# DEVELOPMENT OF ENERGY CALCULATION MODEL OF MULTI-SPLIT TYPE AIR-CONDITIONING SYSTEM

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**Abstract:** An energy calculation model for multi-split type air-conditioning system was developed. A compressor, evaporator and condenser were individually modeled to improve flexibility, since combinatorial analysis is indispensable for multi-split type air-conditioning system. With giving a specific example, procedure to estimate the parameters from the manufacturer's documentation is described. To figure out the characteristics of the multi-split type air conditioning system, sensitivity analysis was performed.

**Key Words:** ***physical model, multi-split type air conditioning system***

## 1 INTRODUCTION

Recently, a great number of commercial buildings in Japan introduce all-in-one air conditioning system on a large scale. Some data shows that about 53.5% (21,221 thousand USRT in the 39,667 thousand USRT) of air-conditioning system is a type of all-in-one air conditioning system at commercial buildings in Japan. Most of them are the multi-split type air conditioning system, which is constructed of outdoor-air processing unit and more than one indoor unit. It is difficult to show the performance of multi-split type air conditioning system with characteristic formula, because characteristics of system alternate with the change of combination of outdoor-air processing unit and indoor units. Therefore optimization of multi-split type air conditioning system is difficult.
In this paper, energy calculation model of multi-split type air-conditioning system is developed to figure out the characteristics of the system. Most of parameters which is necessary to execute model is estimated from the manufacturer's documentation.

## 2 PHYSICAL MODEL OF REFRIGERATION CYCLE

The refrigeration cycle consists of some smaller components. In this study, the model of a compressor, an evaporator, and a condenser are all combined to calculate the refrigeration cycle. A brief calculation flow of each component is described below.

### 2.1 Modeling of compressor

A compression work of the compressor can be generally expressed as a function of inlet and outlet pressures as shown in equation (1). A flow rate of the refrigerant is a function of a density, and it is calculated by solving an equation of state (EOS).

## 2.2 Modeling of evaporator and condenser

In the evaporator, by drawing heat from an ambient air, the two-phase refrigerant could change its state to super heated. Equation (5) shows the heat balance in two-phase region, where $\Delta\theta$ is a logarithmic mean temperature difference (LMTD) expressed in equation (7). Similarly, the heat balance in the super heated region can be expressed in equation (6). Temperatures necessary to calculate the LMTD could be calculated by solving the EOS. Figure 1 shows calculation flow chart of evaporator.
The model of the condenser is almost as same as that of the evaporator.

$$W = \eta_v Vol \frac{pol-1}{pol} P_{in} \left\{ \left( \frac{P_{out}}{P_{in}} \right)^{(pol-1)/pol} - 1 \right\} \quad (1)$$

$$W = G \Delta H_{cmp} \quad (2)$$

$$G = \eta_v \cdot Vol \cdot \rho_{in} \quad (3)$$

$$\rho_{in} = f_{EOS}(P_{in}, H_{in}) \quad (4)$$

$$\begin{aligned} Q_{sh} &= A R_{sh} K_{sh} \Delta\theta_{sh} \\ &= G_{ref}\left(H_{ref,out} - H_{ref,vs}\right) \\ &= G_{air}\left(H_{air,in} - H_{air,sh,out}\right) \end{aligned} \quad (5)$$

$$\begin{aligned} Q_{tp} &= A R_{tp} K_{tp} \Delta\theta_{tp} \\ &= G_{ref}\left(H_{ref,vs} - H_{ref,in}\right) \\ &= G_{air}\left(H_{air,in} - H_{air,tp,out}\right) \end{aligned} \quad (6)$$

$$\Delta\theta = \frac{(T_{1in} - T_{2in}) - (T_{1out} - T_{2out})}{\ln\left((T_{1in} - T_{2in})/(T_{1out} - T_{2out})\right)} \quad (7)$$

$$1.0 = R_{sh} + R_{tp} \quad (8)$$

$$T = f_{EOS}(P, H) \quad (9)$$

$$T_{vs} = f_{EOS}(P_{vs}) \quad (10)$$

*W* : Compression work [kW], $\eta_v$ : Volumetric efficiency [-]
*Vol* : Volume [m$^3$], *pol* : Polytropic index [-]
*P* : Pressure [kPa], *G* : Flow rate [kg/s], $\rho$ : Density [kg/m$^3$]
*H* : Enthalpy [kJ/kg], *A* : Heat transfer area [m$^2$]
*T* : Temperature [K] *R* : Rate of Heat transfer area [-]
*K* : Overall heat transfer coefficient [kW/m$^2$-K]
$\Delta\theta$ : Logarithmic mean temperature difference [K]
$f_{EOS}$ : Equation of state (EOS)

Subscripts:
*ref* : refrigerant, *in* : inlet state, *out* : outlet state
*sh* : super heated, *tp* : two-phase, *sc* : sub cooled
*vs* : saturated vapor state

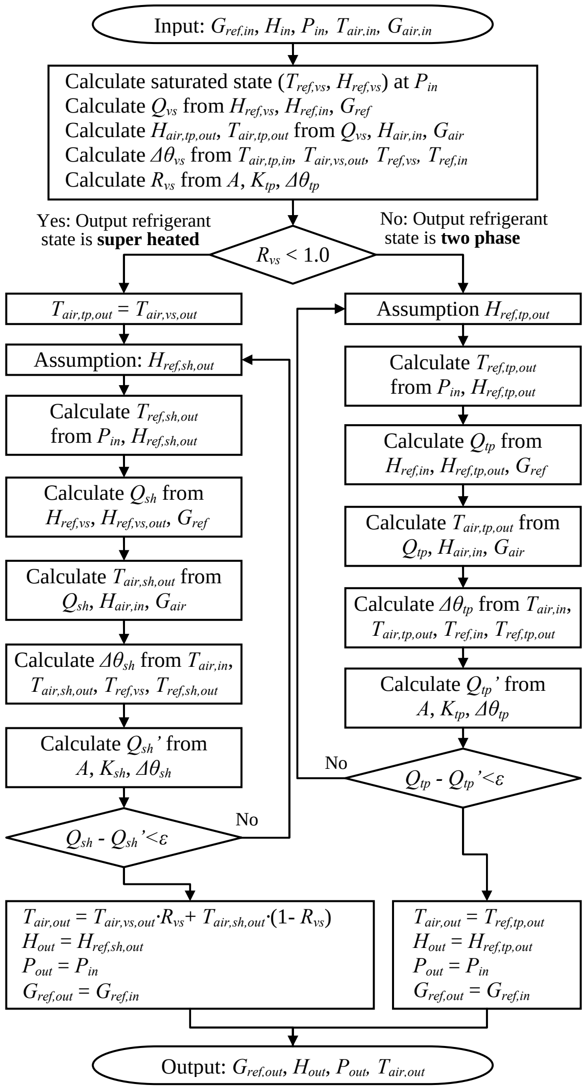


**Figure 1: Calculation flow chart of the evaporator**

## 2.3 Modeling of refrigeration cycle

With using the compressor, the evaporator, and the condenser model, refrigeration cycle model was developed.

### 2.3.1 Assumptions

The model is based on 5 assumptions described below.

1. Evaporating temperature is adjusted as high as possible within the limits of the heat rejection at evaporator.
2. Outlet pressure of compressor is adjusted as low as possible within the limits of the heat rejection at condenser.
3. Super heat degree is controlled at certain value with air flow at evaporator.
4. Refrigerant outlet enthalpy at condenser is controlled same value as inlet enthalpy at evaporator with air flow at condenser.
5. Revolution speed of compressor is controlled as low as possible within the limits of the heat rejection at condenser.

### 2.3.2 Input-output structure

Table 1 shows inputs, outputs and physical parameters of refrigerant cycle model.

**Table 1; Inputs, outputs and physical parameters of refrigerant cycle model**

| | |
|---|---|
| **Inputs** | Outdoor air condition (Dry bulb temperature [°C], Absolute humidity [kg/kg])<br>Room air condition (Dry bulb temperature [°C], Absolute humidity [kg/kg])<br>Heat load at room [kW] |
| **Parameters** | Heat transfer area [$m^2$] and Overall heat transfer coefficients [$kW/m^2$-K]<br>Volume of piston compression [$m^3$], Super heat degree [°C]<br>Upper and lower limit of compressor outlet pressure [kPa]<br>Upper and lower limit of evaporating temperature [°C]<br>Upper and lower limit of compressor revolution speed [rps]<br>Upper limit of air flow rate of HEX [kg/s] |
| **Outputs** | Compression work [kW]<br>Electric consumption of HEX fan [kW]<br>High and low pressure [kPa]<br>Revolution speed of compressor [rps] |

### 2.3.3 Calculation flow

Figure 2 shows a calculation flow chart of the refrigeration cycle. Both evaporating temperature and compressor outlet pressure are calculated iteratively.

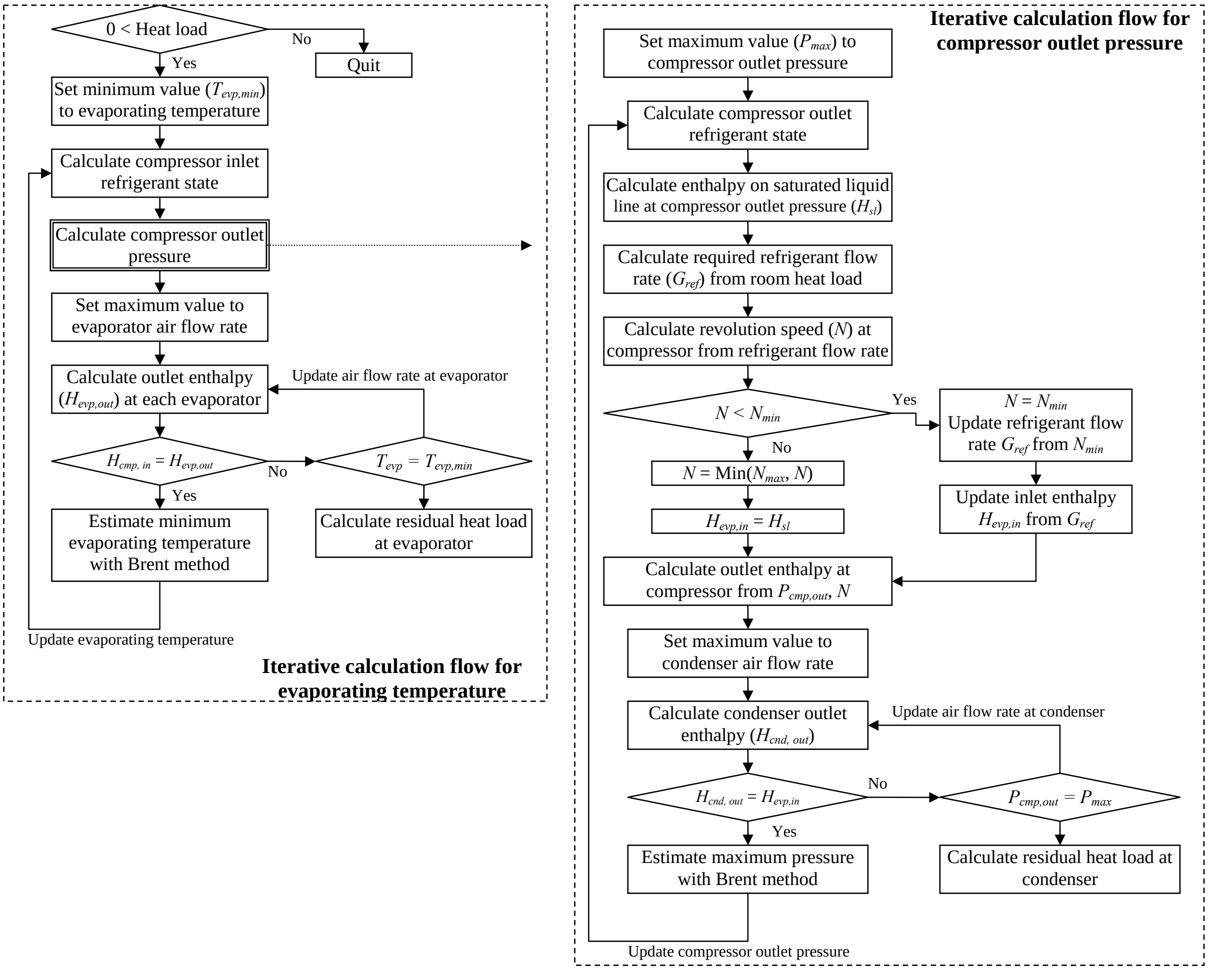


**Figure 2: Calculation flow chart of the refrigeration cycle**

## 3 PROCEDURE TO ESTIMATE THE PHYSICAL PARAMETERS

As described above, the model of refrigeration cycle needs a lot of parameters to execute simulation. Although some of parameters can be obtained from manufacturer's web site, it's not enough to run the model. In this section, with giving a specific example, procedure to estimate the parameters is described. Figure 3 shows an example of multi-split type air-conditioning system to estimate the parameters. 6 indoor units are connected to outdoor-air processing unit.

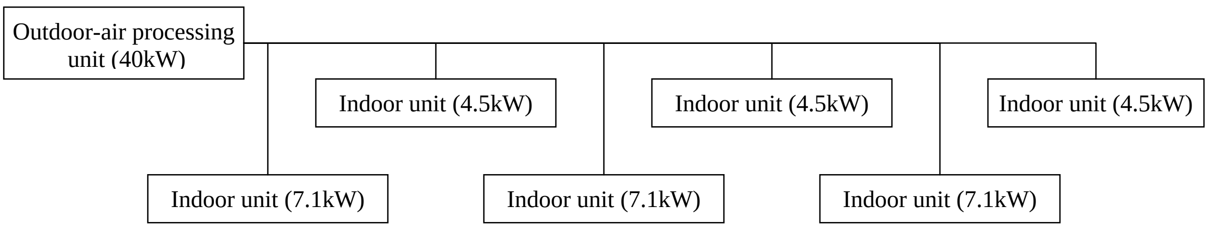


**Figure 3: Sample of multi-split type air-conditioning system to estimate the parameters**

### 3.1 Available information

The values of some physical parameters are described at documentation which can be downloaded for free from manufacturer's web site. Table 2 shows an example of the information obtained from the web site.

**Table 2; Information of refrigeration components obtained from manufacturer's website**

<table>
<tr><th></th><th>Model</th><th>Component</th><th colspan="2">Nominal performance</th></tr>
<tr><td rowspan="6">Outdoor-air processing unit</td><td rowspan="6">RSEYP 400M</td><td rowspan="4">Compressor</td><td>Capacity</td><td>40 [kW]</td></tr>
<tr><td>Electric consumption</td><td>2.0 / 4.5 / 4.5 [kW]</td></tr>
<tr><td>Volume of piston compression</td><td>13.72 / 10.47 / 10.47 [m$^3$/h]</td></tr>
<tr><td>Revolution speed</td><td>6480 / 2900 / 2900 [rps]</td></tr>
<tr><td rowspan="2">Condenser</td><td>Fan air flow rate</td><td>210 [m$^3$/min]</td></tr>
<tr><td>Fan power</td><td>0.75 [kW]</td></tr>
<tr><td rowspan="6">Indoor unit</td><td rowspan="3">FXYSP 45M</td><td rowspan="3">Evaporator</td><td>Fan air flow rate</td><td>11.5 [m$^3$/min]</td></tr>
<tr><td>Fan power</td><td>0.065 [kW]</td></tr>
<tr><td>Capacity</td><td>4.5 [kW]</td></tr>
<tr><td rowspan="3">FXYSP 71M</td><td rowspan="3">Evaporator</td><td>Fan air flow rate</td><td>21 [m$^3$/min]</td></tr>
<tr><td>Fan power</td><td>0.125 [kW]</td></tr>
<tr><td>Capacity</td><td>7.1 [kW]</td></tr>
<tr><td colspan="4">Refrigerant kind</td><td>R410A</td></tr>
</table>

### 3.2 Estimating parameters of compressor

At the normal operating state, in refrigeration cycle of R410A, evaporating temperature is controlled to -5~5 [°C] and superheat degree takes value around 5 [°C]. Assuming that evaporating temperature as 0 [°C] and superheat degree as 5 [°C], inlet refrigerant state of compressor is determined uniquely. In this case, enthalpy, pressure, and density becomes 426.81 [kJ/kg], 798 [kPa], and 29.55 [kg/m$^3$]. Figure 4 shows the determination of compressor inlet refrigerant state at nominal state.
Refrigerant flow rate can be calculated from compression volume and density of compressor inlet refrigerant. Assuming that volumetric efficiency is 0.95, (13.72 + 10.47 + 10.47) [m$^3$/h] / 3600 x 29.55 [kg/m$^3$] x 0.95 = 0.27 [kg/s] is a flow rate of refrigerant in this case.
From nominal capacity and refrigerant flow rate, inlet enthalpy of evaporator can be determined uniquely. In this case, 426.81 [kJ/kg] - 40 [kW] / 0.27 [kg/s] = 278.7 [kJ/kg].
Saturated liquid line and vertical line at evaporator inlet enthalpy intersect at one point, which shows outlet refrigerant state of condenser, 278.7 [kJ/kg] and 2830 [kPa] in this case.
Compression work can be calculated from pressure difference at compressor (Equation 1).
Using ratio of specific heat capacities in place of polytropic index, compression work is calculated to be 10.9 [kW]. Additionally, increased enthalpy at compressor can be calculated from compression work and refrigerant flow rate (Equation 2). In this case, 426.81 [kJ/kg] + 10.9 [kW] / 0.27 [kg/s] = 467.18 [kJ/kg] is outlet enthalpy at compressor.

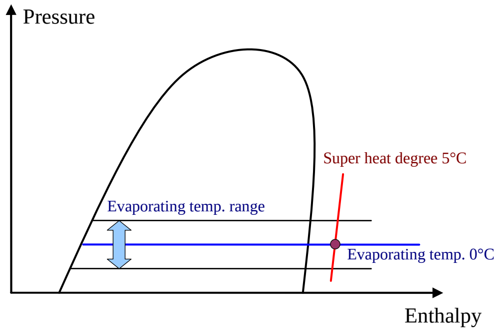


**Figure 4: Determination of compressor inlet refrigerant state at nominal condition**

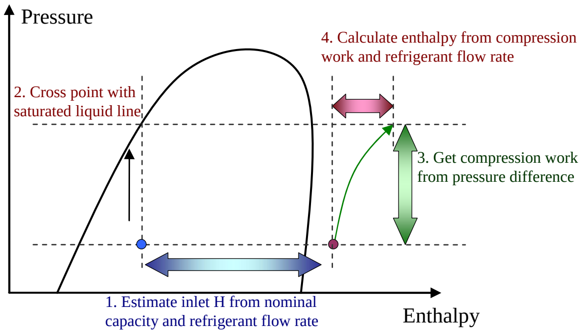


**Figure 5: Determination of HEX inlet and outlet refrigerant state at nominal condition**

### 3.3 Estimating parameters of condenser

As inlet and outlet state and flow rate of refrigerant is identified at foregoing section, heat exchanger duty at condenser can be calculated analytically. Assuming that $K_{sp}$ equals $K_{tp}$, heat transfer area can be determined uniquely with solving equation (5), (6), and (7). From Japanese Industrial Standards (JIS), out door air dry bulb temperature to test nominal performance is 35.0 [°C]. The value of air flow rate is already obtained at section 3.1. In this case, heat transfer area is calculated to be 91.8 [m$^2$] when $K_{sp}$ = $K_{tp}$ = 0.045 [kW/K-m$^2$].

### 3.4 Estimating parameters of evaporator

Heat exchanger duty at evaporator can be obtained as the same way as condenser. However, refrigerant flow rate is determined as a function of ratio of nominal power of out door air processing unit to that of indoor unit. In the case of FXYSP71M, refrigerant flow rate is calculated to be 0.0479 [kg/s] = 0.27 [kg/s] x (7.1 [kW] / 40 [kW]). From Japanese Industrial Standards (JIS), indoor air state to test nominal performance is 27.0 [°CDB] and 19 [°CWB]. In this case, heat transfer area of FXYSP71M and FXYSP45M is calculated to be 8.2 [m$^2$] and 5.4 [m$^2$], respectively when $K_{sp}$ = $K_{tp}$ = 0.045 [kW/K-m$^2$].

## 4 SENSITIVITY ANALYSIS

Using model built at foregoing section, sensitivity analysis was performed to figure out the characteristics of multi-split type air conditioning system.

## 4.1 Response characteristics to changes of outdoor air state

Table 3 shows the response characteristics to changes of outdoor air state. High pressure drops down since heat exchange performance increases with decline of temperature. Because heat load to solve is constant, as pressure difference at compressor and compression work decreases, COP rises from 3.4 to 3.9. Figure 6 shows system COP against out door dry bulb temperature.

**Table 3; Response characteristics to changes of outdoor air state**

| | Outdoor DB temperature | 35 | 34 | 33 | 32 | 31 |
|---|---|---|---|---|---|---|
| RSEYP400M | Outdoor absolute humidity | 0.027 | 0.027 | 0.027 | 0.027 | 0.027 |
| | Indoor heat load | 34.8 | 34.8 | 34.8 | 34.8 | 34.8 |
| | Evaporating temperature | 0.00 | 0.00 | 0.00 | 0.00 | 0.00 |
| | System COP | 2.91 | 2.99 | 3.08 | 3.17 | 3.26 |
| | COP (no fans electric consumption) | 3.40 | 3.52 | 3.64 | 3.77 | 3.91 |
| | Compression work | 10.23 | 9.89 | 9.55 | 9.23 | 8.91 |
| | Revolution speed | 96.6 | 95.4 | 94.1 | 92.9 | 91.8 |
| | Low pressure | 798.3 | 798.1 | 798.0 | 797.9 | 797.8 |
| | High pressure | 2979.0 | 2912.1 | 2844.2 | 2778.5 | 2713.6 |
| | Refrigerant flow rate | 0.242 | 0.238 | 0.235 | 0.232 | 0.229 |
| | Compressor inlet enthalpy | 428.2 | 428.2 | 428.2 | 428.2 | 428.2 |
| | Compressor outlet enthalpy | 470.6 | 469.7 | 468.8 | 468.0 | 467.1 |
| | Condenser outlet DB temperature | 45.7 | 44.6 | 43.5 | 42.4 | 41.3 |
| | Condenser outlet absolute humidity | 0.027 | 0.027 | 0.027 | 0.027 | 0.027 |
| | condenser outlet enthalpy | 284.2 | 282.3 | 280.3 | 278.4 | 276.5 |
| | heat exchange | 45.0 | 44.7 | 44.3 | 44.0 | 43.7 |
| FXYSP71M | Indoor DB temperature | 27 | 27 | 27 | 27 | 27 |
| | Indoor absolute humidity | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| | Air flow rate | 0.42 | 0.42 | 0.42 | 0.42 | 0.42 |
| | Fan power | 0.206 | 0.206 | 0.206 | 0.206 | 0.206 |
| | Outlet air DB temperature | 12.7 | 12.7 | 12.7 | 12.7 | 12.7 |
| | Outlet air absolute humidity | 0.009 | 0.009 | 0.009 | 0.009 | 0.009 |
| | heat load | 7.1 | 7.1 | 7.1 | 7.1 | 7.1 |
| FXYSP45M | Indoor DB temperature | 27 | 27 | 27 | 27 | 27 |
| | Indoor absolute humidity | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| | Air flow rate | 0.23 | 0.23 | 0.23 | 0.23 | 0.23 |
| | Fan power | 0.127 | 0.127 | 0.127 | 0.127 | 0.127 |
| | Outlet air DB temperature | 11.6 | 11.6 | 11.6 | 11.6 | 11.6 |
| | Outlet air absolute humidity | 0.008 | 0.008 | 0.008 | 0.008 | 0.008 |
| | heat load | 4.5 | 4.5 | 4.5 | 4.5 | 4.5 |

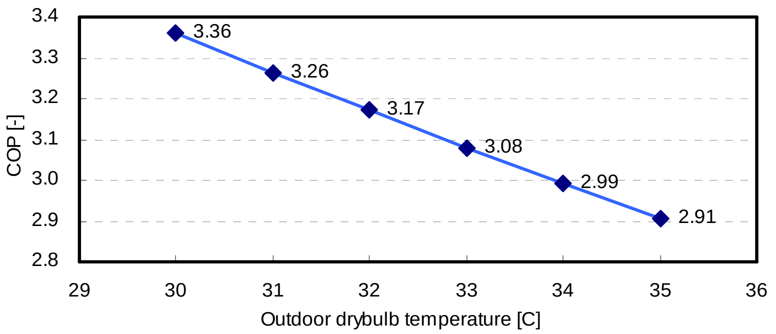


**Figure 6: System COP against out door dry bulb temperature**

## 4.2 Response characteristics to changes of evaporating temperature

Table 4 shows the response characteristics to changes of evaporating temperature. Although developed model can automatically adjusts its evaporating temperature, evaporating temperature is fixed to certain value in this case study.
Since increase of evaporating temperature directly linked to increase of low pressure, pressure difference at compressor decreases. As a result, COP rises from 3.3 to 3.8 with evaporating temperature increases from -5.0 to 1.0. Figure 7 shows system COP against evaporating temperature.

**Table 4; Response characteristics to changes of evaporating temperature**

| | | | | | |
|---|---|---|---|---|---|
| RSEYP400M | Outdoor DB temperature | 30 | 30 | 30 | 30 |
| | Outdoor absolute humidity | 0.027 | 0.027 | 0.027 | 0.027 |
| | Indoor heat load | 27.84 | 27.84 | 27.84 | 27.84 |
| | Evaporating temperature | -5.00 | -3.00 | -1.00 | 1.00 |
| | System COP | 3.29 | 3.47 | 3.65 | 3.83 |
| | COP (no fans electric consumption) | 3.77 | 4.03 | 4.32 | 4.64 |
| | Compression work | 7.39 | 6.91 | 6.44 | 6.00 |
| | Revolution speed | 83.6 | 77.8 | 72.5 | 67.7 |
| | Low pressure | 678.3 | 724.4 | 772.9 | 823.8 |
| | High pressure | 2485.7 | 2481.6 | 2478.6 | 2474.5 |
| | Refrigerant flow rate | 0.177 | 0.176 | 0.176 | 0.175 |
| | Compressor inlet enthalpy | 426.5 | 427.2 | 427.9 | 428.5 |
| | Compressor outlet enthalpy | 468.2 | 466.4 | 464.6 | 462.9 |
| | Condenser outlet DB temperature | 38.3 | 38.2 | 38.1 | 38.0 |
| | Condenser outlet absolute humidity | 0.027 | 0.027 | 0.027 | 0.027 |
| | condenser outlet enthalpy | 269.6 | 269.4 | 269.3 | 269.2 |
| | heat exchange | 35.2 | 34.7 | 34.3 | 33.8 |
| FXYSP71M | Indoor DB temperature | 27 | 27 | 27 | 27 |
| | Indoor absolute humidity | 0.010 | 0.010 | 0.010 | 0.010 |
| | Air flow rate | 0.13 | 0.15 | 0.17 | 0.21 |
| | Fan power | 0.064 | 0.072 | 0.084 | 0.104 |
| | Outlet air DB temperature | -0.3 | 2.5 | 5.5 | 8.5 |
| | Outlet air absolute humidity | 0.004 | 0.005 | 0.006 | 0.007 |
| | heat load | 5.68 | 5.68 | 5.68 | 5.68 |
| FXYSP45M | Indoor DB temperature | 27 | 27 | 27 | 27 |
| | Indoor absolute humidity | 0.010 | 0.010 | 0.010 | 0.010 |
| | Air flow rate | 0.08 | 0.09 | 0.10 | 0.13 |
| | Fan power | 0.044 | 0.050 | 0.058 | 0.070 |
| | Outlet air DB temperature | -0.9 | 1.9 | 4.8 | 7.8 |
| | Outlet air absolute humidity | 0.004 | 0.004 | 0.005 | 0.007 |
| | heat load | 3.6 | 3.6 | 3.6 | 3.6 |

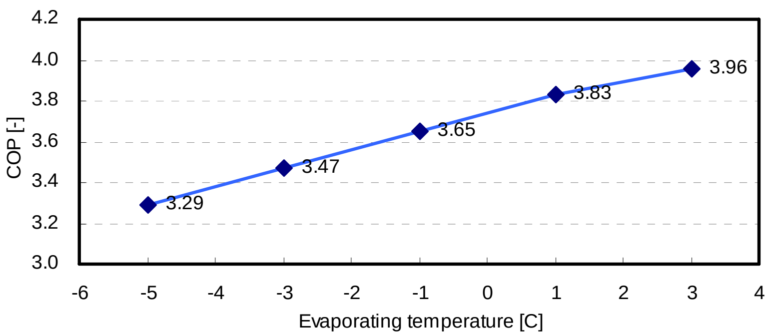


**Figure 7: System COP against evaporating temperature**

## 4.3 Response characteristics to changes of indoor heat load

Table 5 shows the response characteristics to changes of indoor heat load. With decreasing of indoor heat load, evaporating temperature and revolution speed of compressor increase and decrease respectively. Finally, evaporating temperature reach to upper limit (5 [°C] in this case) and revolution speed reach to lower limit (50 [rps] in this case). Keeping with this trend, COP curve shows single-peaked pattern. Figure 8 shows system COP against indoor heat load.

**Table 5; Response characteristics to changes of indoor heat load**

| | | | | | | |
|---|---|---|---|---|---|---|
| RSEYP400M | Outdoor DB temperature | 35 | 35 | 35 | 35 | 35 |
| | Outdoor absolute humidity | 0.027 | 0.027 | 0.027 | 0.027 | 0.027 |
| | Indoor heat load | 34.8 | 30 | 25.2 | 20.4 | 15.6 |
| | Evaporating temperature | 0.0 | 2.5 | 5.0 | 5.0 | 5.0 |
| | System COP | 2.91 | 3.25 | 3.60 | 3.51 | 2.88 |
| | COP (no fans electric consumption) | 3.40 | 3.96 | 4.63 | 4.42 | 3.55 |
| | Compression work | 10.23 | 7.58 | 5.44 | 4.62 | 4.39 |
| | Revolution speed | 96.6 | 74.5 | 56.1 | 50.0 | 50.0 |
| | Low pressure | 798.3 | 863.5 | 933.2 | 933.2 | 933.2 |
| | High pressure | 2979.0 | 2841.9 | 2711.7 | 2593.8 | 2481.3 |
| | Refrigerant flow rate | 0.242 | 0.202 | 0.164 | 0.146 | 0.146 |
| | Compressor inlet enthalpy | 428.2 | 429.0 | 429.7 | 429.7 | 429.7 |
| | Compressor outlet enthalpy | 470.6 | 466.6 | 462.8 | 461.2 | 459.7 |
| | Condenser outlet DB temperature | 45.7 | 43.9 | 42.3 | 40.9 | 39.7 |
| | Condenser outlet absolute humidity | 0.027 | 0.027 | 0.027 | 0.027 | 0.027 |
| | condenser outlet enthalpy | 284.2 | 280.2 | 276.4 | 290.4 | 323.2 |
| | heat exchange | 45.0 | 37.6 | 30.6 | 25.0 | 20.0 |
| FXYSP71M | Indoor DB temperature | 27 | 27 | 27 | 27 | 27 |
| | Indoor absolute humidity | 0.010 | 0.010 | 0.010 | 0.010 | 0.010 |
| | Air flow rate | 0.42 | 0.42 | 0.41 | 0.22 | 0.14 |
| | Fan power | 0.206 | 0.206 | 0.200 | 0.109 | 0.069 |
| | Outlet air DB temperature | 12.7 | 13.4 | 14.0 | 10.9 | 8.1 |
| | Outlet air absolute humidity | 0.009 | 0.010 | 0.010 | 0.008 | 0.007 |
| | heat load | 7.1 | 6.3 | 5.5 | 4.7 | 3.9 |
| FXYSP45M | Indoor DB temperature | 27 | 27 | 27 | 27 | 27 |
| | Indoor absolute humidity | 0.010 | 0.010 | 0.010 | 0.010 | 0.010 |
| | Air flow rate | 0.23 | 0.17 | 0.12 | 0.07 | 0.04 |
| | Fan power | 0.127 | 0.093 | 0.069 | 0.037 | 0.020 |
| | Outlet air DB temperature | 11.6 | 10.6 | 10.0 | 6.5 | 4.4 |
| | Outlet air absolute humidity | 0.008 | 0.008 | 0.008 | 0.006 | 0.005 |
| | heat load | 4.5 | 3.7 | 2.9 | 2.1 | 1.3 |

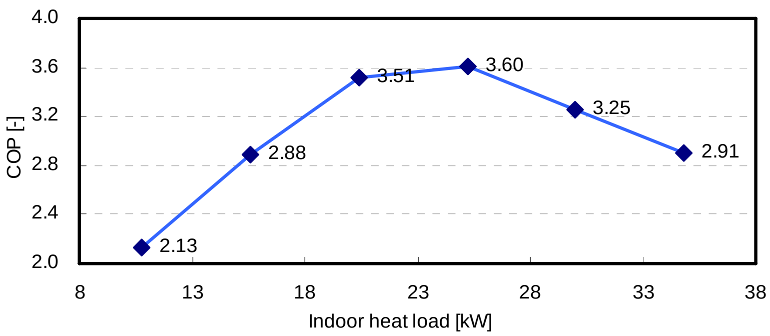


**Figure 8: System COP against indoor heat load**

## 5 CONCLUSION

In this paper, energy calculation model of multi-split type air-conditioning system was developed. With giving a specific example, procedure to estimate the parameters from the manufacturer's documentation was described. To figure out the characteristics of the multi-split type air conditioning system, sensitivity analysis was performed. Theoretical relationship between COP and outdoor dry bulb temperature, evaporating temperature, and indoor heat load was show diagrammatically.

## 6 ACKNOWLEDGMENT

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