# 4. HVAC MODEL DEVELOPMENT AND VERIFICATION

## 4.1 INTRODUCTION

The previous chapter presented the methodology to be followed in developing the energy conversion models related to the air conditioning systems installed in exchanges. This chapter extends upon this by developing a set of models according to the modelling methodology for the process of cooling these buildings. The set of models will enable the complete management of the electrical energy utilised by the air-conditioners.

Conforming to the methodology, the set of models can be broken down into three concurrent *modules*, each describing a different process of the *operational performance* versus *energy cost* relationship described in chapter 3. These three modules are defined as:

- ***Total building heat load***: This is the total heat gained by an exchange building i.e. the total amount of heat energy added to an exchange due to various heat loads or elements e.g. solar radiation, equipment, lights etc.
- ***Required cooling capacity of the air-conditioning equipment***: This specifies the required "size" air-conditioner for an exchange i.e. the cooling capacity required by the air-conditioning equipment for a specific exchange.
- ***Air-conditioner's energy utilisation***: This is the amount of electrical power, and hence energy, consumed by the air-conditioning equipment for a specified exchange and air-conditioner.

## 4.2 BUILDING BLOCKS

To be consistent with the 'Building Block' concept, the outputs of one module are to form the input to the other. This is illustrated in figure 4.1 on the next page. Note that the outputs of the first two modules are thermal energy values (British Thermal Units), whereas those of the third module are electrical energy values (Kilo-watt-hours). From this it is easy to understand the term "energy conversion models".

For the sake of clarity, an additional two modules have been included to provide insight into how the models are to be used to minimise energy costs through DSM activities and tariff selection; it also explains the type of modelling that is to be done i.e. *Linking Energy Models* (discussed in paragraph 3.3.1).

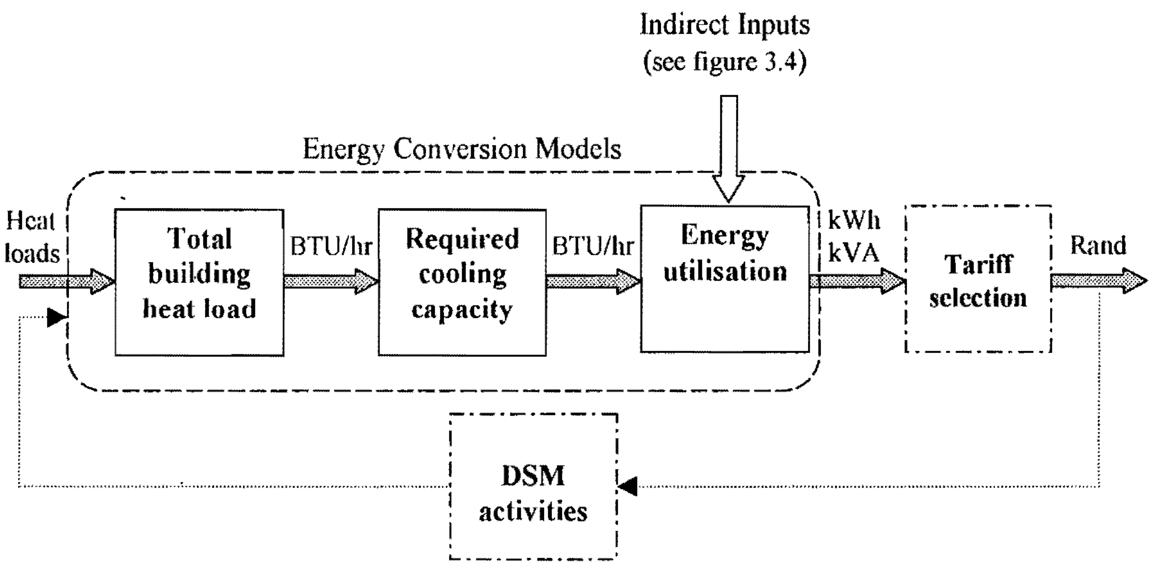


Figure 4.1 Modules forming the model set

Constructing the models in this manner allows experts to use the models relevant to their field of expertise. For example, if the focus were on improving the thermal quality of an exchange, the expert would only need to use the models defining the *total building heat load*. If the models were not constructed in this manner but were combined in a single equation for example, it would be a tedious task extracting the information for a particular field of interest, if even at all possible.

## 4.3 MODELLING ASSUMPTIONS

According to Delport [30, p. 1], when developing models it is usually necessary to make simplifying assumptions – to include all relevant real-world processes will make the model too complex, not only in the development but also in the utilisation of the models. At the same time, it is also important to ensure that an acceptable degree of accuracy is attained by the model outputs, and for this reason assumptions are to be made only if they can be justified. For the modelling at hand, the following assumptions will be made:

- ***There are no temperature gradients:*** it will be assumed that there is a homogeneous temperature throughout the building i.e. the temperature is the same at any region in the building (if one were to measure the temperature on the first floor, for example, and compare it with the temperature on the third floor, there will be no temperature difference). This includes temperatures in adjacent rooms. This assumption is valid as long as there is adequate ventilation throughout the building.
- ***Temperatures are taken at steady state:*** if a parameter that causes the temperature level to vary is changed (e.g. the temperature set point is changed), the interior temperature will undergo a transient and eventually stabilise at some point; the models are developed assuming the temperature is in this region of stability.
- ***Humidity levels remain unchanged:*** it is assumed that the humidity levels inside the exchanges remain constant, even if the dry-bulb temperature is changed. This is a valid assumption as most exchanges have control hardware installed for controlling this.
- ***Air-conditioning equipment is correctly sized:*** for the purpose of the 'energy consumption' module in figure 4.1, it is assumed that the air-conditioner is correctly sized (i.e. its cooling capacity) for the building it has been installed.
- ***Other:*** during the derivation of the models, further assumptions will be made that are more specific to each particular model.

## 4.4 UNITS

For the purpose of this study the following units will be used:

| | | |
|---|---|---|
| **Length** | **Feet** | **[ft]** |
| **Area** | **Feet$^2$** | **[ft$^2$]** |
| **Mass** | **Pounds** | **[lb]** |
| **Temperature** | **Fahrenheit** | **[F]** |
| **Heat** | **British thermal unit** | **[Btu]** |

Making use of these units and not the standard SI units is simply to coincide with tables and charts set up by ASHRAE (American Society for Heat, Refrigeration and Air-conditioning Engineers). Expressing equations in these units simplifies matters considerably since conversions for each element in the tables is not necessary. If it is desired that the final result be in the standard SI unit, then the solution can simply be translated using the following conversions.

$$\mathbf{C = \frac{F - 32}{1.8}}$$

$$\mathbf{0.3\ m = 1\ ft}$$

$$\mathbf{1\ kW = 3410\ Btu/hr}$$

## 4.5 TOTAL BUILDING HEAT LOAD

Figure 4.2 on the next page shows the real-world processes (heat transfers) involved in developing this energy conversion model. Except for $\mathbf{Q_{capacity}}$, which is the cooling load supplied by the air conditioner, all the **Q**'s are either the internal or external heating loads (adding heat) to the exchange. Note that these heat gains can be categorised as being either *internal* (heat sources inside the exchange), or *external heat gains/elements* (occurring outside the building envelope).

Now, the problem at hand is to calculate the total building heat load which consists of all the internal and external heat elements. According to Sauer and Howell [26, pp. 6.1 – 6.4] this is accomplished by summing all the heat elements, internal and external. Thus, from figure 4.2 the total building heat load of a typical exchange is given by equation 4.1.

$$\mathbf{Q_{building} = Q_{solar} + Q_{glass} + Q_{infil} + Q_{wall} + Q_{ceiling} + Q_{part} + Q_{lights} + Q_{people} + Q_{telecom}} \quad [4.1]$$

Where each **Q** is defined as follows:

| | |
|---|---|
| $\mathbf{Q_{building}}$ | total building heat load. |
| $\mathbf{Q_{solar}}$ | heat gain due to solar radiation through windows to the interior. |
| $\mathbf{Q_{glass}}$ | heat gain due to conduction through glass (windows). |
| $\mathbf{Q_{infil}}$ | heat gain due to air infiltrating through cracks (e.g. door edges). |
| $\mathbf{Q_{wall}}$ | heat gain due to conduction through walls. |
| $\mathbf{Q_{ceiling}}$ | heat gain due to conduction through the ceiling. |
| $\mathbf{Q_{part}}$ | heat gain due to conduction through interior wall partitions. |
| $\mathbf{Q_{lights}}$ | heat gain from lights. |
| $\mathbf{Q_{people}}$ | heat gain from people. |
| $\mathbf{Q_{telecom}}$ | heat gain from telecommunication equipment. |

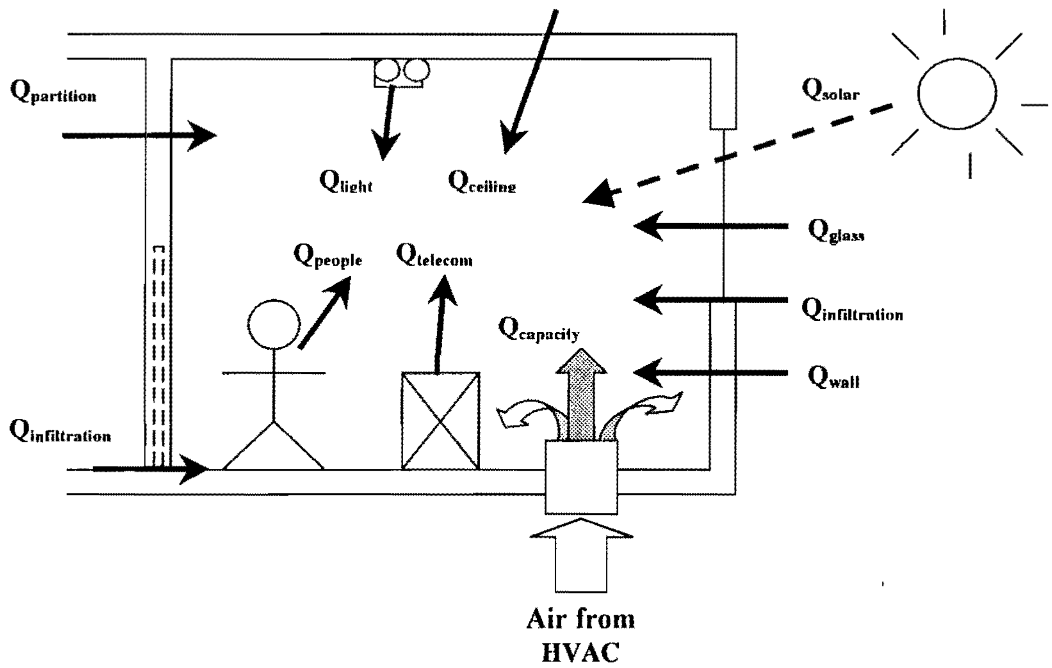


Figure 4.2 Typical heat transfers occurring in an exchange

The assumptions made in paragraph 4.3 enable equation 4.1 to be simplified considerably. That is, the assumption made that there is a homogeneous temperature throughout the building and that no heat transfer occurs through interior partitions (such as from room to room), means that $\mathbf{Q_{part}}$ can be ignored. Most exchanges in South Africa do not have windows in their envelopes, allowing the $\mathbf{Q_{solar}}$ and $\mathbf{Q_{glass}}$ terms to be negated. The equation can thus be rewritten as equation 4.2. All that remains now is to calculate the various heat elements in this equation.

$$\mathbf{Q}_{\text{building}} = \mathbf{Q}_{\text{infil}} + \mathbf{Q}_{\text{wall}} + \mathbf{Q}_{\text{ceiling}} + \mathbf{Q}_{\text{lights}} + \mathbf{Q}_{\text{people}} + \mathbf{Q}_{\text{telecom}} \qquad [4.2]$$

### 4.5.1 Heat Gain Through Walls and Ceilings

A building envelope that consists of walls, roofs, windows and doors is not perfectly heat insulated, as a result heat energy is able to transverse through these structures by means of one or more of the following processes: conduction, convection or radiation. Extrapolating from Young [47, p. 433], when considering the heat transfer through this envelope, a number of considerations have to be taken into account that affect the rate at which heat is added to, or removed from the building:

- *Building size* – The physical dimensions of the building have a direct influence on the rate at which the heat is transferred. The surface area of the walls, roofs, windows, and floors are of importance here. The larger the surface area, the faster the rate of heat transfer.
- *Type of materials used in the construction of the building* – various materials have different properties when it comes to the transfer of heat. The property of a material to resist the flow of heat (from a region of high temperature to a region of low temperature) is known as the *thermal resistance*. The higher the resistance, the slower the transfer of heat.
- *Temperature difference* – this is the change in temperature from one side of a structure (e.g. a wall) to the other side. The higher the temperature gradient, the greater the rate of transfer.

These three factors have a direct influence on the rate at which the heat energy is transferred through a material and can be related by the following equation [48, p. 45]:

$$\mathbf{Q} = \frac{\mathbf{1}}{\mathbf{R}} \times \mathbf{A} \times \mathbf{\Delta T} \qquad [4.3]$$

where $\mathbf{Q}$ is the heat transfer rate [Btu/hr], $\mathbf{R}$ is the thermal resistance [hr-ft$^2$-F/Btu], $\mathbf{A}$ is the surface area [ft$^2$], and $\mathbf{\Delta T}$ is the temperature difference [F].

In many cases the building envelope does not only consist of one type of material, but of many. The thermal resistance of this is a combination of the individual resistances. Take for example a brick wall consisting of bricks, plaster and paint.

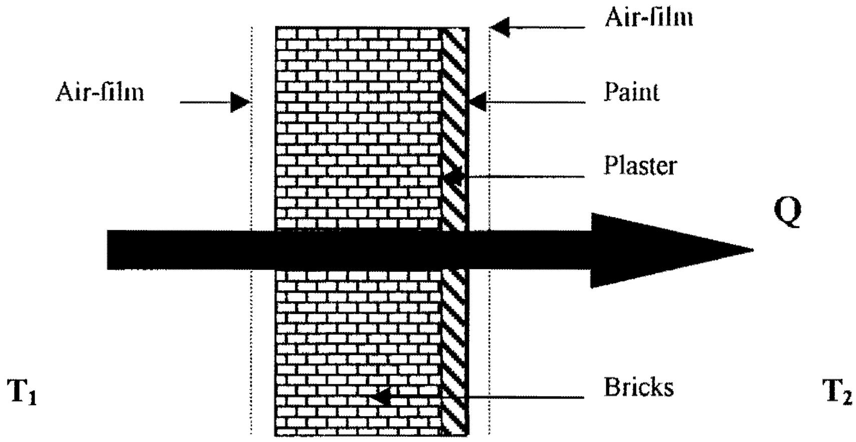


Figure 4.3 Cross-sectional view of a wall

According to Pita [48, p. 46] the total or *overall thermal resistance* is the sum of all the individual resistances of each material. The figure shows that if temperature $T_1$ is greater than $T_2$ the heat will flow through the wall in the direction indicated. Note that there is an air-film on both sides of the wall, the thermal resistance of which must also be taken into account. The overall thermal resistance for this example is therefore:

$$\mathbf{R_T = R_{AIRFILM1} + R_{BRICK} + R_{PLASTER} + R_{PAINT} + R_{AIRFILM2}} \qquad [4.4]$$

ASHRAE have measured the thermal resistances for many building materials, however these values are not given in terms of **R** but rather in terms of the thermal conductance called the *overall heat transfer coefficient* [26, p. 5.7] (***see appendix B for heat transfer coefficients for different building materials – supplied by Pita [48, pp 445 - 449]**) i.e.

$$\mathbf{U = \frac{1}{R_T}} \qquad [4.5]$$

where **U** is the overall heat transfer coefficient [Btu/hr-ft$^2$-F], and $\mathbf{R_T}$ is the total thermal resistance [hr-ft$^2$-F/Btu].

It was mentioned earlier that the rate at which heat is transferred across a material depends on three factors: surface area, thermal resistance and temperature difference. The heat transfer rate for the exchange's walls and ceiling are thus:

$$\mathbf{Q_{wall} = U_w \times A_w \times \Delta T} \qquad [4.6]$$

$$\mathbf{Q_{ceiling} = U_c \times A_c \times \Delta T} \qquad [4.7]$$

where **Q** is the heat transfer rate [Btu/hr], **U** is the overall heat transfer coefficient [Btu/hr-ft$^2$-F], **A** is the surface area [ft$^2$], and **ΔT** is the temperature difference on the opposite sides of the structure [F].

### 4.5.2 Heat Gains Due to Lights

According to Schweitzer and Ebeling [28, p. 2-6] each watt of electrical energy consumed in producing light from fluorescent lamps (used by Telkom) gives off 3.42 Btu/hr. Thus

$$\mathbf{Q_{lights} = 3.42 \times P_L} \qquad [4.8]$$

Where $\mathbf{Q_{lights}}$ is the heat given off due to lighting [Btu/hr], and $\mathbf{P_L}$ is the electrical power consumed by the light sources [W].

### 4.5.3 Heat Gains Due to People

ASHRAE [25, p. 26-8] have devised a list of human activities and their respective heat emissions. For the modelling at hand, where telecommunication exchanges are of interest, it is assumed that the work force is predominantly male, and that the two primary activities carried out by the personnel are (1) "seated, very light work" such as office work, and (2) "light bench work" such as maintenance work done on the telecommunication equipment. These *degrees of activities* give off 444 Btu/hr (130W) and 800 Btu/hr (235W) per adult male respectfully. Therefore

$$Q_{peolpe} = N_p \times \Psi \quad [4.9]$$

Where $\mathbf{Q_{people}}$ is the heat given off by humans [Btu/hr], $\mathbf{N_p}$ is the number of people, and $\Psi$ is the degree of activity [Btu/hr].

### 4.5.4 Heat Gains Due to Infiltration

The infiltration of air through door and window edges obviously also causes an additional heat gain or loss to the exchange. According to ASHRAE [24, p. 25.4] this heat gain to the building can be expressed by the following equation:

$$Q_{infil} = 1.08 \times \beta \times \Delta T \quad [4.10]$$

Where $\mathbf{Q_{infil}}$ is the heat gain due to infiltration [Btu/hr], $\Delta\mathbf{T}$ is the temperature difference [F], and $\beta$ is the cubic feet per minute of infiltrating air [$ft^3$/min].

### 4.5.5 Heat Gains Due to Telecommunication Equipment

It was mentioned in the opening chapter that most of the energy utilised by the telecommunication equipment is converted into heat. More specifically, the average equipment heat load is 2 W/line, which translates into 6.826 Btu/hr/line [41]. Thus,

$$Q_{telecom} = 6.826 \times N_T \quad [4.11]$$

Where $\mathbf{Q_{telecom}}$ is the heat gain due to the telecommunication equipment [Btu/hr], and $\mathbf{N_T}$ is the number of lines.

Now, substituting all these heat loads into equation 4.2 gives the following:

$$\boxed{Q_{building} = [\Delta T \times (U_w A_w + U_c A_c + 1.08 \cdot \beta)] + [3.42 \cdot P_L] + [N_p \cdot \Psi] + [6.826 \cdot N_T]} \quad [4.12]$$

Equation 4.12 can be used to determine the building heat load to an unmanned exchange. However, because the cooling air supplied by the air-conditioners is only concentrated on the exchange rooms themselves and not offices areas, the equation does not completely describe the heat load to a manned exchange. This facet can be included into the model by observing that only the first term in the equation is dependant on temperature, and consequently is the only term that will have a different effect on the heat load. Equation 4.12 can be rewritten to include the effects of having office space in an exchange:

$$\begin{aligned} Q_{building} = &\left[(T_{Out} - T_{Ech}) \times (U_{w\,Out} A_{w\,Out} + U_c A_c)\right] \\ &+ \left[(T_{Offic} - T_{Exh}) \times (U_{w\,Ech} A_{w\,Exh} + 1.08 \cdot \beta_{Exh})\right] \\ &+ \left[3.42 \cdot P_L\right] + \left[N_p \cdot \Psi\right] + \left[6.826 \cdot N_T\right] \end{aligned} \qquad [4.13]$$

where $T_{Out}$ is the outdoor temperature, $T_{Exh}$ is the exchange room temperature, $U_{w\,Out}$ and $A_{w\,Out}$ are the heat transfer coefficient and area of the non-office (exchange room) walls facing the outdoors, $U_c$ and $A_c$ are the heat transfer coefficient and ceiling area of the exchange rooms respectfully, $\beta_{Exh}$ is the infiltration from the offices areas into the exchange room, $T_{offic}$ is the office temperature, $U_{w\,Ech}$ and $A_{w\,Ech}$ are the heat transfer coefficient and areas of the partitioning walls between the office areas and the exchange rooms.

## 4.6 REQUIRED COOLING CAPACITY

According to Schweitzer and Ebeling [28, p. 2-3], the principle used to determine the cooling capacity required by an air conditioner is "*load*" determination i.e. the purpose of an air-conditioner to remove the heat supplied by the various heat loads (as those depicted in figure 4.2) so as to maintain a prescribed temperature. The problem is, how much of this heat must the air-conditioner remove – with too much removed the temperature will become too low; conversely, with too little heat removed the temperature will be too high.

The problem can be solved by observing the first law of thermodynamics that states: *The energy added to a system less the energy removed from the system equals the energy change in the system* [48, p. 31]. This can be expressed by the following:

$$\textbf{Energy Gained} - \textbf{Energy Lost} = \textbf{Change in Energy} \qquad [4.14]$$

It was mentioned that the role of the air-conditioner is to maintain a constant temperature. This implies that the energy content of the air contained within the building remains unchanged. Hence, the 'Change in Energy' term in equation 4.14 can be equated to zero, implying:

$$\textbf{Energy Gained} = \textbf{Energy Lost} \qquad [4.15]$$

For the modelling at hand, the 'Energy Gained' term in equation 4.15 is the sum of the heat gains (Q's) given in figure 4.2; and the 'Energy Lost' term is the energy that the air-conditioner is to remove. From this it is clear that the air-conditioner is to remove an amount of energy (heat) equal to that supplied by the various heat loads. Thus, noting that the total building heat load is calculated in equation 4.12 and 4.13 as $\mathbf{Q_{building}}$, the cooling capacity required by an air conditioner to maintain an energy balance is:

$$\boxed{\mathbf{Q}_{\mathbf{capacity}} = \mathbf{Q}_{\mathbf{building}}} \qquad [4.16]$$

Where $\mathbf{Q_{capacity}}$ is the cooling capacity of an air-conditioner [Btu/hr].

## 4.7 ENERGY UTILISATION

The building heat load will determine how "hard" the air conditioner must operate in order to keep the exchange at a predetermined temperature. More specifically, the cooling capacity [Btu/hr] required to establish an energy balance is dependant upon the building heat load. Once the cooling capacity is known, the electrical input energy to the air conditioning system can be determined using its C.O.P (Coefficient Of Performance), which is defined as [3, p. 16]:

$$\mathbf{C.O.P} = \frac{\mathbf{Useable\ Output\ Energy}}{\mathbf{Electrical\ Input\ Energy}} \qquad [4.17]$$

Grobler [23, p. 80] applies this definition to an air conditioning system; however, the units of his definition have been manipulated to suit the content of this study i.e.

$$\mathbf{C.O.P} = \mathbf{0.293} \times \frac{\mathbf{Q_{capacity}}}{\mathbf{P_{aircon}}} \qquad [4.18]$$

Where $\mathbf{Q_{capacity}}$ is the cooling capacity of the air conditioner [Btu/hr], $\mathbf{P_{aircon}}$ is the power consumed by the air conditioner [W], and the coefficient is as a result of balancing units.

Most air-conditioning systems installed in exchanges throughout South Africa are centralised HVAC systems. These systems consist of two or more air-conditioning units, of which each might have a different C.O.P. With this taken into account, equation 4.19 provides the energy conversion model enabling the calculation of the power consumed by an air-conditioning system.

$$\mathbf{C.O.P} = \mathbf{0.293} \times \frac{\sum_{i=1}^{n} \mathbf{Q_{capacity_i}}}{\sum_{i=1}^{n} \mathbf{P_{aircon_i}}} \qquad [4.19]$$

Where $\mathbf{n}$ is the number of air-conditioning units contained in the HVAC system.

It has been established thus far that in order to maintain an energy balance, the cooling capacity of the air conditioner ($\mathbf{Q_{capacity}}$) must equal the building heat load ($\mathbf{Q_{building}}$). It has also been established that the C.O.P is the ratio of this cooling capacity, to the electrical input energy, in this case $\mathbf{P_{aircon}}$. Therefore, from equation 4.16 the C.O.P is also:

$$\mathbf{C.O.P} = \mathbf{0.293} \times \frac{\mathbf{Q_{building}}}{\mathbf{P_{aircon}}} \qquad [4.20]$$

From this, and equation 4.19 the following useful formula is constructed:

$$P_{tot\ aircon} = 0.293 \times \frac{Q_{building}}{0.293 \times \frac{\sum_{i=1}^{n} Q_{capacity_i}}{\sum_{i=1}^{n} P_{aircon_i}}} \quad [4.21]$$

or

$$P_{tot aircon} = Q_{building} \times \left( \frac{\sum_{i=1}^{n} P_{aircon_i}}{\sum_{i=1}^{n} Q_{capacity_i}} \right) \quad [4.22]$$

The "Electrical Input Energy" used in the equations 4.17 and 4.18 to describe the C.O.P was defined as being *real power* (Watts). However to be more accurate, the *apparent power* (VA's) should be used. Fortunately this feature can be included into equation 4.22 using the *power factor* (the ratio between real power and the apparent power) i.e.

$$\boxed{P_{tot aircon} = \left( Q_{building} \times \left( \frac{\sum_{i=1}^{n} P_{aircon_i}}{\sum_{i=1}^{n} Q_{capacity_i}} \right) \right) \times pf^{-1}} \quad [4.23]$$

Where $\mathbf{P_{tot\ aircon}}$ is the total power consumed by the air conditioning system [VA], $\mathbf{Q_{building}}$ is the total building heat load as given by equations 4.12 or 4.13 [Btu/hr], $\mathbf{P_{aircon\ i}}$ is the power consumed by each air conditioning unit [W], $\mathbf{Q_{capacity\ i}}$ is the cooling capacity of each air conditioning units [Btu/hr], and **pf** is the power factor of the entire air-conditioning system.

The energy utilisation of an air-conditioning system installed in a telecommunication exchange can now be completely described by the models defined by equations 4.12, 4.13, 4.16, and 4.23. Since the output of model 4.23 is dependant on models 4.12 or 4.13, the energy utilisation can be described in terms of aspects such as outdoor weather conditions, building envelope, type of building materials, lights, people and telecommunication equipment.

## 4.8 MODEL VERIFICATION

For the purpose of verifying the proposed energy conversion models, input values gathered from typical exchanges under normal operating conditions were obtained. More specifically two exchanges were used for this purpose; namely *case study one* (CS1) and *case study two* (CS2). For explanatory reasons, CS1 and CS2 made use of an unmanned and manned exchange respectively.

### 4.8.1 Case Study Details

| | **Description** | |
|---|---|---|
| **Detail** | **Case Study 1** | **Case Study 2** |
| ***Name*** | Hillcrest Exchange | Bronberg Exchange |
| ***Location*** | Hillcrest, Pretoria | Sunnyside, Pretoria |
| ***Type*** | Unmanned | Manned |
| ***Audit dates*** | 5 Oct 2000 – 6 Nov 2000 | 23 Jan 2001 – 31 Jan 2001 |
| ***Air-conditioner type*** | 3 x 4.8kW stand-alone units. | 6 x 18.5kW chillers. |
| ***C.O.P*** | 1.49 | 1.62 |
| ***Power Factor*** | 1 | 0.71 |
| ***Wall type*** | 8 in. common brick walls with no insulation. | 12 in. concrete + 8 in. common brick with no insulation. |
| ***Outer wall area [$ft^2$]*** | 2 110 | 5920 |
| ***Office Wall area [$ft^2$]*** | - | 5850 |
| ***Set point [F]*** | 70 | 68 |
| ***Power for lighting [W]*** | 1 430 | 20 000 |
| ***Number of people*** | 1 | 20 |
| ***Degree of activity*** | "Seated, very light work" | "Light bench work" |
| ***Install lines*** | 5 750 | 122 000 |

Table 4.1 Case study details

### 4.8.2 Verification of Models: *Case Study One*

Figure 4.4 below presents a visual comparison between the measured and simulated (using models 4.12, 4.16, and 4.23) power consumptions of the air-conditioning system installed in Hillcrest exchange. The real world values tabulated in table 4.1 were used as the inputs to the models.

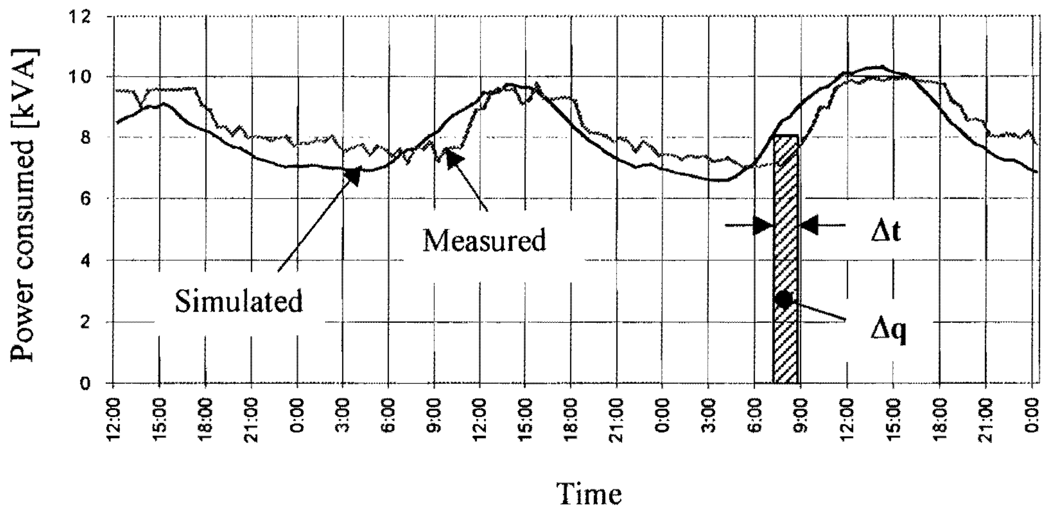


Figure 4.4 Comparison between measured and simulated air-conditioner power consumptions for two and one half days at Hillcrest

Note that the power consumed by the air-conditioners have a cycling period of one day. The reason for this is largely due to the outdoor temperature – at approximately midday the ambient temperature is usually the hottest, thereafter it gradually cools until the next morning when the temperature starts to increase again. Since the building heat load is strongly dependant on the temperature difference between the outdoor and indoor temperature (see $\Delta T$ in equation 4.12), and because the indoor temperature is to remain constant (at 70 °F), it is easy to understand that the building heat load, and hence 'power consumed' (see equation 4.23) tracks the profile of the outdoor temperature.

It is also noted in the figure that a phase-shift, which the models do not take into account exists between the simulated and measured readings because of time delays that occur in the transfer of heat through materials, such as through walls and air. Modifying an equation presented by Halliday and Resnick [49, p. 361] to suit this study, this phase-shift can be corrected by the inclusion of equation 4.24 (presented below) into the models; the formula is graphically explained in the figure.

$$\Delta t = \frac{\Delta q}{U \times A \times \Delta T} \qquad [4.24]$$

where $\Delta t$ is the time delay [hours], $\Delta q$ is the energy transferred during the period $\Delta t$ [Btu], $U$ is the heat transfer coefficient [Btu/hr-ft$^2$-F], $A$ is the surface area [ft$^2$], and $\Delta T$ is the change in temperature [F].

A more detailed evaluation of the data obtained from the audit showed the results listed in table 4.2 below. Note that the percentage errors are marginal, and thus the simulated results generated by the models are a true representation of the actual, real-world process.

| | **Energy Consumption [kWh]** | **Maximum Demand [kVA]** |
|---|---|---|
| ***Measured*** | 6 074 | 9.95 |
| ***Simulated*** | 6 002 | 10.33 |
| ***Difference [%]*** | 1.2 | 3.8 |

Table 4.2 Comparison between measured and simulated results

### *4.8.3* Verification of Models: *Case Study Two*

Case study 2 was used to verify the models for a manned exchange. Bronberg exchange was used for this purpose. Once again, for verification purposes figure 4.5 is presented to provide visual confirmation that the models are a true representation of the real world. As with the case above, the values given in table 4.1 are used as the inputs to the models (in this case equations 4.13, 4.16, and 4.23). An evaluation of the data showed the results listed in table 4.3 below.

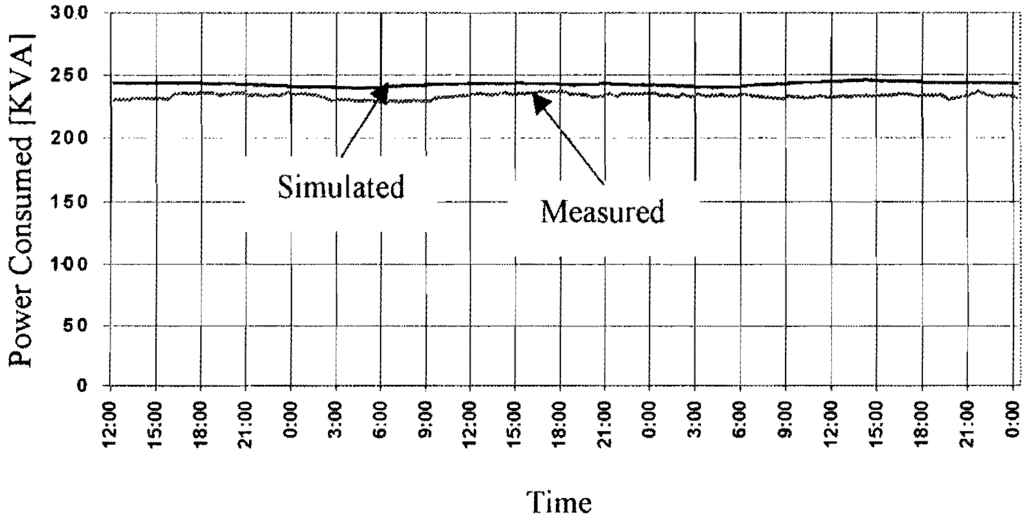


Figure 4.5 Comparison between measured and simulated air-conditioner power consumptions for two and one half days at Bronberg

| | **Energy Consumption [kWh]** | **Maximum Demand [kVA]** |
|---|---|---|
| ***Measured*** | 133 271 | 236.99 |
| ***Simulated*** | 140 837 | 247.96 |
| ***Difference [%]*** | 5.7 | 4.6 |

Table 4.3 Comparison between measured and simulated results

From the case studies it is clear that the proposed energy conversion models are feasible and are able to accurately predict the processes involved in cooling exchanges. It is noted that the load profiles of the simulated results are very similar to the actual (measured) profiles. However, more applicable to DSM is that the simulated results for the *energy consumption* and *maximum demand* are in the order of 5% of the actual values.

## 4.9 SUMMARY

The purpose of the chapter was to develop energy conversion models related to the air-conditioning equipment installed in telephone exchanges that could be used in DSM. Three areas (defined as modules at the beginning of the chapter) of the cooling process were modelled; these collectively described the relationship between *operational performances* and *energy utilisation* of the air-conditioning equipment. As such the models provide a useful tool in the implementation of Energy Management and hence DSM.

The models were verified using two typical exchanges under normal operating conditions. These two exchanges, of which one was a manned exchange and the other an unmanned exchange, formed the basis for two case studies. The results of these case studies showed that the models are extremely accurate at predicting the energy utilisation (consumption and maximum demand levels) of air-conditioning equipment.

Now that the models have been developed and verified, the next step is to illustrate the manner in which they can be used to aid the DSM process. The next chapter is focused on just this, and will clarify how the models can be used as a tool to lower the energy costs incurred as a result of the cooling process.