HASP/ACSS: SIMULATION PROGRAM FOR ENERGY CONSUMPTION OF AIR CONDITIONING SYSTEMS

Tatsuo Inooka*, Yo Matsuo**, Koichi Yokoyama***

* Nikken Sekkei Ltd, Planners/Architects/Engineers
** The University of Tokyo
*** Shin-Nippon Air Conditioning Engineering Co., Ltd.

This simulation program HASP/ACSS has been newly developed to compute yearly energy consumption of air conditioning systems. Various types of systems, equipment, controlling methods, etc. can be evaluated by solving heat balance equations of the whole system hour by hour. Solving process, mathematical models of equipment, computable systems, computation examples, etc. are described.

INTRODUCTION

Development of HASP/ACSS was started by the Committee for Developing the Standard Simulation Program for Air Conditioning Systems (Chairman: Prof. Yo Matsuo), the Japan Building Mechanical Engineers Association and has been completed in June 1985.

This program forms the enclosed portion in Fig. 1. The simulation of air conditioning systems is carried out by the use of thermal loads, meteorological data, equipment specification, and operation and control data. Of them, the thermal loads are separately computed by a dynamic thermal load computation program in advance.

The program is composed of two main parts: the part of data input and check and the part of computation of energy consumption. A capacity of the program written by FORTRAN-77 is as shown in Table 1.

Table 1 Capacity of HASP/ACSS

	Input/ Check	Computation	Total
No. of Steps	6,500	9,500	16,000
Memory	375 kB	464 kB	651 kB*
(Data domain)	(67 kB)	(122 kB)	(138 kB)*
No. of subroutines	53	97	141*

* include figures common to both the parts.

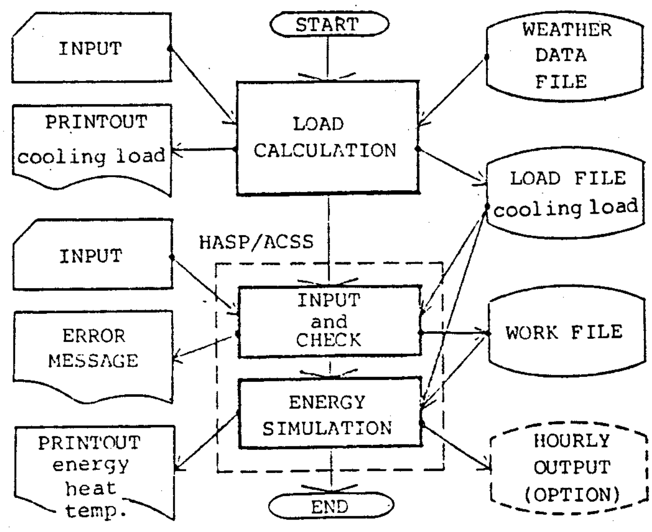


Fig. 1 Job Flow

FUNDAMENTAL CONCEPTS

Energy Simulation

The program HASP/ACSS is purposed to simulate yearly energy consumption of air conditioning systems and, in the course of the simulation, know hourly values of equipment operation and control, room temperature and humidity, and heat extraction from rooms.

In HASP/ACSS, heat balances of all elements of an air conditioning system, such as rooms, air conditioners, room terminal units, refrigerators, boilers, cooling towers and thermal storages, are expressed by means of fundamental equations, which are simultaneously solved so that temperature, humidity, flow rate and calorific values can maintain a balance under hourly loading and operating conditions.

Basic Composition of Subsystems

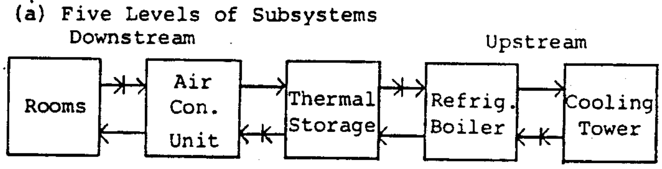
Fundamental equations for an air conditioning system include many unlinear factors, such as performance affected by temperature change and operation discontinuously altered by control. Introducing equations which express all such unlinear factors of all components an air conditioning system complicates the computation too much. Therefore, the subsystems are divided into five levels as shown in Fig. 2 and component models in each level are expressed by individual equations.

Each subsystem takes supply water temperature and flow rate from upper stream and load conditions from lower streams. After computation for processing downstream loads, a subsystem sends flow rate and temperature conditions to lower streams and necessary load conditions to upper stream. This computation starts with the lowest subsystem and proceeds to upper subsystem one after another.

The supply water temperature and flow rate given from upper stream are preset values as assumed initial values. When a subsystem upstream is underloaded, it is not necessary to repeat the computation of the subsystem downstream because the assumed supply water temperature is achieved. However, when the load is excessive, computation of e subsystem downstream has to be performed again because the supply water temperature changes. The solution converges to a certain condition by such iterative process.

Information between subsystems upstream and downstream is transferred through piping systems. The piping systems have high flexibility in connection and, for example, subsystems can be connected as shown in Fig. 2-b.

(a) Five Levels of Subsystems



(b) Example of Connection

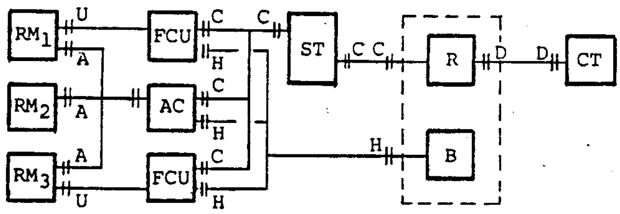


Fig. 2 Composition of System

MATHEMATICAL MODELS OF MAJOR EQUIPMENT

humidifying Coil

Basic Models Chilled water coil requiring dehumidification has to be modelled by simultaneously solving two state equations of temperature and humidity. For this, Inoue and Lee proposed separation of dry coil and wet coil in the computation[(2)] and Matsuo modified their proposal so as to correct outlet conditions during overloading by the use of an equivalent simple coil[(3)]. This Matsuo's model was adopted as the basic model of coils in an air conditioner.

When air state (T_{A1}, X_{A1}, H_{A1}) at the inlet of coil and water temperature, t_{W1}, at that inlet are given, it is solved as follows whether the target temperature, T_{A3}, at the outlet of coil can be reached under air volume G_A, flow rate G_W and coil area S_0. The air at the inlet is a mixture of temperature and humidity, T_{RN}, X_{RN}, of return air (weighting mean of air volume of rooms) and those, T_0, X_0, of outdoor air. Letting volume of outdoor air be G_0,

$$T_{A1} = T_{RN} + (T_0 - T_{RN})G_0/G_A$$
$$X_{A1} = X_{RN} + (X_0 - X_{RN})G_0/G_A \qquad \cdots\cdots\cdots (1)$$

T_{A3} is assumed to be equal to the outlet air temperature necessary for keeping room temperature T_d as preset and controlled at a room k. Letting load demand and air volume of the room k be $Q_{S,k}$ and $G_{R,k}$ respectively,

$$T_{A3} = T_d - \{Q_{S,k} + WZ_{0,k}(T_B - T_d)\}/0.288G_{R,k} \qquad (2)$$

WZ_0: Weighting factor (WF) of change in room temperature; T_B: Datum temperature

Humidity, X_{A3}, of outlet air, enthalpy H_{A3}, and air state (T_{A2}, X_{A2}, H_{A2}) on the boundary of dry coil and wet coil are determined as state points on the psychrometric diagram if relative humidity of air in wet coil is taken as ψ (Fig. 3). Hence, load, Q_D, on dry coil and load, Q_W, on wet coil are:

$$Q_D = 0.288G_A\ (T_{A1} - T_{A2})$$
$$Q_W = 1.2G_A\ (H_{A2} - H_{A3}) \qquad \cdots\cdots\cdots\cdots (3)$$

On the other hand, water temperatures, T_{W2}, on the boundary and, T_{W3}, at the outlet are:

$$T_{W2} = T_{W1} + Q_W/G_W\ , \qquad T_{W3} = T_{W2} + Q_D/G_W \qquad \cdots\cdots (4)$$

Thus, dry coil area S_D and wet coil area S_W necessary to eliminate the load are obtained as follows:

$$S_D = Q_D/\{U_D\cdot\Delta t_e\}$$
$$\Delta t_e = (T_{A1} - T_{W3} - T_{A2} + T_{W2})$$
$$/\log_e\{(T_{A1} - T_{W3})/(T_{A2} - T_{W2})\}$$
$$S_W = Q_W/\{U_W\cdot\Delta h_e\}$$
$$\Delta h_e = (H_{A2} - H_{W2} - H_{A3} + H_{W1})$$
$$/\log_e\{(H_{A2} - H_{W2})/(H_{A3} - H_{W1})\} \qquad \cdots\cdots (5)$$

where, U_D, U_W: Heat transmission coefficient of dry coil and wet coil. If $S_D + S_W > S_0$, the load is excessive and if $S_D + S_W \leq S_0$, the load is within the capacity of an air conditioner.

Where coils in an air conditioner are dealt with as an equivalent simple coil without separation into dry coil and wet coil, the above relationship is as shown in Fig. 4. The heat balance equations in this case are:

$$Q = 1.2G_A\ (H_{A1} - H_{A3})$$
$$= G_W\ (T_{W3} - T_{W1}) = G_W'\ (H_{W3} - H_{W1})$$
$$= U_e\cdot S\ (H_{A1} - H_{W3} - H_{A3} + H_{W1})$$
$$/\log_e\{(H_{A1} - H_{W3})/(H_{A3} - H_{W1})\} \qquad \cdots\cdots (6)$$

U_e: Apparent heat transmission coefficient; G_W': Apparent flow rate

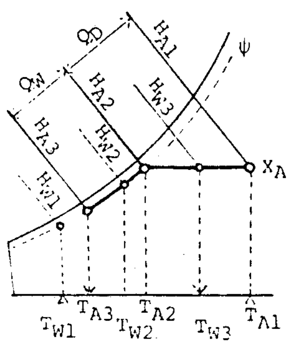
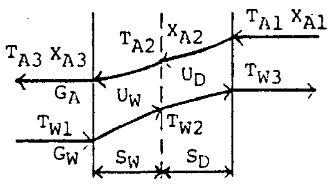


Fig. 3 State Point of Dehumidifying Coil

(a) Dry Coil and Wet Coil



(b) Equivalent Simple Coil

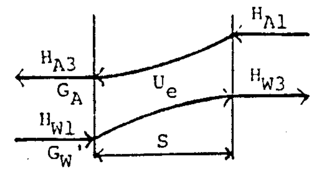


Fig. 4 Model of Dehumidifying Coil

Correction of Temperature and Humidity at the Outlet when overloaded Since the coil area $S_D + S_W$ is reduced to S_0, heat exchanged $Q_D + Q_W$ is decreased to Q^*, and accordingly air state and water temperature at the outlet become T_{A3}^*, X_{A3}^*, H_{A3}^* and T_{W3}^*. If Eq. (6) is simultaneously solved assuming that the air and water state (G_W, G_W', G_A, U_e) at the inlet are unchanged,

$$H_{A3}^* = \left\{ H_{A1}(G_A/G_W' - 1) + H_{W1}(1 - E)\right\} / \left\{G_A/G_W' - E\right\} \quad \ldots\ldots (7)$$

where, $E = \exp\left\{ U_e \cdot S_0 (1/G_A - 1/G_W')\right\}$

T_{A3}^* and X_{A3}^* can be obtained from relative humidity as state points on the psychrometric diagram. The room temperature and humidity are corrected by the outlet air state thus obtained. Temperature and humidity, $T_{RM,i}$, $X_{RM,i}$, at Room i are:

$$T_{RM,i} = (Q_{S,i} + WZ_{0,i} \cdot T_B + 0.288 G_{R,i} \cdot T_{A3}^*) / (WZ_{0,i} + 0.288 G_{R,i})$$

$$X_{RM,i} = (Q_{L,i} + WL_{0,i} \cdot X_B + 727 G_{R,i} \cdot X_{A3}^*) / (WL_{0,i} + 727 G_{R,i}) \quad \ldots\ldots\ldots (8)$$

These $T_{RM,i}$ and $X_{RM,i}$ are reflected to temperature and humidity of return air upon the next iteration.

Correction of Flow Rate when underloaded Where the load is within the capacity of an air conditioner, the target values of air state at the outlet can be attained and the preset room temperature can be maintained. In this connection, if flow rate is variably controlled, flow rate $G_W'^*$ to upper stream and water temperature T_{W3}^* at the outlet vary. Eq. (6) is simultaneously solved with other conditions unchanged. To make explicit solution possible,

$$Z = (H_{A1} - H_{W3}^*)/(H_{A3} - H_{W1})$$

$$a = (Q_D + Q_W)/\left\{U_e \cdot S_0 (H_{A3} - H_{W1})\right\} \quad \ldots\ldots (9)$$

From Eq. (9),

$$a \cdot \log Z = Z - 1 \quad \ldots\ldots\ldots\ldots (10)$$

H_{W3}^* and G_W can be obtained by solving Eq. (10) with the approximate explicit function of Eq. (11).

$$Z = f(a) \doteqdot - 0.23184a + 1.34982a^2 - 0.07011a^3 - 0.05757a^4 \quad \ldots\ldots (11)$$

Thermal Storage

Model of a thermal storage is a well-mixed storage having n pieces of vessels, each V in volume (Fig. 5). If flow rate and temperatures of water from upper and lower streams are W_J, W_K, T_J and T_K, flow direction between vessles is determined by difference between W_J and W_K and flow rate in the storage is $W = |W_J - W_K|$.

In the case of $W_K > W_J$, each of the vessles is expressed by the following differential equations. (Similar equations can be applied also to the case of $W_J > W_K$.) Heat loss from the storage is disregarded in the equations for simplification.

$$V \cdot dT_n/d\tau = W_K\ (T_K - T_n)$$

$$V \cdot dT_i/d\tau = W\ (T_{i+1} - T_i)$$

$$V \cdot dT_1/d\tau = W\ (T_2 - T_1) + W_J\ (T_J - T_1) \quad \ldots\ldots (12)$$

n, i, 2, 1: No. of vessels; T_1, T_2, T_i, T_n: Water temperature of vessels; τ: Time

Eq. (12) is numerically solved for time interval Δt by difference calculus. Since general forward difference can be explicitly solved, equipment on the upper stream can be separated from that on the lower stream in the model. However, n or Δt cannot be freely preset because of the limitation of $\max\{W_J, W_K\} \cdot \Delta t/V \leq 1$ for securing the stability.

On the other hand, although backward difference is stable on any conditiion, equipment on the upper and lower streams has to be wholly dealt with letting T_J and T_K implicit.

In the present program, backward difference scheme is used on the basis of the Matsuo's model(3) in which heat quantities, Q_J, Q_K, from the upper and lower streams are used instead of inflow water temperature. Since values computed for subsystems on the upper and lower streams can be used for Q_J and Q_K, equipment on the upper and lower streams can be dealt with independently of each other. And heat balance can be maintained in the model.

Different from actual temperatures, inflow water temperatures are expressed by adding Q_J and Q_K to temperature of water discharged from a thermal storage.

$$T_K = T_1^t + \Delta T_K, \quad \Delta T_K = Q_K/W_K$$

$$T_J = T_n^t + \Delta T_J, \quad \Delta T_J = Q_J/W_J \quad \ldots\ldots\ldots (13)$$

Thus, the backward difference scheme is:

$$(T_n^t - T_n^{t-\Delta t})/\Delta t = (T_1^t + \Delta T_K - T_n^t) W_K/V$$

$$(T_i^t - T_i^{t-\Delta t})/\Delta t = (T_{i+1}^t - T_i^t) W/V$$

$$(T_1^t - T_1^{t-\Delta t})/\Delta t = \left\{(T_2^t - T_1^t) W + (T_n^t + \Delta T_J - T_1^t) W_J\right\}/V \quad (14)$$

where $\max\{W_J, W_K\} > V$, this model sometimes leads to vessel water temperature deviated from actual inflow water temperature. However, no divergence of solution occurs as is the case with use of forward difference.

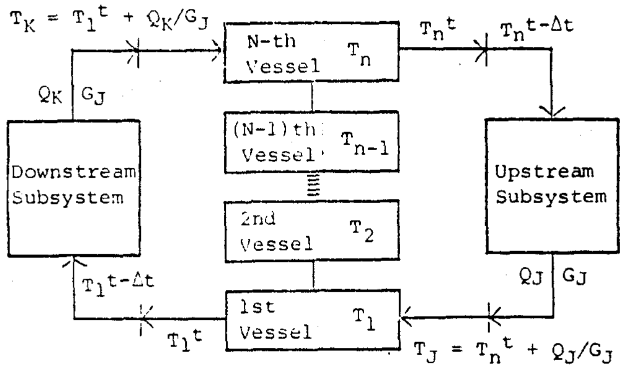


Fig. 5 Model of Thermal Storage

FUNCTIONS OF SUBSYSTEMS

The above-mentioned concept and algorithm are developed into a program applicable to evaluation of various types of systems, equipment, control methods, etc. This section describes in what manner models of individual subsystems are composed, specified and solved.

Air Conditioning Subsystem

Types of Air Conditioners   Air conditioning methods of CAV (left on Fig. 6), VAV, DD and PAC can be computed by this program. FCU, TRH and HPU can be dealt with as a room terminal unit. The system has large degree of freedom because the supply air and the outdoor air can be controlled in an air conditioner as shown in Tables 2 and 3.

Combined Use of System   A room terminal unit can be combined with the primary air conditioner (Fig. 6). This means that two conditioning systems are present at a room. Heat balance equation is given in Eq. (15) below and solved assuming that the primary air conditioner has priority in processing and that the secondary unit processes excessive or deficient heat.

$$0.288G_{AC}(T_{RM} - T_{AC}) + 0.288G_{UT}(T_{RM} - T_{UT}) = Q_S + WZ_0\ (T_B - T_{RM}) \qquad \text{..........} \ (15)$$

G_{AC}, T_{AC}: Air quantity and temperature at the outlet of primary air conditioner; G_{UT}, T_{UT}: Air quantity and temperature at the outlet of secondary unit; Q_S: Load demand; T_{RM}: Room temperature; T_B: Datum temperature; WZ_0: WF of room temperature variation

the primary air conditioner, T_{AC} necessary for obtaining the preset T_{RM} is computed by disregarding the second term on the left side of Eq. (15). If the coil is underloaded, it can be dealt with by the primary air conditioner only; if load on the coil is excessive, T_{AC} is modified into T_{AC}' and then T_{UT} of the secondary unit is obtained from the known T_{AC}'. And, if the unit is underloaded, the preset T_{RM} is obtained; if that load is excessive, T_{RM} varies and is reflected to computation of the primary air conditioner and the unit as return air temperature upon the next iteration.

Intake of Outdoor Air   In addition to the minimum outdoor air, the items shown in Table 3 can be preset for controlling outdoor air. The time schedule is used for shutting outdoor air at the rise time, etc. Additional control can be provided to the items of enthalpy exchanger and maximum outdoor air. Fig. 7-a shows an example of judgement of system on-off by detection of outdoor air conditions. Fig. 7-b is an example of proportional control, in which outdoor air is introduced at Points A, B and C when controlled by T_{AC}, humidity and enthalpy respectively.

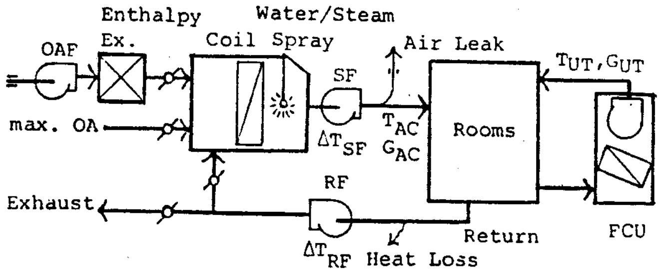


Fig. 6 Air Conditioner Model (CAV + FCU)

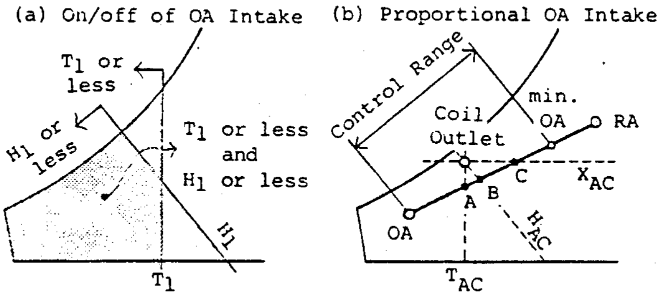


Fig. 7 Examples of OA Intake Control

Table 3 Outdoor Air Control

	Controlling Method	Additional Data
Basic Control	1. Minimum OA	-
	2. Time schedule	Time pattern
	3. Enthalpy exchanger	Efficiency
	4. Maximum OA	-
Additional Control	5. On-Off control	Operation range
	6. Proportional control	-

Methods 5 and 6 are options for Methods 3 and 4.

Table 2 Temperature and Humidity Control of Supply Air of Air Conditioner and Room Terminal Unit

Controlling Method	Control Code			Additional	Applicable Systems					
	TM1	HM1	TM2	Data	CAV	VAV	DD	PAC	UINT	OAAC
1. Return thermostat/humidistat	TRM	XRM	-	-	□	△	△	□	—	—
2. Room thermostat/humidistat	TRM	XRM	TRM	-	○	○	○	○	○	—
3. Keeping supply air constant	FIX	FIX	FIX	Outlet conditions	○	○	○	○	—	○
4. Securing min. supply air to rooms in VAV	MIN	-	MIN	-	—	○	○	—	—	—
5. Securing max. supply air to rooms in VAV	MAX	-	MAX	-	—	○	○	—	—	—

thods 4 and 5 are applied also to trol of cold/hot air temp. of DD.   TM2 is for control of hot air temp. of DD.   △ : Applicable to HM1 only   □ : Inapplicable to combined System

Heat Loss from Ducts, Air Leak and Heat from Fan
Heat loss from ducts is obtained by two approaches of heat loss, Q_{L1}, through duct walls and proportional air conditioning load Q_{L2}. For air leak, air flow passing through air supply fan and coil is increased at a rate of leakage. Heat from fan leads to air temperature rise Δt_F.

Heat Source Subsystem

Machine Models Models of refrigerator (Fig. 8), heat pump, absorption refrigerator and boiler were prepared by the use of characteristic equations based on their specifications. The maximum ability f_C and input, f_P, therefor are expressed as a function of which parameters are water temperatures, T_K and T_J, at the inlets downstream and upstream.

$$f_P(T_K, T_J) = \sum_{i=0}^{2} \sum_{j=0}^{2} C_{ij} \cdot T_K{}^i \cdot T_J{}^j \qquad \text{....... (16)}$$

Characteristic of partial load is a function which takes load factor X as a parameter.

$$f_X(X) = \sum_{i=0}^{2} C_i \cdot X^i \qquad \text{.......... (17)}$$

Standard values of the characteristic equations are incorporated into the program. But the user can also input any value at his option; if at least 9 sets of parameters and characteristic values are inputted, approximation equations are produced by multiple regression.

Definition of Group Plural units of heat source machines are defined as a group which is a unit for control (Fig. 9). Plural connections downstream can be made to the group. The connections are classified into cooling and heating depending on the symbol (+, -) of heat, Q_K, downstreatm.

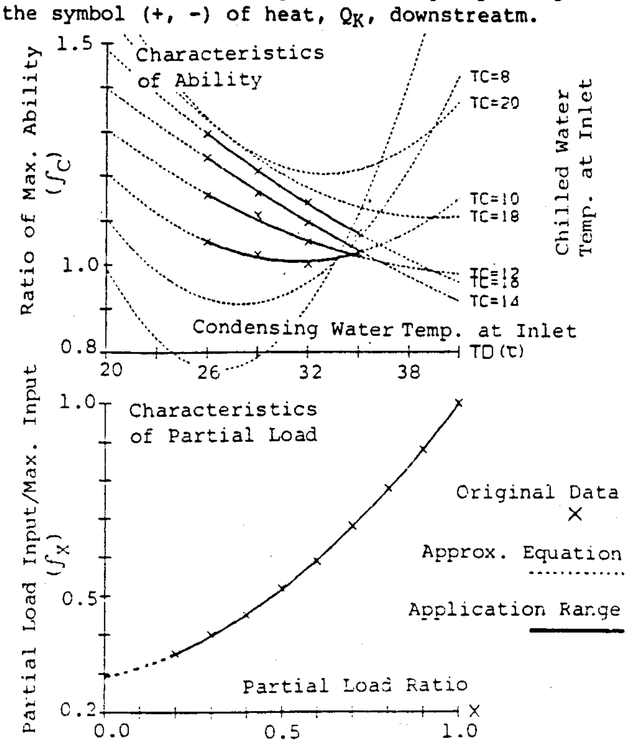


Fig. 8 Examples of Characteristics of Equipment

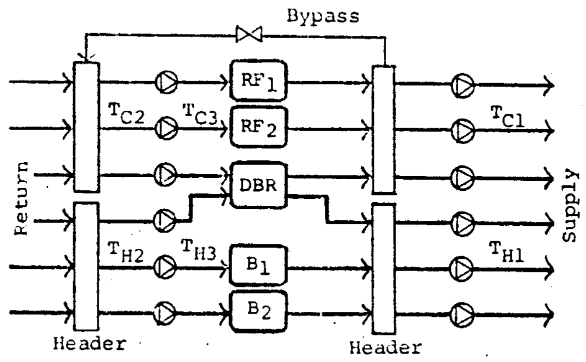


Fig. 9 Model of Heat Source Group

Bypass Control If flow rate, ΣG_P, of the primary pump is larger than circulation flow rate, ΣG_K, on the secondary side, its difference is flown by bypass. Temperature, T_{C2}, of return water at the header including the bypass is obtained and then water temperature, T_{C3}, at the inlet of machine is determined adding heat from the primary pump.

$$T_{C2} = T_{C1} + \Sigma Q_K/G, \quad G = \max\{\Sigma G_{p,i},\ \Sigma G_K\} \qquad \text{.. (18)}$$

$$T_{C3} = T_{C2} + \sum_i \{P_i \cdot x_i \cdot 860\}/G \qquad \text{.......... (19)}$$

T_{C1}: Water temperature at the outlet of machine obtained at previous step; P_i: Input power of the primary pump; x_i: Heat factor of pump; i: Machine no.; ΣQ_K: Thermal load on the secondary side

Number of Machines to be operated

Letting the preset water temperature at the outlet of the group of heat source be T_{CS}, load, Q_{CR}, to be dealt with by that group is:

$$Q_R = G(T_{C3} - T_{CS}) \qquad \text{.......... (20)}$$

The number, M, of machines to be operated can be determined so as to meet the following conditions.

$$\Sigma^{M} Q_{C,i} \geqq Q_R \text{ where } Q_{C,i} = Q_{0,i} \cdot f_C(T_{C3}, T_D) \qquad \text{.. (21)}$$

$$\Sigma^{M} G_{P,i} \geqq \Sigma G_K \qquad \text{.......... (22)}$$

$Q_{0,i}$: Rated ability of machine i; T_D: Condensing water temperature at the inlet

M can be determined by repeating Eqs. (18) and (21) according to the operation priority of machines in the group set in advance.

Judgement of Overload If $\Sigma Q_{C,i} \geqq Q_R$, water temperature at the outlet can meet the preset value. Letting load factor and rated input power be $X = Q_R/\Sigma Q_{C,i}$ and $P_{0,i}$ respectively, input power, P_i, to machines under partial loading is:

$$P_i = P_{0,i} \cdot f_P(T_{C3}, T_D) \cdot f_X(X) \qquad \text{.......... (23)}$$

If $\Sigma Q_{C,i} < Q_R$, water temperature, T_{C1}, at the outlet varies as follows.

$$T_{C1} = T_{C3} - \Sigma Q_{C,i}/G \qquad \text{.......... (24)}$$

Heat Recovery Heat recovery has CHR mode (recovery of hot water mainly by cooling) and HCR (mainly by heating). After ascertaining the operating conditions by computation of the principal mode, heat recovery is made within the limit of ·haust heat. Exhaust heat, Q_D, of double bundle frigerator DBR (Fig. 9) is:

$$Q_D = Q_C \cdot X + P \cdot 860 \cdot x_p \quad \cdots\cdots\cdots\cdots (25)$$

Q_C: Cooling ability; X: Load factor; P: Input power; x_p: Heat factor

Series Operation If there are common connecting systems on the downstream side, series operation can be carried out as shown in Fig. 10. In this case, water temperature at the outlet of the primary group of heat source is considered equal to that at the inlet of the secondary group.

Cooling Tower Subsystem

Machine Model The model of cooling tower is expressed as follows on the basis of equations for psychrometric heat transfer.

$$Q_D = 1.2G(H_{A2} - H_{A1}) = 1.2C_x \cdot G(WB_2 - WB_1) \quad .. (26)$$

$$Q_D = L(T_{D2} - T_{D1}) \quad \cdots\cdots\cdots\cdots (27)$$

$$Q_D = U_z \cdot t_e$$

$$t_e = (T_{D2} - WB_2 - T_{D1} + WB_1) / \log e\{(T_{D2} - WB_2)/(T_{D1} - WB_1)\} \quad \ldots (28)$$

WB_1, WB_2: Wet-bulb temperatures at the inlet and outlet of cooling tower; H_{A1}, H_{A2}: Saturation enthalpy at WB_1 and WB_2; T_{D2}, T_{D1}: Water temperature at the inlet and outlet of cooling tower; G: Air volume; L: Flow rate; Uz: Psychrometric heat transfer coefficient

C_x in Eq. (26) is apparent specific heat, i.e., $C_x = (H_{A2} - H_{A1})/(WB_2 - WB_1)$. If both H_{A2} and WB_2 are unknown, the simultaneous equations Eqs. (26), (27) and (28) cannot be solved. Therefore, differentiated value* $C_x = di/dt$ is obtained at WB1 from the approximation equation** of saturated steam pressure proposed by Matsuo. This means that the model shown in Fig. 11 is computed.

* Saturated steam pressure:

$$P_S(T) = 760 \cdot e^{A(T)}$$

$$A(T) = 6.18145 \cdot 10^{-12} T^5 - 3.429809 \cdot 10^{19} T^4 + 1.113417 \cdot 10^{-6} T^3 - 2.986334 \cdot 10^{-4} T^2 + 7.265429 \cdot 10^{-2} - 5.111336$$

Saturated absolute humidity:

$$X_S(T) = 0.622\, P_S(T)/\{760 - P_S(T)\}$$

Saturated enthalpy:

$$i_{(T,X)} = 0.24T + X\,(597.3 + 0.441T)$$

$$^{**}\; C_x = di/dt = 0.24 + 0.622\{P_S(T) \cdot A'(T) \cdot L(T) \cdot D(T) + 0.441\, P_S(T) \cdot D(T) + P_S(T) \cdot L(T) \cdot P_S(T) \cdot A'(T)\} / (D(T))^2$$

where, $L(T) = 597.3 + 0.441T$
$D(T) = 760 - P_S(T)$

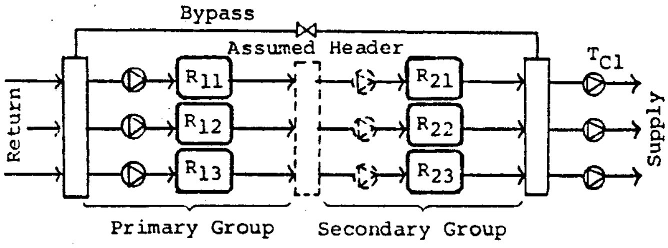


Fig. 10 Series Connection of Heat Source Group

Uz Value Considering it difficult for the user to preset the value of Uz, Uz is produced by simultaneously solving Eqs. (26) and (27) from cooling capacity Q_D, flow rate L, air volume G (all the above as specified for product), outdoor air wet-bulb temperature WB_1, and condensing water temperature T_{D2} at the inlet. In the simulation, T_{D1} can be got by solving the simultaneous equations from known Uz value. Also, the lower limit, T_{DL}, of T_{D1} can be set. If T_{D1} solved is lower than T_{DL}, T_{DL} is maintained with the aid of a bypass (Fig. 11-c).

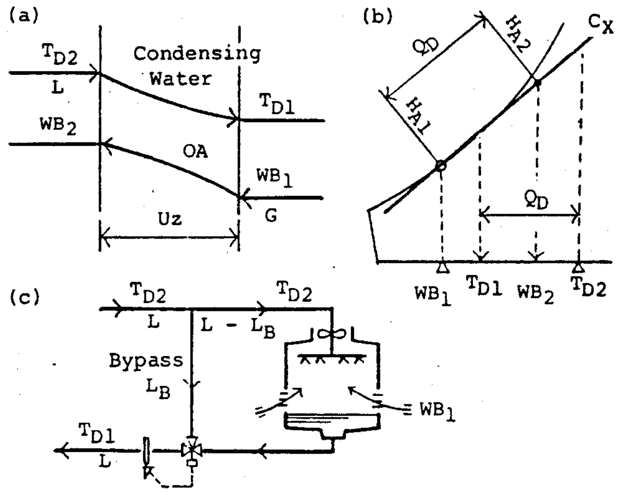


Fig. 11 Model of Cooling Tower

Thermal Storage Subsystem As shown in Fig. 12, operation of upstream heat source is controlled by hourly setting time and water temperature to be controlled.

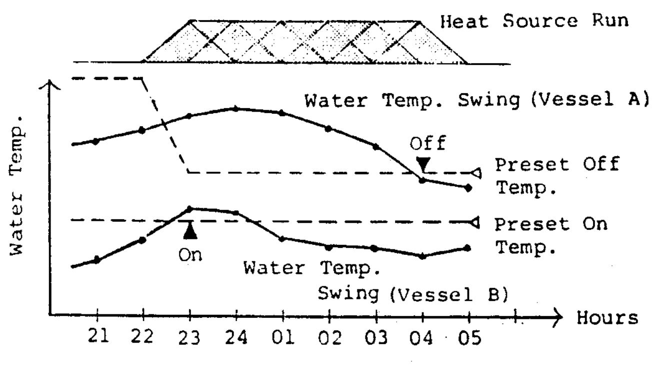


Fig. 12 Operation Control of Thermal Storage

Piping Subsystem In piping subsystem, the number of pumps to be operated can be controlled. Heat loss from piping and heat from pumps are computed. The approach is basically the same as for the subsystems for ducts and fans.

For piping, temperature drop under non-operation and thermal storage load under operation are produced for heat capacity preset concerning the whole piping including heat source machines (Fig. 13).

Letting representative heat capacity, its water temperature, circulation water amount and return water temperature be Q_{CC}, T_{CC}, G_K and T_{C2}, temperature, T_{C2}', of return water to the upstream heat source is equal to water temperature of the mixture of T_{C2} and T_{CC}.

$$T_{C2}' = (T_{C2} \cdot G_K \cdot \Delta t + Q_{CC} \cdot T_{CC}) / (G_K \cdot \Delta t + Q_{CC}) \quad (29)$$

Δt: Interval of computation time

Operation Time Operation time can be inputted for every air conditioner and terminal room unit. In the combined system, the program can run properly even if there is a time lag between operation of air conditioner and terminal room unit (Fig. 14).

Heat extraction and room temperature and humidity are computed in the same manner as conducted for dynamic thermal load computation. Although operation is discontinuous at the time of start and stoppage, it is solved by converging the computation twice in operation time.

EXAMPLE OF SIMULATION

The air conditioning system and building selected as models for simulation are shown in Figs. 15 and 16. The air conditioning system has two types, i.e., (1) Conventional system (CAV + CWV) and (2) Energy saving system (VAV + VWV). Simulation result is given in Table 4 and part of the output in Table 5.

The model is the same as used in Reference (1). The simulation result reveals nearly mean values of the simulation by the programs proposed in Reference (1).

The computing time by the use of IBM S/4341·M/2 was about 11 minutes for (1) Conventional system and about 21 minutes for (2) Energy saving system. The simulation time accounts for about two to three times the thermal load computation time conducted in advance.

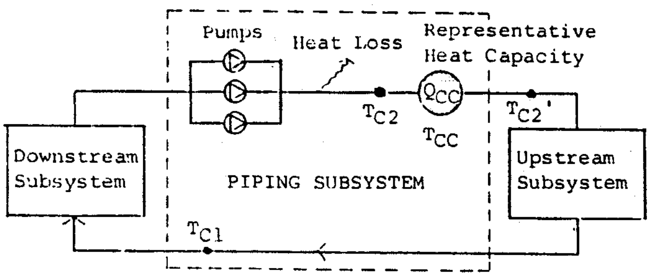


Fig. 13 Model of Piping System

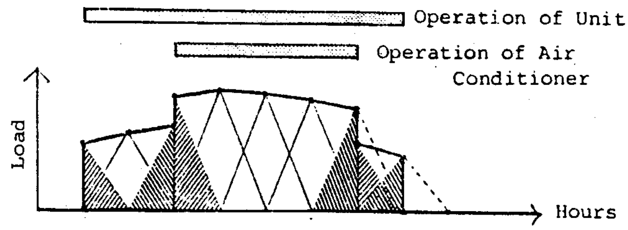


Fig. 14 Example of Preset Operation Time

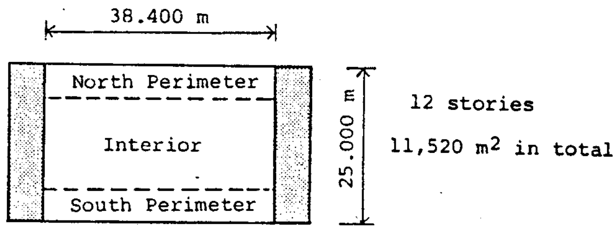


Fig. 15 Model Building (Plan)

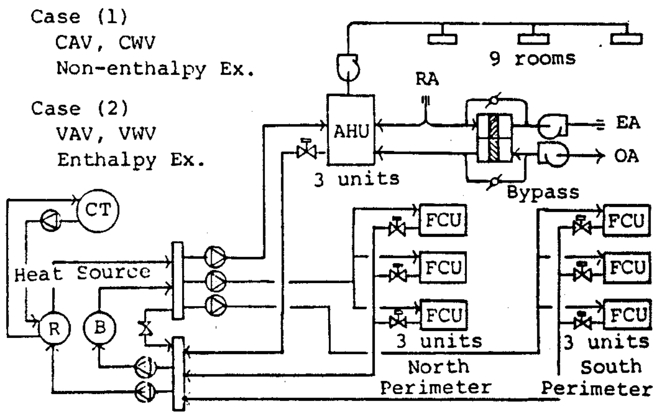


Fig. 16 System Model for Simulation Example

Table 4 Results of Yearly Simulation

Model System	Total Heat Exchanged				Total Consumption				Energy per primary air-conditioned area (Mcal/m²·a)
	Air Handling Unit		Fan Coil Unit		Electricity		Gas (1000 m³)	Water (kℓ)	
	Cooling (Gcal)	Heating (Gcal)	Cooling (Gcal)	Heating (Gcal)	Source (MWh)	Pumps and Fans (MWh)			
(1) Conventional S.	577	10	17	155	220	194	19	1,470	106
(2) Energy Saving S.	499	2	16	99	185	184	12	1,290	90

Table 5 Example of Output

```
----- HASP/ACSS/8506  ANNUAL OUTPUT  HASP/ACSS FINAL TEST  60.06.30  CASE-2  ENERGY SAVING SYSTEM
ANNUAL TOTAL/AVE   (365 DAYS)

UNIT ( TEMP=DEG.C,HUMI=G/KG (PER),Q=GCAL/Y,GA=GCM/Y,GW=GCM/Y,ELC=MWH,GAS=KCM/Y,OIL=KL/Y,STM=TON/Y,WTR=KTON/Y),(RUNNING HOUR)

RM=N01*  1 (2745.0) TR=21.57  XR= 9.86 ( 61)   (TR=23.13  XR=10.12 ( 57))  QCS=    4.302 QHS=    7.080 QCL=    3.431 QHL=    0.279
  =S01*  1 (2745.0) TR=23.37  XR= 9.83 ( 55)   (TR=24.82  XR=10.07 ( 52))  QCS=    6.425 QHS=    2.181 QCL=    3.425 QHL=    0.242
   101*  1 (2745.0) TR=29.73  XR=11.06 ( 42)   (TR=29.10  XR=10.88 ( 43))  QCS=   29.488 QHS=    0.042 QCL=   15.119 QHL=    0.015
   N02*  1 (2745.0) TR=21.97  XR=11.06 ( 67)   (TR=23.40  XR=11.58 ( 64))  QCS=   45.357 QHS=   63.436 QCL=   31.259 QHL=   11.000
RM=S02*  1 (2745.0) TR=23.94  XR=10.95 ( 59)   (TR=25.23  XR=11.42 ( 57))  QCS=   70.406 QHS=   18.307 QCL=   31.558 QHL=   10.249
RM=102*  1 (2745.0) TR=34.08  XR=12.82 ( 38)   (TR=32.93  XR=12.65 ( 40))  QCS=  411.178 QHS=    0.144 QCL=  151.459 QHL=    0.897
RM=N12*  1 (2745.0) TR=21.45  XR= 9.47 ( 59)   (TR=23.07  XR= 9.74 ( 55))  QCS=    4.492 QHS=    8.123 QCL=    3.787 QHL=    0.270
RM=S12*  1 (2745.0) TR=22.98  XR= 9.46 ( 54)   (TR=24.43  XR= 9.71 ( 51))  QCS=    6.377 QHS=    2.954 QCL=    3.777 QHL=    0.248
RM=112*  1 (2745.0) TR=26.30  XR=10.38 ( 49)   (TR=26.01  XR=10.25 ( 49))  QCS=   22.912 QHS=    0.692 QCL=   15.139 QHL=    0.019
UT=FC1*  1 (2305.0) QC=    3.872  QH=  77.429                               EMC=    0.000 EAC=    9.957
UT=FC2*  1 (2305.0) QC=   11.264  QH=  21.736                               EMC=    0.000 EAC=    9.957
AC=AC1*  1 (2745.0) QC=   39.584  QH=   0.115  SP=   0.000                  EMC=    0.698 EAC=    9.980 WTR=    0.006
AC=AC2*  1 (2745.0) QC=  419.076  QH=   0.333  SP=   0.000                  EMC=    6.978 EAC=  105.622 WTR=    0.178
AC=AC3*  1 (2745.0) QC=   40.384  QH=   1.659  SP=   0.000                  EMC=    0.698 EAC=   11.504 WTR=    0.001
MC=RF1     ( 945.0) QC=  533.938  QH=   0.000  XQ=   0.575                  EMC=  115.699 EPP=    9.450                NGT=    0.000
MC=B01     (1340.0) QC=    0.000  QH=  87.053  XQ=   0.067                  EMC=    0.673 EPP=    8.978 GAS=   12.030
CT=CT1     ( 945.0) QC=  652.089                                            EMC=    6.993 EPP=    0.000 WTR=    1.092 NGT=    0.000
PP=PC1     (2285.0) GW=   25.394   N= 1.0/  1                                             EPP=   27.292                NGT=    0.000
PP=PC2     (1845.0) GW=    2.696   N= 1.0/  1                                             EPP=    3.831                NGT=    0.000
PP=PC3     (1845.0) GW=    4.254   N= 1.0/  1                                             EPP=    6.050                NGT=    0.000
PP=PD1     ( 945.0) GW=   36.056   N= 1.0/  1                                             EPP=   35.059                NGT=    0.000
*ENRGY
 TOTAL     ELC=  369.419 GAS=   12.030 OIL=    0.000 WTR=    1.277            Heat Source
 SUB       EMC=  131.738 EPP=   90.660 EAC=  147.020 NGT=    0.000
```

CONCLUSIONS

(1) In HASP/ACSS, subsystems were classified into five levels, the degree of freedom in selection of systems being thereby increased.

Strictly speaking, air conditioning systems are different from each other because every designer has different concept. Although the program HASP/ACSS may not meet all requirements of all designers, it has considerably high flexibility. In addition, HASP/ACSS is of such structure as to easily cope with addition or change of function in the future.

(2) In HASP/ACSS, iteration was used for solving accurate heat balances. Since a system of normal design is not subjected to excessive loads for the greater part of running time, the program was conceived so as not to incur repetition of computation under minor loads.

(3) When loads are excessive, repetition of computation is necessary because supply conditions of upper stream changes. Where VAV, DD, etc. are overloaded, the number of repetitions of computation is liable to increase because temperature and flow rate change at the same time. Also in the combined system, the number of repetitions increases because the ratio of sharing loads between the primary air conditioner and the secondary unit changes. When upstream equipment is under excessive loads, the number of repetitions becomes larger because lower streams are largely affected.

In the example of simulation conducted for a combined VAV and FCU, excessive loading occurred at a refrigerator upstream during the rise time and the summer peak hours. Computation was repeated over 30 times at peak times and 2.5 times on yearly average.

(4) While sensible heat and latent heat are independently solved in the thermal load computation, they are simultaneously computed for the model of coil in the simulation. Temperature and humidity give great influence on the performance of coil. In computing room humidity, WF for humidity is a problem to improve in future.

ACKNOWLEDGEMENTS

In the course of the development the authors were afforded various advice and cooperation by and therefore wish to express sincere thanks to members of the committee stated in the opening paragraph and, particularly for management of the committee, the Tokyo Electric Power Company, Inc.

REFERENCES

(1) MATSUO, Y. "Survey of Simulation Technology in Japan and Asia," Building Energy Simulation Conference, Seattle, 1985.

(2) INOUE, U. and LEE, H. "Air Conditioning Handbook," Maruzen, Tokyo, 1982.

(3) MATSUO, Y., YOKOYAMA, K. and INOOKA, T. "An Energy Simulation Program on Air Conditioning Systems, Parts 1 to 3," Summaries of Technical Papers of Annual Meeting, Society of Heating, Air Conditioning and Sanitary Engineers of Japan, Tokyo, 1985.