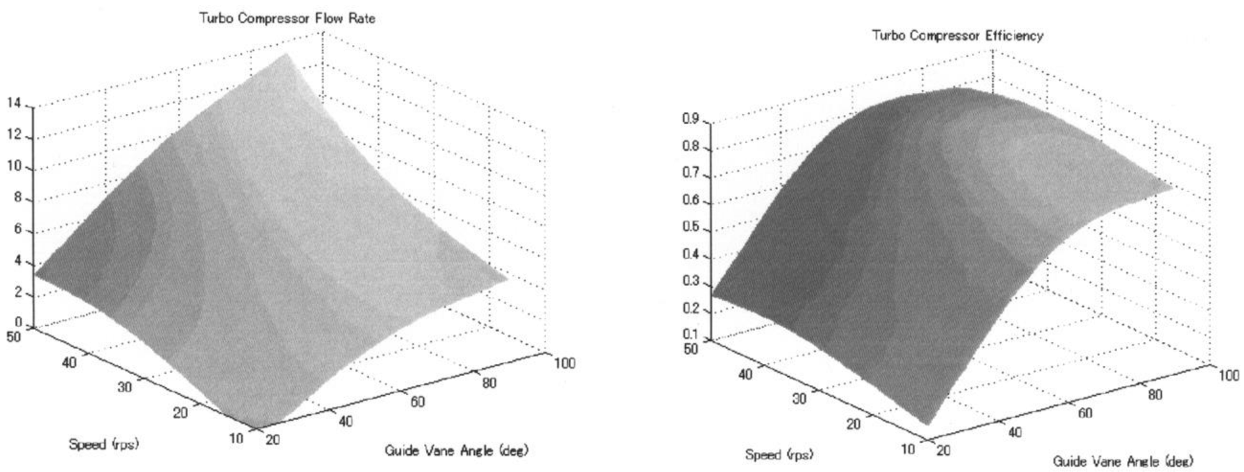


Fig.7 An example of turbo chiller characteristic chart

3. Discussion

Dynamic simulation models of various kinds of chillers, that is, the absorption chiller, the reciprocal chiller and the turbo chiller, have been presently developed. "Types", or component models, for HVACSIM$^+$(J), Japanese version of HVACSIM$^+$, having then been prepared and some of the simulation results were compared with the experimental data. It has been proved that those simulation models fairly reproduce actual dynamic behavior of chillers. Results also proved that the dynamic performance during transient conditions such as start-up, shut-down and rapid change of chiller loads can be precisely evaluated. Partial load characteristics and performance change of the chillers due to the change of external conditions such as outside air temperatures and chiller outlet water temperature can also be evaluated.

This achievement by authors is expected to favor promoting the following kinds of energy and environmental issues of building systems.

a) Correct estimation of energy consumption at the transient conditions of chillers that have been usually ignored, especially of the absorption chiller, will contribute to precisely evaluate a long-range energy performance of them.

b) Seasonal performance of the chiller / heat pump systems, which are affected by cooling and heating load changing pattern, the outside air conditions and the chiller outlet water temperature that should be seasonally reset for optimal operation, could be provided with sufficient precision. This will encourage HVAC designers as well as manufacturers to contribute for establishing real energy efficient systems.

c) Sensitivity analysis will be performed by simulations so that which part of the system should be modified in order to raise coefficient of performance effectively with minimum cost. The heat and power, or the cogeneration, system would be more precisely evaluated when combined with dynamic gas turbine or engine models.

d) The HVACSIM$^+$ that had previously suffered the lack of dynamic models of chillers and subsystems of various types has now gained strong tools to evaluate energy efficiency of the total HVAC system. The HVACSIM$^+$ (J) is also equipped with dynamic water and ice thermal storage models to enable optimization of HVAC energy plant.

e) It is further expected that the present achievement would lead to develop chiller control optimization, optimal tuning of control parameters and fault detection and diagnosis system of chillers and subsystems.

However, developed chiller models are not always fully justified by the experimental data as well as actual operational data in the building systems. Also, details of simulation parameters as described in a series of equations in the present paper should be given by manufacturers, which might be the critical path for now but would be solved in the near future when the building energy consumption became the critical issue for energy conservation and global environment.

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