BUILDING ENERGY AND SYSTEM SIMULATION PROGRAMS: MODEL DEVELOPMENT, COUPLING AND INTEGRATION

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ABSTRACT

This paper presents some results of the development and application work of the Building Equipment section at EMPA related to integrated building and HVAC simulation environments.

Routines for thermal comfort evaluations with DOE2 are developed and COMVEN, the simulation code of the COMIS multizone air flow program, has been adapted as a type for TRNSYS. The application potential. for this coupling is demonstrated for a retrofit study of a school building, naturally ventilated through a glazed double facade.

Secondly a general physical model for closed circuit cooling towers is described including comparison with measured manufacturer data sets.

KEYWORDs: Building simulation, coupling, integration, TRNSYS, DOE-2, COMIS, multizone airflow modelling, combined heat and air transport, passive cooling, closed circuit evaporative cooling tower, cooled ceiling

1 INTRODUCTION

The Building Equipment Section at EMPA has been involved in building and HVAC simulation for more than a decade. Among the wide range of programs worked with, the most important are DOE-2, TRNSYS, SUPERLITE and COMIS. The work covered both the use and the development of programs, for research and consulting. This paper presents some results of this work, describing a new model as well as the work on coupling and integrating these individual programs.

In many design cases, energy as well as occupant comfort are the relevant criteria which are studied using computer simulation programs. Comfort evaluations cover air quality, thermal, visual and acoustical comfort. For all these individual aspects, specific simulation programs are available today, but very few programs allow for the integrated evaluation of several or all relevant parameters. The more, heat transport, ventilation as well as lighting are physically coupled and therefore must be integrally modelled in the simulation process (Fig. 1). Section 2 and 3 shortly outline some aspects of the actual work at EMPA in this context.

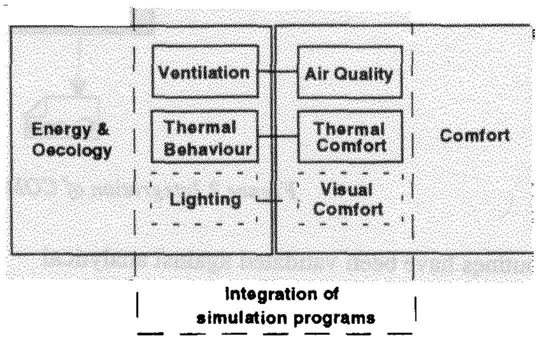

Figure 1 Integrated program tools for integrated design

In building equipment systems simulation, for many plant types several physical models are available. The purpose of most of these models is component development. Therefore, to use the model, detailed information about the construction of the plant is required. This is in contrast to the HVAC planner's needs. There, normally the component nominal condition is known and the system behaviour under different operating conditions is of interest. Therefore, the purpose of the model presented in section 4 is to describe the system behaviour physically correctly, but with parameters which are related to the planning task.

2 EVALUATION OF THERMAL COMFORT WITH DOE-2

One of the shortcomings of the present version of DOE-2 is that the applied weighting factor method does not allow for the direct output of inside surface temperatures. This feature is needed to consider the radiant temperatures as one of the key elements in thermal comfort calculation.

At EMPA, an attempt was made aimed at determining the inside surface temperatures by a set of routines on the basis of the wall response factors and wall heat gains.

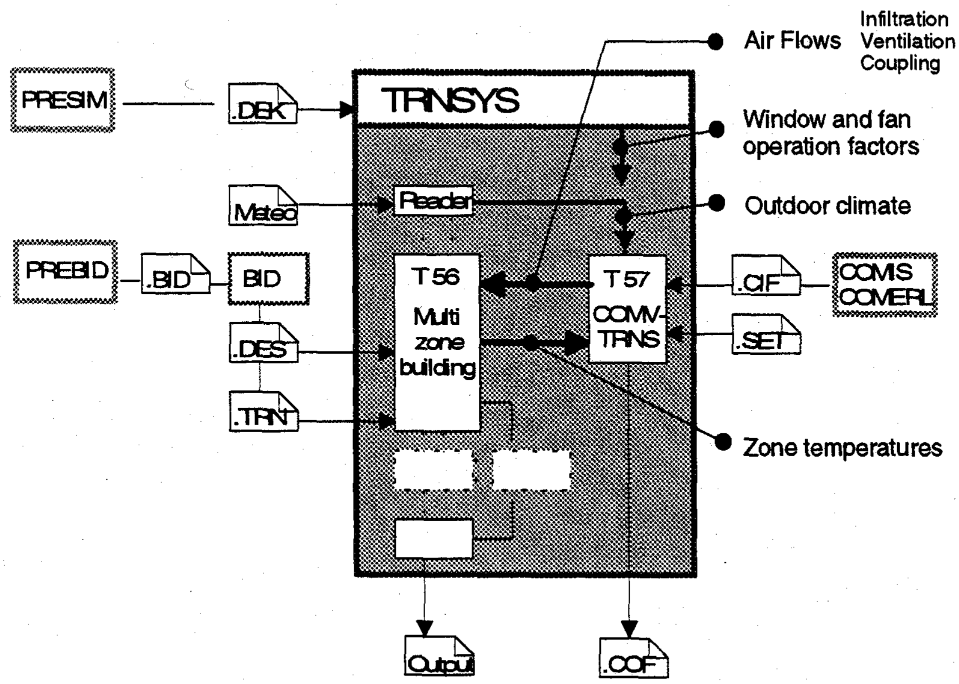

Figure 2 Integration of COMVEN as a Type into TRNSYS

These routines have been validated against analytical solutions as well as by comparison with the measured data sets used in the validation efforts within IEA-ECB Annex 21. The routines developed are presently integrated into DOE-2 in close co-operation with the Simulation Research Group at the LBL [1]. Routines for view factors calculation are planned to be incorporated. For a certain point in the room under investigation, these are needed for a detailed thermal comfort evaluation on the basis of the operative temperature.

3 COMBINED MODELLING OF HEAT AND AIR TRANSPORT

3.1 Integration of COMVEN into TRNSYS

Many building simulation programs are not very well adapted to the simulation of natural ventilation. On the other hand, multizone air flow models normally require the room air temperatures as input values. Therefore the modelling of thermally induced driving forces is limited because in many applications the room air temperatures are not known a priori. In such cases, the integrated use of a multizone air flow model in a building simulation code allows for the treatment of combined heat and air transport problems [2]. Therefore, one step in the coupling and integration task is the adaptation of COMVEN as Type 57 for the building and systems simulation code TRNSYS, to be used in combination with the TRNSYS multizone building Type 56. COMVEN is the air flow simulation code of COMIS, presently under further development in the frame of IEA-ECB Annex 23 'Multizone Air Flow Modelling'. Figure 2 shows the TRNSYS program with the two Types mentioned, the parameters of their data connection, as well as the related input and output files and the respective pre-processor programs. Details on the coupling, including an example case, are given in [3].

3.2 Retrofit with a glazed double facade

For a four-storey school building, retrofit concepts have been worked out on the basis of a glazed double facade, built up over the original structure which remains practically unchanged. This approach is effective in respect to construction costs and to ecological aspects. Figure 3a and b, respectively, show one of the proposed constructions with a flow pattern at a certain time. On each side of a room, there is a section of the double facade where the space is open in vertical direction, acting as a ventilation shaft. In the middle section, the original window is removed and replaced by a window in the outer facade.

While the potential for reduced transmission losses in winter time is quite obvious, more concern was related to the overheating risk in summer and thus to the possibilities to ventilate and naturally cool the building satisfactorily. Thus, the aim of the simulation study presented in this paper was to provide data on the thermal comfort and the corresponding ventilation situation in the different rooms for a typical hot summer period, and to establish strategies for the operation of the different windows and openings for an optimum indoor environment.

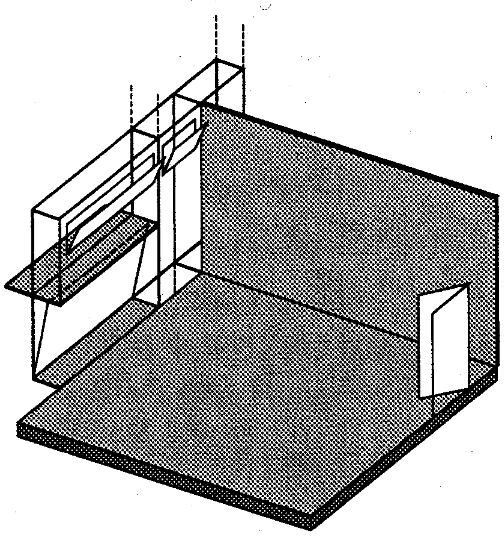

Figure 3a Isometric drawing of one room with the double facade spaces

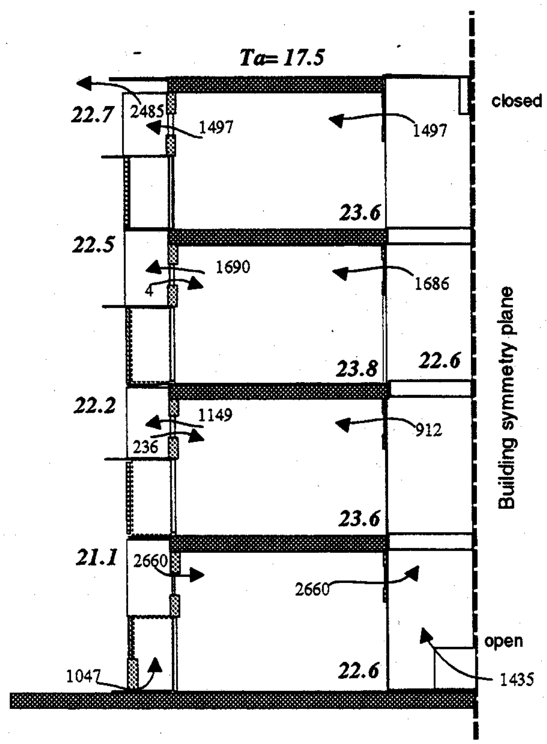

Figure 3b Cross-section of the building section modelled, with air flow [kg/h] and temperature [^{o}C] values at 6 am (see Fig. 4)

Simulations have been made for a typical hot summer period. The result parameters are the air flows per opening in the room as well as the temperatures in the individual zones. Figure 4 shows for a three-day period the calculated temperatures and air flows for the room and the double façade space of the second floor as well as for the staircase, together with the opening schedules for the windows, the openings to the double façade space and the openings in the staircase to outside.

Thermal comfort conditions were checked by comparing the resultant air temperatures with the design conditions. The air quality aspects are covered by mean age of air values per room or by defining CO_2 sources according to the occupant presence and checking the resulting concentrations. Wind effects as well as the influence of the second, north oriented building half have not been considered in the simulations.

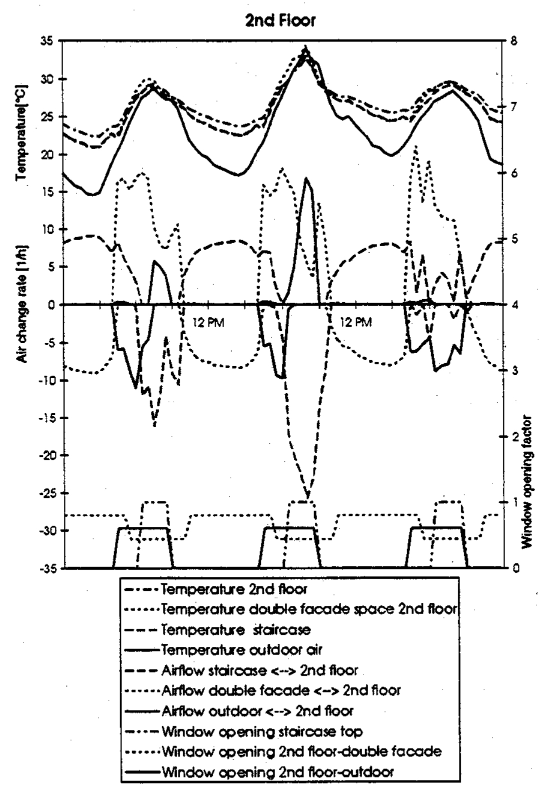

Figure 4 Temperatures and air flows to and from the 2nd floor room and opening schedules for the different openings. The bottom staircase opening is opened as soon as the top opening is closed (and vice-versa)

Within the iterative solution process, oscillations may occur in stack (buoyancy) dominated driving force conditions. In such cases, changes in the air temperatures and thus in the stack pressures lead to reversed flow directions in a critical zone and in consequence the room temperatures change again significantly. These oscillations from one iteration step to the next may lead to numerical convergence problems, which can be overcome by introducing an element which numerically damps the air flow data connection between Type 56 and Type 57 (COMV-TRNS).

4 MODEL FOR CLOSED CIRCUIT EVAPORATIVE COOLING TOWER

4.1 The importance of cooling towers

Presently cooled ceilings are increasingly used for cooling purposes in office and also industrial buildings. Because cooled ceilings are able to carry off considerable heat loads at relatively small temperature differences between the room air and the cooled ceiling surface, it is possible to run the system with supply water temperatures between 18 and 20 °C. In many climates, cooling towers are able to deliver water at such temperature levels during a large proportion of the cooling period. In Europe, closed circuit cooling towers are mostly used this purpose due to soiling problems, to avoid high maintenance costs. For such systems a general physical model for a closed circuit evaporative cooling tower was developed and compared with measurements. The model is expressed purely in terms of design parameters as heat load, mass flow etc. After a set-up procedure with the nominal conditions the model calculates the water outlet temperature under different operating conditions (e.g. air flow rate through the tower or wet bulb temperatures). The model has been integrated in the building simulation program TRNSYS and will also be integrated in a future version of the program DOE-2.

4.2 The model

In the closed cooling tower model (Fig. 5), the heat exchange between the cooling water and the spray water film (Fig. 6) can be expressed by

$$U \cdot (\theta_c - \theta_s) \cdot dA = \dot{m}_c \cdot c_c \cdot d\theta_c \qquad (1).$$

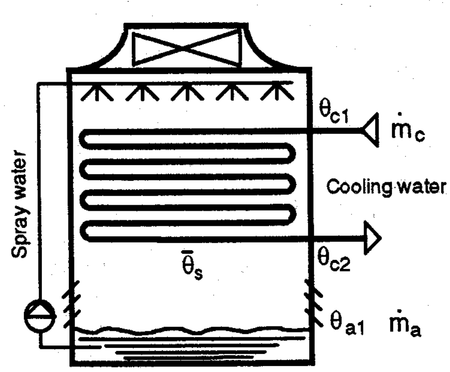

Figure 5 Schematic of the closed cooling tower

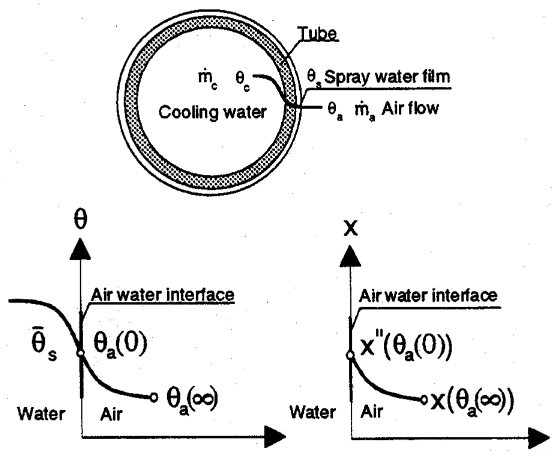

***Figure 6** Heat and mass transfer in the cooling tower heat exchanger tube*

Assuming a constant overall heat transfer coefficient and a constant spray water temperature (Fig. 7), we obtain after integration of Eqn. (1)

$$\kappa = \frac{U \cdot A}{\dot{m}_c \cdot c_c} = \ln\left[\frac{\theta_{c1} - \bar{\theta}_s}{\theta_{c2} - \bar{\theta}_s}\right] \qquad (2).$$

The same relation can also be formulated for the nominal condition:

$$\kappa_0 = \frac{U_0 \cdot A}{\dot{m}_{c0} \cdot c_c} = \ln\left[\frac{\theta_{c1_0} - \bar{\theta}_{s0}}{\theta_{c2_0} - \bar{\theta}_{s0}}\right] \qquad (3).$$

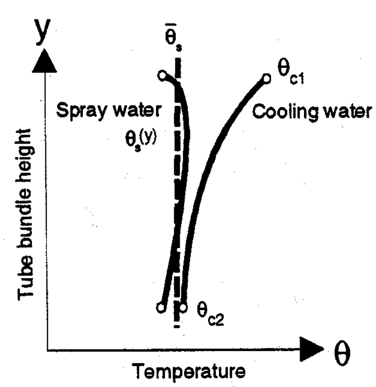

***Figure 7** Cooling and spray water temperature, respectively, in relation to the tube bundle height (qualitative)*

If we assume that the nominal overall heat transfer coefficient U_0 changes only in function of the cooling water mass flow, the following approach defines the U-Value for any other condition:

$$U = U_0 \cdot f(\dot{m}_c) \qquad (4)$$

and therefore

$$\kappa = \kappa_0 \cdot f(\dot{m}_c)\cdot \frac{\dot{m}_{c0}}{\dot{m}_c} \qquad (5).$$

After rearrangement of Eqn. (2), and with Eqn. (5), we obtain for the cooling water outlet temperature

$$\theta_{c2} = \overline{\theta}_s + (\theta_{c1} - \overline{\theta}_s)\ \exp\left(-\kappa_0 \cdot f(\dot{m}_c)\frac{\dot{m}_{c0}}{\dot{m}_c}\right) \qquad (6).$$

In Eqn. (6), the quantity κ_0 cannot be determined because the spray water temperature for the nominal conditions $\overline{\theta}_{s0}$ is unknown. But for a large number of κ_0 the second term on the right side of Eqn. (6) can be neglected. The following consideration shows, that an overestimation of κ_0 by any factor χ gives a small influence on the fluid side energy balance for $\kappa > 3$. For this we approximate Eqn. (6) with

$$\tilde{\theta}_{c2} = \overline{\theta}_s + (\theta_{c1} - \overline{\theta}_s)\cdot \exp\left(-\kappa_0 \cdot \chi \cdot f(\dot{m}_c)\frac{\dot{m}_{c0}}{\dot{m}_c}\right) \qquad (7).$$

The relative error of the cooling water energy balance gives an indication on the accuracy of this approximation:

$$\frac{\dot{Q}_c - \tilde{\dot{Q}}_c}{\dot{Q}_c} = \frac{\dot{m}_c \cdot c_c \cdot (\theta_{c1} - \theta_{c2}) - \dot{m}_c \cdot c_c \cdot (\theta_{c1} - \tilde{\theta}_{c2})}{\dot{m}_c \cdot c_c \cdot (\theta_{c1} - \theta_{c2})} \qquad (8).$$

After inserting Eqn. (6) and Eqn. (7) into Eqn. (8) the relative error is described by

$$\left|\frac{\dot{Q}_c - \tilde{\dot{Q}}_c}{\dot{Q}_c}\right| = \frac{e^{-\chi\cdot\kappa} - e^{-\kappa}}{1 - e^{-\kappa}} \qquad (9).$$

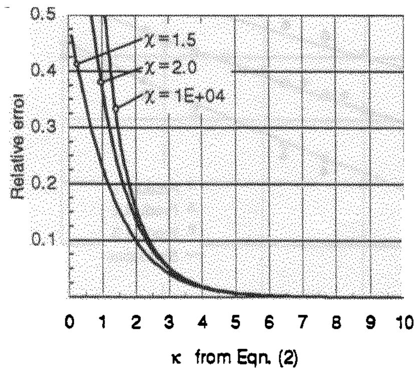

***Figure 8** Relative error of the cooling water energy balance as a function of* κ

As shown in Fig. 8, a very large overestimation of the real κ_0 $(\chi >> 1)$ gives an acceptable accuracy if $\kappa > 3$. Most cooling towers used in HVAC applications comply with this condition. Thus, with $\chi >> 1$ in Eqn. (7) the cooling water outlet temperature is approximated by

$$\theta_{c2} = \overline{\theta}_s \qquad (10).$$

In the next section the air temperature at the air-water-interface and the cooling water outlet temperature are calculated .

At the air-water-interface the air and water temperature are equivalent as shown in Fig. 6

$$\theta_a(0) = \overline{\theta}_s$$

Considering Eqn. (10) and Eqn. (11)

$$\theta_a(0) = \theta_{c2} \qquad (12).$$

The air stream through the cooling tower takes energy from the spray water by convection and evaporation:

$$\dot{q} = \alpha_a \cdot [\theta_a(0) - \theta_a(\infty)] + K \cdot [x_a^{*}(0) - x_a(\infty)] \cdot r \qquad (13).$$

Eqn. (13) can also be written on the basis of enthalpy terms as follows:

$$\dot{q} = K \left\{ \frac{\alpha_a}{c_{pw} \cdot K} \cdot [h_a^{*}(\theta_{c2}) - h_a(\theta_a(\infty))] \right\}$$

$$K \cdot \left\{ \left(1 - \frac{\alpha_a}{c_{pw} \cdot K}\right) \cdot [x_a^{*}(\theta_{c2}) - x_a(\theta_a(\infty))] \cdot r \right\}$$

The dimensionless term in Eqn. (14) is called the Lewis number:

$$Le = \frac{\alpha_a}{c_{pw}\ K}$$

If the air and the spray water temperature do not differ too much, the Lewis number becomes $Le \approx 1$ and we obtain from Eqn. (14)

$$\dot{q} = K \cdot [h_a^{*}(\theta_{c2}) - h_a(\theta_a)]$$

Eqn. (16) describes the energy flow from the spray water film to the air stream through the cooling tower. Therefore, the following energy balance must be satisfied:

$$\dot{m}_a \cdot dh = K \cdot \left[h_a^{''}(\theta_{c2}) - h_a(\theta_a)\right] \cdot dA \qquad (17).$$

We obtain after integration of Eqn. (17)

$$\zeta = \frac{K \cdot A}{\dot{m}_a} = \ln\left[\frac{h_a^{''}(\theta_{c2}) - h_a(\theta_{a1})}{h_a^{''}(\theta_{c2}) - h_a(\theta_{a2})}\right] \qquad (18)$$

or for the initial conditions

$$\zeta_0 = \frac{K_0 \cdot A}{\dot{m}_{a0}} = \ln\left[\frac{h_a^{''}(\theta_{c2_0}) - h_a(\theta_{a1_0})}{h_a^{''}(\theta_{c2_0}) - h_a(\theta_{a2_0})}\right] \qquad (19).$$

The coefficient K can be expressed with $Le = 1$ according to Eqn. (15) as

$$K = \frac{\alpha_a}{c_{pw}} \qquad (20).$$

Eqn. (20) shows that the magnitude K depends on the air side heat transfer coefficient. Assuming that the nominal air side heat transfer coefficient changes only in function of the air mass flow, the following exponential approach can be applied:

$$\alpha_a = \alpha_{a0} \cdot \left(\frac{\dot{m}_a}{\dot{m}_{a0}}\right)^{\gamma} \qquad (21)$$

or as a consequence

$$K = K_0 \cdot \left(\frac{\dot{m}_a}{\dot{m}_{a0}}\right)^{1} \qquad (22).$$

For tube bundle heat exchangers, a value of $\gamma = 0.6$ can normally be used [5].

After inserting Eqn. (22) into Eqn. (18) and simplifying with Eqn (19), we rearrange for $h_a^{''}(\theta_{c2})$, and obtain

$$h_a^{''}(\theta_{c2}) = \frac{h_a(\theta_{a2}) - h_a(\theta_{a1}) \cdot \exp\left[-\zeta_0 \cdot \left(\frac{\dot{m}_a}{\dot{m}_{a0}}\right)^{\gamma-1}\right]}{1 - \exp\left[\quad \left(\quad\right)^{\gamma-}\right]} \qquad (23)$$

or further simplified with the air flow side energy balance

$$h_a^{''}(\theta_{c2}) = h_a(\theta_{a1}) + \frac{Q_c}{\dot{m}_a \cdot \left(1 - \exp\left[-\zeta_0 \cdot \left(\frac{\dot{m}_a}{\dot{m}_{a0}}\right)^{\gamma-1}\right]\right)} \qquad (24).$$

The saturated air temperature can be calculated using Eqn. (24) . This is the air temperature at the air-water-interface and therefore according to Eqn. (12) also the cooling water outlet temperature.

Thus, the performance of a closed circuit cooling tower can be easily calculated on the basis of five parameters only. These parameters are available to the planner already in an early stage of the planning process.

4.3 Example:

Nominal condition:

Cooling capacity	500	kW
Cooling water inlet	39	$°C$
Cooling water outlet	34	$°C$
Wet bulb temperature	21	$°C$
Air mass flow	12.1	kg/s

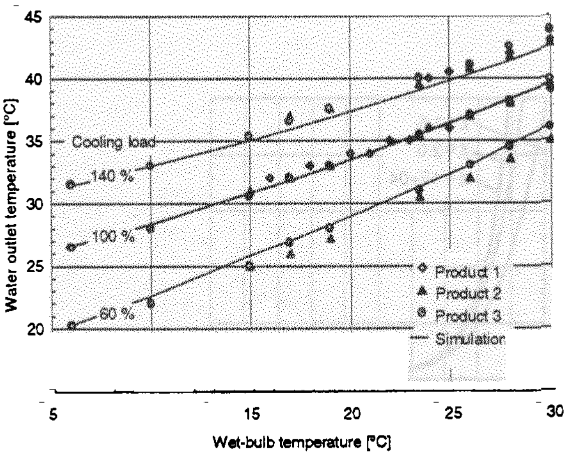

Figure 9 Simulation results for a closed circuit cooling tower compared with data sets from manufacturer's product catalogs

Fig. 9 shows the comparison of the calculated cooling water outlet temperatures as a function of the air wet bulb temperature with manufacturer's measurements. The agreement is fully satisfactory for the application of the model in practice.

5 CONCLUSION

The planning process of today calls on one hand for tools which allow for the integrated determination of the thermal and visual comfort as well as of the indoor air quality. On the other hand, system components modelling ought to comply to the needs of the planner by using parameters which reflect the design problem and which can easily be specified on the basis of manufacturer's catalog data.

The evaluation of design concepts for naturally ventilated and cooled buildings can be greatly improved by simulation which allow for the study of the complex interaction phenomena between the thermal behaviour and the natural ventilation and lighting of a building.

A model for closed circuit evaporative cooling towers, versatile and adjusted to the practitioner's needs, is available within TRNSYS and a future version of DOE-2 for the simulation and dimensioning of highly energy efficient room cooling systems.

6 ACKNOWLEDGEMENTS

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8 NOMENCLATURE

A	Area	m^2
c	Specific heat	$J/(kg \cdot K)$
h	Enthalpy	J/kg
K	Mass transfer coefficient	$kg/(s \cdot m^2)$
Le	Lewis number	-
$\dot{m}$	Mass flow	kg/s
$\dot{Q}$	Rate of heat	W
$\dot{q}$	Specific rate of heat	W/m^2
r	Enthalpy of vaporization	J/kg
U	Overall heat transfer coefficient	$W/(m^2 \cdot K)$
x	Humidity ratio, kg water vapour per kg of dry air	kg/kg
y	Coordinate	m
α	Heat transfer coefficient	$W/(m^2 \cdot K)$
γ	Exponent	-
θ	Temperature	$°C$
κ	Exchanger heat transfer units	-
ξ	Dimensionless number	-

Subscripts:

a	Air
c	Cooling water
s	Spray water
pw	Moist air at constant pressure
1	Inlet
2	Outlet

Superscripts:

0	Nominal condition
"	Saturated condition