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THE SIMULATION MODEL OF INDUSTRIAL CONDITIONING SYSTEMS

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The Simulation Model of Industrial Air Conditioning Systems

SYNOPSIS

This paper describes a simulation program which was developed for the modelling of air-conditioning systems and conditioned spaces in industrial buildings. The program can be used for a design of systems for new buildings and for analysis of existing ones. By viewing the building as a dynamic entity, it is possible to investigate how thermal capacity of the building elements acts on both the conditioned space and the performance of the air-conditioning system.

The program simulates three important aspects of a building. Simulation of the central air-conditioning system determines enthalpy, temperature and humidity of the air at different points of the system. System performance, capacity requirements and energy consumption under various circumstances are also determined. Simulation of the conditioned space shows its temperature and humidity, when different central air-conditioning systems, airflows and supply-air temperatures are being used. Simulation of the control system makes it possible to investigate the effect of different control methods and devices on the performance of the components of the central air-conditioning system and on the conditions within the conditioned space.

Conditions in the simulated space, capacity requirements of the central air-conditioning system and the air mixture conditions regarding fresh and recirculated air have been compared with measured values. The measurements have shown that simulation gives a realistic indication of how the simulated aspects behave under the given circumstances.

The temperatures of the simulated space have also been compared with the values calculated by the Ventac-program. The average difference was 0,2 ^{o}C.

List of symbols

A_i = area of the windows [m^2]
A_s = area of the walls [m^2]
c_w = heat capacity of the wall [kJ/kgK]
DH = output
e_n = error signal
h_s = heat transfer coefficient of the walls [W/m^2K]
i_s = enthalpy of the room air [kJ/kg]
i_{sp} = enthalpy of the supply air [kg/kg]
i_u = enthalpy of the outdoor air [kg/kgK]
K_p = propotional gain factor
K_i = integral gain factor
$\dot{m}_h$ = vapour from the open containers [kg/s]
$\dot{m}_{hk}$ = vapour from the machines [kg/s]
$\dot{m}_{ih}$ = moisture from the human bodies [kg/s]
$\dot{m}_{iv}$ = mass flow of the supply air [kg/s]
m_s = mass of the room air [kg]
$\dot{m}_{sp}$ = mass flow of the supply air [kg/s]
$\dot{m}_v$ = mass flow of the infiltration air [kg/s]
m_w = mass of the wall [kg]
t = time
T_s = indoor temperature [^oC]
T_u = outdoor temperature [^oC]
T_w = temperature of the walls [^oC]
U_i = thermal transmittance coeff. of the windows [W/m^2K]
U* = thermal transmittance coefficient of the walls without the effect of the inner heat transfer coefficient [W/m^2K]
x_s = humidity of the indoor air [kg/kg]
x_{sp} = humidity of the supply air [kg/kg]
x_u = humidity of the outdoor air [kg/kg]
ϕ_{rad} = radiating heating load [W]
ϕ_s = convective heating (or cooling) load [W]

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1 Scope of the Model

In the research and design work of industrial air conditioning systems a need for a simulation program became evident. Because suitable and commercially available programs were not found, it was decided to develop such a program. Using now the newly developed IVSIMUL-model it is possible to simulate

- indoor temperature and humidity
- temperature, humidities and enthalpy at different points of the air conditioning system
- heating and cooling effects and energy consumption
- effects of different control systems
- effect of the thermal capacity of the building envelope

The program consists of three different parts which are linked together. One part which is linked to the dynamic room model simulates the behaviour of the air conditioning unit. The control section makes it possible to simulate the operation of the control system.

2 Room model

In many fields of industry it is important that the air-conditioning system maintains the required indoor humidity and temperature. The simulation of the air conditioning system is also based on the knowledge of humidity, temperature and enthalpy at different points of the system. It's obvious, therefore, that a room model must be created in such a way, that humidity is also taken into account.

Enthalpy equation of the room air is written:

$$\frac{di_s}{dt} m_s = \phi_s + \dot{m}_{sp} (i_{sp} - i_s) + h_s A_s (T_w - T_s) + U_i A_i (T_u - T_s) + \dot{m}_v (i_u - i_s) \qquad (1)$$

The heat balance of the inner surface of the exterior wall is given by the equation (2). The equation is simplified, because it is assumed that the wall consists of only one layer.

$$h_s A_s (T_s - T_w) + \phi_{rad} = U^* A_s (T_w - T_u) + \frac{d T_w}{dt} m_w c_w \qquad (2)$$

The humidity balance equation (3) must be solved together with the equations (1) and (2).

$$\frac{dx_s}{dt} = \dot{m}_{ih} + \dot{m}_{hk} + \dot{m}_h + \dot{m}_v (x_v - x_s) + \dot{m}_{iv} (x_{sp} - x_s) \quad (3)$$

3 Control simulation

Figures 1 and 2 illustrate an example of a control loop simulation. In both cases the required room temperature is 20 °C. When a proportional control is used, a difference exists between the actual and required temperature. When a proportional and integral (PI) control is being used, the required room temperature can be achieved. A step by step calculation of the output from the PI-controller is carried out using the formula (4).

$$DH = K_p(e_n - e_{n-1}) + K_i e_n t \quad (4)$$

It is also possible to simulate the proportional control with hysteresis, which appears in real systems when self acting thermostatic valves are being used.

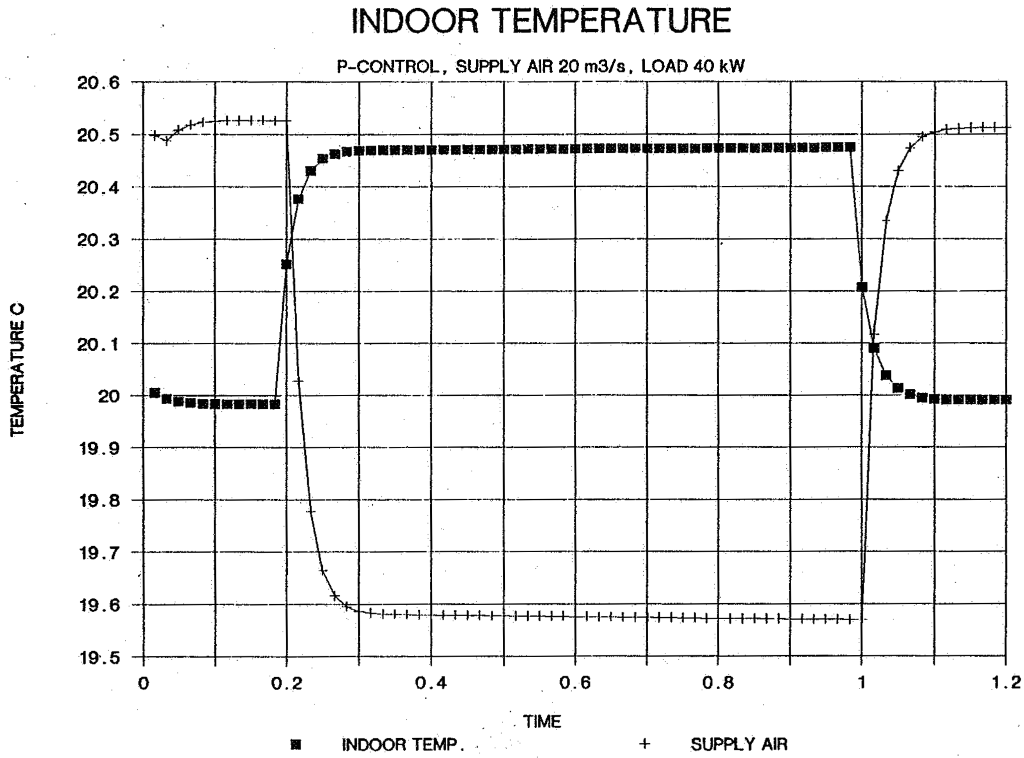

Figure 1. Simulated indoor temperature, P-Control.

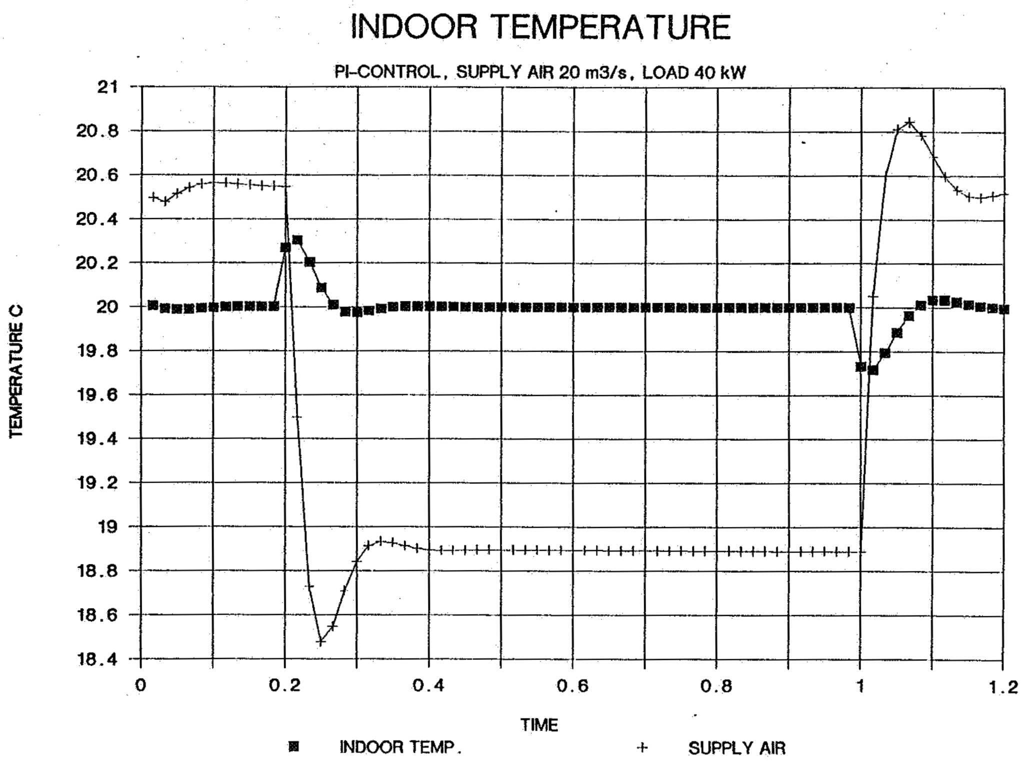

Figure 2. Simulated indoor temperature, PI-Control.

4 Comparison of the simulated and measured values

Simulated values of a printing shop were compared with its measured values. The measured values were

- indoor temperature and humidity in four points
- temperature and humidity of the air in different places of the air conditioning units
- water temperature before and after the heating coils
- energy consumption of the air conditioning
- energy consumption of the radiators
- power demand of the printing machines

The difference between the simulated values and the measured mean daily temperatures was 0,4 ^{o}C on the average. The biggest difference was 3,5 ^{o}C. An example of one day is illustrated in the figure 3.

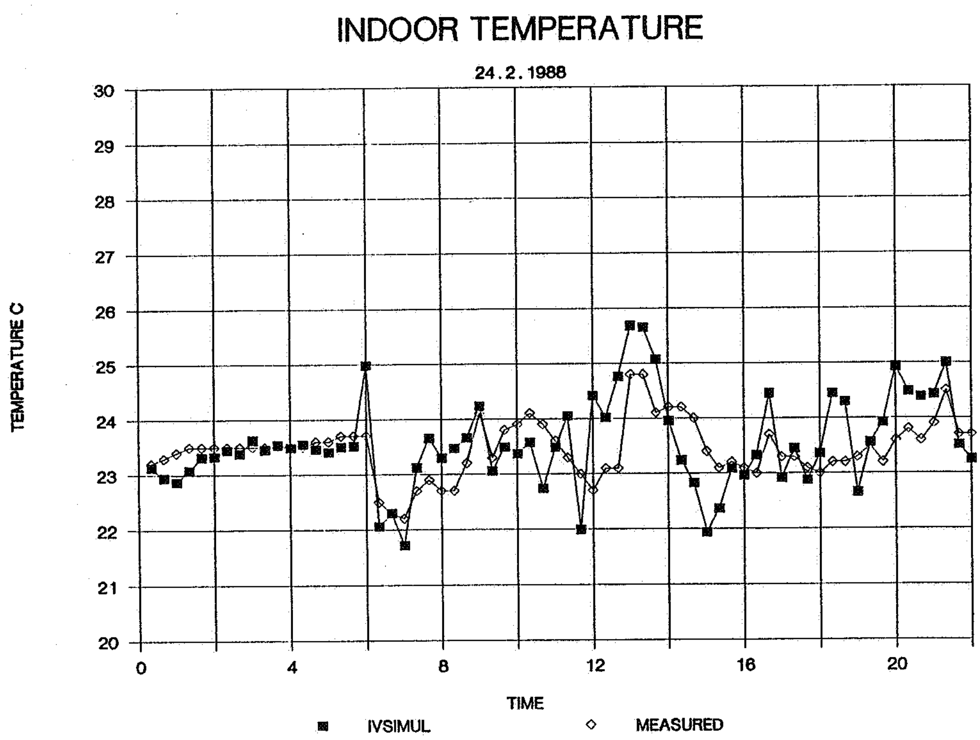

Figure 3. Measured and simulated indoor temperature.

The simulated indoor humidity was somewhat higher than the measured value. The average difference was 0.0018 kg/kg. Reliability of the moisture measurements is not, however, as good as that of the temperature measurements.

The simulated temperature and humidity of the mixed indoor and outdoor air were close to the measured values. The average difference between the simulated and measured effect of the heating coil was 4,9 %. An example of the measured and calculated effect can be seen in the figure 4.

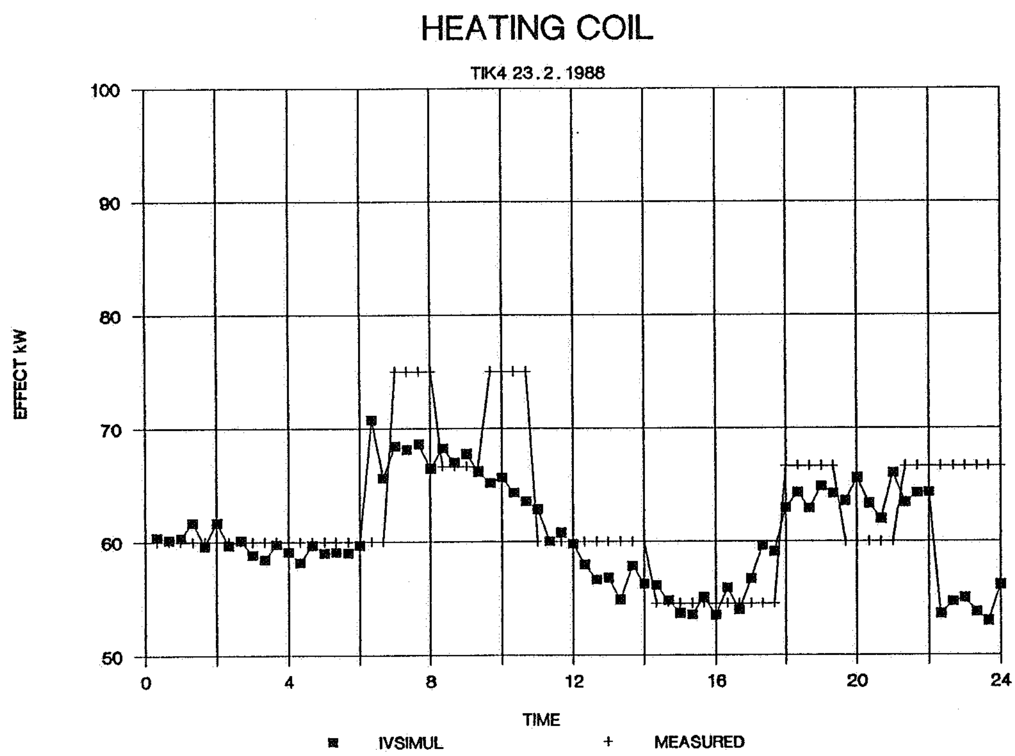

Figure 4. Measured and simulated effect of the heating coil.

5 Comparison of the IVSIMUL and Ventac programs

The IVSIMUL-program was also compared with the Ventac-program. A small hypothetical industrial building was used as a test building . Two cases were used in the comparison. Another was a massive building with minor internal load and the other was a light building with heavy internal load. The floor area of the building was 200 m^2. Window area to the south was 10 m^2 as it was to the north as well. Lighting load was 20 W/m^2. Internal load from machinery was 1 kW in the heavy building and in the light building 5 kW. Supply air flows were 1 l/s m^2 and 10 l/s m^2 accordingly. Working hours were from 7.00 to 16.00 and a presence of 10 workers was assumed. Supply air temperature was 18 ^oC and the air conditioning was operating continuously. There was no moisture or latent heating load.

Calculations showed that the results from the tested programs were near to each other. Differences of the mean daily temperatures of typical days from January to June were from -0,1 to 0,6 ^oC in the heavy industrial building and from -0,3 to 0,4 ^oC in the light industrial building. The standard deviations of the daily differences were from 0,2 to 0,4 ^oC in the light building and from 0,2 to 0,7 ^oC in the heavy building (figures 5 and 6).

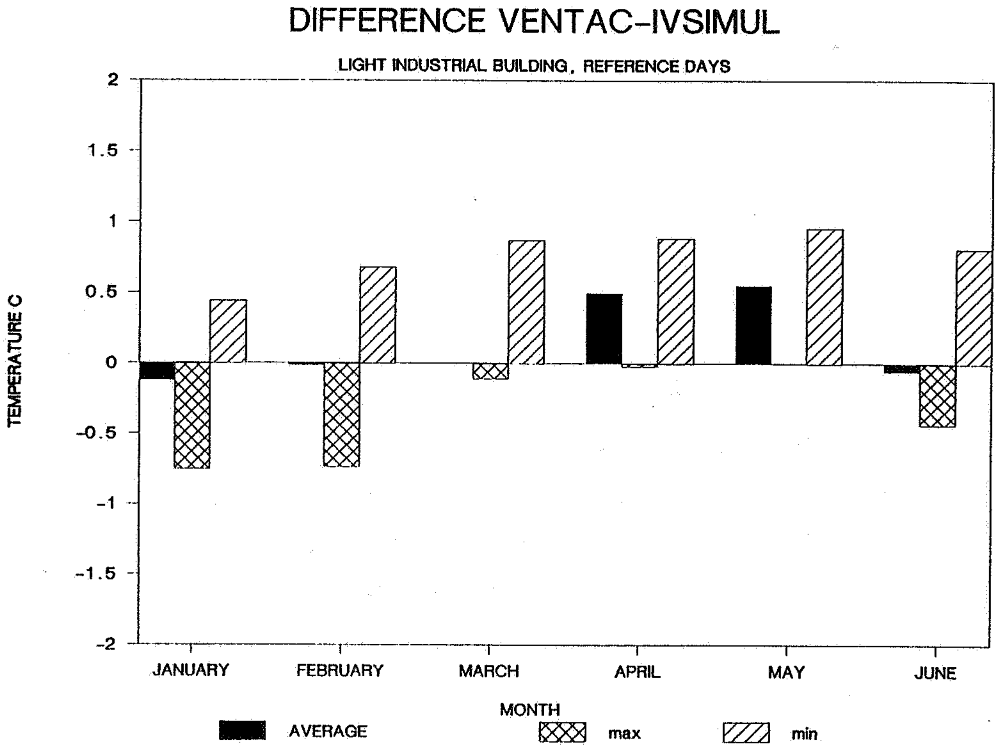

Figure 5. Comparison of the Ventac and IVSIMUL-programs.

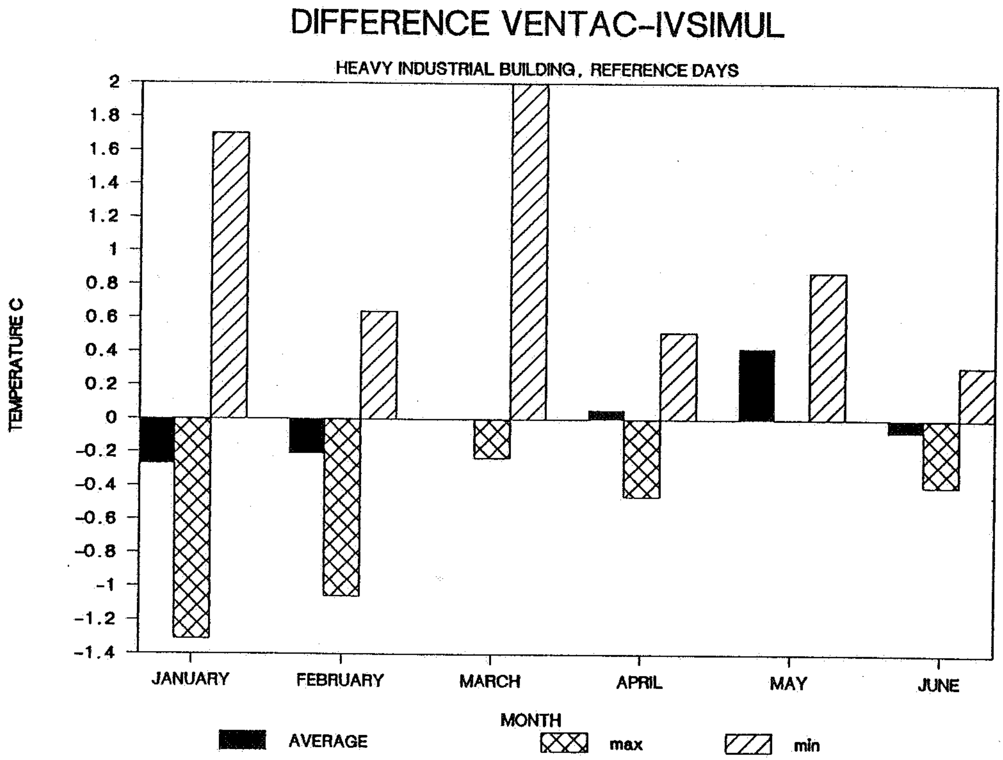

Figure 6. Comparison of the Ventac and IVSIMUL-programs.

6 Conclusions and further development

The IVSIMUL-program seems to give reasonably good results for practical purposes. Several simplifications, however, have been made and they should be born in mind when using the program. The program will be used in the future for analysis of new and existing industrial buildings. More accurate description of the cooling coils will be one of the future enhancements. The control system part of the program could also be extended. The possibility of linking the IVSIMUL thermal simulation model to a concentration model developed by PI-Consulting Ltd is under consideration.

Acnowledgements

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