Verification and application of a component-based simulation technique for air-handling systems

P.G. Rousseau,[1] E.H. Mathews[2] and A.G. Malan[2]
(First received March 1995; Final version August 1995)

Nomenclature

$a, b_{0.3}$	regression coefficients determined from curve fits
A_f	coil face area [m^2]
A_o	total outside surface area [m^2]
C_o	pressure loss coefficient
C_p	specific heat of the fluid [J/kgK]
D	fan rotor diameter [m]
m_{ai}	air mass flow rate into a component [kg/s]
m_{ae}	air mass flow from a component [kg/s]
m_{li}	measured liquid mass flow rate [kg/s]
h_{ai}	entering enthalpy [J/kgK]
h_{ae}	exiting air enthalpy [J/kgK]
h_o	outside convection heat transfer coefficient calculated from measured parameters [W/m^{2}K]
$h_{o(fit)}$	outside convection heat transfer coefficient obtained from curve fit [W/m^{2}K]
K_h	fan pressure head coefficient
K_f	fan flow coefficient
n	fan rotational speed [r.p.m.]
P_{gage}	gauge pressure relative to atmosphere [Pa]
P_{am}	total pressure rise through fan [Pa]
P_{al}	measured pressure drop on the air side [Pa]
$P_{al(fit)}$	air side pressure drop calculated from curve fit [Pa]
Q_a	measured total heat gain to the moist air [W]
Q_{as}	measured sensible heat gain to the moist air [°C]
$Q_{a(pred)}$	predicted total heat gain to the moist air [W]
$Q_{as(pred)}$	predicted sensible heat gain to the moist air [W]
Q_{total}	total heat gain to air [W]
$Q_{sensible}$	sensible heat gain to air [W]
S	measured sensible heat ratio
$S_{(pred)}$	predicted sensible heat ratio
t_{ai}	measured dry bulb temperature of entering air [°C]
t_{aiwb}	measured wet bulb temperature of entering air [°C]
t_{li}	measured liquid temperature at inlet [°C]
t_{ae}	measured dry bulb temperature of leaving air [°C]
t_{aewb}	measured wet bulb temperature of leaving air [°C]
t_{le}	measured liquid temperature at outlet [°C]
UA	overall heat transfer coefficient
V_{ai}	measured inlet air velocity [m/s]
ρ_{ai}	air density [kg/m^3]
η_{fan}	fan efficiency
η_{motor}	motor efficiency

[1]Department of Mechanical Engineering, Potchefstroom University for CHE, Private Bag X6001, Potchefstroom, 2520 South Africa

[2]Centre for Experimental and Numerical Thermoflow (CENT), Department of Mechanical and Aeronautical Engineering, University of Pretoria, Pretoria, 0002 South Africa. The authors are also consultants for Transfer of Energy, Momentum and Mass International (Pty) Ltd (TEMMI).

Abstract

Forty percent of the total energy used in developed countries is expended in buildings. More than half of this is used to condition the indoor environment. It is estimated that as much as 30% of the energy used for air-conditioning may be saved through the use of improved design tools. Such tools should allow simulation of the building and the air-conditioning system in an integrated manner. This paper describes the verification of a new component based air-handling system simulation technique. The new technique is based on mathematical models each of which represents a different component of the air-handling system. It facilitates the simulation of the building, the heating, ventilation, and air-conditioning (HVAC) system and its controller in an integrated manner. Each individual mathematical component model, as well as the integrated system simulation, was verified by comparing predicted and measured data of the air-handling system of the Merensky Library at the University of Pretoria. An average error of 8% was recorded. Building indoor air temperatures were predicted to within 5% of the measured values. The simulation technique was then applied in a retrofit of this system. The study, which consisted of the addition of evaporative pre-coolers to the air-handling system, showed predicted energy savings of between 12% and 40%. The required capital expenditure has a predicted payback period of less than 3 years. Having been verified and its application possibilities established, the new technique can now be successfully applied to the design and retrofit of other HVAC systems.

Introduction

In 1984 a study by Tuddenham [1] on statistics of 15 countries including the USA, UK, Germany, Australia and South Africa revealed that 40% of their total energy consumption is used in buildings. According to more recent sources [2;3] buildings are responsible for more than 50% of the total energy used in Europe and the United States. In the United Kingdom more than 60% of this building energy is used for air-conditioning.[3] These studies all show that a significant portion of the global energy consumption is expended in conditioning the indoor environment. It is clear that energy efficiency is of major importance in this field. It is commonly believed that some of this energy can be saved through the use of improved design and retrofit techniques.[4] Although it is difficult to determine an exact figure, many researchers believe that HVAC energy savings of at least 30% can be accomplished.[5;2] More optimistic figures are as high as 70% or higher.[6] It appears therefore that significant energy savings may be realised with the aid of an improved HVAC design tool.

Existing HVAC design tools

Literature reviews by ASRAE [7] and Athienitis [8] have shown that present-day HVAC system design tools do not simulate all the components of a building and HVAC system in an integrated manner. This segregated approach has certain deficiencies. It requires fixing the indoor air temperature at an early stage in the design process although fixed air temperatures rarely occur in practice. Many of the most promising energy saving strategies like intermittent operation of the system, dead-band control, and night cooling are also not facilitated.[9]

The actual performance of the HVAC system can therefore only be investigated by simulating the system in an integrated manner. Any HVAC thermal design tool should therefore account for the integrated performance of the complete HVAC system including the building and the controller.[4;10;11;12]

What does integrated simulation mean?

To clearly define what is meant by integrated simulation it is necessary to consider the interaction between the different components of the integrated building system. These include the building, the air-handling unit, the plant and the system controller.

The building is influenced by a number of thermal loads. These include internal loads such as lighting and people, external loads which are a result of the outdoor environment and loads due to the air supplied to the zone.

The air-handling unit, which is responsible for supplying the conditioned air to the building and extraction of 'old' air, is influenced by the building inside conditions (in the form of return air from the building and controller set-points) and the condition of the outdoor air (in the form of fresh air supply). The operation of the air-handling unit is also influenced by the plant as this will limit the capacity of the unit. The building is thus influenced indirectly by the plant through the supply air produced by the air-handling unit.

The operation of the plant is directly and indirectly influenced by the building and air-handling unit through controllers. Its operation is also affected by the outdoor air condition which determines the efficiency of the condenser. This can either be through the operation of an air-cooled condenser or in the case of a water-cooled condenser through the operation of the cooling tower through which the condenser water flows.

It is clear that the different components of the integrated building system interact with each other. Integrated simulation takes into account this interdependence by completing the calculations for each model, including the controller algorithms, at each time step before moving onto the next time step. There is therefore no decoupling of the calculations and each of these calculations is carried out in the same time loop.

Why will integrated simulation lead to energy savings?

Simulating the building, HVAC system and its control in an integrated manner can make an important contribution towards better energy efficiency. It will not only enable the

designer to achieve better designs, but also to better the energy efficiency of existing air-conditioning systems. The reason for this is three-fold:

- If the energy cost savings potential could be illustrated more clearly to building owners and managers, it would be afforded more attention. Schliesing [13] concluded that '... the single most important need in building operation is more convincing evidence of the economic aspects ...'. A well-verified integrated simulation tool will provide such evidence.
- Simulation is seen as the most flexible [10] and probably also the most acceptable way to encourage and enforce building energy norms. The norms will compare the simulated energy consumption of any new budding with that of a so-called 'base building'.
- ASRAE [7] maintains that the results achieved with integrated simulation '... would definitely be superior to those presently available ...'. Holte,[l 1] a consulting HVAC engineer with 25 years of experience, also states: 'It is my firm belief that, for us to build the buildings that we must have, we will have to integrate all building components and systems' and '... a change in one can affect the others. It is this interaction that we must fully understand ...'. Only by simulation will we be able to investigate all these complicated interactions.

This paper describes the theory, verification and application of a new simulation tool that can aid in achieving these goals. The air handling system of the Merensky Library at the University of Pretoria was used for the purpose of this study.

Merensky Library HVAC system and operation

The building is served by three separate air-handling plants, each plant supplying conditioned air to all six floors of the building. Each floor is in turn divided into three fictitious zones, each theoretically served by a separate dual conduit air-handling unit. For simulation purposes the layout of the three units was exactly alike. Figure 1 shows the layout of such an air-handling plant.

As shown in Figure 1, only a portion of the air fed to the zone actually passes through the cooling coil. This is referred to as the cold supply air which is a mixture of fresh outdoor air and return air from the zone. The warm supply air consists of return air.

The mixture of fresh and return air supplied to the coil is controlled by the so-called economiser which consists of two dampers. The economiser compares the return and outdoor air temperatures and mixes the supply air so that the lowest possible cooling load is required from the coil.

The cold supply air temperature from the cooling coil is controlled at approximately 11°C. This is the task of the bypass valve situated in the chilled water supply which is maintained at approximately 8°C.[14]

The hot supply air damper ensures that the total air flow rate to the zone remains constant. This is done by maintaining a constant pressure in the mixed air duct in front of the outlet diffuser. Since the pressure in the zone is relatively constant, a constant pressure before the diffuser ensures a fixed pressure drop across the diffuser. The diffuser outlet area is also fixed and therefore the flow rate remains constant.

The pressure at the outlet of the cold supply air fan is maintained at a constant value of approximately 300 Pa by vortex dampers situated in the fan inlet. The mixture of cold and hot supply air to the zone is controlled by a damper in each supply. The cold supply air damper is controlled such as to maintain the zone air temperature at approximately 23°C. The following section discusses the technique used to simulate the air-handling system.

HVAC system simulation technique

Simulation philosophy

The HVAC system simulation makes use of building blocks each of which represents a different component. Each component has one or more inputs and outputs between which the working fluid, in this case air, is processed. These components include HVAC system components such as cooling coils, ducts, fans, dampers, etc. The building zone itself is also viewed as a component in the system simulation.

The system simulation procedure balances the system network such that the output parameters from each building block equals the input parameters of the next building block. The mathematical component models ensure the correct relation between the input and output variables of each building block.

The HVAC component models are based on the so-called simplified fundamental principle approach. These models are completely component based and employ the principles of mass and energy conservation together with thermodynamic and psychrometric properties to predict flow rates, pressures and temperatures into and out of each component. In some cases empirical correlations are used instead of complex theory such as heat transfer coefficients for heat exchangers. This approach facilitates the simulation of almost any equipment configuration which is ideal for design and probably the only approach applicable to retrofit studies.[15]

The controller is simulated in the same time loop on top of the combined system in an integrated manner. The outputs of the system simulation (such as temperatures, pressures and flow rates) are used as inputs to the controller. The outputs of the controller, such as damper and fan-speed settings, are based on the controller input variables and the control algorithms. Solving all these variables in an iterative manner results in a completely integrated simulation of the building, HVAC system and controller.

It is important to note that a quasi-steady approach is followed. This means that the steady state solution of the

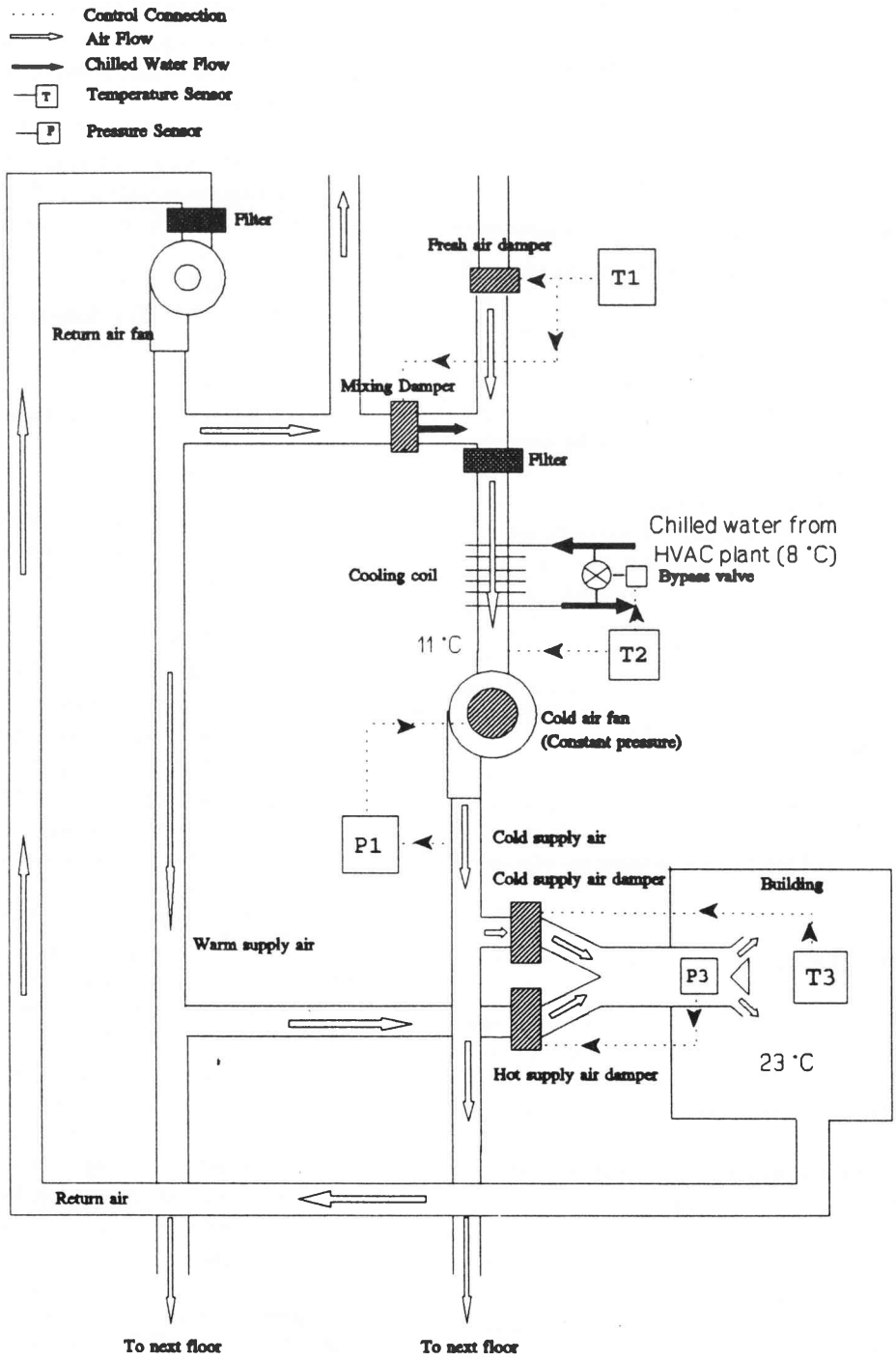


Figure 1 Schematic representation of the dual conduit air-handling unit

system equations is obtained for each time step throughout a 24 h day. In this case a relatively constant indoor air temperature is maintained in the building and the effect of the thermal storage is therefore negligible.

Figure 2 shows the process followed to employ this simulation philosophy. The first step is to determine the empirical correlation coefficients required in the pre-compiled [9] component models for each specific component in the system. This characterisation is done from either measured or catalogue data.

Once the coefficients for the correlations are determined, it is wise to verify each model separately. This is done by inserting measured input variables (such as temperature, pressure, etc.) into the mathematical model. The simulated output variables (again temperatures, pressures, etc.) are then compared to the measured output variables and the accuracy established.

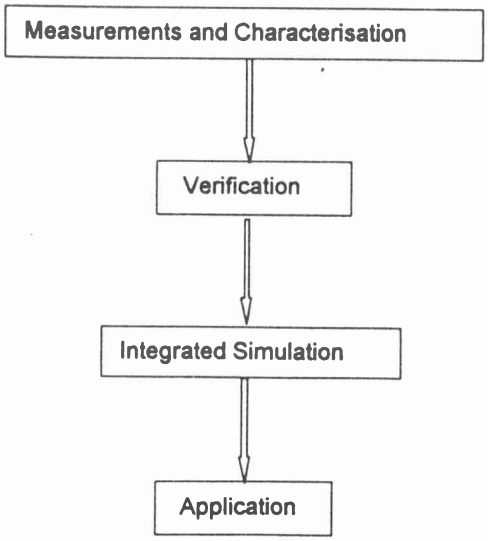


Figure 2 Schematic representation of the simulation methodology

With the models verified, they are linked together in the system network in the same geometrical configuration as the actual HVAC system being studied. The network is then applied by solving for the dynamic variables at each point in the integrated system.

Retrofit or design studies can now easily be performed by *inter alia* swapping components, inserting new components, or changing the control algorithms.

The theory used to model each of the component models is discussed in the next section.

Theory of the HVAC component models

In this section a summary of the equations describing each mathematical model will be given. For a detailed derivation of these equations see Rousseau.[9]

Cooling coil

The total heat transfer rate (Q_a) is given by

$$Q_a = m_{ai}\,(h_{ae} - h_{ai}) \tag{1}$$

with m_{ai} the air mass flow rate and h_{ai} and h_{ae} the inlet and outlet air enthalpies, respectively. It can also be found from

$$Q_a = \frac{h_o A_o}{S}\,\frac{(t_{le} - t_{ai}) - (t_{li} - t_{ae})}{\ln\left(t_{le} - t_{ai}\right) / \left(t_{li} - t_{ae}\right)} \tag{2}$$

with h_o the heat transfer coefiicient, A_o the overall heat transfer area, S the sensible heat ratio, t_{ai} and t_{ae} the air inlet and outlet temperatures, respectively, and t_{li} and t_{le} the inlet and outlet temperatures of the chilled water supplied by the air-conditioning plant. The heat transfer coefficient is given by

$$h_o = a_o + a_1 V_{ai} + a_2 V_{ai}^2 \tag{3}$$

with a_o to a_2 suitable regression coefficients determined from measured data.

The total heat extracted from the air must also be equal to the heat absorbed by the chilled water. Therefore

$$Q_a = m_{li} C_p\,(t_{li} - t_{le}) \tag{4}$$

with m_{li} the water flow rate and t_{li} and t_{le} the inlet and outlet water temperatures, respectively. Equations (1), (2) and (4) are solved together with the appropriate psychrometric equations in an iterative manner to determine the water and psychrometric air outlet conditions for any given set of inlet conditions.

The air pressure drop through the cooling coil (P_{al}) is given by

$$P_{al} = b_o + b_1 V_{ai} + b_2 V_{ai}^2 \tag{5}$$

with b_o to b_2 suitable regression coefficients.

Fan

The pressure rise (P_{am}) through the fan is determined from

$$P_{am} = k_h \rho_{ai} n^2 D^2 \tag{6}$$

with ρ_{ai} the inlet air density, n the rotational speed of the fan and D the rotor diameter. The pressure head coefficient (K_h) is a non-dimensional parameter determined through basic dimensional analysis techniques. For a specific fan, a correlation can be found between this coefficient and the so-called flow coefficient (K_f), which is also non-dimensional. A simple polynomial correlation is usually sufficient in the following form:

$$K_h = a_o + a_1 K_f + a_2 K_f^2 + a_3 K_f^3 \tag{7}$$

with a_o to a_3 again suitable regression coefficients. The flow coefficient (K_f) is defined as

$$K_f = \frac{m_{ai}}{\rho_{ai} n^2 D^3} \tag{8}$$

with m_{ai} the air mass flow rate. The total heat absorbed by the air passing through a fan with the motor mounted in the air stream is

$$Q_a = \frac{(1 - \eta_{fan}\eta_{motor})\, m_{ai} P_{am}}{\rho_{ai}\eta_{fan}\eta_{motor}} \quad (9)$$

with η_{fan} the efficiency of the fan and η_{motor} the efficiency of the motor.

The fan efficiency can also be correlated with K_f as follows:

$$\eta_{fan} = b_o + b_1 K_f + b_2 K_f^2 + b_3 K_f^3 \quad (10)$$

with $b_o ... b_3$ suitable regression coefficients determined from measured data. The equation relating the power of the fan-motor combination is as follows:

$$Pwr = \frac{m_a dP_a}{\rho_{ai}\eta_{fan}\eta_{motor}} \quad (11)$$

Ducts, filters and dampers

The model for each of these components essentially gives the pressure drop (dP_a) as a function of the air speed in the following form:

$$dP_a = \frac{1}{2}\rho_{ai} C_o V_{ai}^2 \quad (12)$$

with V_{ai} the air speed and C_o the pressure loss coefficient. The value of C_o is dependent on each specific component. C_o can be found from either measured data, catalogue data or from empirical tables such as those listed in ASHRAE's Fundamentals handbook.[7]

Building zone

A comprehensive simulation of the building zone would require a large number of inputs (such as the wall geometry, window sizes, climatic conditions, etc.) as well as a very detailed load calculation procedure. In order to simplify the simulation of the building zone for the purpose of this investigation, the cooling loads were not calculated but again determined from measurements for a typical summer day. This of course implies that the building simulation will only be valid for the typical building operating conditions during which the measurements were conducted.

The inputs to the building model are the air flow rate into the zone as well as the condition of the air entering the zone. The specified building loads are then used in the model to determine the condition of the air leaving the zone and entering the HVAC system.

The air inlet enthalpy (h_{ai}) and outlet enthalpy (h_{ae}) are related by the equation for the total cooling load (Q_{total}) as follows:

$$Q_{total} = m_{ai}\,(h_{ae} - h_{ai}) \quad (13)$$

with C_p the specific heat of air.

Measurements and characterisation

This section focuses on the derivation of the empirical correlation coefficients needed in the mathematical models discussed above. These coefficients were all derived from measurements conducted on the actual building and HVAC system components. The values for the correlation coefficients are given in Appendix A.

Cooling coil

In order to obtain the correlation coefficients for the heat transfer coefficient in equation (3) and the pressure drop in equation (5) the following parameters were measured on the cooling coil at different operating conditions: the air mass flow rate, the inlet and outlet conditions of the air (both temperature and relative humidity), the pressure drop through the coil, the mass flow rate of the chilled water and the water inlet and outlet temperatures. The face area and overall heat transfer area must also be known.

By employing equations (2) and (4) together with the appropriate psychrometric relations, h_o can be derived for different values of V_{ai}. Figure 3 shows the measured values.

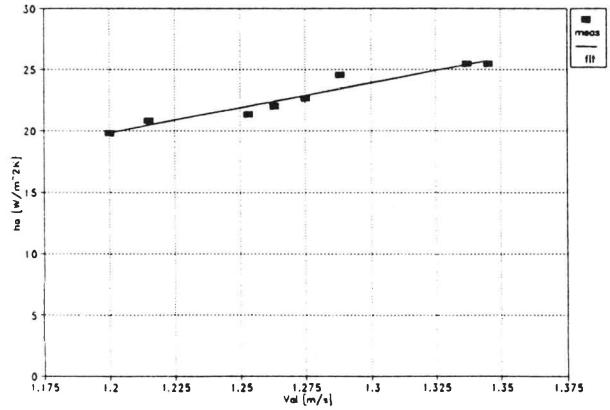


Figure 3 Values for h_o versus V_{ai} obtained from both the measurements and the curve fit

Figure 4 shows measured values for P_{ai} versus V_{ai} as well as the curve fit.

Fan

In order to obtain the correlation coefficients which define the relation between K_h and K_f in equation (7) as well as the correlation between η_{fan} and K_f in equation (10), the following parameters were measured on the return air and cold air fans at different operating points: the air mass flow rate, the static pressure rise through the fan, the fan rotational speed and the electric power used by the fan motor. The rotor diameter and fan outlet area must also be known.

By employing equations (6) and (8) the measured values for K_h versus K_f were determined. Figure 5 shows the relation as well as the curve fit. Figure 6 shows values for η_{fan} versus K_f as well as the curve fit.

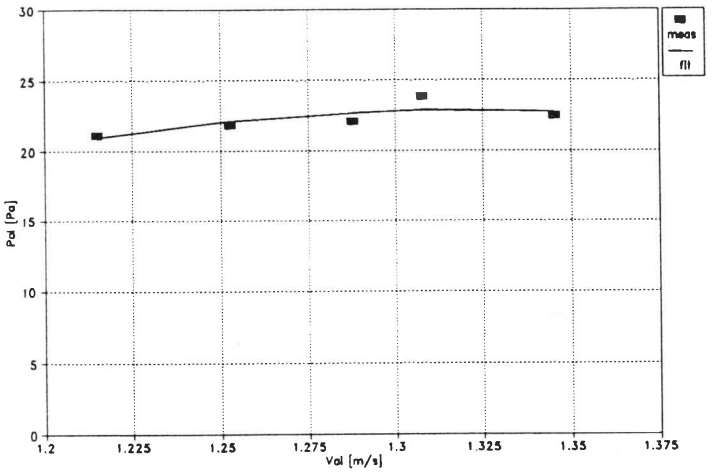


Figure 4 Values for P_{ai} versus V_{ai} obtained from both measurements and the curve fit

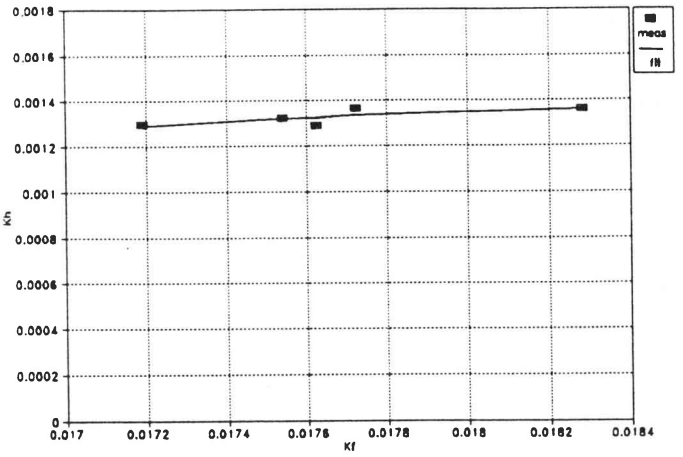


Figure 5 Values for K_h versus K_f obtained from both measurements and the curve fit

Ducts, filters and dampers

The pressure loss coefficient C_o for each duct section as well as each filter was determined by measuring the flow rate and pressure drop. Equation (12) was then employed to solve for C_o.

The pressure and flow rates through each damper are dictated by the damper setting. The values of C_o for the dampers were therefore variables in the simulation and determined by the system controller. The final values for were obtained as part of the simulation results.

Building zone

In order to obtain the total and sensible heat loads of the zone, the air inlet and outlet conditions (both temperature and relative humidity), as well as the air mass flow rates, were measured.

By employing equations (13) and (14) together with the appropriate psychrometric relations, the total and sensible heat loads were determined. The measurements, which were conducted on a typical summer day, indicated that the loads remained relatively constant for the greatest part of the day. For this reason a constant value was specified for both the total and sensible building loads. The zones had an average total and sensible heat load of 94 kW and 84.6 kW, respectively.

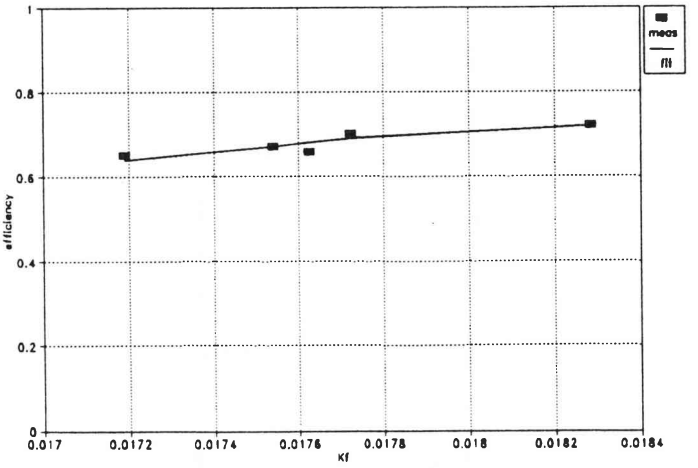


Figure 6 Values for η_{fan} versus K_f obtained from both measurements and the curve fit

Having modelled the components in the HVAC system the simulation can now be constructed and implemented.

Integrated HVAC system simulation

From the system operation, as explained in the paragraphs above, it is clear that the controller ensures certain fixed values of pressure, flow and temperature at specific points. For this reason the simulation of the controller was given special attention.

As stated in Section 4, the pressure and flow rate through each damper is dictated by the damper setting. This is simulated in the damper model by varying the loss coefficient C_o.

The controller is simulated by varying the damper settings in an iterative manner until the specified values of temperature, pressure, etc. are obtained. This means that at first the damper settings are guessed and then, for that iteration, assumed to be fixed. The system network equations are then solved to find the flow rates, temperatures, pressures and power consumption of each component. If the resultant controlled flow rates are not correct, the next iteration is initiated by changing the damper settings. The network is again solved and checked. This process continues until all the specifications are met. Once the solution is complete the cooling load on the coil and the energy consumption of the fans can be extracted.

The simulation procedure is broken up into two main parts, namely, (i) air flow solver; and (ii) temperature solver.

Solving airflows

A building block diagram such as the one in Figure 7 is useful in simplifying the development of the integrated sim-

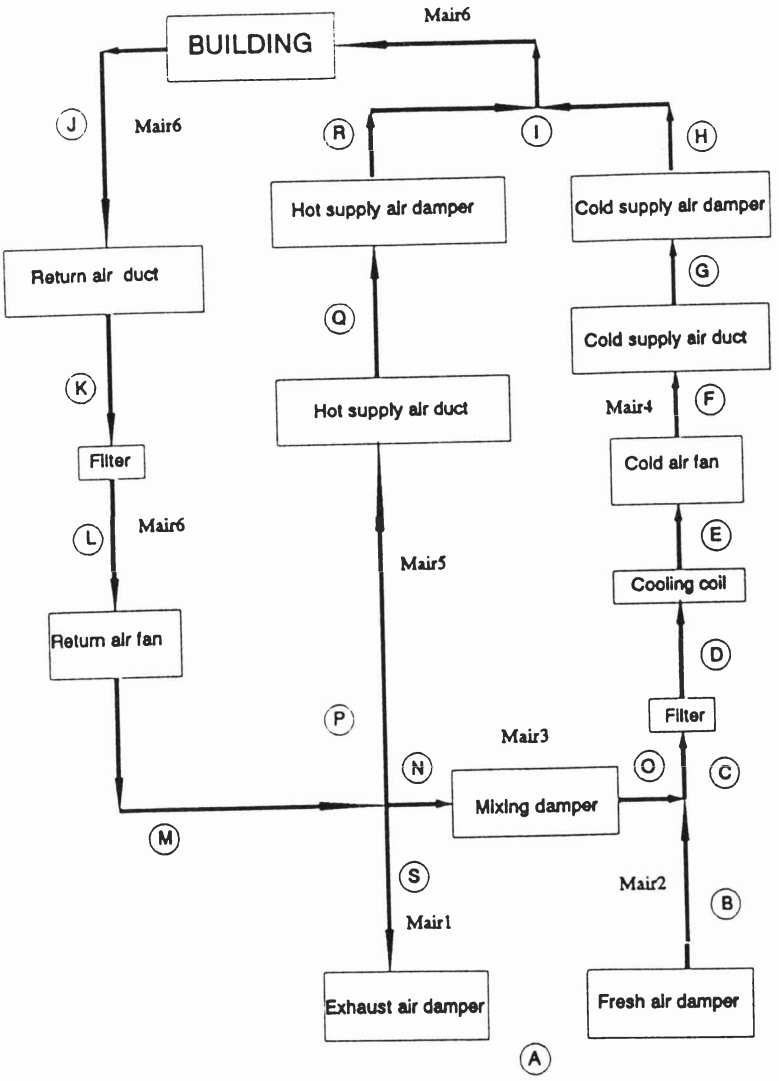


Figure 7 Building block diagram of the air-handling unit

ulation. Note that the diagram is a representation of the system layout shown in Figure 1. If one however considers its operation, as described in the previous paragraph, this diagram may, for the purpose of the simulation, be simplified considerably.

The cold supply air damper setting has to be guessed in order to solve the air flows for the first iteration. The cold air fan controller maintains point F at a constant pressure, while I is kept at a constant pressure by the hot supply air damper controller. The pressure drop from points F to I would be $P_{\mathrm{F_{gage}}} - P_{\mathrm{I_{gage}}}$ which is also the sum of the pressure drops through the elements between points F to I.

$$P_{\mathrm{F_{gage}}} - P_{\mathrm{I_{gage}}} = \frac{1}{2}\rho_{\mathrm{ai}} C_{\mathrm{oFG}} V^2_{\mathrm{air_{PG}}} + \frac{1}{2}\rho_{\mathrm{ai}} C_{\mathrm{oGI}} V^2_{\mathrm{air_{GI}}} \quad (14)$$

The mass flow through the cold supply air duct is given by

$$m_{\mathrm{air}} = (\text{Air density}) * (\text{Flow velocity}) * (\text{Flow area}) \quad (15)$$

With the pressures known, the air flow through the cold supply air duct is given by

$$m^2_{\mathrm{air_4}} = \frac{P_{\mathrm{F_{gage}}} - P_{\mathrm{I_{gage}}}}{\frac{C_{\mathrm{oFG}}}{2\rho_{\mathrm{ai}}(\text{Flow area})^2_{\mathrm{FG}}} + \frac{C_{\mathrm{oGI}}}{2\rho_{\mathrm{ai}}(\text{Flow area})^2_{\mathrm{GI}}}} \quad (16)$$

It can be assumed that the air loss through doors and windows in the building is minimal which means that m_{air1} must equal m_{air2} to satisfy mass conservation. There are thus five unknown variables left in the air loop namely m_{air2}, m_{air3}, m_{air5}, m_{air6} and the damper setting (C_{oQR}).

The block diagram can therefore now be simplified as shown in Figure 8. Since we are at present only interested in determining the flow rates through each component, we may discard the components through which the flow rate is known, i.e. m_{air4}.

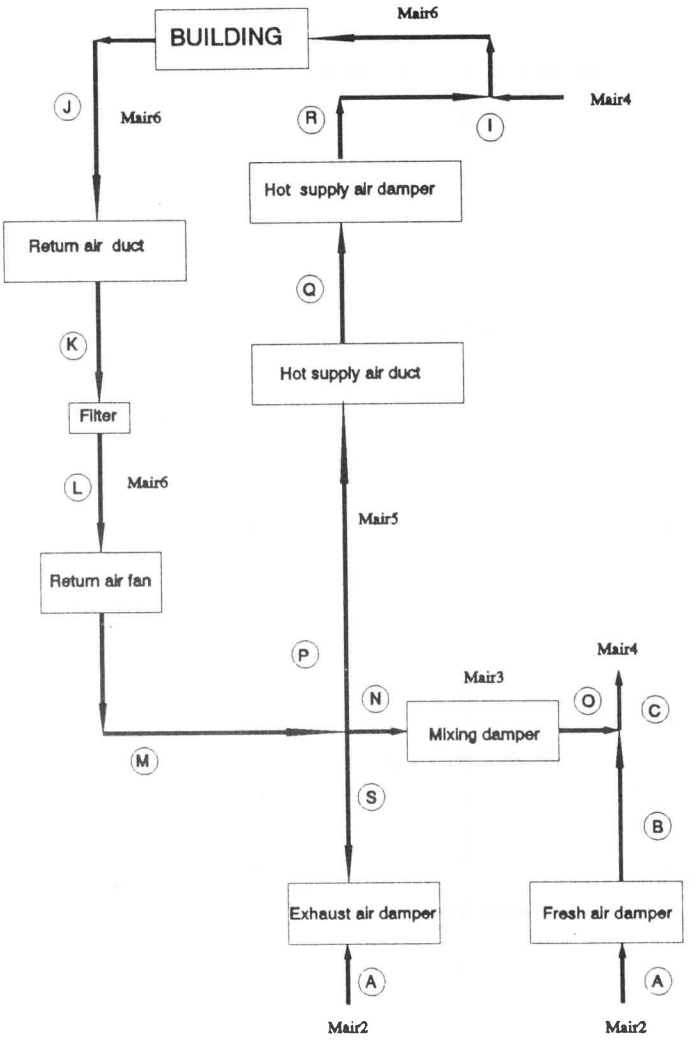


Figure 8 First simplified circuit

On careful examination of this diagram, it can be reduced to two simple flow paths incorporating various branches. The flow patterns are shown in Figure 9.

Using Kirchoffs [16] voltage and current law analogues, the following equations can be written from Figure 9.

$$m_{air4} = m_{air2} + m_{air3} \quad (17)$$

$$m_{air6} = m_{air2} + m_{air5} + m_{air3} \quad (18)$$

$$m_{air6} = m_{air4} + m_{air5} \quad (19)$$

$$0 = P_{A_{gage}} - P_{al_{AB}} + P_{al_{ON}} + P_{al_{MJ}} - P_{J_{gage}} \quad (20)$$

$$0 = P_{A_{gage}} - P_{al_{AS}} + P_{al_{PR}} - P_{I_{gage}} \quad (21)$$

These equations may be solved simultaneously using the Secant numerical method [17] and by keeping in mind that the pressure drop through each component is known as a function of its flow rate. The next step is to determine the temperatures at each node.

Solving temperatures

In solving the temperatures a successive approximation numerical method is required. The process involves guessing the air wet bulb and dry bulb temperature at any node in the network, and then marching around the network back to the initial node. If the guessed temperature differs from the newly calculated one, the process is repeated, using the newly calculated temperature as a starting point. When the calculated temperatures stabilise the building inside temperature is evaluated.

A 486 Personal Computer takes in excess of ten seconds to complete the whole operation, i.e. flows and temperatures.

Verification of integrated simulation

Each component model was first verified separately by comparing simulating output variables to the measured output variables. An average error of 8% with a maximum error of 25% was recorded. The integrated simulation was then verified by comparing the predicted indoor air temperatures with measured values for a chosen day. Figure 10 shows how the predicted temperatures compared with the actual conditions on a typical summer day. The average deviation was less than 5%.

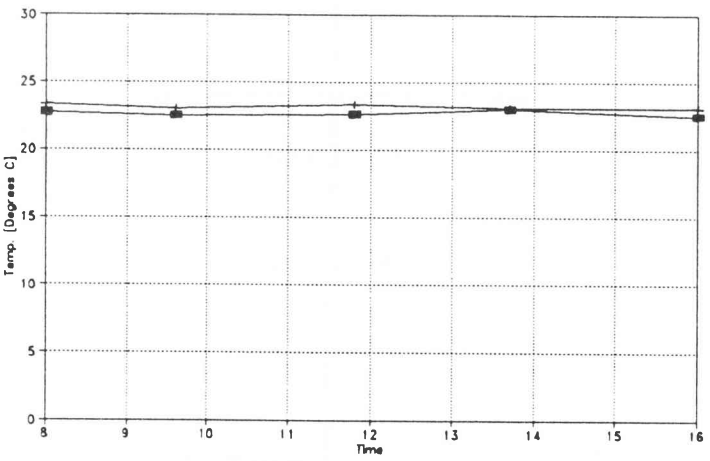


Figure 10 Predicted and measured temperatures inside the building on a typical day

Application study

The application study comprised proposing an economically viable method to reduce the HVAC system's energy consumption. By experimentation with the simulation software it was found that the addition of indirect evaporative pre-coolers at point D (Figure 7) would substantially better the system's energy efficiency. The tables list a few of the simulated outputs for the retrofitted air-handling plants.

The retrofit cost was estimated at R75 000 and, with the average energy savings being almost 30%, would have a payback period of 3 years.

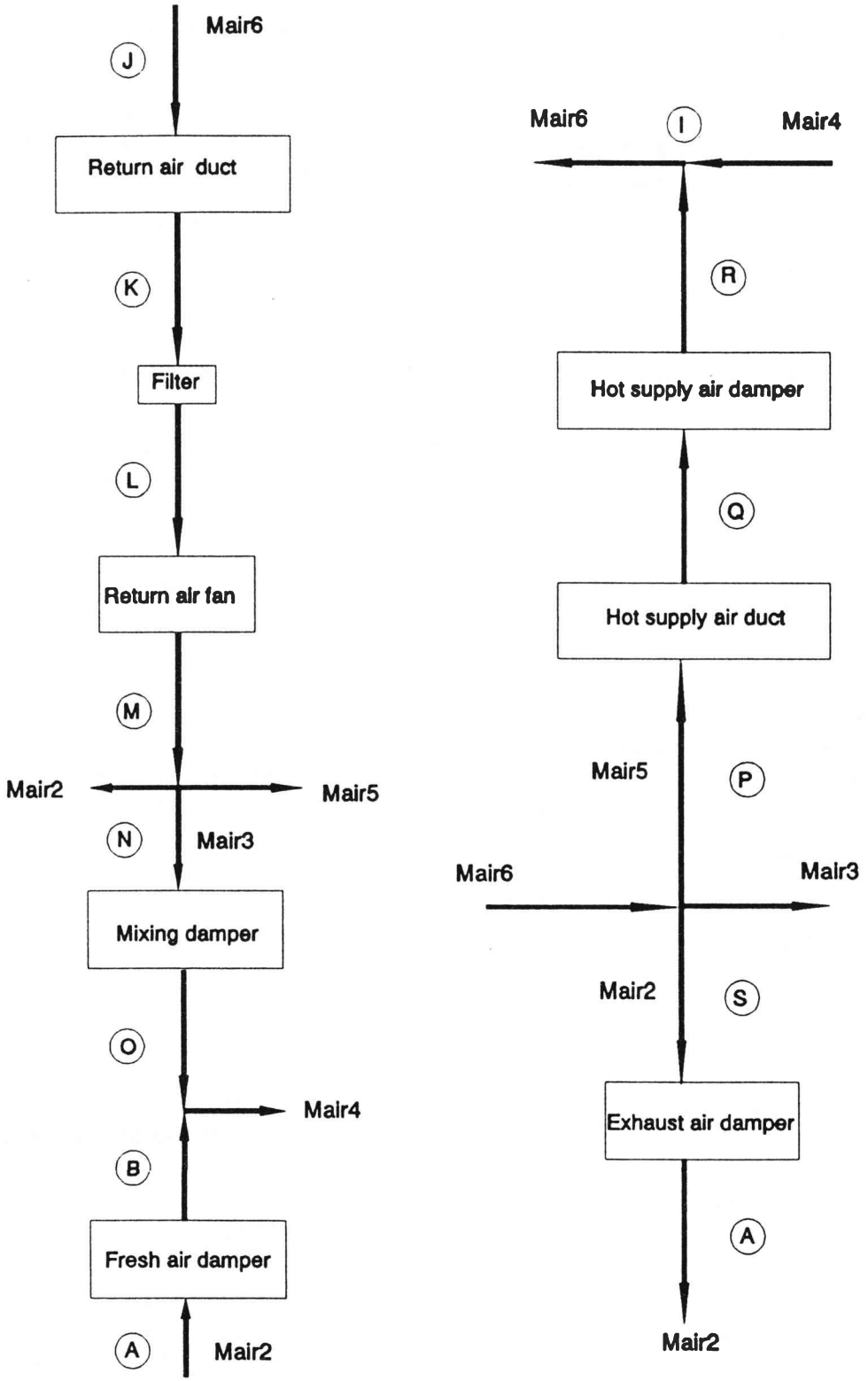


Figure 9 Final simplification of the system block diagram

♦ South-Eastern Plant

Outside temp. [°C]	Outside humidity	Pred. inside temp. [°C]	Pred. inside humidity	Energy saved* [kW]	% Energy saved*
16	55%	23.13	37.07%	3.19	23.64
20	45%	22.97	39.00%	6.64	31.47
26	40%	22.84	40.94%	8.25	20.34
28	35%	22.89	40.73%	8.25	20.01

♦ South-Western Plant

Outside temp. [°C]	Outside humidity	Pred. inside temp. [°C]	Pred. inside humidity	Energy saved* [kW]	% Energy saved*
16	55%	22.73	42.80%	13.38	42.80
20	45%	22.75	42.46%	12.33	27.95
26	40%	23.02	41.14%	8.45	17.67
28	35%	23.00	40.90%	7.7	16.02

♦ Central Plant

Outside temp. [°C]	Outside humidity	Pred. inside temp. [°C]	Pred inside humidity	Energy saved* [kW]	% Energy saved*
16	55%	22.71	41.77%	13.77	34.29
20	45%	22.76	42.55%	12.77	29.02
26	40%	22.91	40.40%	9.00	29.02
28	35%	22.90	41.28%	8.53	17.71

* The energy consumption of the chiller plant was approximated by multiplying the cooling coil load with the plant's effective COP which was determined from measurements.

Conclusion

The aim of this project was the verification and application of a component-based simulation technique for air-handling systems. The air-handling plant of the Merensky Library at the University of Pretoria was used as subject for this study.

The so-called simplified fundamental principal component based models, which form the building blocks for the new technique were successfully verified. They gave an average error of only 8%. Integrated simulation software was then developed with which the indoor air temperature was simulated to within 5% of the measure values.

An energy savings application study was performed. It comprised the addition of evaporative precooling to the air-handling units. The simulation predicted average energy savings of 30%, which would result in a payback period of less than three years.

Having been successfully verified and its application possibilities established, the new technique can now be applied to the design and retrofit of other HVAC systems to aid in attaining more energy efficient designs.

References

[1] Tuddenham D. Computers in air-conditioning load estimation. In: *Air-conditioning System Design for Buildings*, AFC Sheratt (ed.). McGraw-Hill, London, 1983.

[2] Clarke JA. *Energy simulation in building design*, Adam Hilger, Bristol, 1985.

[3] Gandrille T & Hammond GP. An intermediate-level model of external convection for building energy simulation. *Energy and Buildings*, 1988, **12**, 53–66.

[4] Rousseau PG & Mathews EH. *Needs and trends in building and HVAC thermal design tools.* Centre for Experimental and Numerical Thermoflow, University of Pretoria, 1993.

[5] Bevington R & Rosenfeld AH. Energy for buildings and homes. *Scientific American*, 1990, 39–45.

[6] Aires-Barros L. The energy efficiency revolution: formula for success, *International Journal of Global Energy Issues*, 1990, **2**, 111–116.

[7] American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. *Fundamentals Handbook.* Atlanta, 1989.

[8] Athienitis AK, Stylianou M & Shou J. A methodology for building thermal dynamics and control applications. *ASHRAE Transactions*, 1990, Pt 2, 839–848.

[9] Rousseau PG. Integrated building and HVAC thermal simulation. PhD thesis, University of Pretoria, 1994.

[10] Taylor ST. An overview of standard 90.1-HVAC systems and equipment. ASHRAE J., 1990, 26–35.

[11] Holte DE. All building systems must be fully integrated. *ASHRAE Journal Official Product and Show Guide*, 1993, S232.

[12] Thumann A. *Fundamentals of energy engineering.* Prentice-Hall, New Jersey, 1984.

[13] Schliesing JS, O'Neill PJ, Halverson MA & Bramley MR. Building operation: opportunities for improvement. ASHRAE J., 1993, 38–43.

[14] Von Willich & Vrba. Specifications for the air-conditioning installations, Merensky Library, University of Pretoria.

[15] American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc. *Annotated guide to models and algorithms for energy calculations relating to HVAC equipment.* Atlanta, 1990.

[16] Edminister JA. *Theory and problems of electric circuits.* McGraw-Hill, 1983.

[17] Mathews JH. *Numerical methods.* Prentice-Hall International Editions, 1987.

Appendix A

Appendix A contains regression coefficients used to model the heat exchangers and fans of the air-handling units of the Merensky Library at the University of Pretoria.

Heat exchanger: Cooling and dehumidifying coil

$h_o = a_o + a_1 V_{ai} + a_2 V_{ai}^2$

a_o	a_1	a_2
−26.109	36.075	1.829

$P_{al} = b_o + b_1 V_{ai} + b_2 V_{ai}^2$

b_o	b_1	b_2
−276.11	452.49	−171.19

Fans

Each air-handling unit has two centrifugal double inlet backward curved blade fans i.e. cold and return air fan.

Cold air fan

$K_h = a_o + a_1 K_f + a_2 K_f^2 + a_3 K_f^3$

a_o	a_1	a_2	a_3
0.09465	−28.188	2815.9	−93613

$\eta_{fan} = b_o + b_1 K_f + b_2 K_f^2 + b_3 K_f^3$

b_o	b_1	b_2	b_3
−5.3259	1368.8	−109047	2838933

Return air fan

$K_h = a_o + a_1 K_f + a_2 K_f^2 + a_3 K_f^3$

a_o	a_1	a_2	a_3
0.17181	−30.642	1765.1	−33797.8

$\eta_{fan} = b_o + b_1 K_f + b_2 K_f^2 + b_3 K_f^3$

b_o	b_1	b_2	b_3
105.48	−18250	1054205	−20204702