APPENDIX A

SIMULATION MODELS

This appendix presents all the existing simulation models in QUICKcontrol and typical system curves of certain equipment.

A.1. PUMPS

A.1.1 Assumptions:

- No psychrometric property relations are needed to calculate the liquid density. The density is taken throughout as $\rho_l = 1000$ kg/m^3. Similarly the specific heat at constant pressure will be taken throughout as $c_{pl} = 4187$ J/kg°C.
- The drive motor may not be situated within the liquid flow.
- The dynamic pressure difference over the pump is assumed to be negligible.

A.1.2 Parameters:

a_j	Correlation coefficients for the K_h versus K_f relation with j = 0 to 2.
b_j	Correlation coefficients for the η_{pump} versus K_f relation with j = 0 to 2.

A.1.3 Inputs:

A.1.3.1Simulation:

m_l	Mass flow rate of liquid at inlet [kg/s].
T_{li}	Temperature of liquid entering at inlet [°C].

A.1.3.2Interface:

D	Rotor diameter [m].
N	Rotational speed of the pump [rpm].
η_{motor}	Efficiency of the drive motor.
H	Pressure head [m].
q	Flow rate [kg/s].
η_{pump}	Pump efficiency.

A.1.4 Outputs:

dP_l	Static pressure rise [Pa].
T_{le}	Temperature of liquid leaving at outlet [°C].
Pwr	Pumping power required [W].

A.1.5 Internal variables:

H_1	Extra pressure head point [m].
H_2	Extra pressure head point [m].

K_f	Dimensionless flow coefficient.
K_h	Dimensionless pressure head coefficient.
η_{pump}	Efficiency of the pump.
η_1	Extra efficiency point.
η_2	Extra efficiency point.
Q_l	Rate of heat gain to the liquid [W].

A.1.6 Explicit equations:

Three different pressures and efficiencies at three different flows are needed to calculate the correlation coefficients. A flow variation of 20% above and below the interface input value is assumed. The extra points are then calculated as follow:

$$H_1 = a_h(0.8q)^2 + b_h(0.8q) + c_h$$
$$\eta_1 = a_\eta(0.8q)^2 + b_\eta(0.8q) + c_\eta$$
$$H_2 = a_h(1.2q)^2 + b_h(1.2q) + c_h$$
$$\eta_2 = a_\eta(1.2q)^2 + b_\eta(1.2q) + c_\eta$$

with

$a_h = -0.63125$

$b_h = 1.18125$

$a_\eta = -0.06188$

$b_\eta = 0.33625$ for $0 < q < 3$.

$a_h = -0.25914$

$b_h = 0.591667$

$a_\eta = -0.01816$

$b_\eta = 0.190308$ for $3 < q < 7.5$.

$a_h = -0.3248$

$b_h = 3.736$

$a_\eta = -0.00648$

$b_\eta = 0.133$ for $7.5 < q < 15$.

$a_h = -0.100$
$b_h = 2.197$
$a_\eta = -0.00507$
$b_\eta = 0.159$ for $15 < q < 20.$

$a_h = -0.02674$
$b_h = 0.639$
$a_\eta = -0.00089$
$b_\eta = 0.05089$ for $20 < q < 40.$

$a_h = -0.01505$
$b_h = 0.88875$
$a_\eta = -0.00028$
$b_\eta = 0.029263$ for $40 < q < 80.$

$a_h = -0.00164$
$b_h = 0.262$
$a_\eta = -0.000047$
$b_\eta = 0.010467$ for $80 < q < 150.$

These values were calculated as the average values of the coefficients of a wide range of centrifugal pump curves. These results are shown in figures A.1 to A.14. c_h and c_η are now calculated from the one given operating point obtained from the supplier or measurements. This implies only one operating point is needed to obtain the mathematical model of a specific pump.

And

$$c_h = H - (a_h q^2 + b_h q)$$
$$c_\eta = \eta - (a_\eta q^2 + b_\eta q)$$

For each of these three points the K_h and K_f value must be calculated as follow:

$$K_{h1} = \frac{gH_1}{N^2 D^2}$$

$$K_{f1} = \frac{m_{l1}}{\rho_1 N D^3}$$

$$K_h = \frac{gH}{N^2 D^2}$$

$$K_f = \frac{m_l}{\rho_1 N D^3}$$

$$K_{h2} = \frac{gH_2}{N^2 D^2}$$

$$K_{f2} = \frac{m_{l2}}{\rho_1 N D^3}$$

Using these values the correlation coefficients can be calculated as follow:

$$a_2 = \frac{(K_{h2} - K_{h1})(K_f - K_{f1}) - (K_{f2} - K_{f1})(K_h - K_{h1})}{(K_{f2}^2 - K_{f1}^2)(K_f - K_{f1}) - (K_{f2} - K_{f1})(K_f^2 - K_{f1}^2)}$$

$$a_1 = \frac{(K_h - K_{h1}) - (K_f^2 - K_{f1}^2)a_2}{(K_f - K_{f1})}$$

$$a_0 = K_{h1} - K_{f1}a_1 - K_{f1}^2 a_2$$

$$b_2 = \frac{(\eta_2 - \eta_1)(K_f - K_{f1}) - (K_{f2} - K_{f1})(\eta - \eta_1)}{(K_{f2}^2 - K_{f1}^2)(K_f - K_{f1}) - (K_{f2} - K_{f1})(K_f^2 - K_{f1}^2)}$$

$$b_1 = \frac{(\eta - \eta_1) - (K_f^2 - K_{f1}^2)b_2}{(K_f - K_{f1})}$$

$$b_0 = \eta_1 - K_{f1}b_1 - K_{f1}^2 b_2$$

These coefficients can now be used to calculate the necessary outputs with the following explicit equations:

$$K_f = \frac{m_l}{\rho_l N D^3}$$

$$K_h = a_0 + a_1K_f + a_2K_f^{\,2}$$

$$dP_l = K_h\rho_l N^2 D^2$$

$$\eta_{\text{pump}} = b_0 + b_1K_f + b_2K_f^{\,2}$$

$$Q_l = \frac{(1-\eta_{pump})m_l dP_l}{\rho_l\eta_{pump}}$$

$$T_{le} = T_{li} + \frac{Q_l}{m_l c_{pl}}$$

$$Pwr = \frac{dP_l m_l}{\rho_l\eta_{motor}\eta_{pump}}$$

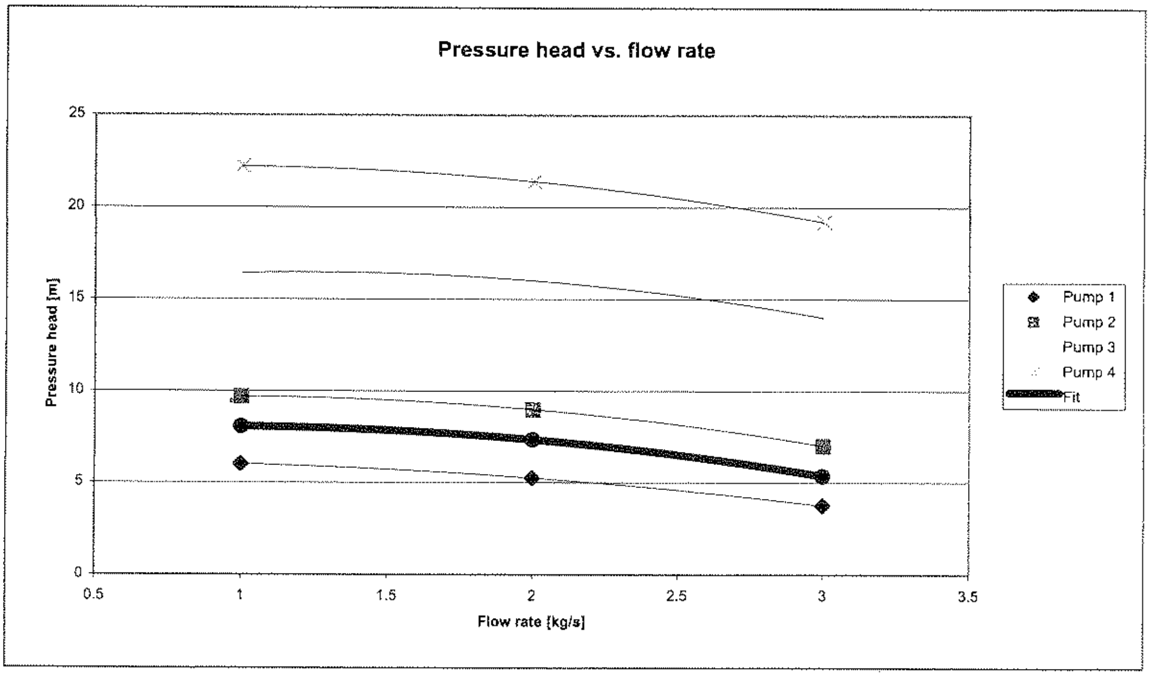

Figure A.1: Pressure head correlation for flow between 0 and 3 kg/s

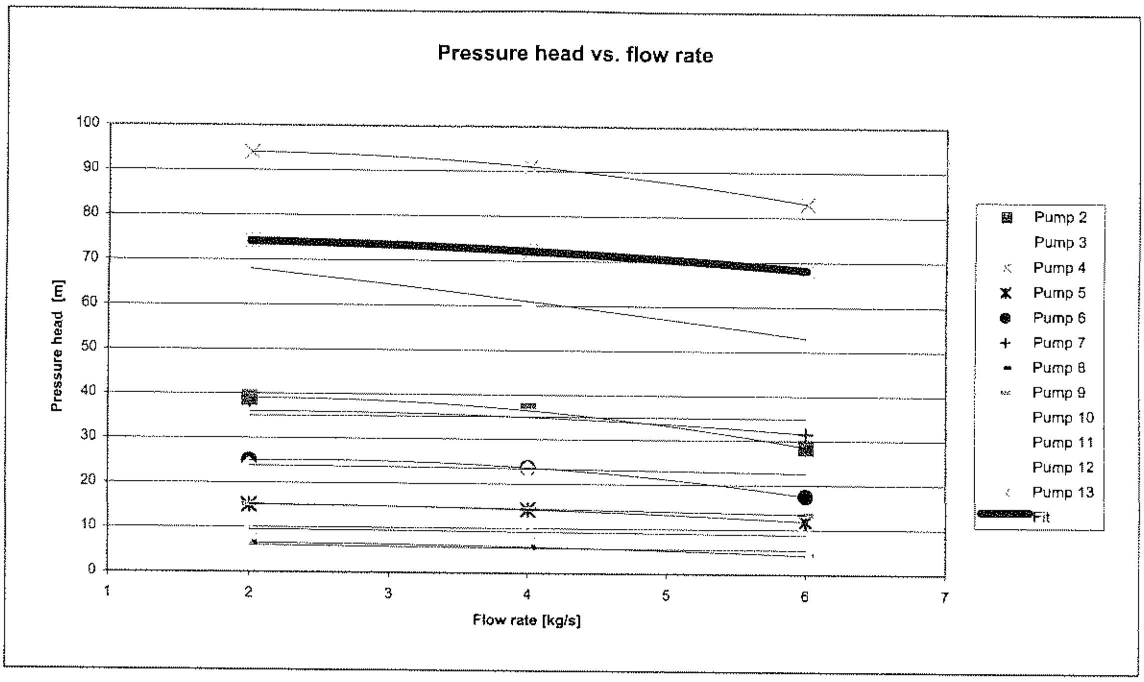

Figure A.2: Pressure head correlation for flow between 3 and 7.5 kg/s

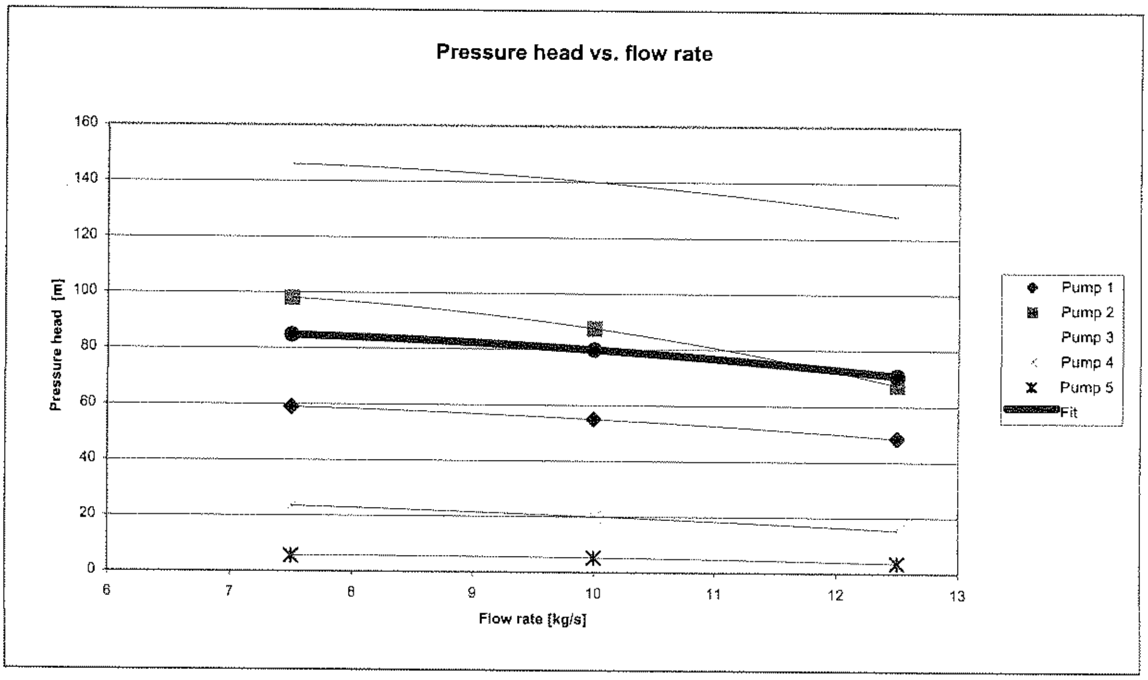

Figure A.3: Pressure head correlation for flow between 7.5 and 15 kg/s

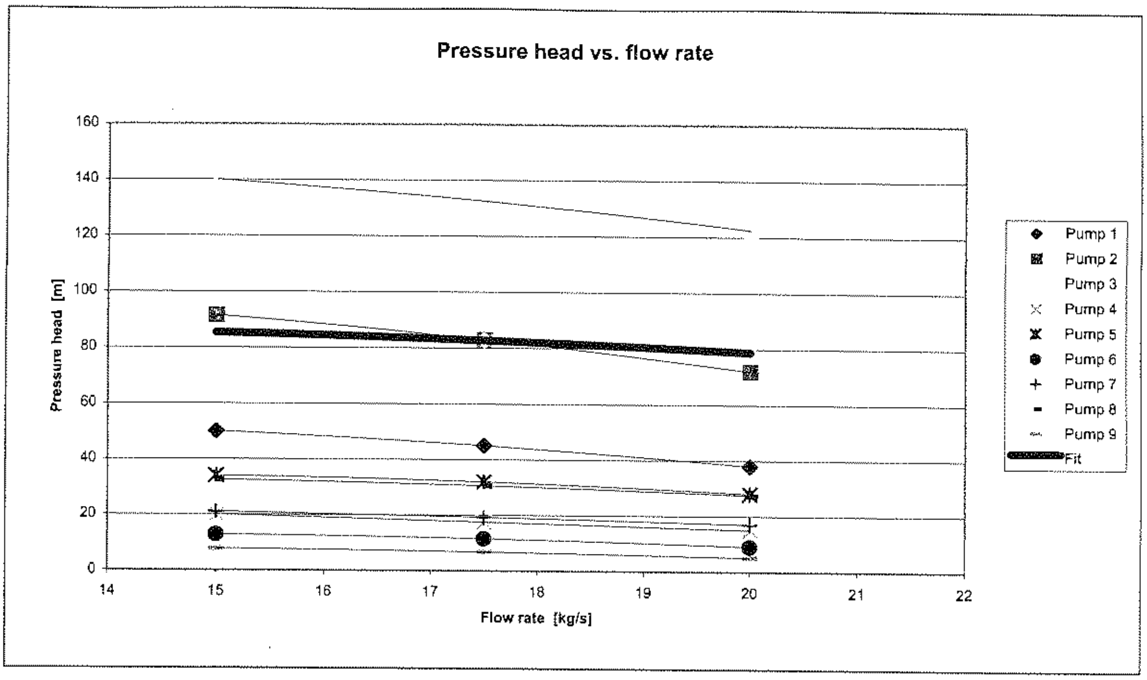

Figure A.4: Pressure head correlation for flow between 15 and 20 kg/s

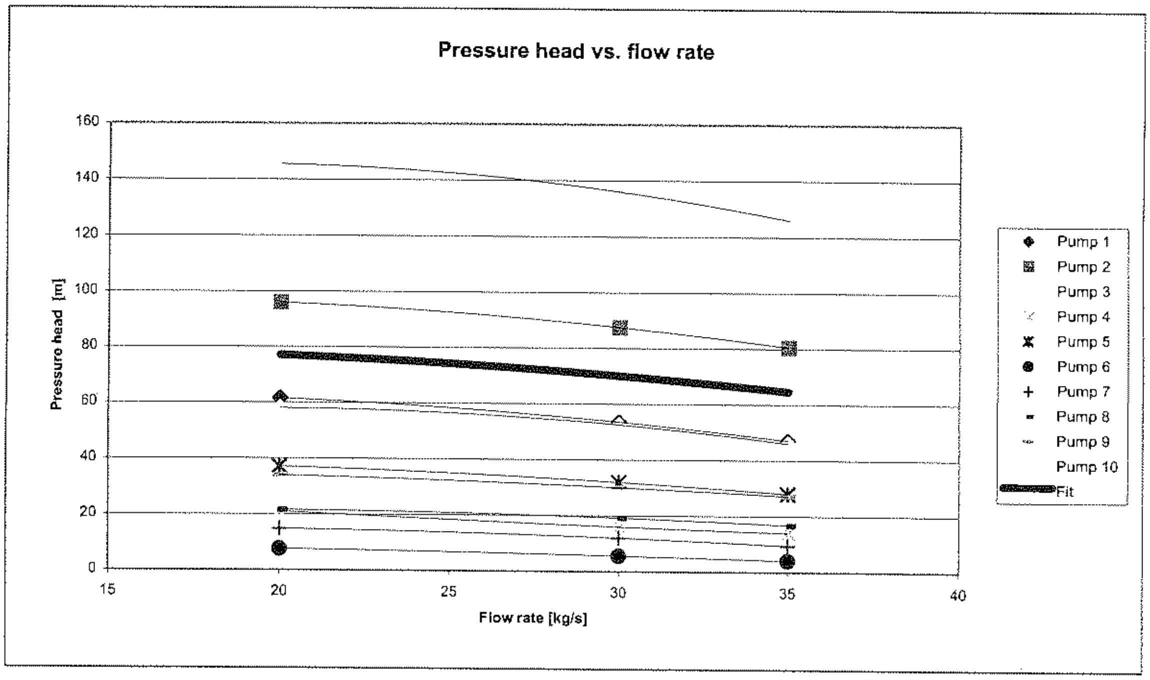

Figure A.5: Pressure head correlation for flow between 20 and 40 kg/s

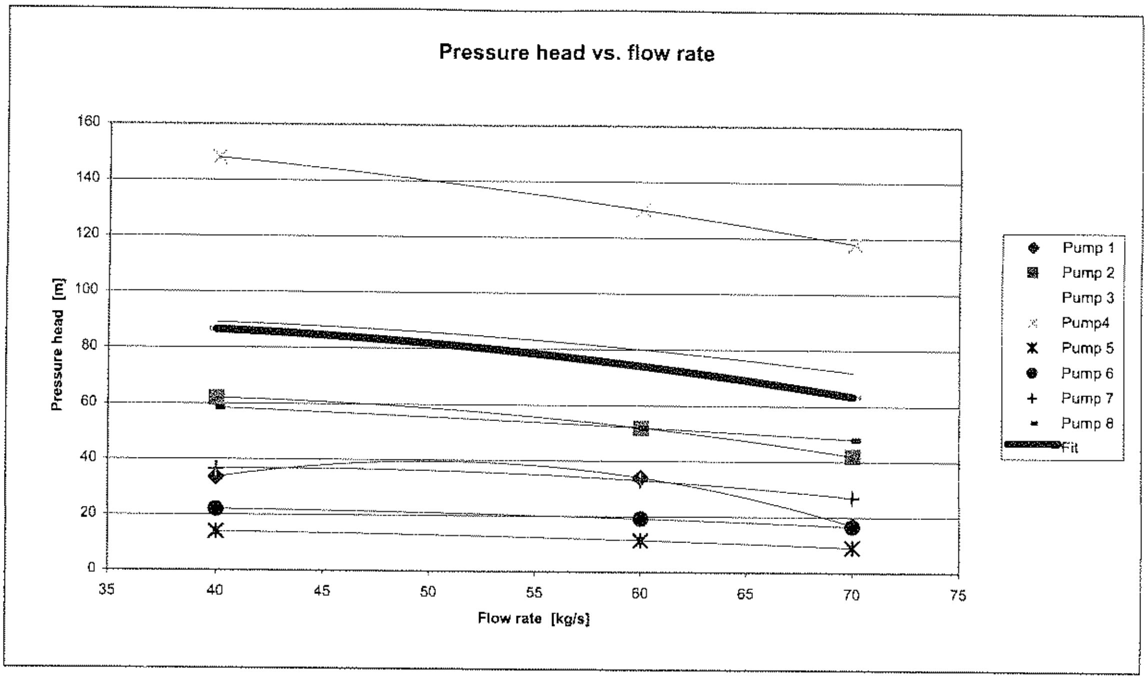

Figure A.6: Pressure head correlation for flow between 40 and 70 kg/s

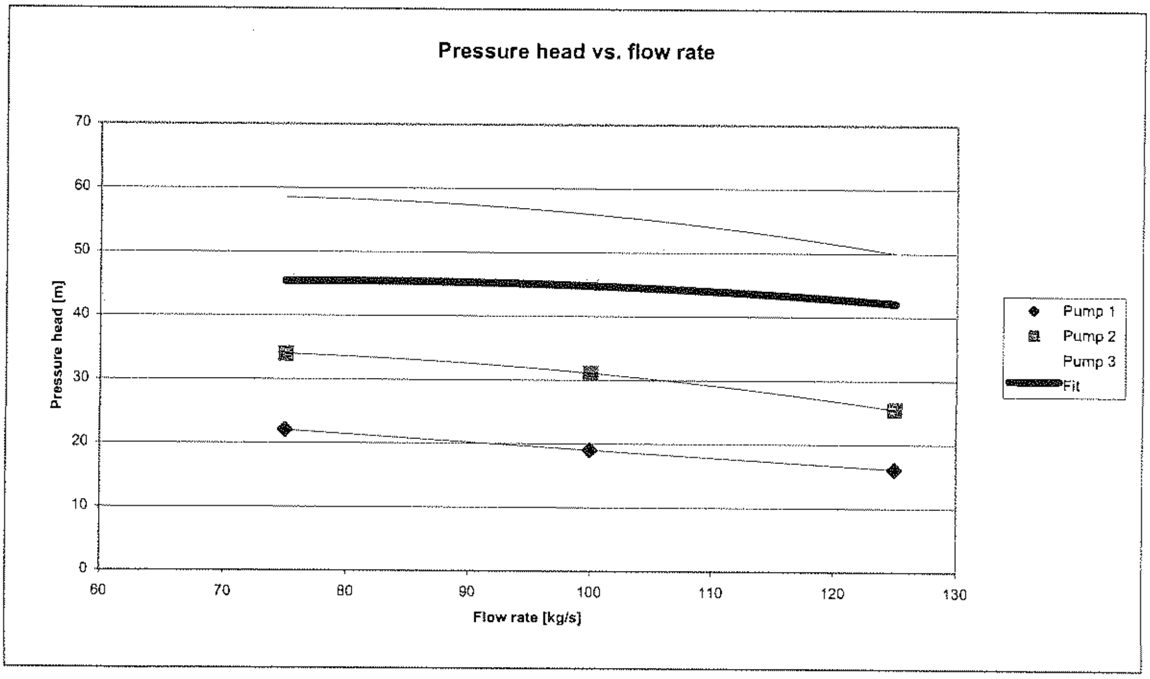

Figure A.7: Pressure head correlation for flow between 70 and 150 kg/s

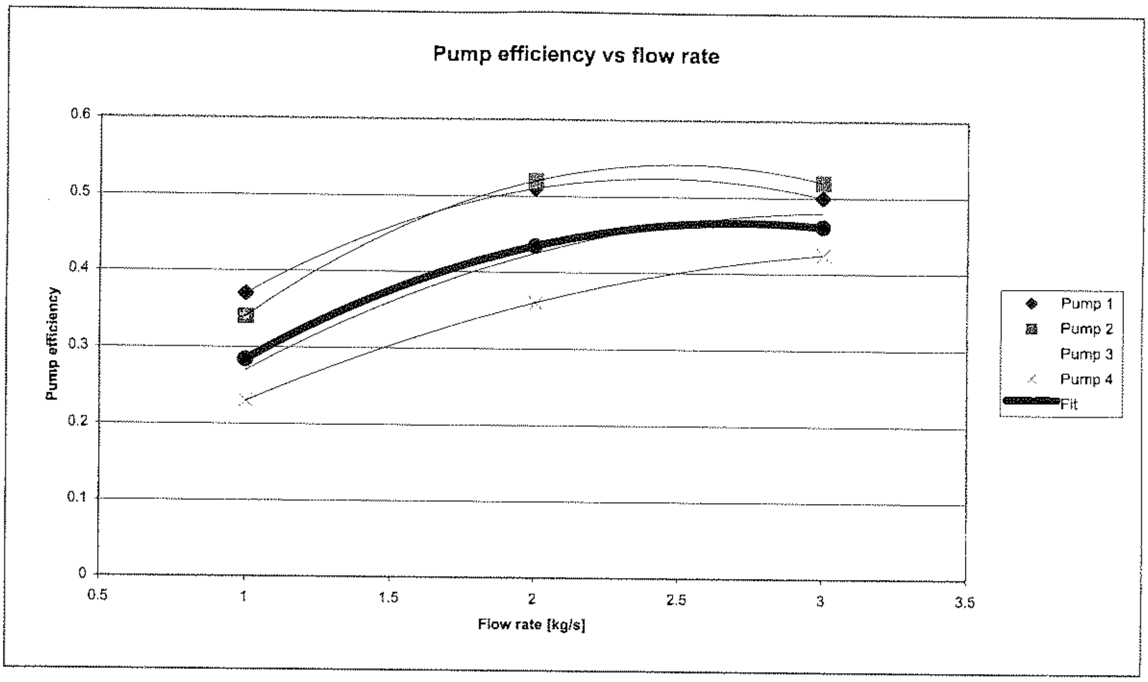

Figure A.8: Efficiency correlation for flow between 0 and 3 kg/s

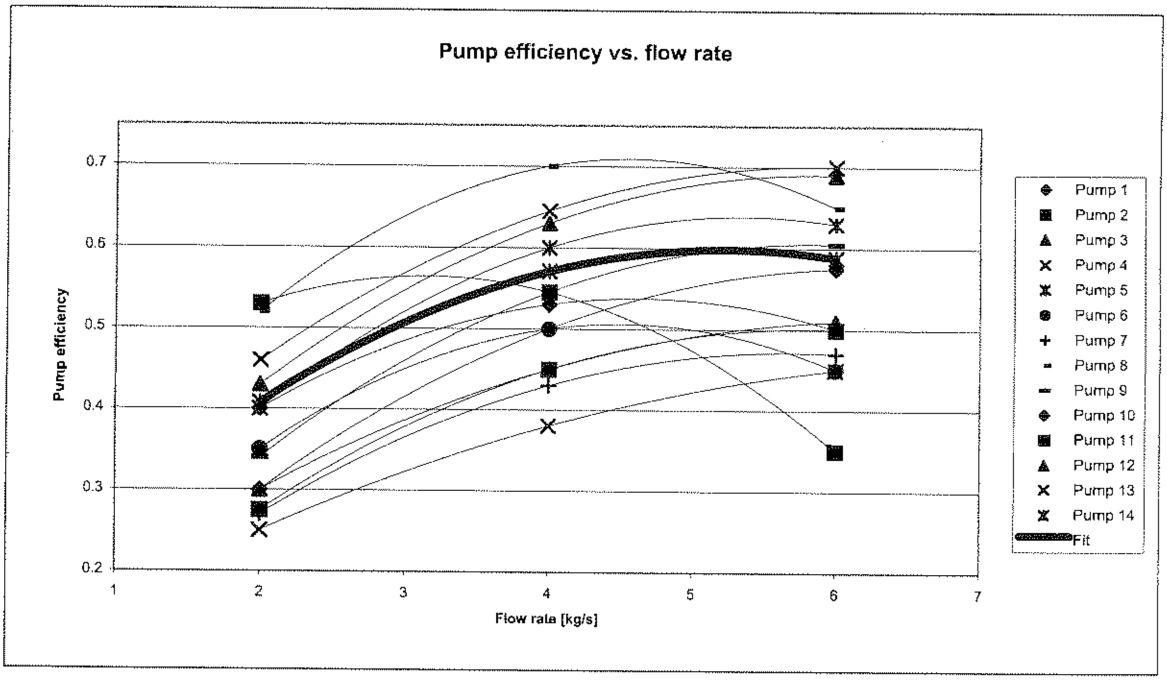

Figure A.9: Efficiency correlation for flow between 3 and 7.5 kg/s

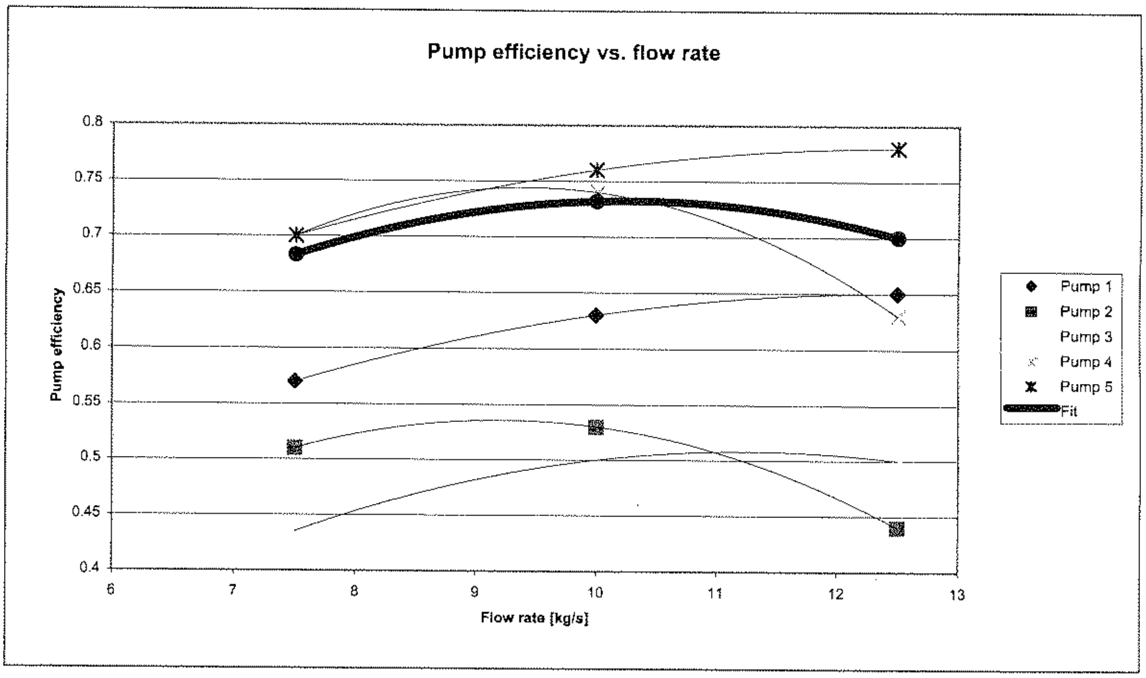

Figure A.10: Efficiency correlation for flow between 7.5 and 15 kg/s

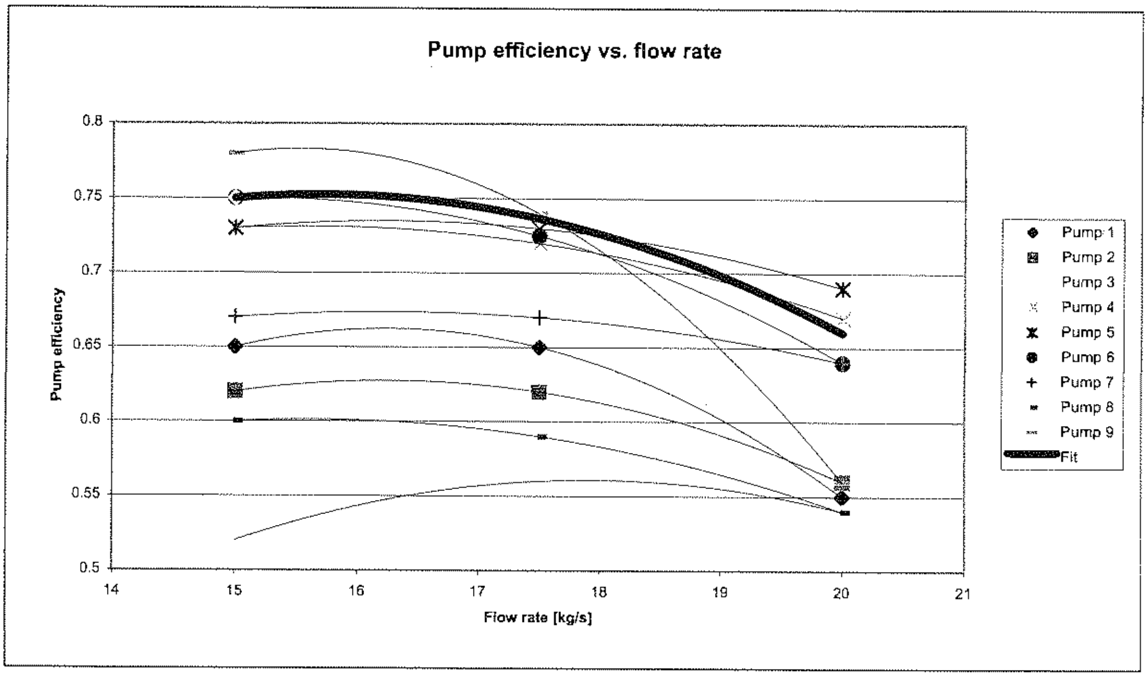

Figure A.11: Efficiency correlation for flow between 15 and 20 kg/s

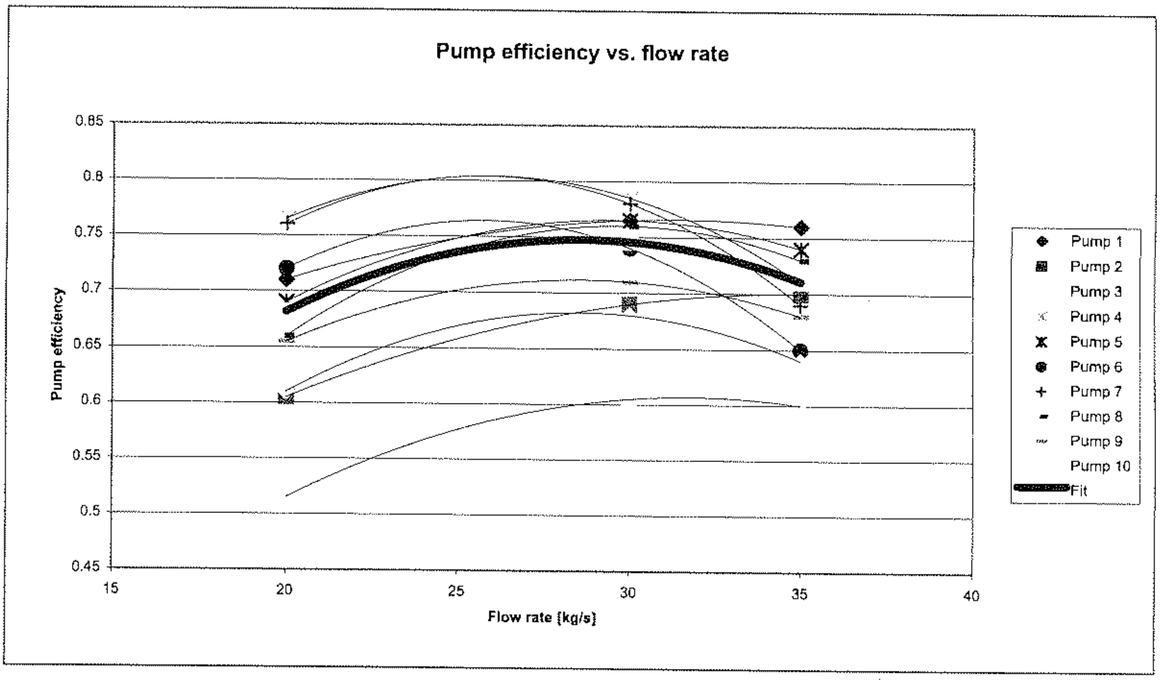

Figure A.12: Efficiency correlation for flow between 20 and 40 kg/s

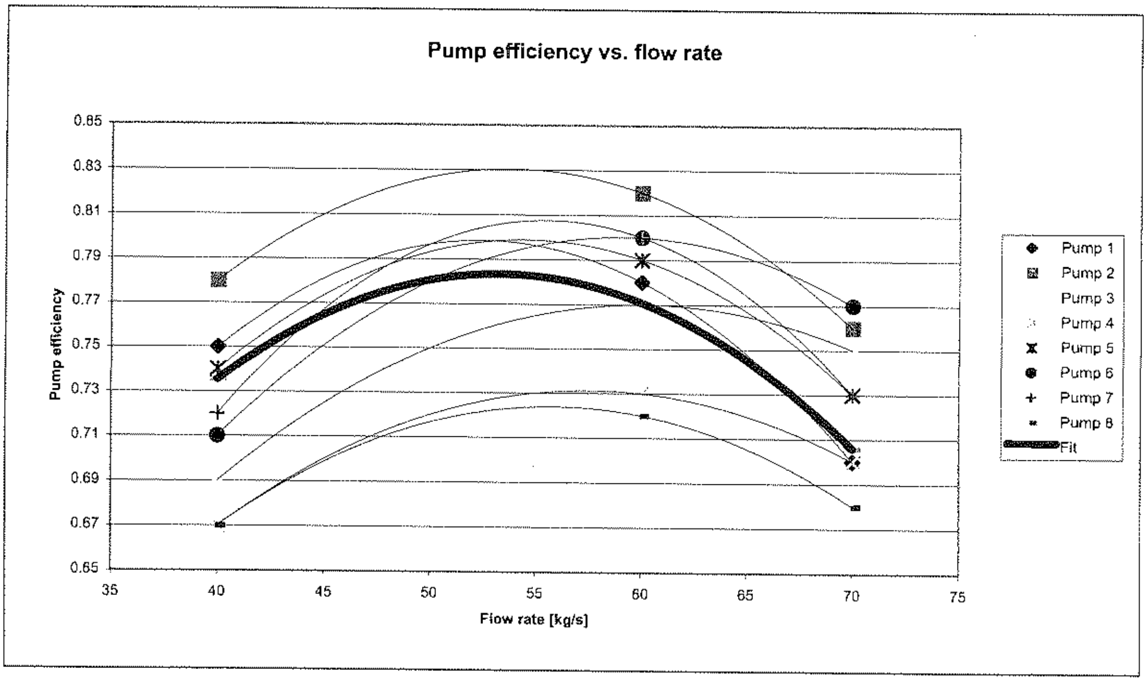

Figure A.13: Efficiency correlation for flow between 40 and 70 kg/s

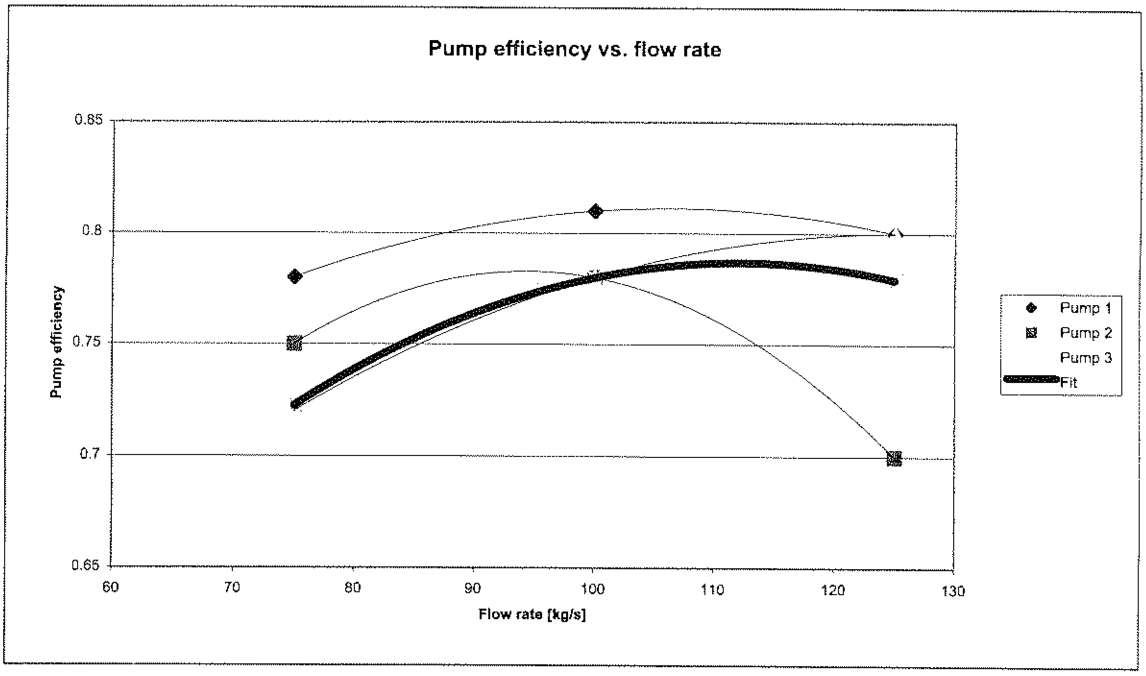

Figure A.14: Efficiency correlation for flow between 70 and 150 kg/s

A.2. FAN

A.2.1 Parameters:

a_j	Correlation coefficients for the K_h versus K_f relation with j = 0 to 2.
b_j	Correlation coefficients for the η_{fan} versus K_f relation with j = 0 to 2.

A.2.2 Inputs:

A.2.2.1Simulation:

m_a	Mass flow rate of dry air at inlet [kg/s].
w_{ai}	Humidity ratio of air entering at the inlet [kg_{vapour} / $kg_{dry\ air}$].
h_{ai}	Specific enthalpy of air entering at the inlet [J/$kg_{dry\ air}$].

A.2.2.2Interface:

D	The rotor diameter [m].
N	Rotational speed of the fan [rpm].
η_{motor}	Efficiency of the drive motor.
P_{total}	Total pressure [Pa].
q	Volume flow rate [m^3/s].
η_{fan}	Fan efficiency.

A.2.3 Outputs:

dP_a	Static pressure rise [Pa].
w_{ae}	Humidity ratio of air entering at the inlet [$kg_{vapour} / kg_{dry\ air}$].
h_{ae}	Specific enthalpy of air entering at the inlet [$J/kg_{dry\ air}$].
Pwr	Power required [W].

A.2.4 Internal variables:

P_1	Extra pressure points [Pa].
P_2	Extra pressure points [Pa].
K_f	Dimensionless flow coefficient.
K_h	Dimensionless pressure head coefficient.
η_{fan}	Efficiency of the fan.
η_1	Extra efficiency point.
η_2	Extra efficiency point.
ρ_a	Air density [kg/m^3]
Q_a	Rate of heat gain to the air [W].

A.2.5 Explicit equations:

Three different pressures and efficiencies at three different flows are needed to calculate the correlation coefficients. A flow variation of 10% above and below the interface input value is assumed. The extra points are then calculated as follow:

$$P_1 = a_h(0.9q)^2 + b_h(0.9q) + c_b$$

$$\eta_1 = a_\eta(0.9q)^2 + b_\eta(0.9q) + c_\eta$$

$$P_2 = a_h(1.1q)^2 + b_h(1.1q) + c_h$$

$$\eta_2 = a_\eta(1.1q)^2 + b_\eta(1.1q) + c_\eta$$

with

$a_h = -823.114$

$b_h = 354.706$

$a_\eta = -95.3q^2 + 108.78q - 33.777$

$b_\eta = 18.281q^2 - 24q + 10.802$ for $0 < q < 1$ [m^3/s].

$a_h = -147.577$

$b_h = 378.984$

$a_\eta = -0.2432q^2 + 1.4215q - 2.176$

$b_\eta = 0.3379q^2 - 2.2688q + 4.4056$ for $1 < q < 5$ [m³/s].

$a_h = -33.262$

$b_h = 309.9086$

$a_\eta = 0.0074q - 0.0801$

$b_\eta = -0.054q + 0.7516$ for $5 < q < 15$ [m³/s].

$a_h = -5.47298$

$b_h = 158.6346$

$a_\eta = -0.00299$

$b_\eta = 0.1219$ for $15 < q < 30$ [m³/s].

$a_h = -2.14676$

$b_h = 74.34644$

$a_\eta = -0.0014$

$b_\eta = 0.081011$ for $30 < q < 60$ [m³/s].

These values were calculated as the average values of the coefficients of a wide range of backward curved centrifugal fan curves or as a function of flow. The a and b coefficients as a function of flow can be seen in figures A.15 to A.20. The accuracy of this can be seen in figures A.21 to A.30. c_h and c_η are calculated from the one given operating point obtained from the supplier or measurements. This implies that only one operating point is needed to obtain the mathematical model of a specific fan.

And

$$c_h = P - (a_h q^2 + b_h q)$$

$$c_\eta = \eta - (a_\eta q^2 + b_\eta q)$$

For each of these three points the K_h and K_f value must be calculated as follow:

$$K_{h1} = \frac{P_1}{\rho_a N^2 D^2}$$

$$K_{f1} = \frac{\rho_a (0.9q)}{\rho_a N D^3}$$

$$K_h = \frac{P}{\rho_a N^2 D^2}$$

$$K_f = \frac{\rho_a q}{\rho_a N D^3}$$

$$K_{h2} = \frac{P_2}{\rho_a N^2 D^2}$$

$$K_{f2} = \frac{\rho_a (1.1q)}{\rho_a N D^3}$$

Using these values the correlation coefficients can be calculated as follow:

$$a_2 = \frac{(K_{h2} - K_{h1})(K_f - K_{f1}) - (K_{f2} - K_{f1})(K_h - K_{h1})}{(K_{f2}^2 - K_{f1}^2)(K_f - K_{f1}) - (K_{f2} - K_{f1})(K_f^2 - K_{f1}^2)}$$

$$a_1 = \frac{(K_h - K_{h1}) - (K_f^2 - K_{f1}^2)a_2}{(K_f - K_{f1})}$$

$$a_0 = K_{h1} - K_{f1}a_1 - K_{f1}^2 a_2$$

$$b_2 = \frac{(\eta_2 - \eta_1)(K_f - K_{f1}) - (K_{f2} - K_{f1})(\eta - \eta_1)}{(K_{f2}^2 - K_{f1}^2)(K_f - K_{f1}) - (K_{f2} - K_{f1})(K_f^2 - K_{f1}^2)}$$

$$b_1 = \frac{(\eta - \eta_1) - (K_f^2 - K_{f1}^2)b_2}{(K_f - K_{f1})}$$

$$b_0 = \eta_1 - K_{f1}b_1 - K_{f1}^2 b_2$$

These coefficients can now be used to calculate the necessary outputs with the following explicit equations:

$$\rho_a = \zeta(h_{ai}, w_{ai})$$

$$K_f = \frac{\rho_a q}{\rho_a N D^3}$$

$$\mathrm{K_h = a_0 + a_1 K_f + a_2 K_f{}^2}$$

$$P_{total} = K_h \rho_a N^2 D^2$$

$$\eta_{\mathrm{fan}} = \mathrm{b_0 + b_1 K_f + b_2 K_f{}^2}$$

$$Q_a = \frac{(1-\eta_{fan}) m_a P_{total}}{\rho_a \eta_{fan}} \qquad \text{motor outside airstream}$$

$$Q_a = \frac{(1-\eta_{fan}\eta_{motor}) m_a P_{total}}{\rho_a \eta_{fan} \eta_{motor}} \qquad \text{motor inside airstream}$$

$$\mathrm{w_{ae} = w_{ai}}$$

$$h_{ae} = h_{ai} + \frac{Q}{m_a}$$

$$Pwr = \frac{P_{total} m_a}{\rho_a \eta_{motor} \eta_{fan}}$$

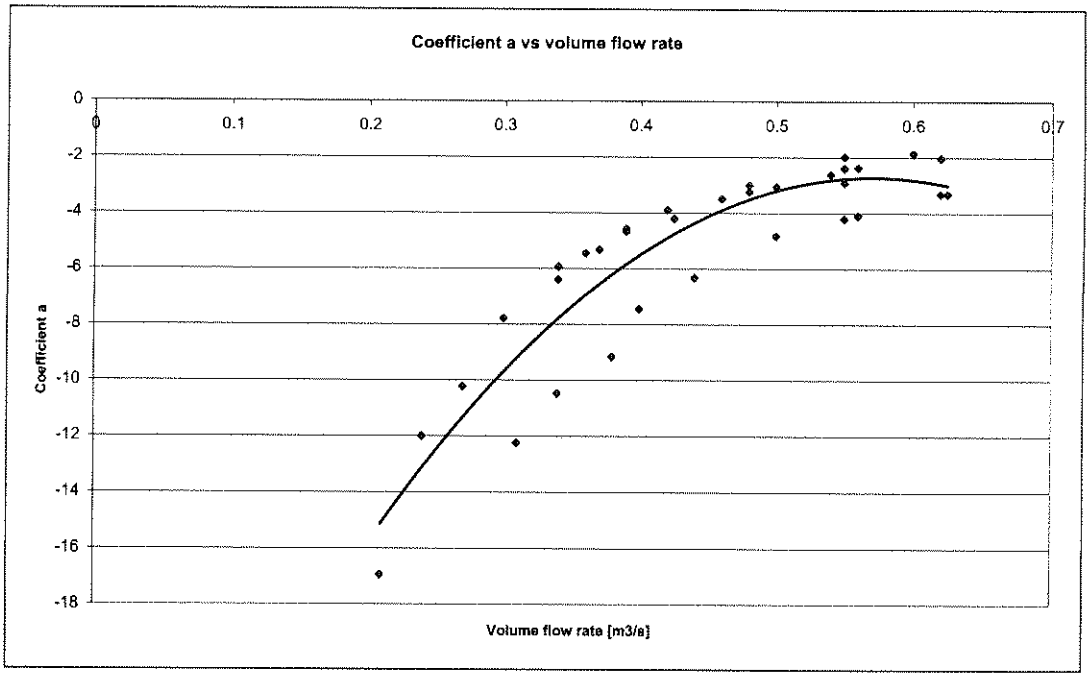

Figure A.15: Efficiency's a coefficient as a function of volume flow rate for flow between 0 and 1 m^3/s

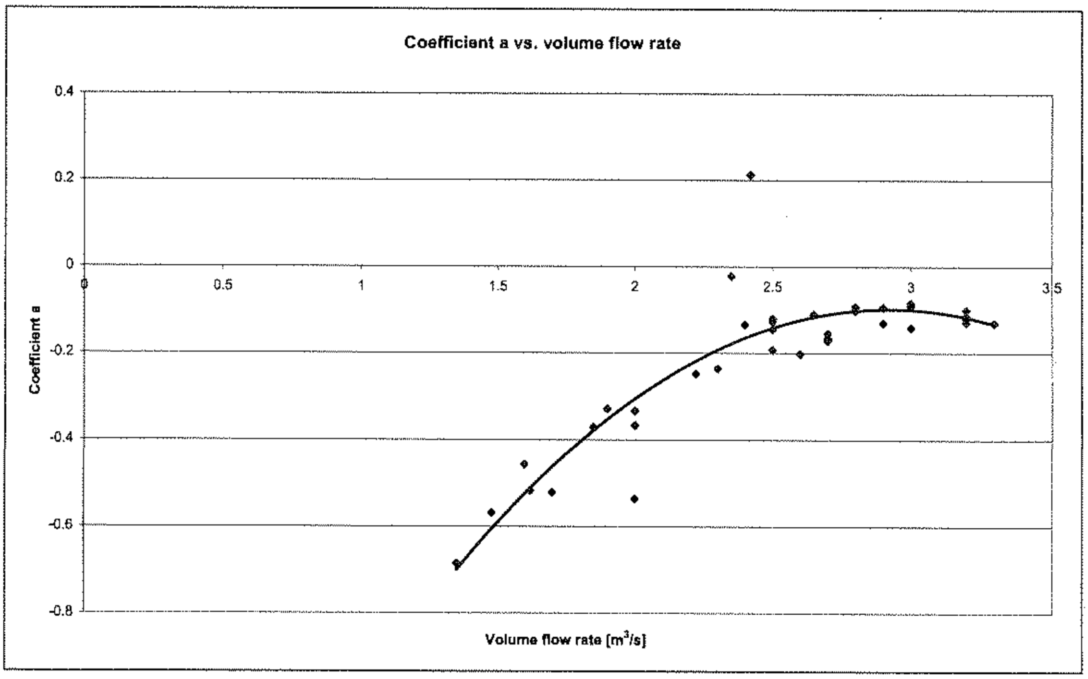

Figure A.16: Efficiency's a coefficient as a function of volume flow rate for flow between 1 and 5 m^3/s

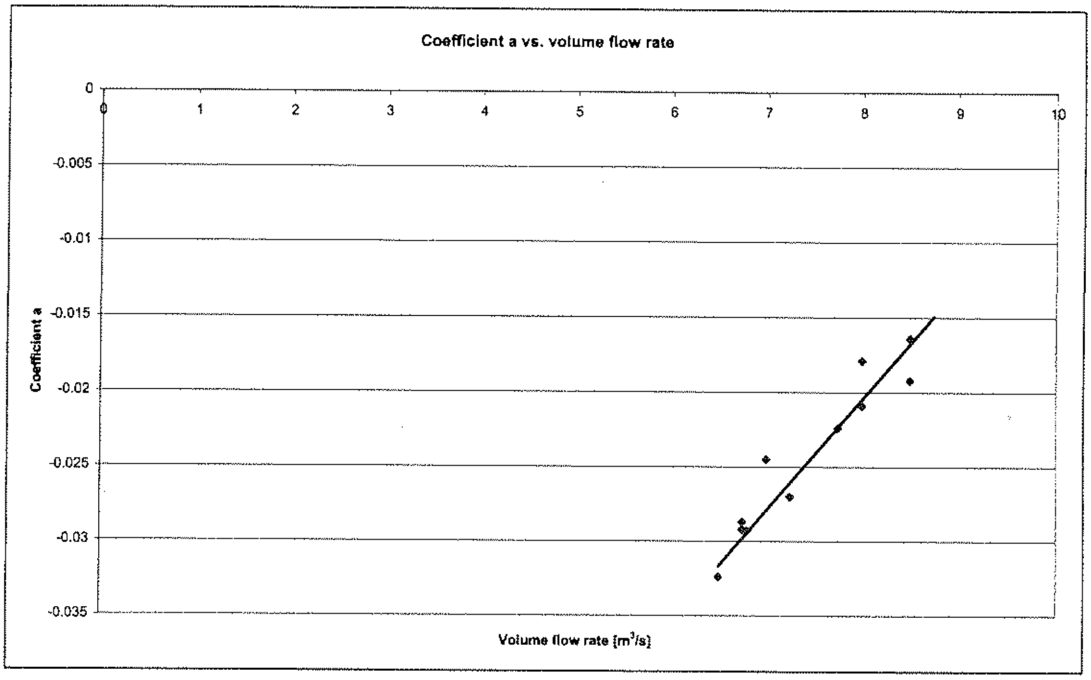

Figure A.17: Efficiency's a coefficient as a function of volume flow rate for flow between 5 and 15 m^3/s

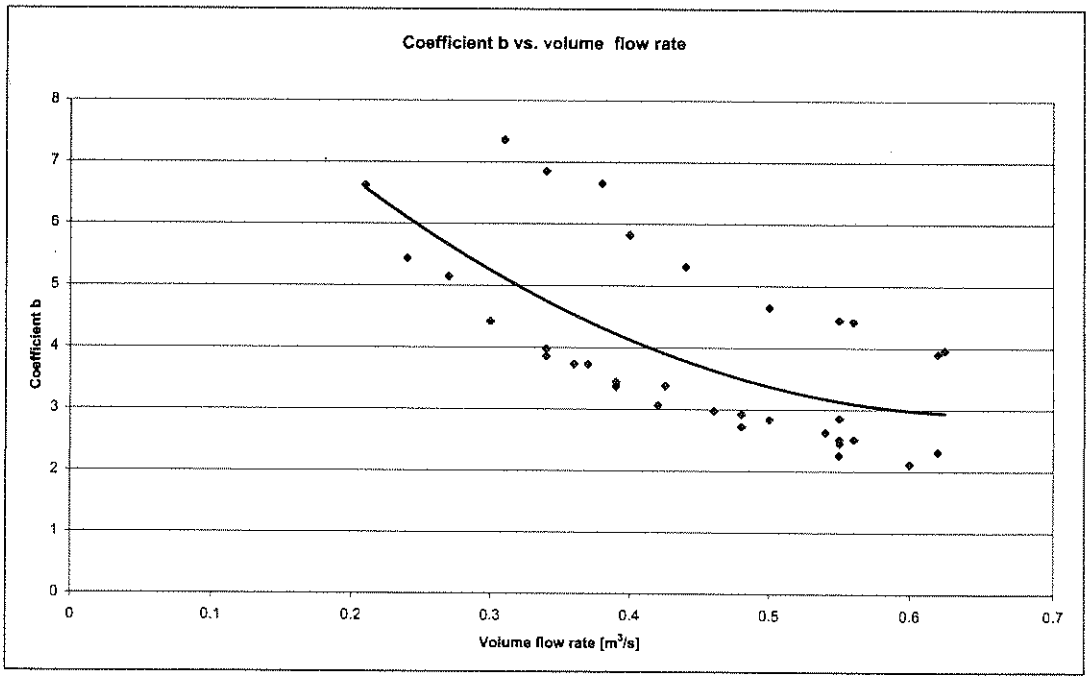

Figure A.18: Efficiency's b coefficient as a function of volume flow rate for flow between 0 and 1 m^3/s

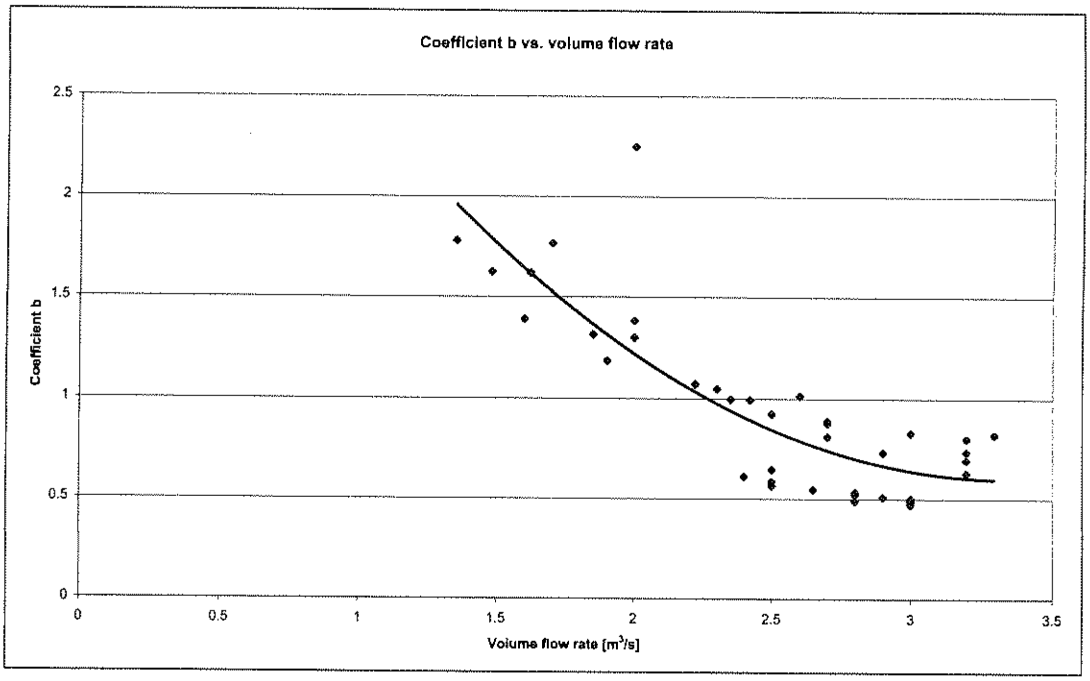

Figure A.19: Efficiency's b coefficient as a function of volume flow rate for flow between 1 and 5 m^3/s

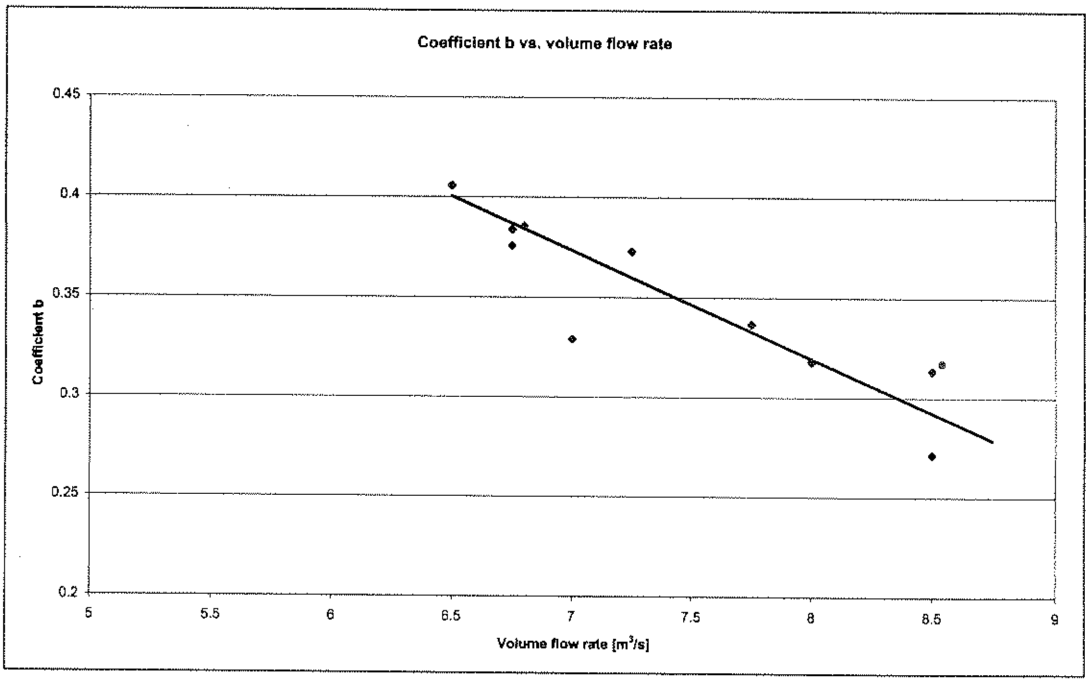

Figure A.20: Efficiency's b coefficient as a function of volume flow rate for flow between 5 and 15 m^3/s

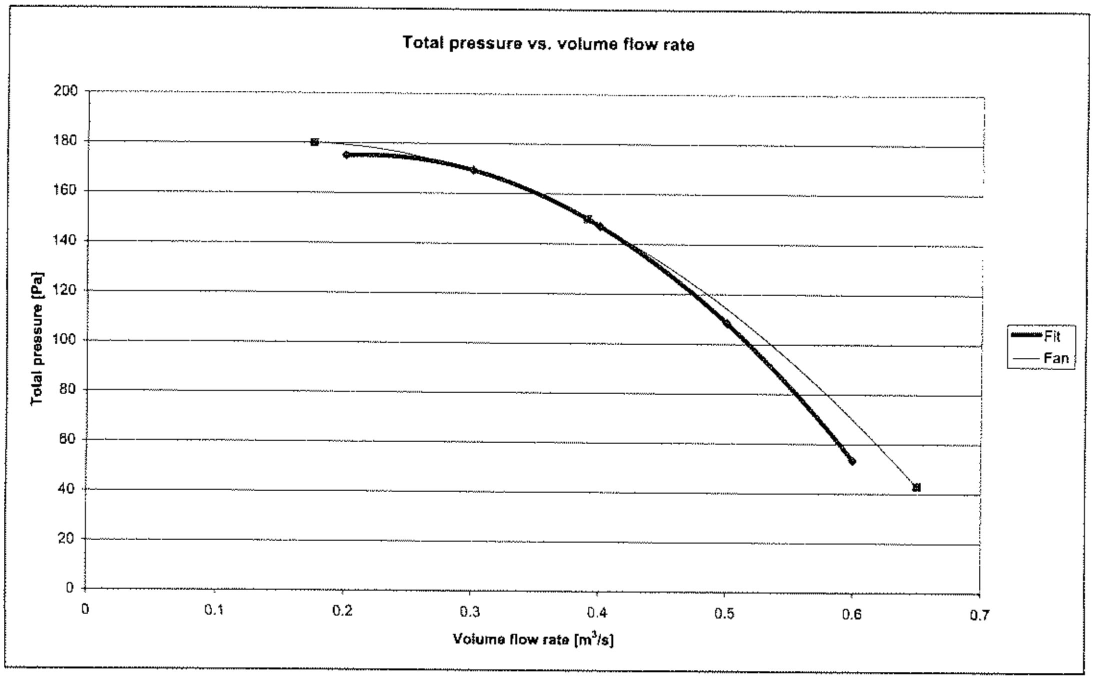

Figure A.21: Total pressure correlation for a volume flow rate between 0 and 1 m^3/s

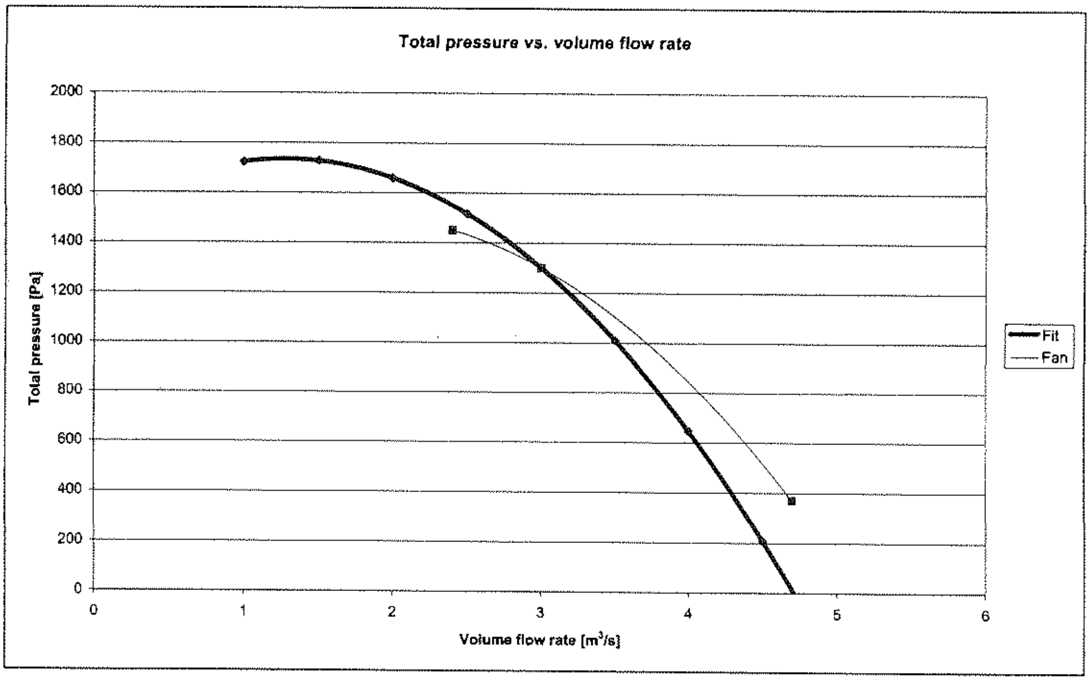

Figure A.22: Total pressure correlation for a volume flow rate between 1 and 5 m^3/s

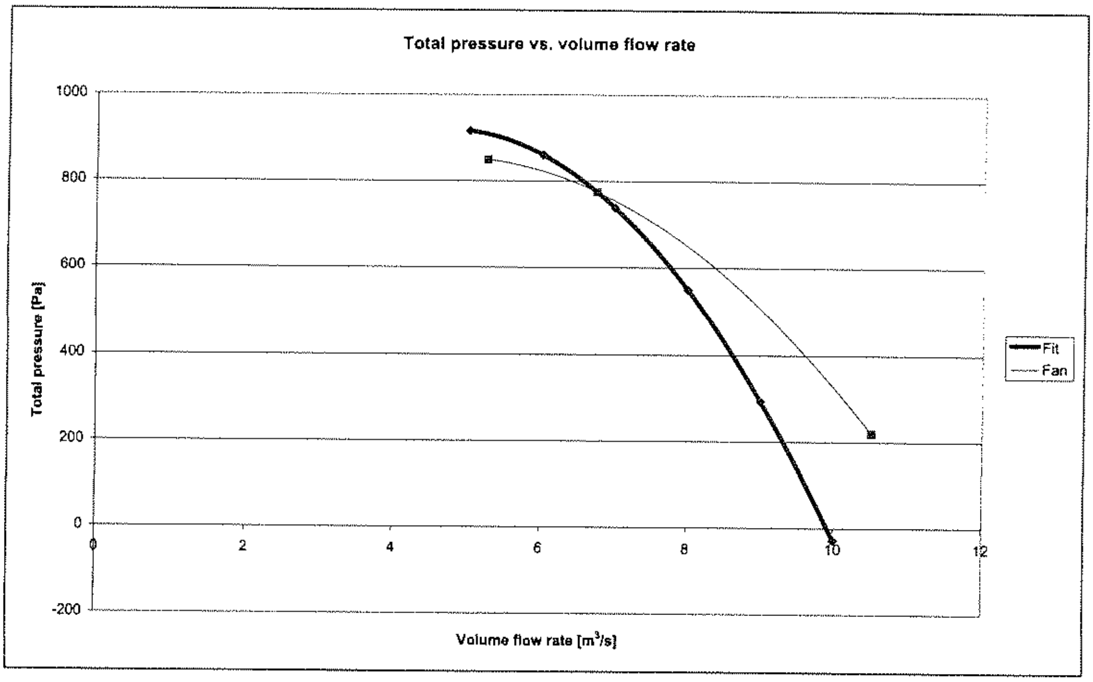

Figure A.23: Total pressure correlation for a volume flow rate between 5 and 15 m³/s

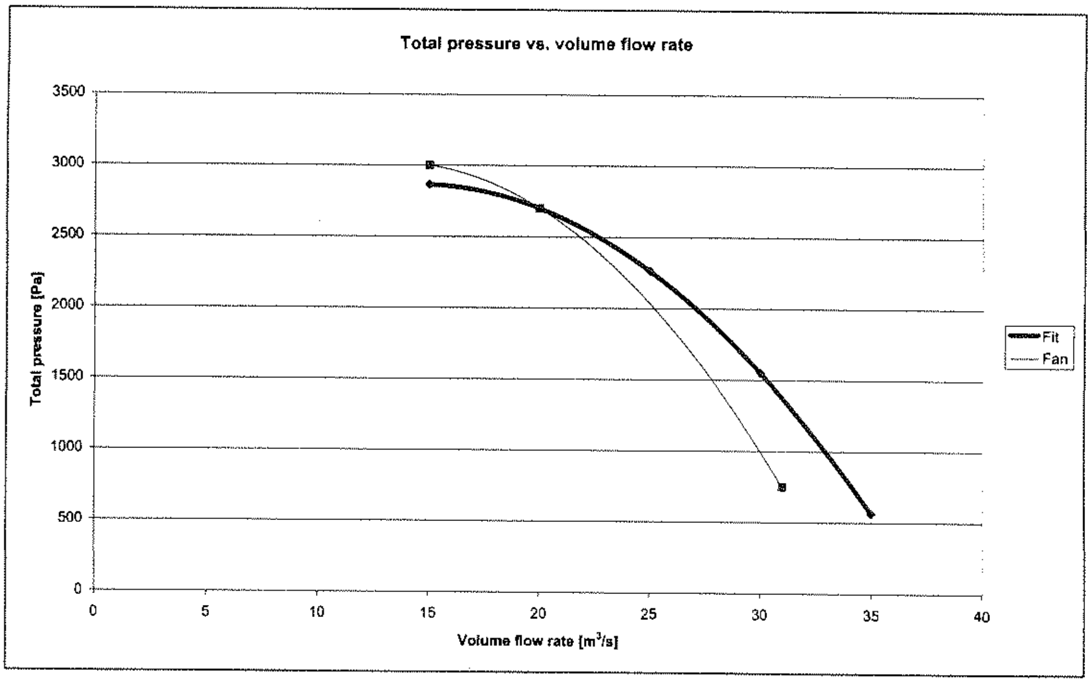

Figure A.24: Total pressure correlation for a volume flow rate between 15 and 30 m³/s

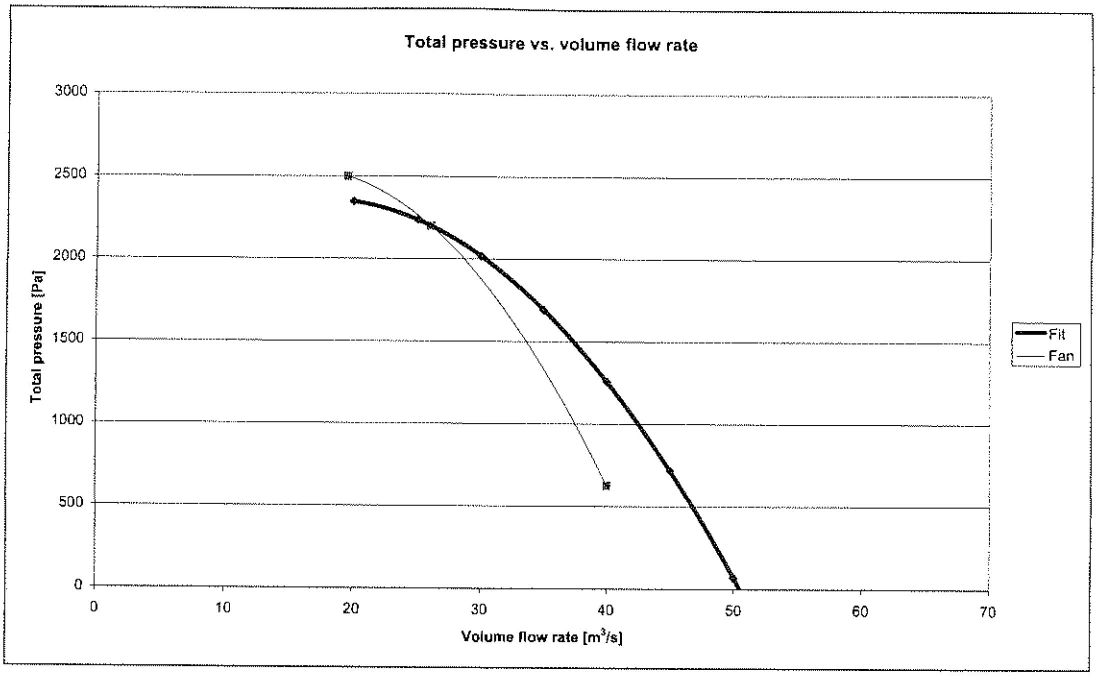

Figure A.25: Total pressure correlation for a volume flow rate between 30 and 60 m^3/s

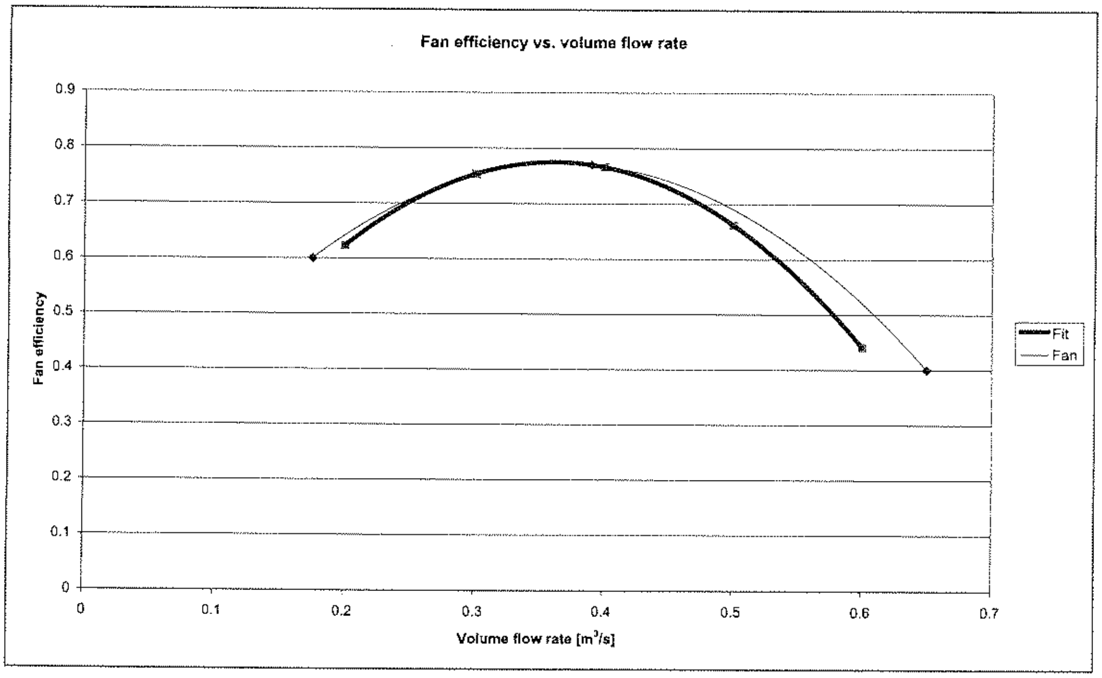

Figure A.26: Fan efficiency correlation for a volume flow rate between 0 and 1 m^3/s

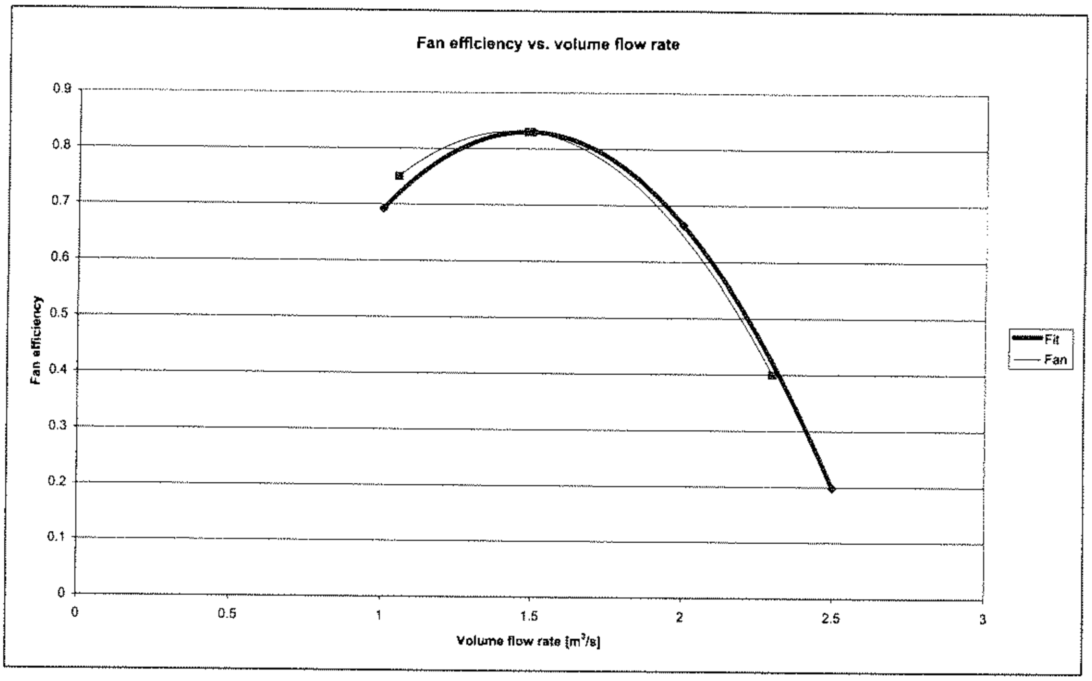

Figure A.27: Fan efficiency correlation for a volume flow rate between 1 and 5 m^3/s

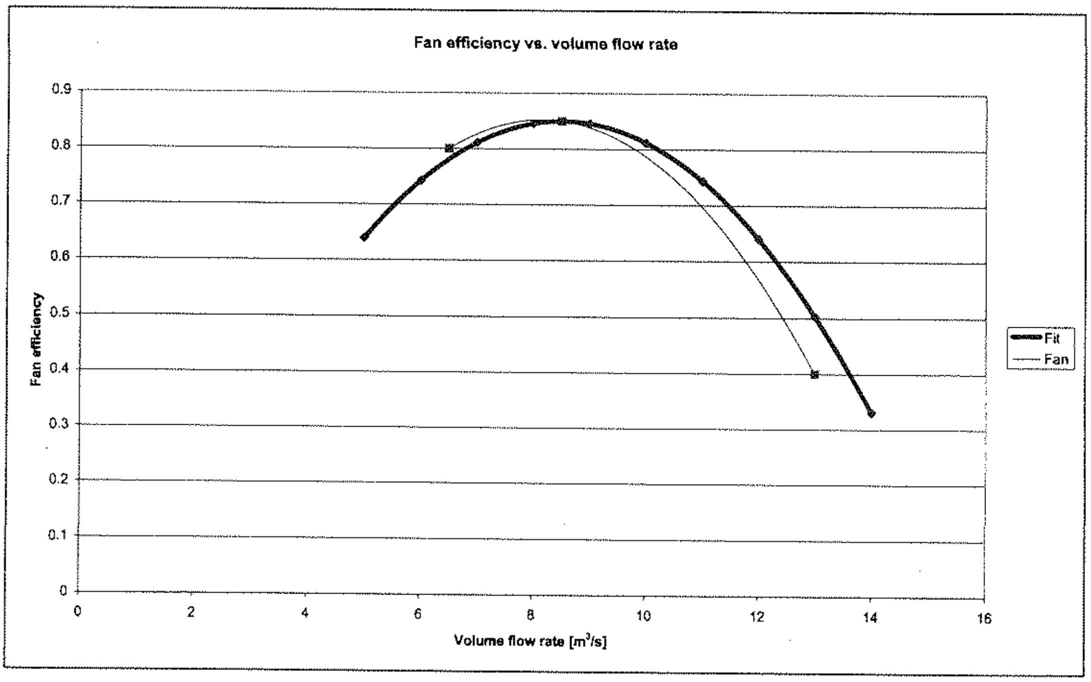

Figure A.28: Fan efficiency correlation for a volume flow rate between 5 and 15 m^3/s

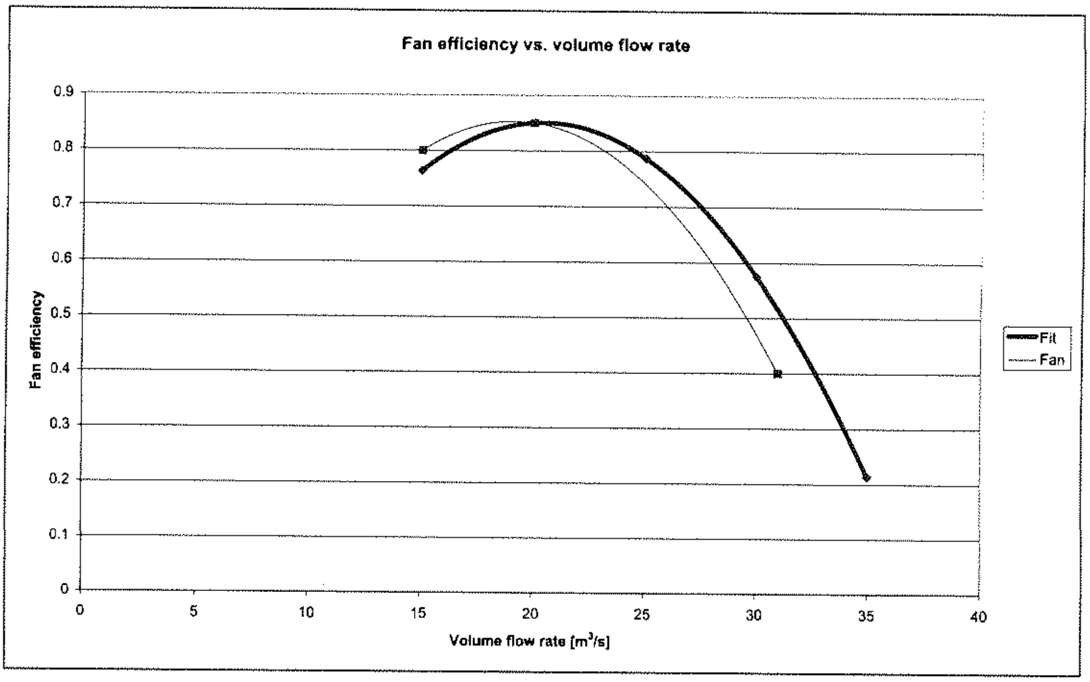

Figure A.29: Fan efficiency correlation for a volume flow rate between 15 and 30 m^3/s

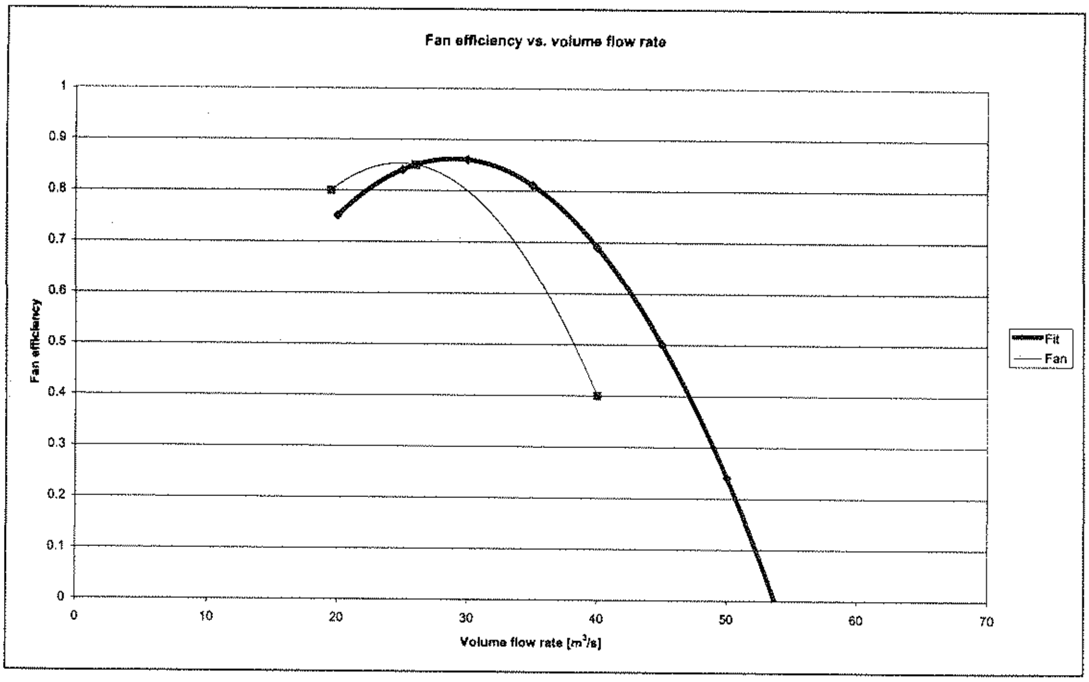

Figure A.30: Fan efficiency correlation for a volume flow rate between 30 and 60 m^3/s

A.3. AIR COOLED LIQUID CHILLER

A.3.1 Parameters:

f	Active fraction of total capacity.
a_n	Regression coefficients for cooling capacity.
b_n	Regression coefficients for power.

A.3.2 Inputs:

A.3.2.1 Simulation:

$m_{li(evap)}$	Mass flow rate of water through the evaporator [kg/s].
$T_{li(evap)}$	Temperature of water entering evaporator [°C].
$w_{ai(cond)}$	Humidity ratio of air entering the condenser [kg_{vapour} / $kg_{dry\ air}$].
$h_{ai(cond)}$	Specific enthalpy of air entering the condenser [J/$kg_{dry\ air}$].

A.3.2.2 Interface:

Cc	Cooling capacity at the expected operational temperatures of the chiller [kW].
P	Compressor power at the expected operational temperatures of the chiller [kW].
T_{aa}	Expected dry bulb temperature of the air entering the condenser [°C].
T_{ee}	Expected temperature of water exiting evaporator [°C].

A.3.3 Outputs:

$T_{le(evap)}$	Temperature of water leaving evaporator [°C].
Q_e	Cooling capacity [kW].
Pwr	Power consumed by the compressor and fan [kW].

A.3.4 Internal variables:

$T_{ai(cond)}$	Dry bulb temperature of air entering condenser[°C].

A.3.5 Explicit equations:

$$T_{ai(cond)} = \zeta\left(h_{ai(cond)}, w_{ai(cond)}\right)$$

$$Q_e = f\left(a_0 + a_1 T_{le(evap)} + a_2 T_{ai(cond)}\right)$$

With

$$a_2 = -0.0112Cc + 0.024$$

$$a_1 = 0.0265Cc - 0.2547$$

$$a_0 = Cc - \left(T_{ee}a_1 + T_{aa}a_2\right)$$

Coefficients a_2 and a_1 were found to be linear over a wide range of chillers with respect to cooling capacity as can be seen in figure A.31 and A.32. a_0 is calculated from the one given operating point obtained from the supplier or measurements. This implies that only one operating point is needed to obtain the mathematical model of a specific chiller.

$$Pwr = f\left(b_0 + b_1 T_{li(evap)} + b_2 T_{ai(cond)}\right)$$

With

$$b_2 = 0.0079P + 0.3051$$

$$b_1 = 0.0184P - 0.1403$$

$$b_0 = P - \left(T_{ee} b_1 + T_{aa} b_2\right)$$

Coefficients b_2 and b_1 were found to be linear over a wide range of chillers with respect to compressor power as can be seen in figures A.33 and A.34. b_0 is calculated from the one given operating point obtained from the supplier or measurements. This implies that only one operating point is needed to obtain the mathematical model of a specific chiller.

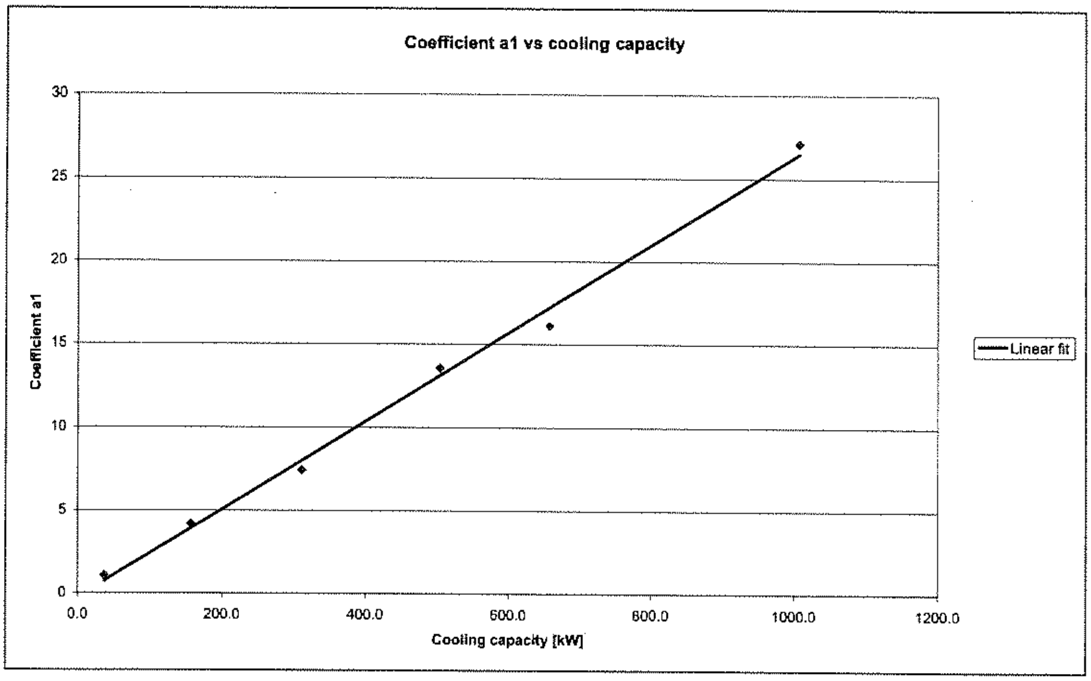

Figure A.31: Coefficient a1 as a function of cooling capacity

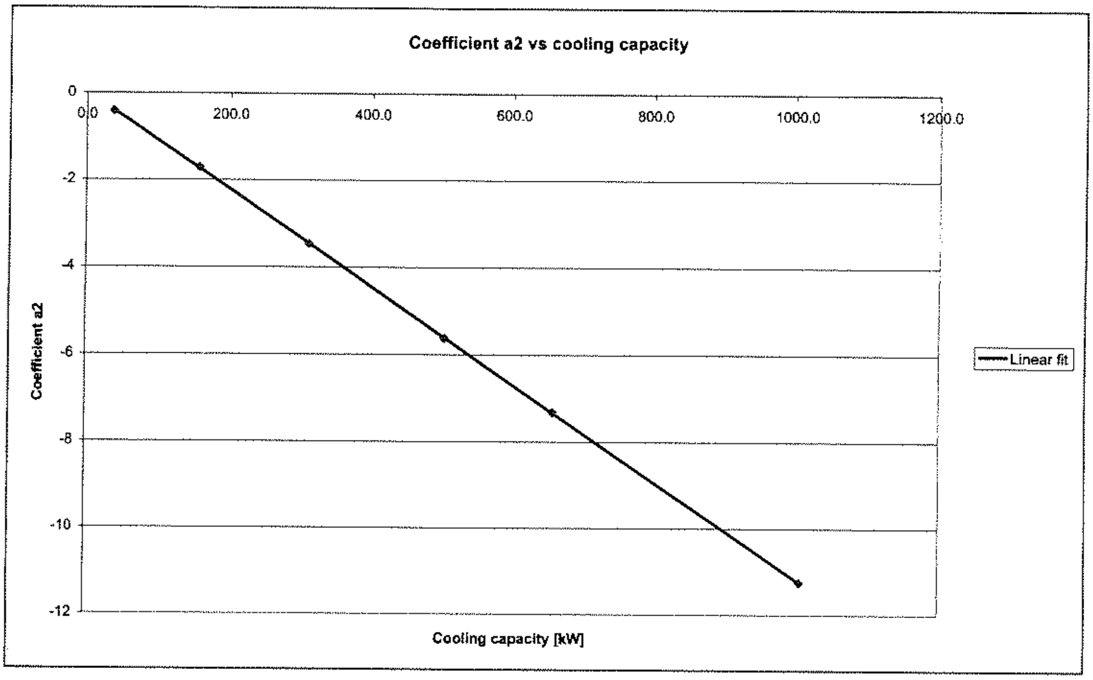

Figure A.32: Coefficient a2 as a function of cooling capacity

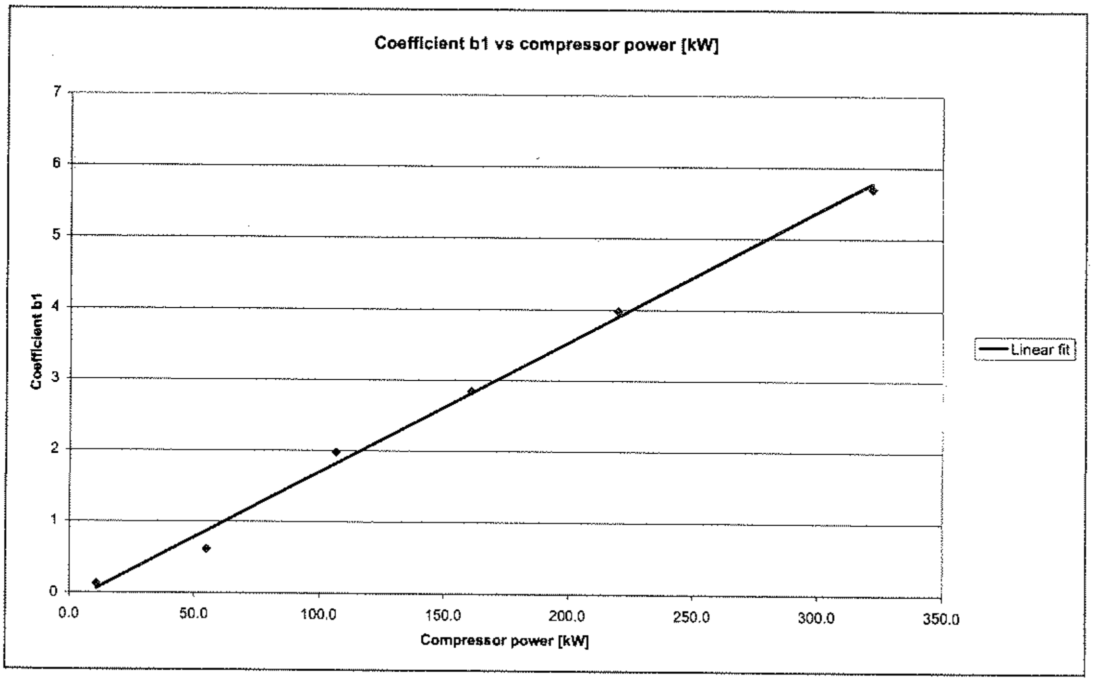

Figure A.33: Coefficient b1 as a function of compressor power

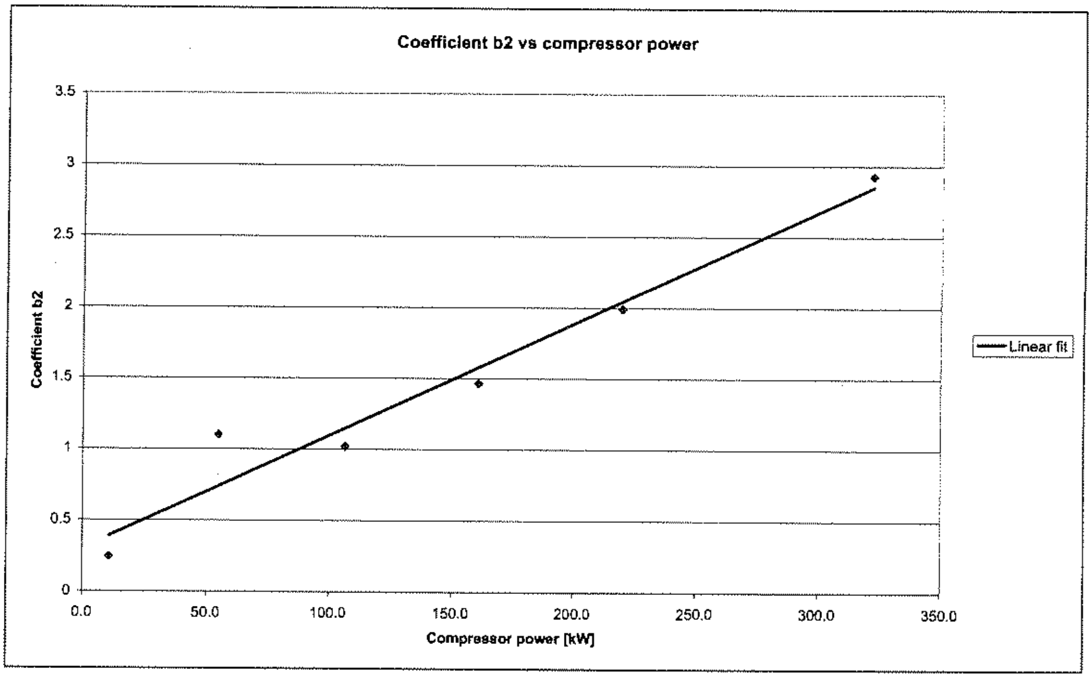

Figure A.34: Coefficient b2 as a function of compressor power

A.4. WATER COOLED LIQUID CHILLER

A.4.1 Parameters:

f	Active fraction of total capacity.
a_n	Regression coefficients for cooling capacity.
b_n	Regression coefficients for power.

A.4.2 Inputs:

A.4.2.1 Simulation:

$m_{l(cond)}$	Mass flow rate of water through the condenser [kg/s].
$m_{l(evap)}$	Mass flow rate of water through the evaporator [kg/s].
$T_{li(cond)}$	Temperature of water entering condenser [°C].

$T_{li(evap)}$	Temperature of water entering evaporator [°C].

A.4.2.2Interface:

Cc	Cooling capacity at the expected operational temperatures of the chiller [kW].
P	Compressor power at the expected operational temperatures of the chiller [kW].
T_{ce}	Expected temperature of water exiting condenser [°C].
T_{ee}	Expected temperature of water exiting evaporator [°C].

A.4.3 Outputs:

$T_{le(cond)}$	Temperature of water leaving condenser [°C].
$T_{le(evap)}$	Temperature of water leaving evaporator [°C].
Q_e	Cooling capacity [kW].
Pwr	Power consumed by the compressor [kW].

A.4.4 Explicit equations:

$$Q_e = f\left(a_0 + a_1 T_{le(cond)} + a_2 T_{le(evap)}\right)$$

With

$$a_2 = 0.0266Cc + 2.8714$$

$$a_1 = -0.01Cc + 0.2289$$

$$a_0 = Cc - \left(T_{ce}a_1 + T_{ee}a_2\right)$$

Coefficients a_2 and a_1 were found to be linear over a wide range of chillers with respect to cooling capacity as can be seen in figure A.35 and A.36. a_0 is calculated from the one given operating point obtained from the supplier or measurements. This implies that only one operating point is needed to obtain the mathematical model of a specific chiller.

$$Pwr = f\left(b_0 + b_1 t_{li(cond)} + b_2 t_{li(evap)}\right)$$

With

$$b_2 = 0.007P + 0.3549$$

$$b_1 = 0.0124P + 0.4207$$

$$b_0 = P - \left(t_{ce} b_1 + t_{ee} b_2\right)$$

Coefficients b_2 and b_1 were found to be linear over a wide range of chillers with respect to compressor power as can be seen in figure A.37 and A.38. b_0 is calculated from the one given operating point from the supplier or measurements. This implies that only one operating point is needed to obtain the mathematical model of a specific chiller.

$$T_{le(cond)} = T_{li(cond)} - \frac{Q_e + Pwr}{m_{l(cond)} c_{pl}}$$

$$T_{le(evap)} = T_{li(evap)} - \frac{Q_e}{m_{l(evap)} c_{pl}}$$

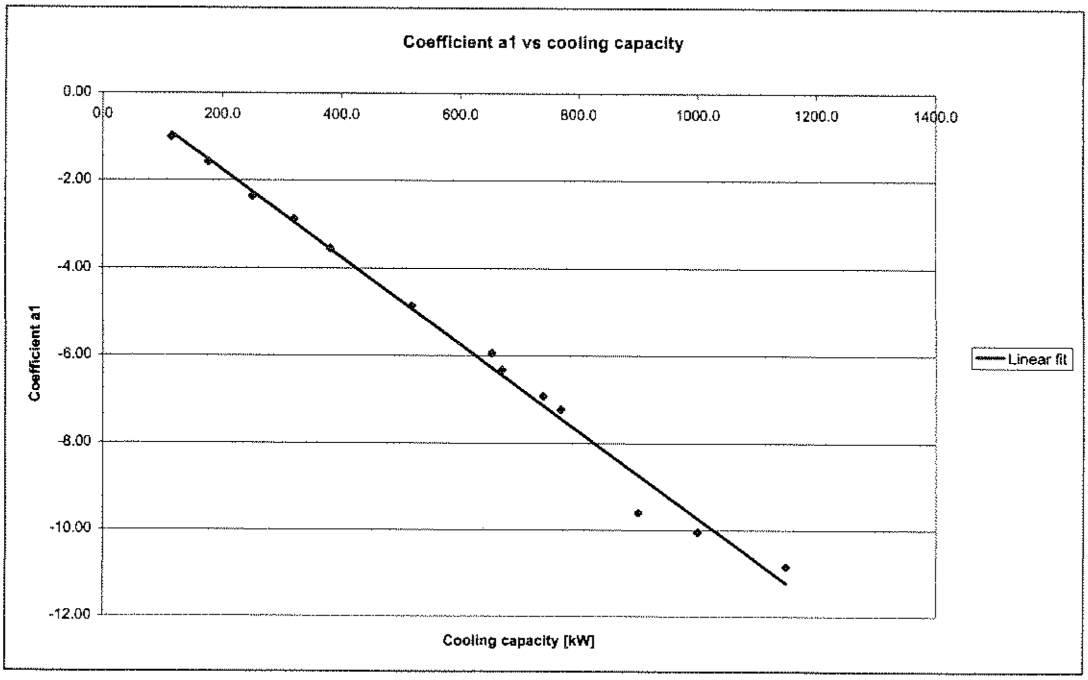

Figure A.35: Coefficient a1 as a function of cooling capacity

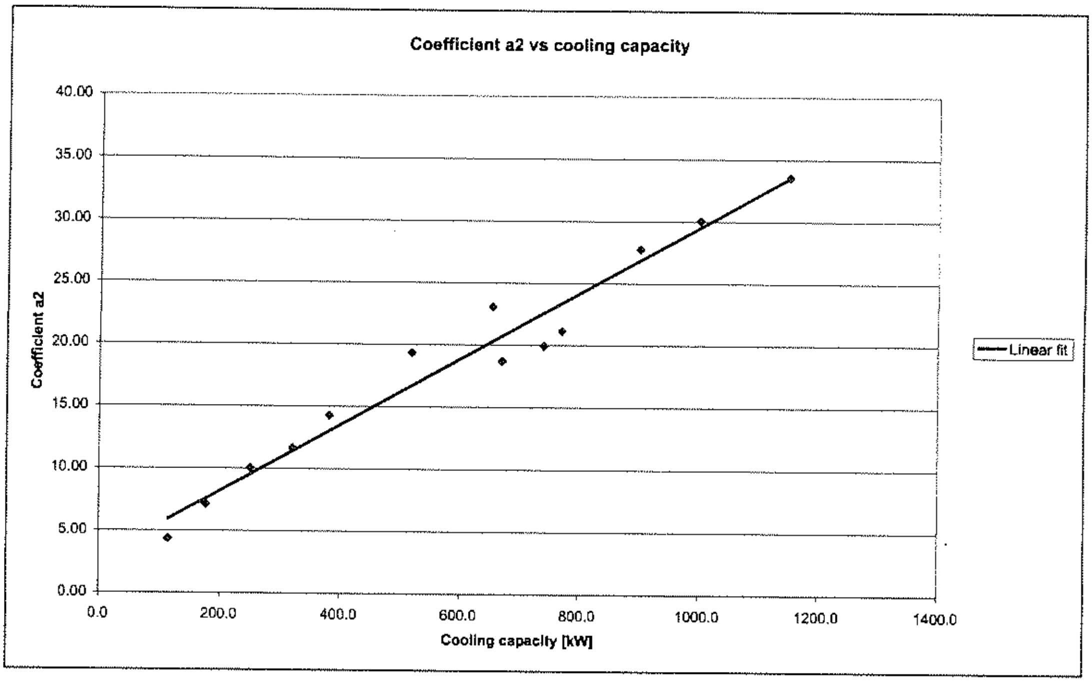

Figure A.36: Coefficient a2 as a function of cooling capacity

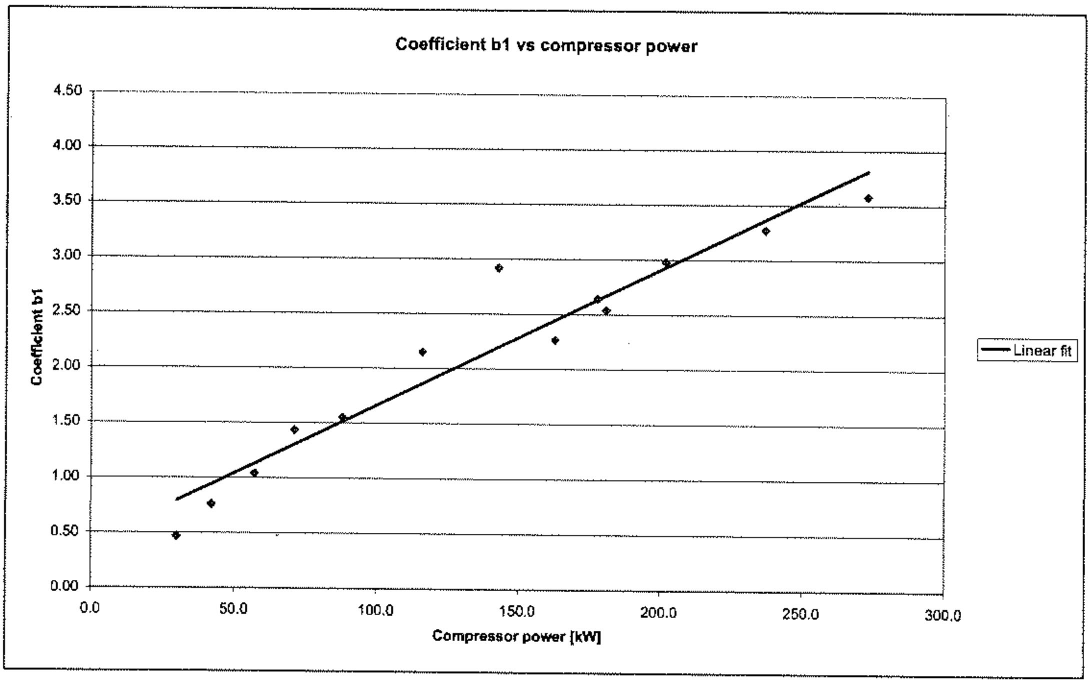

Figure A.37: Coefficient b1 as a function of compressor power

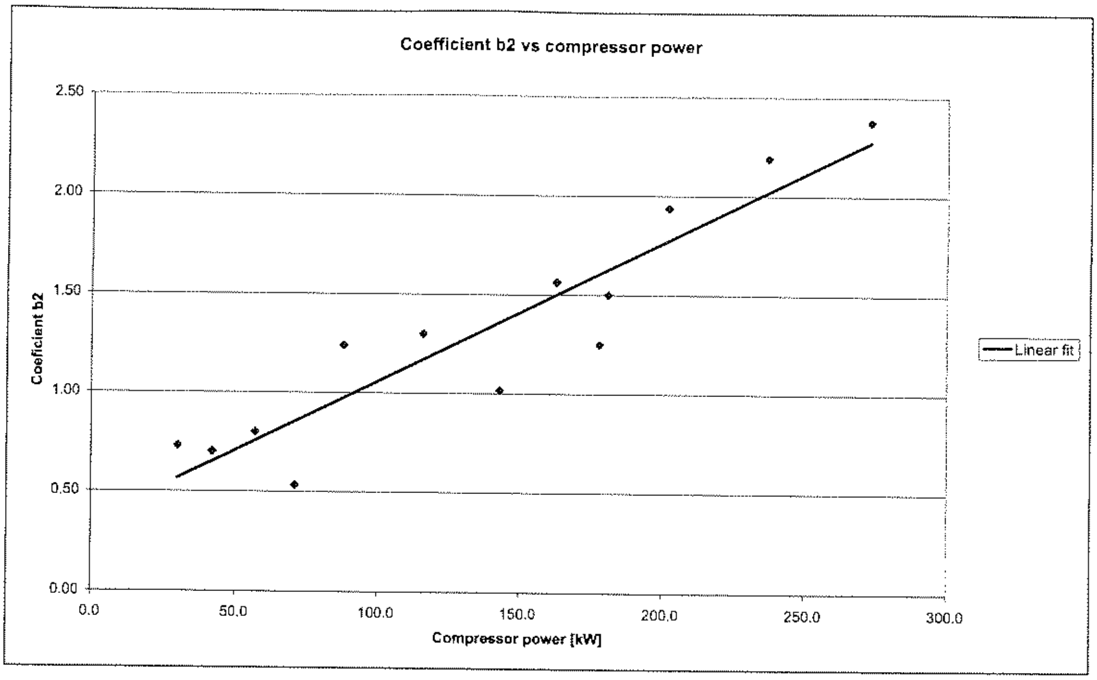

Figure A.38: Coefficient b2 as a function of compressor power

A.5. COIL

A.5.1 Parameters:

A_f	Coil face area [m^2].
R	Ratio A_o/A_f.
f	Fraction of m_l that is not bypassed and passes through the coil.
a_j	Correlation coefficients for the h_o versus V_a relation with $j = 0$ to 2.

A.5.2 Inputs:

m_a	Mass flow rate of dry air at the inlet [kg/s].
w_{ai}	Humidity ratio of air entering at the inlet [kg_{vapour}/kg_{dryair}].
h_{ai}	Specific enthalpy of air entering at the inlet [J/kg_{dryair}].
m_l	Mass flow rate of liquid at inlet [kg/s].
T_{li}	Temperature of liquid entering at inlet [°C].

A.5.3 Outputs:

w_{ae}	Humidity ratio of air leaving at the outlet [kg_{vapour}/kg_{dryair}].
h_{ae}	Specific enthalpy of air leaving at the outlet [J/kg_{dryair}].
T_{le}	Temperature of liquid leaving at outlet [°C].

A.5.4 Internal variables:

A_o	Total outside heat transfer area [m^2].
ρ_a	Air density [kg/m^3].
T_{ai}	Dry-bulb temperature of air at the inlet[°C].
V_a	Air face velocity [m/s].
h_o	Outside surface convection heat transfer coefficient [$W/m^2 °C$].
T_{ae}	Dry-bulb temperature of air at the outlet [°C].
w_{adp}	Humidity ratio of saturated air at t_{li} [$kg_{vapour}/kg_{dry\ air}$].
h_{ad}	Specific enthalpy of air at the outlet [$J/kg_{dry\ air}$].
S	Sensible heat ratio.

A.5.5 Explicit equations:

$$m_l = fm_l$$

$$\rho_a = \zeta(h_{ai}, w_{ai})$$

$$T_{ai} = \zeta(h_{ai}, w_{ai})$$

$$V_a = \frac{m_a}{\rho_a A_f}$$

$$h_o = a_0 + a_1 V_a + a_2 V_a^{\ 2}$$

$$A_0 = RA_f$$

If $T_{li} \geq \zeta$ (T_{ai}, saturated) the coil is dry which means that $S = 1$, $w_{adp} = w_{ai}$ and $h_{ad} = h_{ai}$ and the following equations apply.

$$w_{ae} = w_{ai}$$

$$T_{ae} = \frac{\left[e^{h_o A_o\left(\frac{1}{m_a c_{pa}} - \frac{1}{m_l c_{pl}}\right)} - 1\right] T_{li} - \left[\frac{m_a c_{pa}}{m_l c_{pl}} - 1\right] T_{ai}}{e^{h_o A_o\left(\frac{1}{m_a c_{pa}} - \frac{1}{m_l c_{pl}}\right)} - \frac{m_a c_{pa}}{m_l c_{pl}}}$$

$$h_{ae} = \zeta(T_{ae}, w_{ae})$$

$$T_{le} = T_{li} - \frac{m_a c_{pa}}{m_l c_{pl}}(T_{ae} - T_{ai})$$

Else the coil is wet and the following equations apply

$$w_{adp} = \xi(T_{li}, saturated)$$

A.5.6 Simultaneous equations:

$$h_{ae} = h_{ai} + \frac{h_o A_o (T_{le} - T_{ai}) - (T_{li} - T_{ae})}{m_a S \ln \frac{(T_{le} - T_{ai})}{(T_{li} - T_{ae})}}$$

$$T_{le} = T_{li} - \frac{m_a}{m_l c_{pl}} (h_{ae} - h_{ai})$$

$$T_{ae} = T_{ai} - \frac{w_{ai} - w_{ae}}{w_{ai} - w_{adp}} (T_{ai} - T_{li})$$

$$w_{ae} = \zeta(T_{ae}, h_{ae})$$

$$h_{ad} = \zeta(T_{ai}, w_{ae})$$

$$S = \frac{h_{ad} - h_{ae}}{h_{ai} - h_{ae}}$$

A.5.7 Explicit equation:

$$T_{le} = f T_{le} + (1 - f) T_{li}$$

A.6. COOLING TOWER

A.6.1 Parameters:

a_j	Correlation coefficients for the UA versus m_l and t_{li} relation with j = 0 to 5
m_a	Mass flow rate of outdoor air through the tower [kg/s]
Pwr_{fan}	Power required by the built-in fan [W]
Pwr_{pump}	Power required by the built-in pump [w]

A.6.2 Inputs:

w_{ai}	Humidity ration of air entering at inlet [kg $_{vapour}$/kg $_{dry\ air}$]
h_{ai}	Specific enthalpy of air entering at inlet [J/kg $_{dry\ air}$]

m_l	Mass flow rate of liquid at inlet [kg/s]
T_{li}	Temperature of liquid entering at inlet [°C]

A.6.3 Outputs:

w_{ae}	Humidity ratio of air leaving at outlet [kg $_{\text{vapour}}$/kg $_{\text{dry air}}$]
h_{ae}	Specific enthalpy of air leaving at outlet [J/kg $_{\text{dry air}}$]
T_{le}	Temperature of liquid leaving at outlet [°C]
Pwr	Input power required [W]

A.6.4 Internal variables:

h_{si}	Specific enthalpy of saturated air entering at t_{le} [J/kg $_{\text{dry air}}$]
h_{se}	Specific enthalpy of saturated air entering at t_{li} [J/kg $_{\text{dry air}}$]

A.6.5 Explicit equations:

$$UA = a_0 + a_1 m_l + a_2 T_{li} + a_3 m_l{}^2 + a_4 T_{li}{}^2 + a_5 m_{li} T_{li}$$

$$h_{se} = 10^3(16.66326 + 4.701617 T_{li} - 0.11237 T_{li}{}^2 + 0.004991 T_{li}{}^3 - 0.17197 p_{barom} - 0.01364 p_{barom} T_{li} + 0.000493 p_{barom} T_{li}{}^2 - 2.9x10^{-5} p_{barom} T_{li}{}^3)$$

$$Pwr = Pwr_{fan} + Pwr_{pump}$$

A.6.6 Simultaneous equations:

$$h_{si} = 10^3(16.66326 + 4.701617 T_{le} - 0.11237 T_{le}{}^2 + 0.004991 T_{le}{}^3 - 0.17197 p_{barom} - 0.01364 p_{barom} T_{le} + 0.000493 p_{barom} T_{le}{}^2 - 2.9x10^{-5} p_{barom} T_{le}{}^3)$$

$$Q_a = UA\left[\frac{(h_{si} - h_{ai}) - (h_{se} - h_{ae})}{\ln\dfrac{(h_{si} - h_{ai})}{(h_{se} - h_{ae})}}\right]$$

$$Q_a = m_a(h_{ae} - h_{ai})$$

$$Q_a = m_l c_{pl}(T_{li} - T_{le})$$

A.6.7 Explicit equations:

$$T_{ae} = \xi(h_{ae}, saturated)$$

A.7. HEATER

A.7.1 Parameters:

Q_a^*	Heater capacity [W]

A.7.2 Inputs:

m_a	Mass flow rate of air entering the heater [kg/s]
h_{ai}	Specific enthalpy of air entering heater [J/kg $_{\text{dry air}}$]
w_{ai}	Humidity ration of air entering the heater [kg $_{\text{vapour}}$/kg $_{\text{dry air}}$]

A.7.3 Outputs:

h_{ae}	Specific enthalpy of air leaving the heater [J/kg $_{\text{dry air}}$]
w_{ae}	Humidity ratio of air leaving the heater [kg $_{\text{vapour}}$/kg $_{\text{dry air}}$]

A.7.4 Explicit equations:

$$h_{ae} = h_{ai} + \frac{Q_a}{m_a}$$

$$w_{ae} = w_{ai}$$

A.8. PID CONTROLLER

A.8.1 Parameters:

t	Present time
Δt	Size of simulation time step
θ_{lo}	Throttling range (proportional band) low of controlled variable
θ_{hi}	Throttling range (proportional band) high of controlled variable
ϕ_{lo}	Low potential of final control element
ϕ_{hi}	High potential of final control element
k_I	Integral control factor
k_D	Derivative control factor

A.8.2 Inputs:

θ	Value of controlled variable

A.8.3 Outputs:

ϕ	Potential of final control element

A.8.4 Internal variables:

θ_{sp}	Set-point value of controlled variable
ϕ_{sp}	Set-point potential of final control element
k_P	Proportional control factor
ε	Error function

A.8.5 Explicit equations:

$$\theta_{sp} = \frac{\theta_{lo} + \theta_{hi}}{2}$$

$$\phi_{sp} = \frac{\phi_{lo} + \phi_{hi}}{2}$$

$$k_P = \frac{\phi_{lo} - \phi_{hi}}{\theta_{lo} - \theta_{hi}}$$

$$\varepsilon_t = \theta_t - \theta_{sp}$$

$$\begin{aligned} &if = \theta_{t-\Delta t} \le \theta_{lo} \quad then \quad \phi_t = \phi_{lo} \\ &else \quad if \quad \theta_{t-\Delta t} \ge \theta_{hi} \quad then \quad \phi_t = \phi_{hi} \\ &else \quad \phi_t = k_P \varepsilon_{t-\Delta t} + k_I \sum_{t=0}^{t=t-\Delta t} [\varepsilon \Delta t]_t + k_D \left[\frac{\varepsilon_{t-\Delta t} - \varepsilon_{t-2\Delta t}}{\Delta t} \right] \end{aligned}$$

A.9. STEP CONTROLLER

A.9.1 Parameters:

t	Present time.
Δt	Size of simulation time steps.
n	Number of steps.
$\theta_{L,1\ldots n}$	Loading set-points of controlled variable.
$\theta_{U,1\ldots n}$	Unloading set-points of controlled variable.
$\phi_{L,1\ldots n}$	Loading potential steps of final control element.

$\phi_{U,1\ldots n}$ Unloading potential steps of final control element.

A.9.2 Inputs:

θ Value of controlled variable.

A.9.3 Outputs:

ϕ Potential of final control element.

A.9.4 Explicit equations:

$$\text{If } \theta_{t-\Delta t} \geq \theta_{t-2\Delta t} \text{ and } \theta_{t-\Delta t} \geq \theta_{L,i} \text{ then } \phi_t = \phi_{L,i}$$

$$\text{Else if } \theta_{t-\Delta t} \prec \theta_{t-2\Delta t} \text{ and } \theta_{t-\Delta t} \prec \theta_{L,i} \text{ then } \phi_t = \phi_{U,i}$$

APPENDIX B

CONFERENCE FACILTIES

This appendix presents all the zone verification results and the new control results of application 2, chapter 4.

B.1. VERIFICATION RESULTS

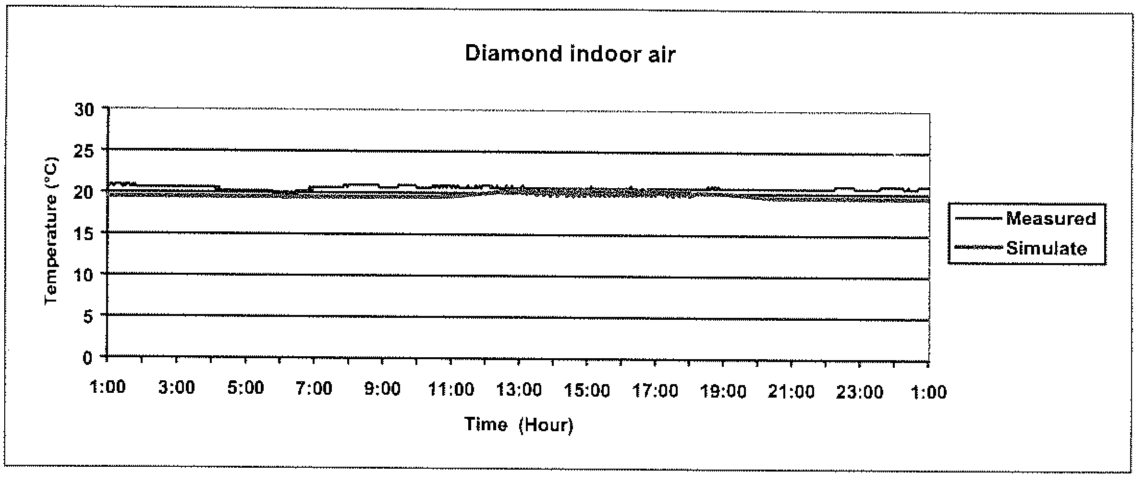

Figure B.1: Diamond indoor air verification study

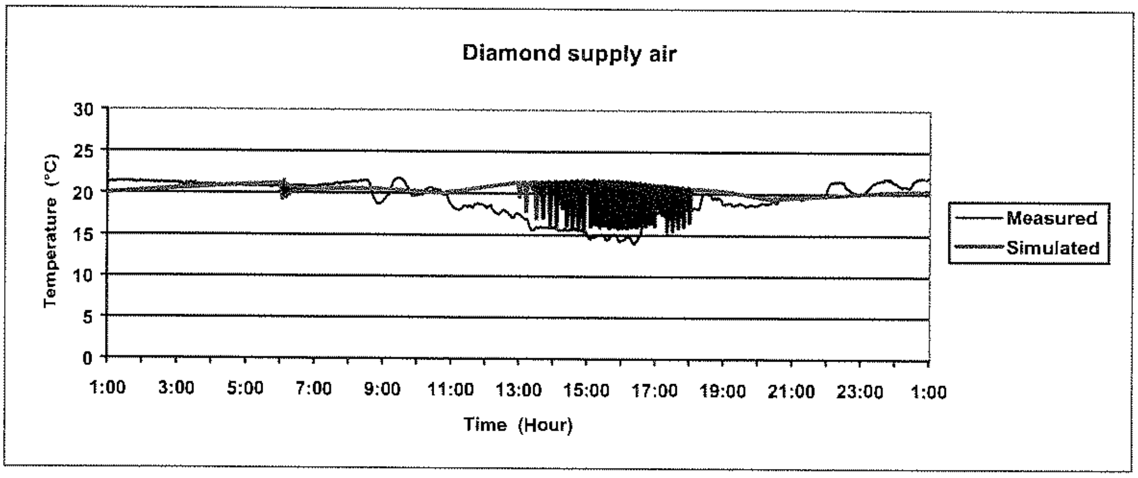

Figure B.2: Diamond supply air verification study

	MAX ERROR (°C)	AVERAGE ERROR (°C)	% OF TIME WITHIN 2 °C
SUPPLY AIR	6.8	1.5	75
INDOOR AIR	1.4	0.9	100

Table B.1: Summary of Diamond results

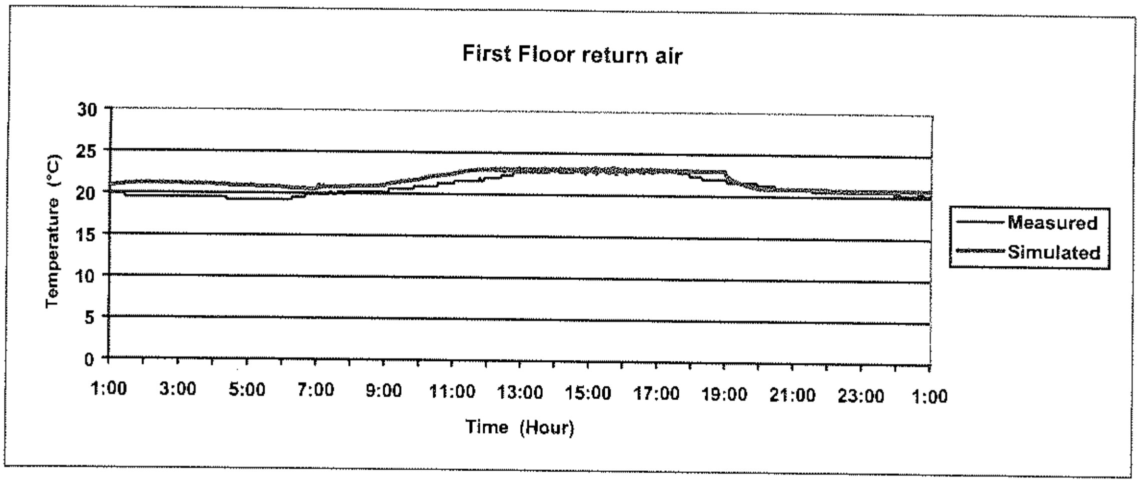

Figure B.3: First Floor return air verification study

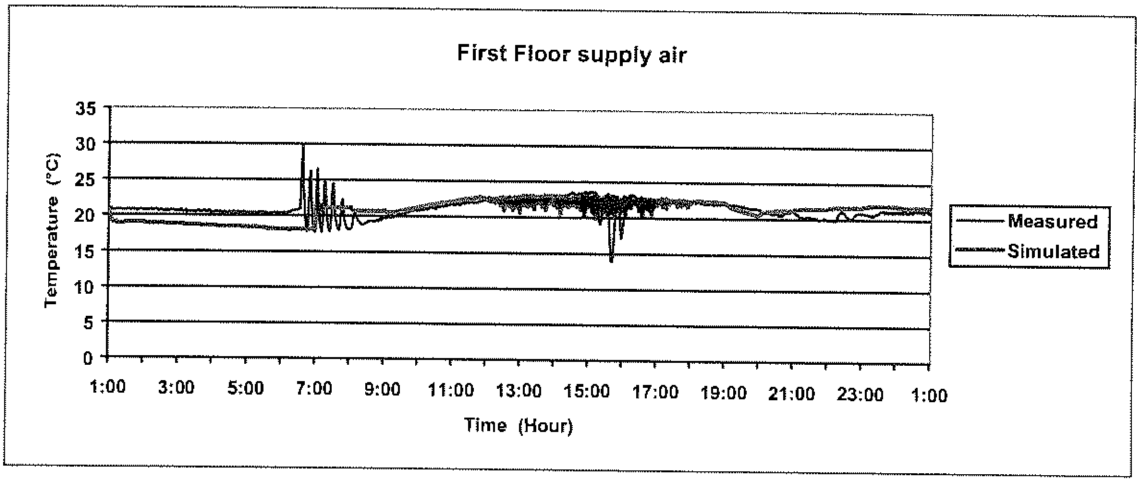

Figure B.4: First Floor supply air verification study

	Max error (°C)	**Average error (°C)**	**% of time within 2 °C**
Supply air	15	1.4	81
Return air	2.4	0.6	100

Table B.2: Summary of First Floor results

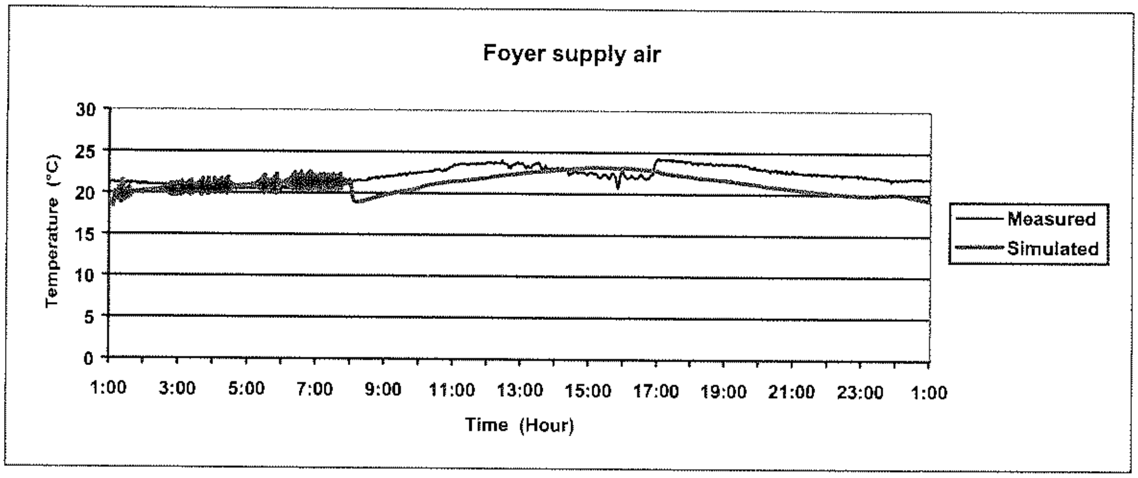

Figure B.5: Foyer supply air verification study

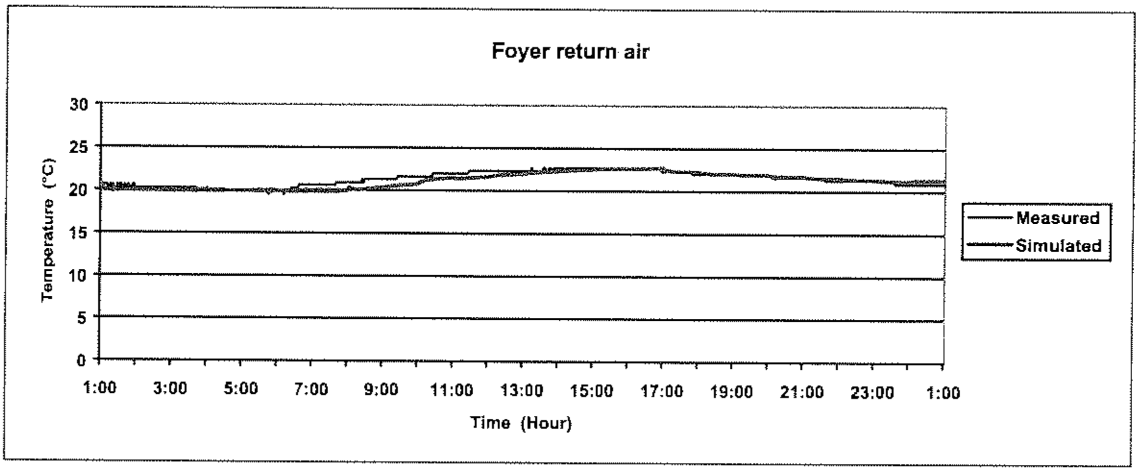

Figure B.6: Foyer return air verification study

	Max error (°C)	**Average error (°C)**	**% of time within 2 °C**
Supply air	3.4	1.3	72
Return air	4.9	0.2	100

Table B.3: Summary of Foyer results

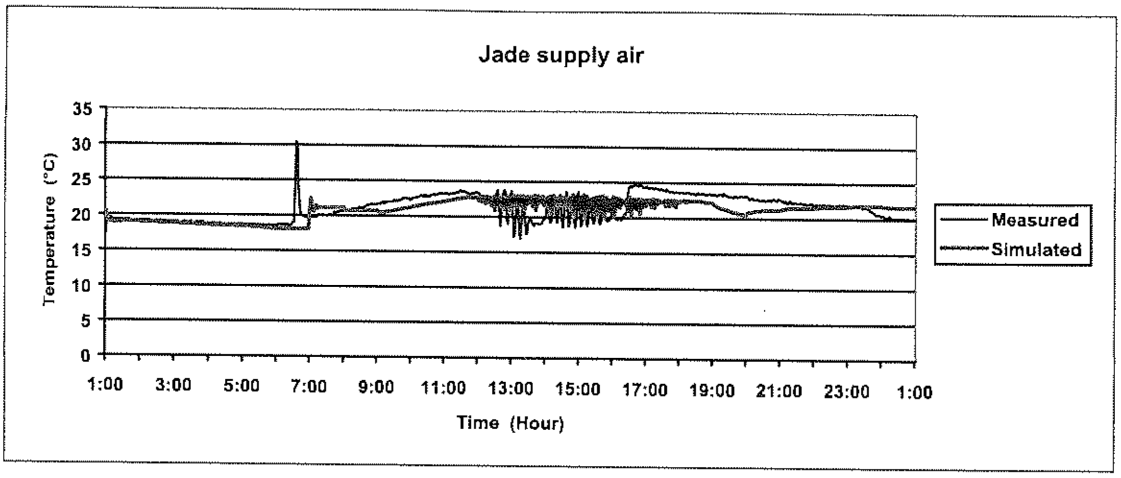

Figure B.7: Jade supply air verification study

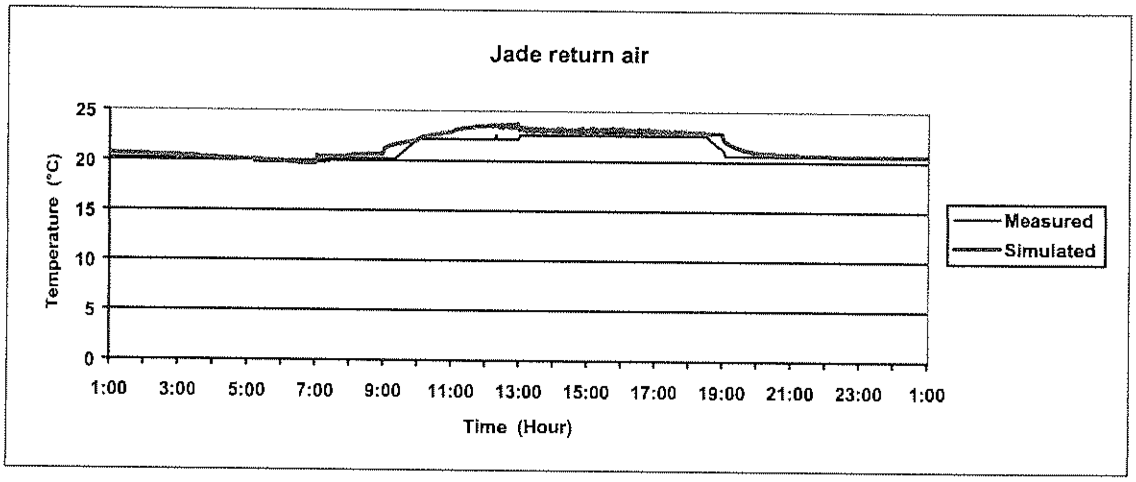

Figure B.8: Jade return air verification study

	Max error (°C)	**Average error (°C)**	**% of time within 2 °C**
Supply air	12	1.7	88
Return air	1.5	0.4	100

Table B.4: Summary of Jade results

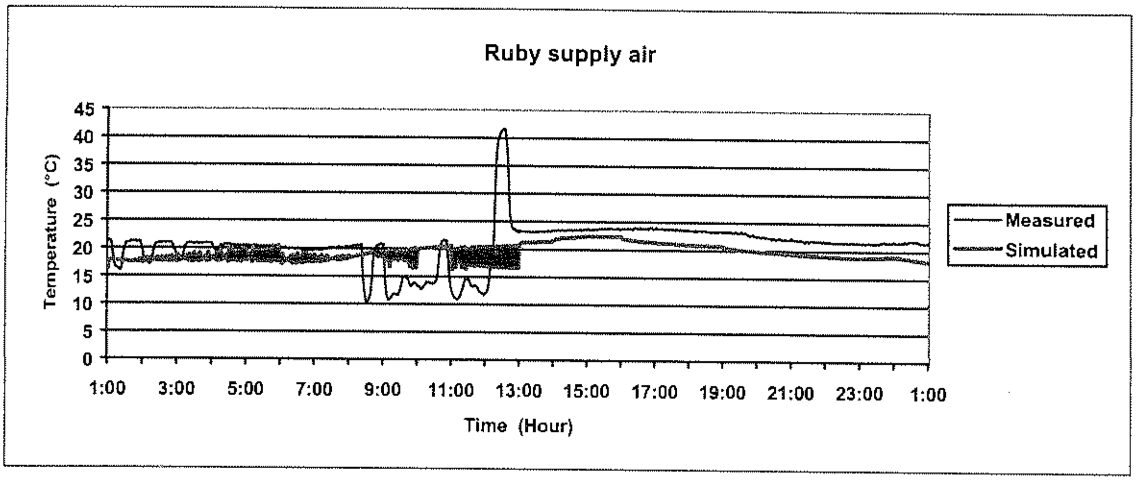

Figure B.9: Ruby supply air verification study

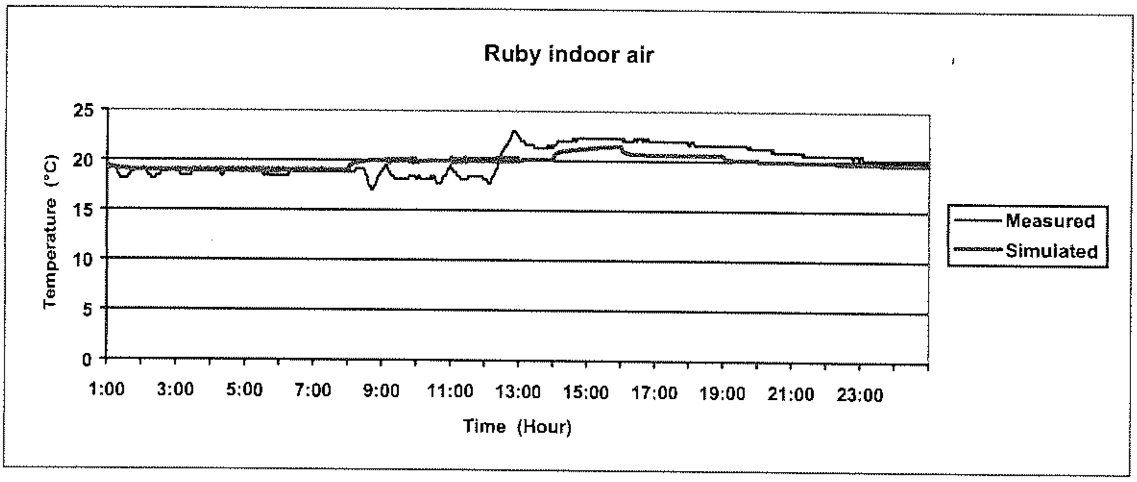

Figure B.10: Ruby indoor air verification study

	Max error (°C)	**Average error (°C)**	**% of time within 2 °C**
Supply air	20	3.1	97
Return air	3	1	100

Table B.5: Summary of Ruby results

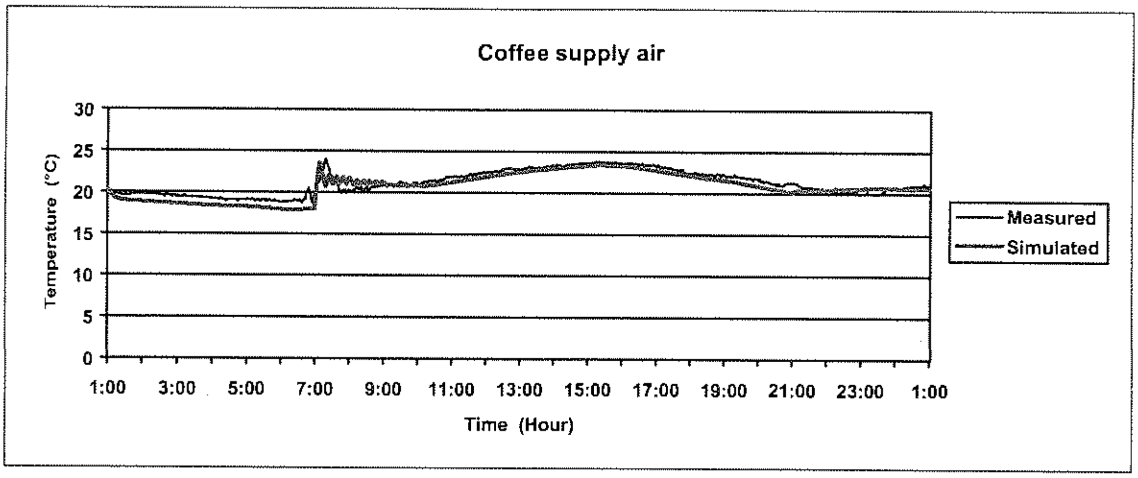

Figure B.11: Coffee supply air verification study

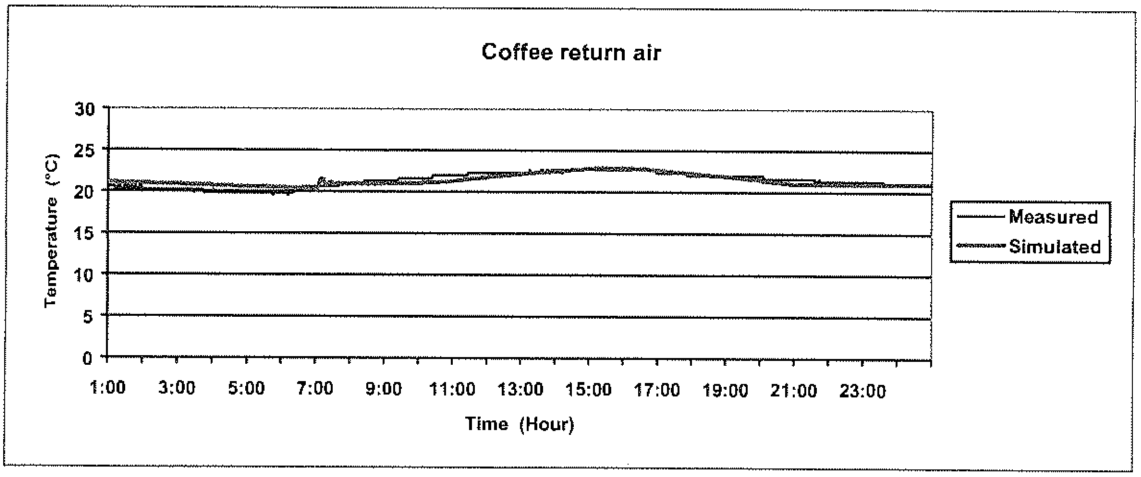

Figure B.12: Coffee return air verification study

	Max error (°C)	Average error (°C)	% of time within 2 °C
Supply air	3	0.6	99
Return air	1	0.4	100

Table B.6: Summary of Coffee results

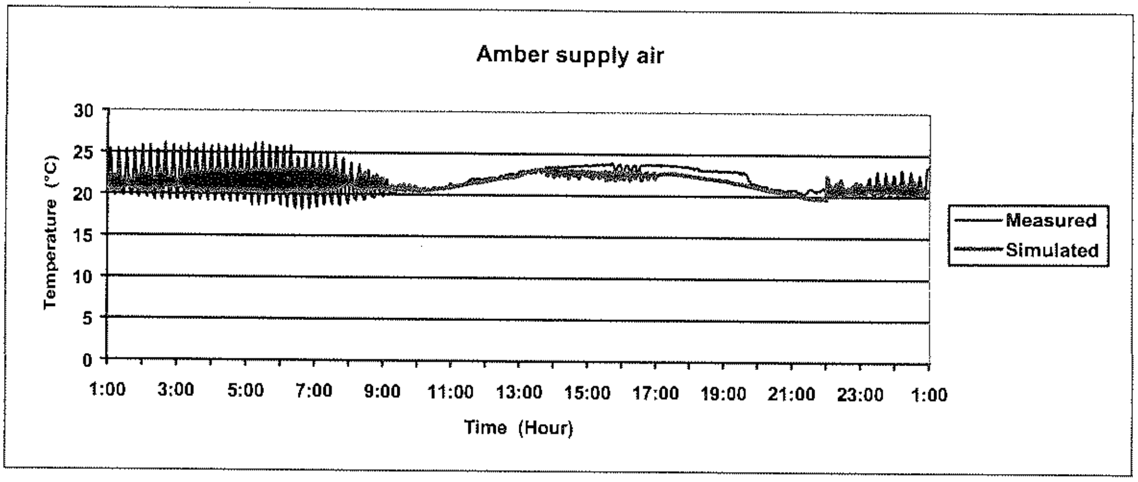

Figure B.13: Amber supply air verification study

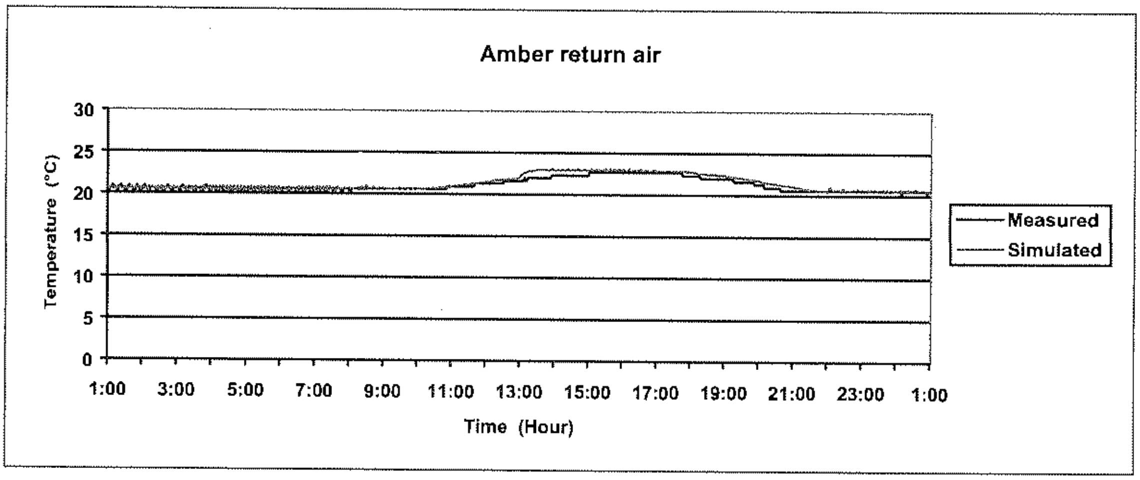

Figure B.14: Amber return air verification study

	Max error (°C)	Average error (°C)	% of time within 2 °C
Supply air	6.3	1.6	75
Return air	1.8	0.6	100

Table B.7: Summary of Amber results

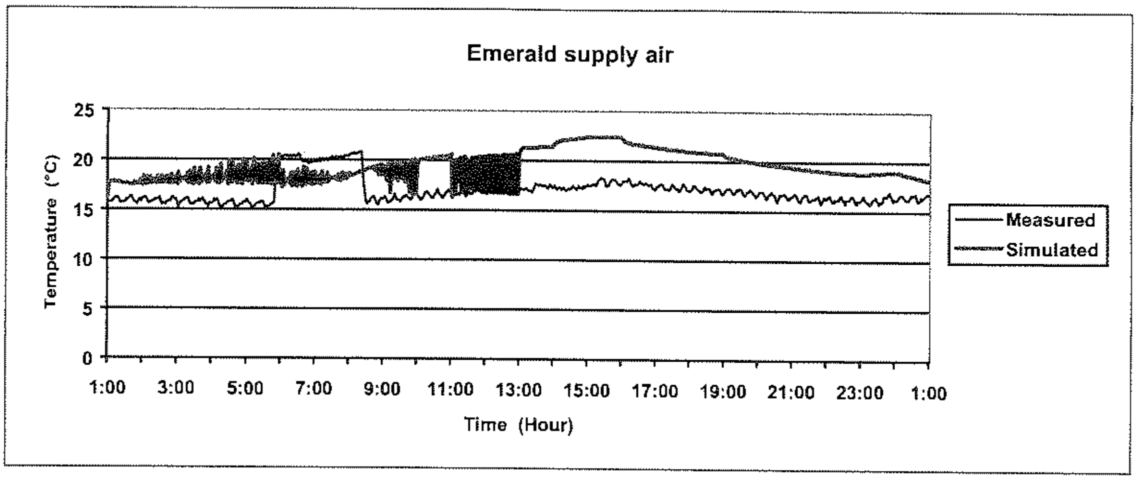

Figure B.15: Emerald supply air verification study

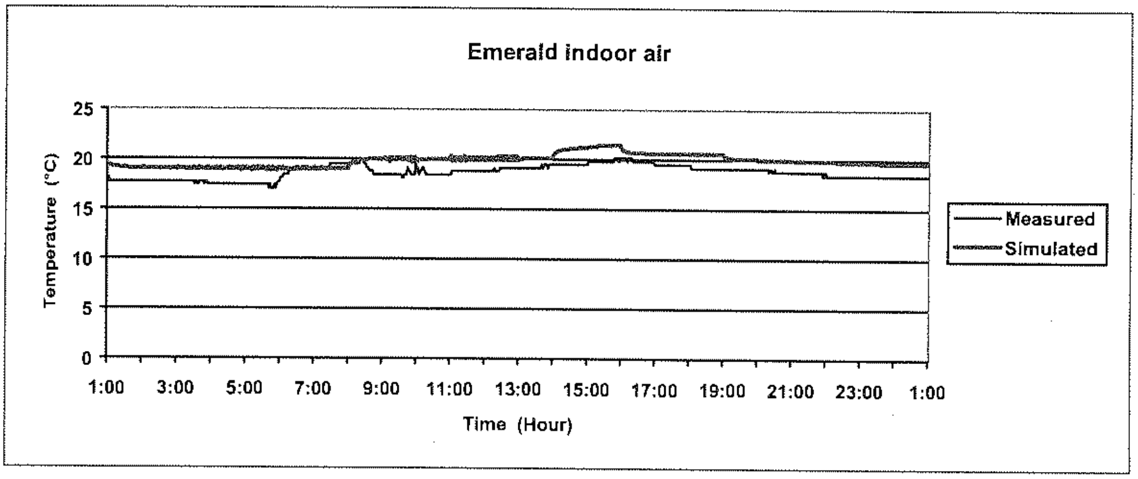

Figure B.16: Emerald indoor air verification study

	Max error (°C)	**Average error (°C)**	**% of time within 2 °C**
Supply air	5.4	2.8	27
Return air	2.1	1.1	100

Table B.8: Summary of Emerald results

B.2. ENERGY AND POWER RESULTS

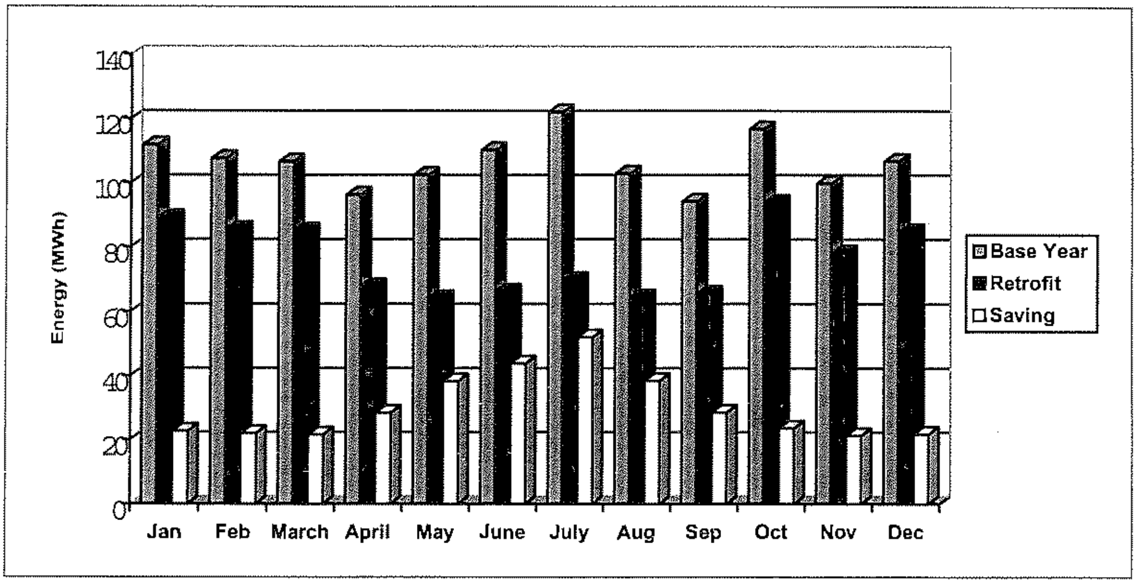

Figure B.17: Fan scheduling energy consumption result

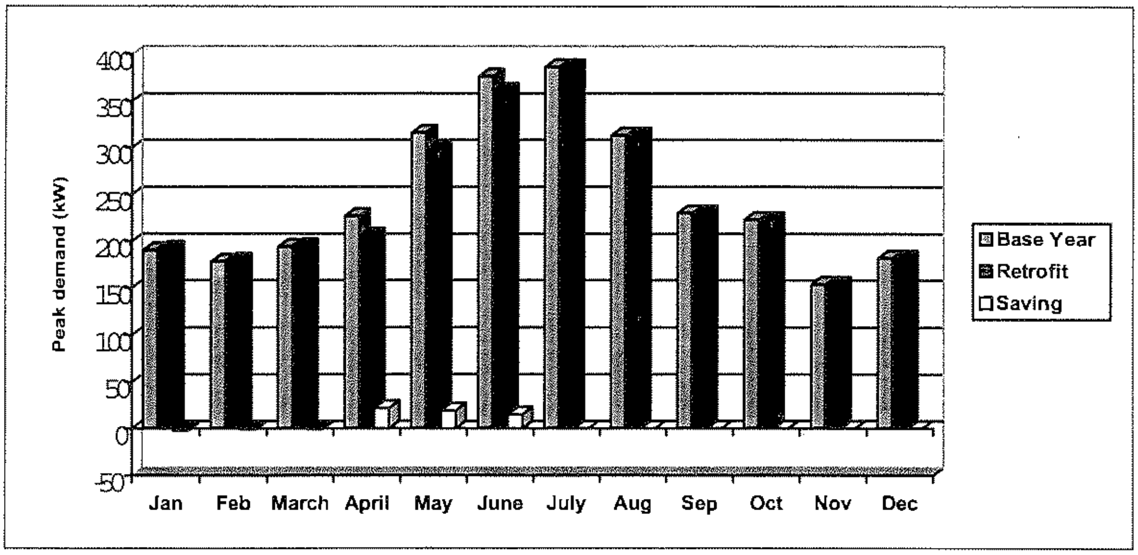

Figure B.18: Fan scheduling peak demand simulation result

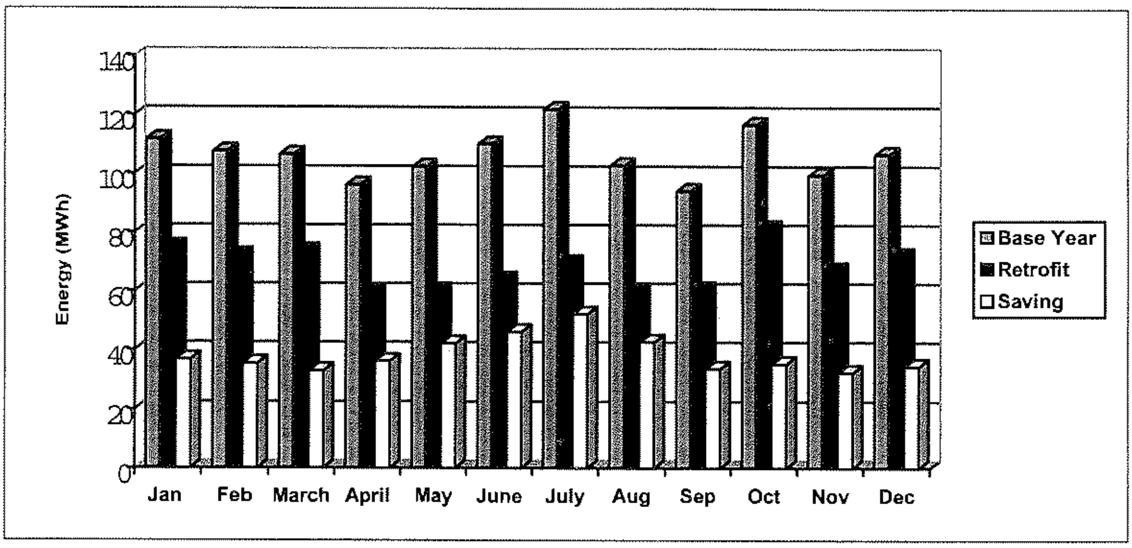

Figure B.19: Fan scheduling, economiser and setpoint energy simulation result

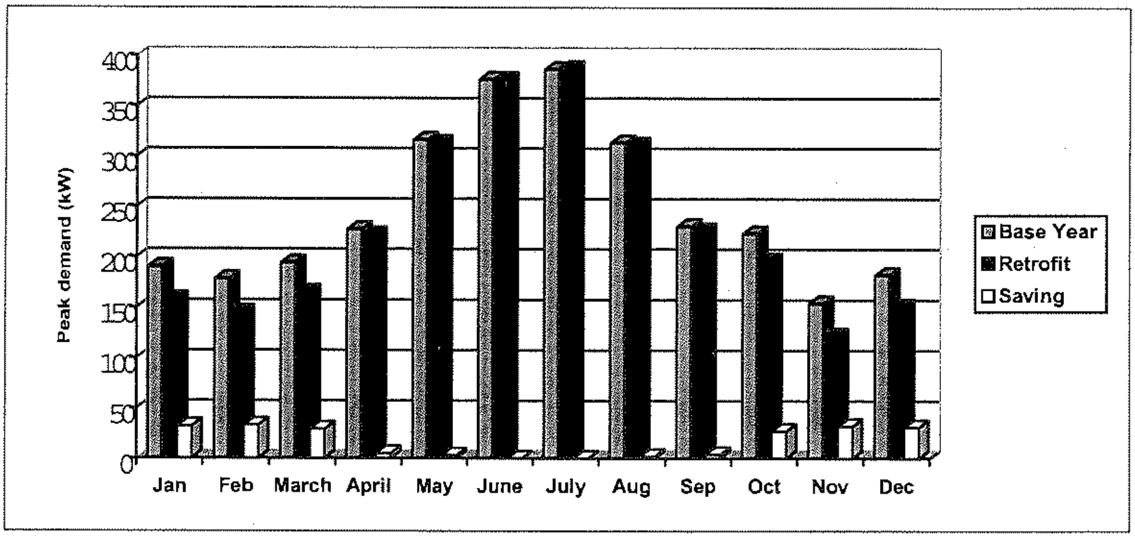

Figure B.20: Fan scheduling, economiser and setpoint peak demand simulation result

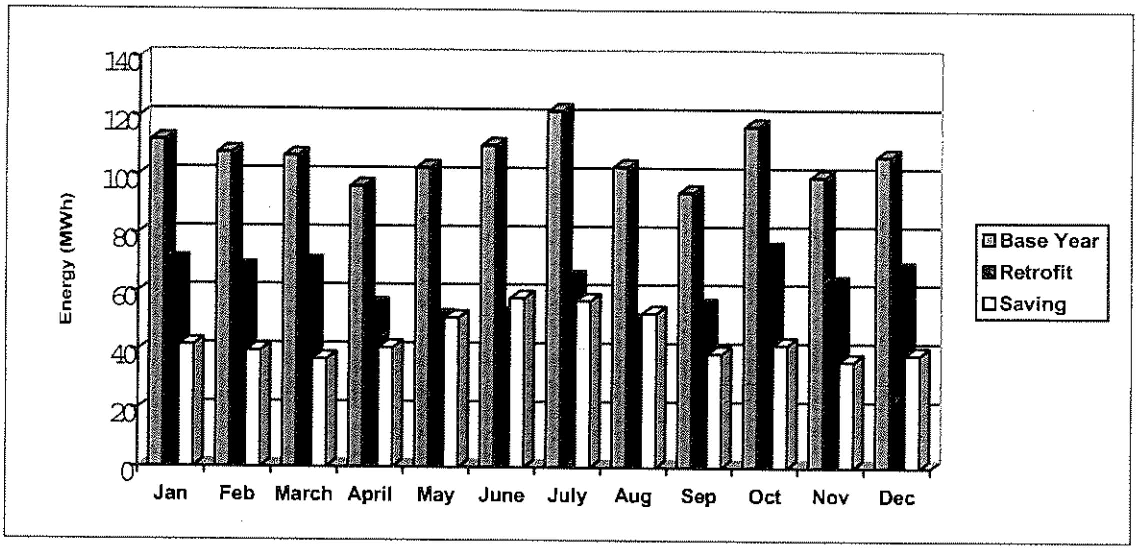

Figure B.21: Fan scheduling, economiser and setback control energy simulation result

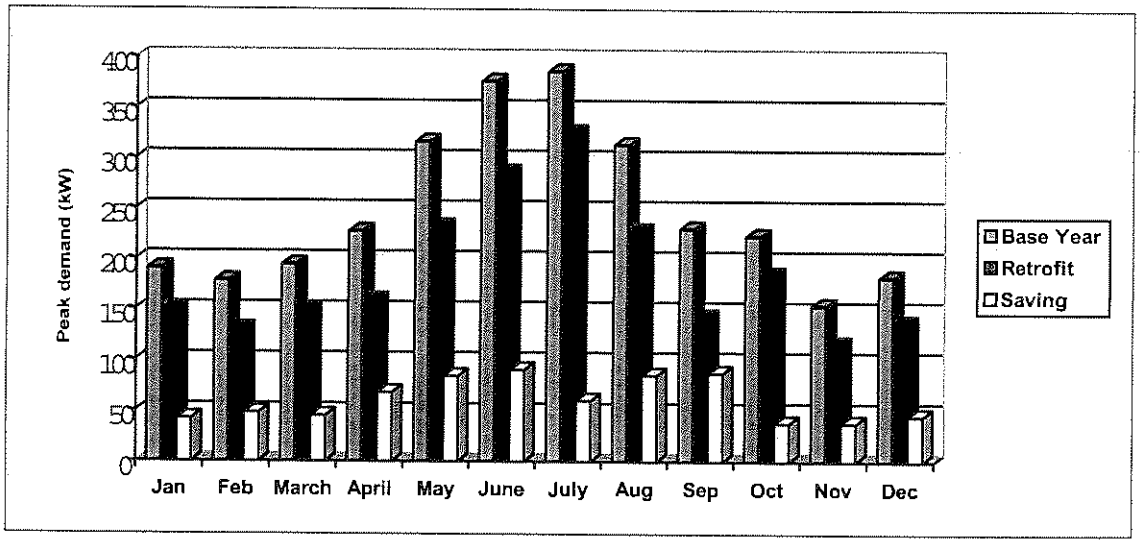

Figure B.22: Fan scheduling, economiser and setback control demand simulation result

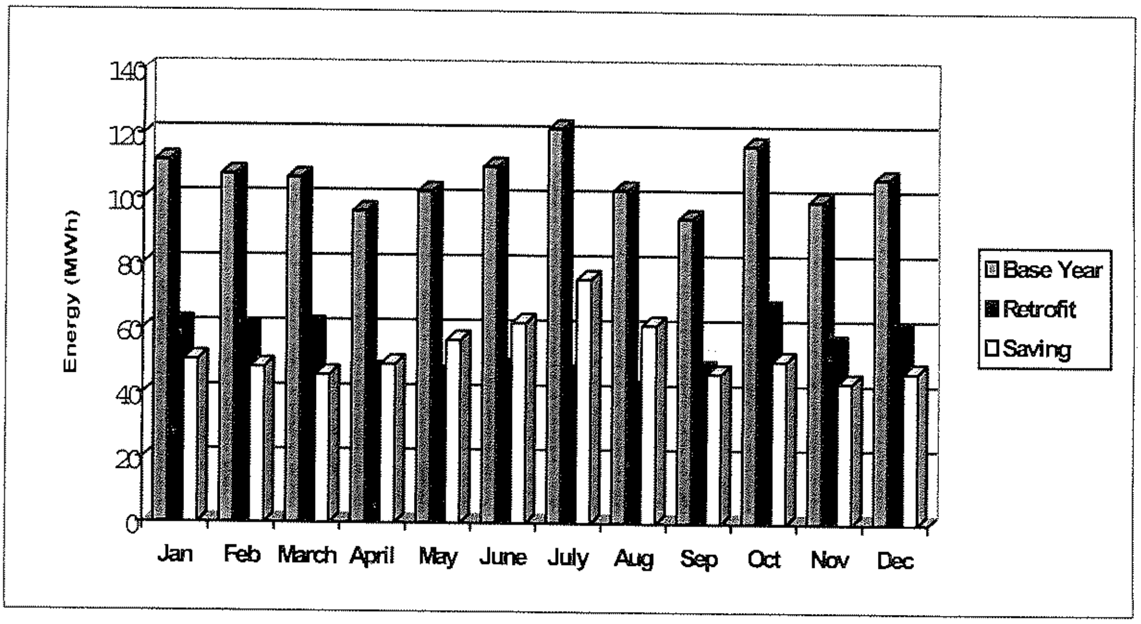

Figure B.23: Fan scheduling, economiser, setback and fan control energy simulation result

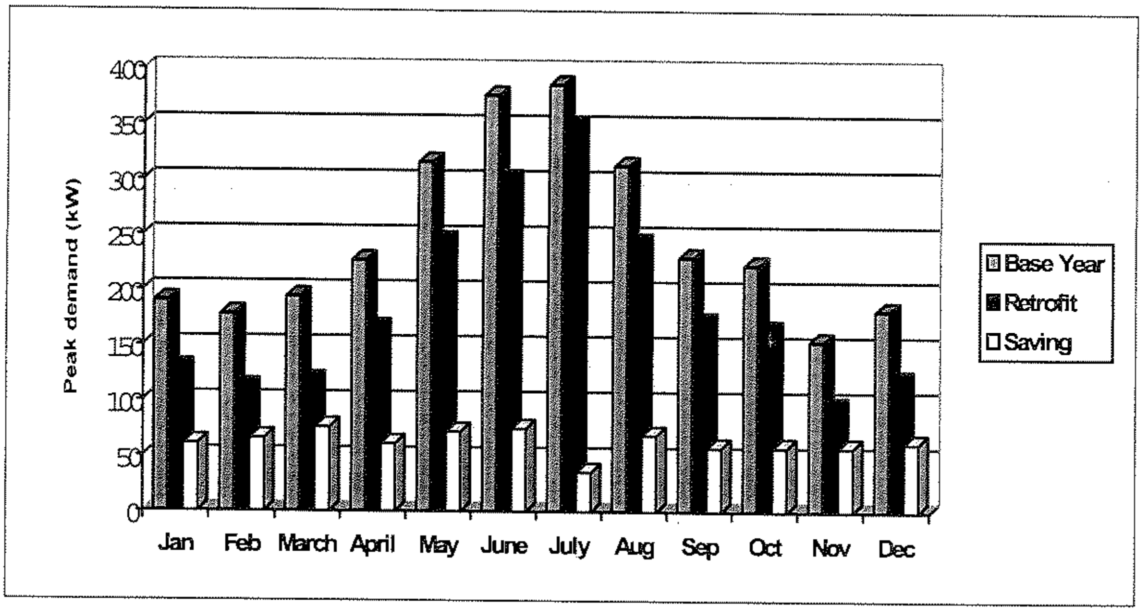

Figure B.24: Fan scheduling, economiser, setback and fan control simulation result

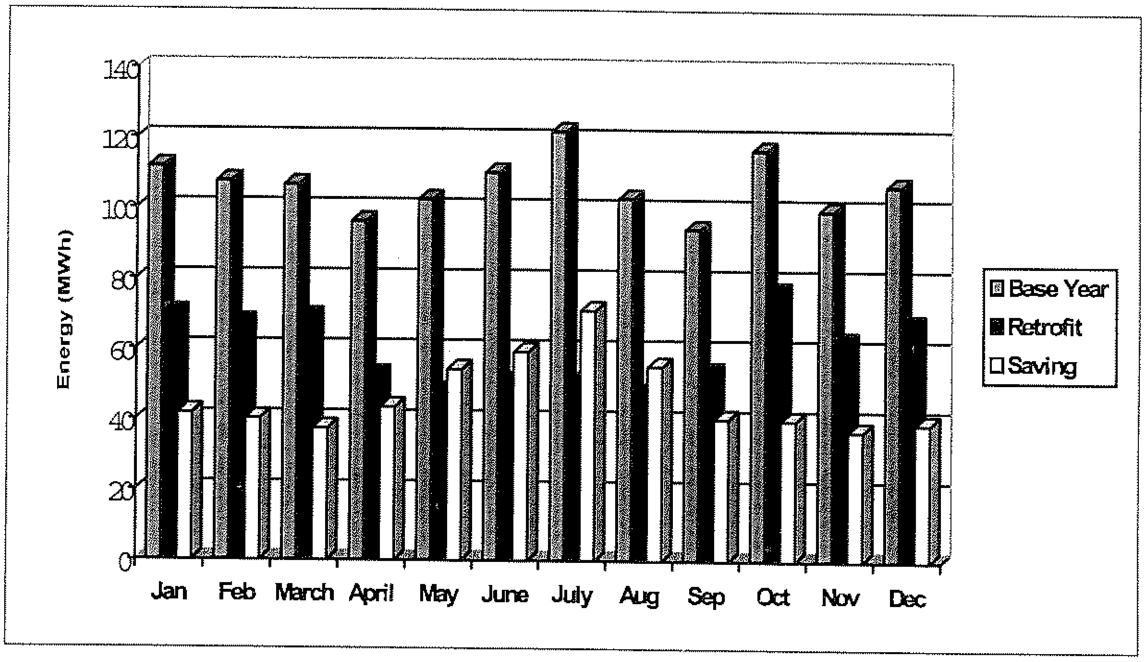

Figure B.25: Fans scheduling, economiser, setpoint and boiler control energy simulation result

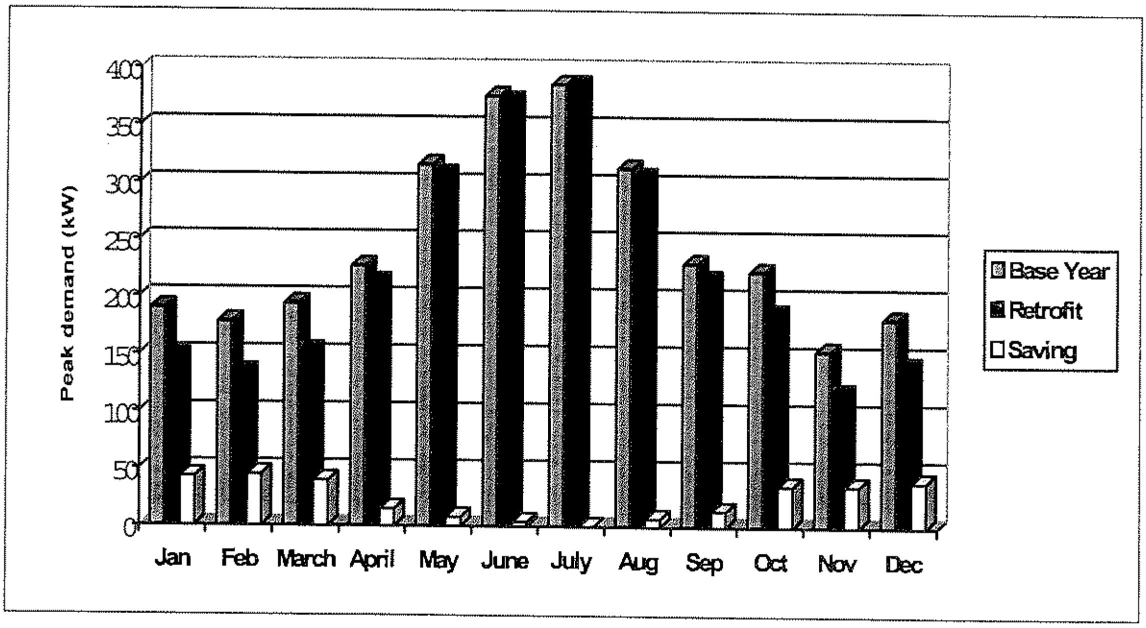

Figure B.26: Fan scheduling, economiser, setpoint and boiler control peak simulation result

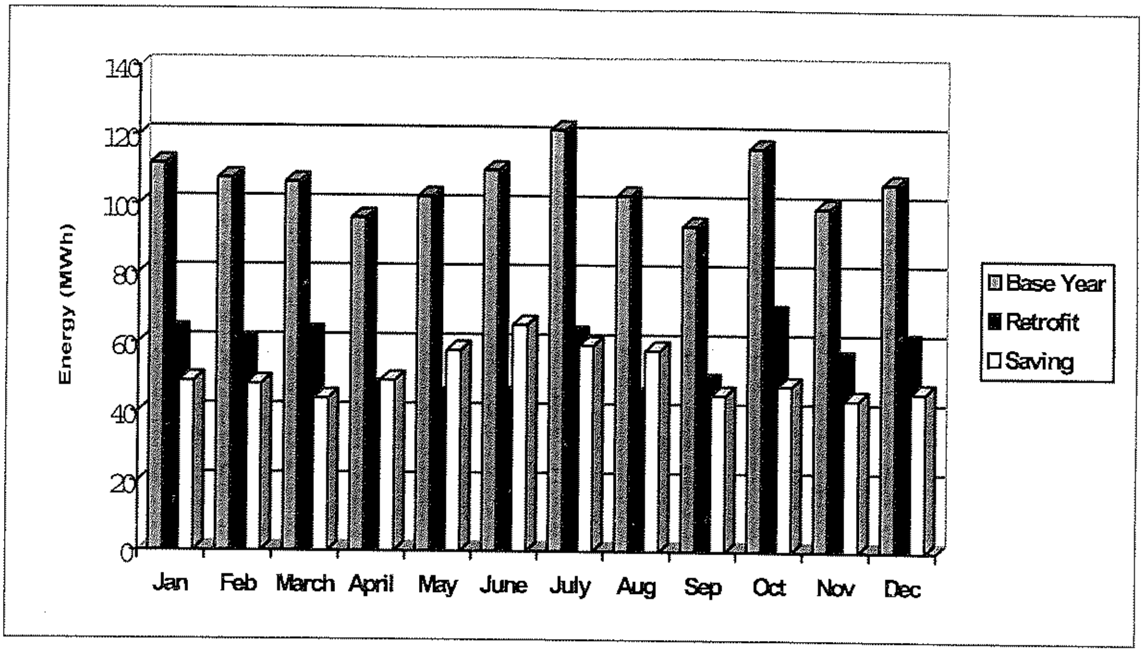

Figure B.27: Fan scheduling, economiser, setback and boiler energy simulation result

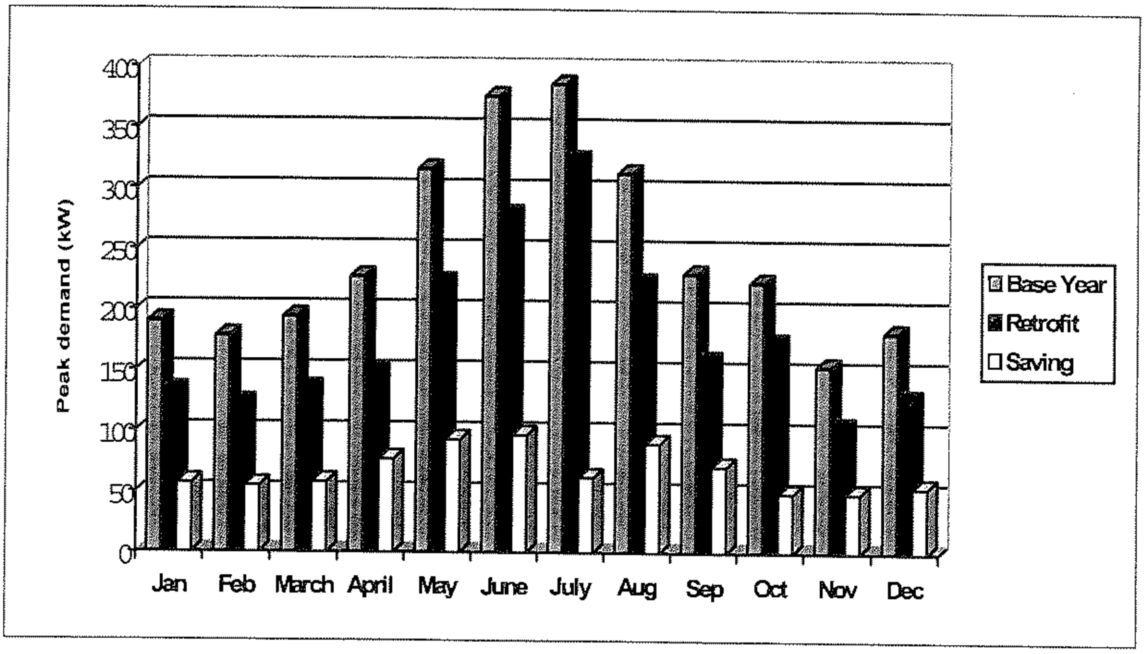

Figure B.28: Fan scheduling, economiser, setback and boiler control peak simulation result

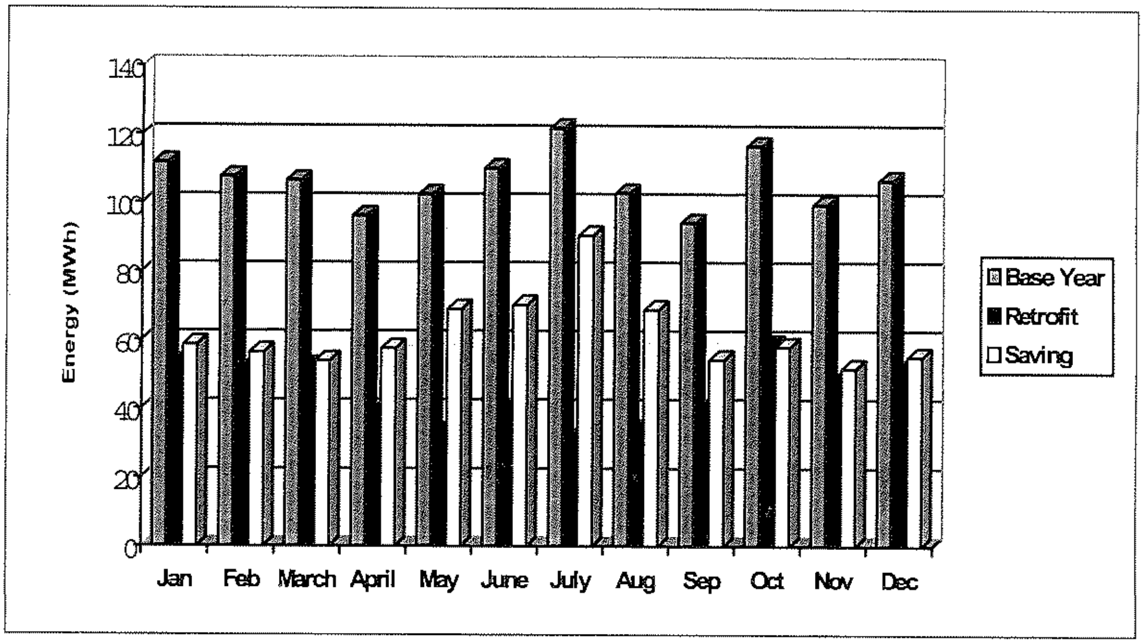

Figure B.29: Fan scheduling, economiser, setback, fan and boiler control energy simulation result

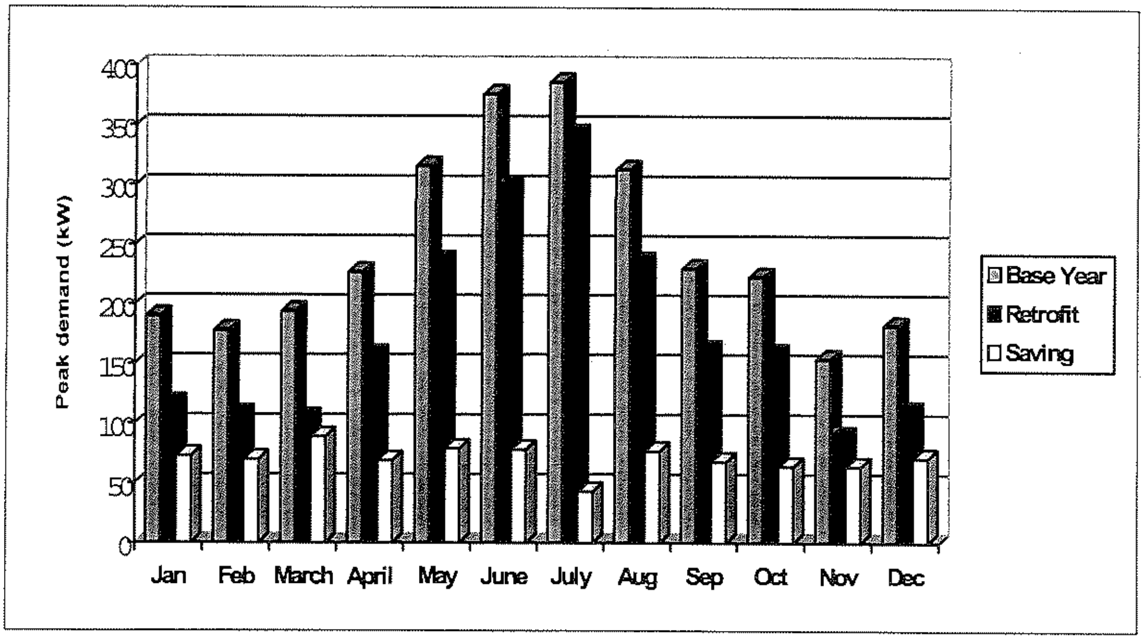

Figure B.30: Fan scheduling, economiser, setback, fan and boiler control peak simulation result

APPENDIX C

THE POTENTIAL FOR DSM ON MINE COOLING SYSTEMS

This appendix presents all the equipment specifications and the verification results of application 1, chapter 8.

C.1. EQUIPMENT SPECIFICATIONS

A more detailed look at the pre-cool towers and the surface cooling plant can be seen in Figure C.1 and Figure C.2. Scheme A is shown in Figure C.2.

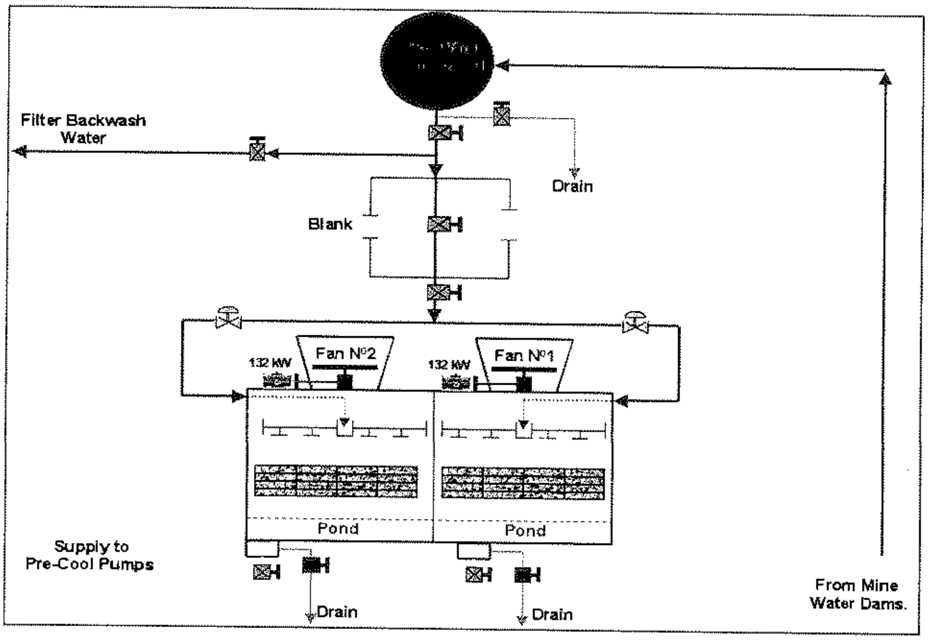

Figure C.1: Pre-cooling tower configuration

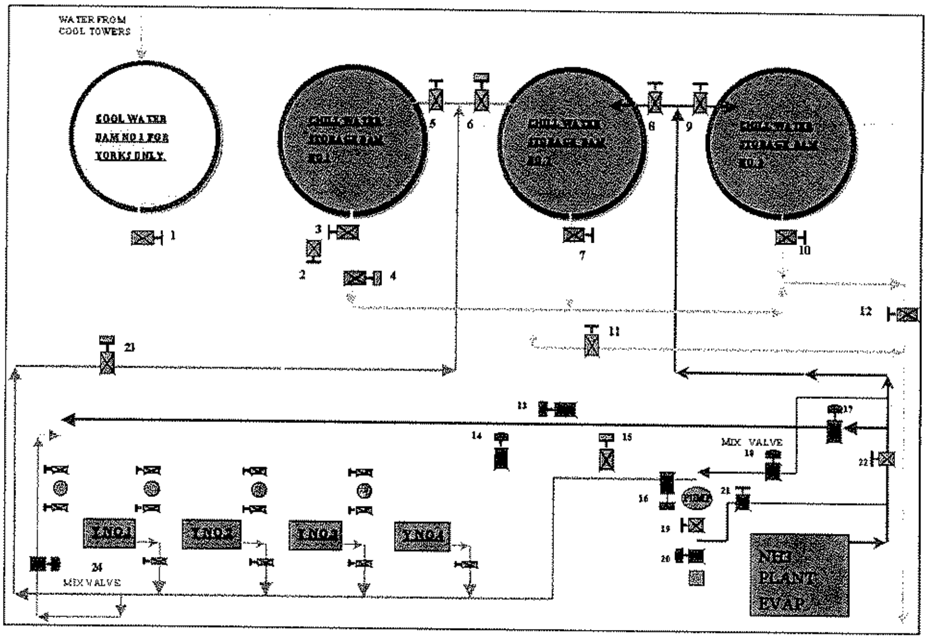

Figure C.2: Surface cooling plant

C.1.1 Dams

Dam	Purpose	Volume [m^3]	Open/ Closed	Controlled by
Mine Water	Collect water from mine	3817.7	Open	-
Hot Water	Hot water pumped from Mine water dam	2709	Open	Level
Pre-Cool	Collect water from Pre-cool Towers	318.1	Open	Level
Cool Water	Store water after Pre-cool Towers	2709	Closed	Level
Chill 1	Store water after York chillers	2709	Closed	Level
Chill 2	Store water after York or Howden chillers	2709	Closed	Level
Chill 3	Store water after Howden chiller	2709	Closed	Level

Table C.1: Summary of dam

C.1.2 Pumps

Table C.2 presents a summary of the pumps used in the surface cooling cycle. The various pump curves are shown in the following pages.

Pump description	Pump (Type)	Impeller diameter (m)	Flow [l/s]	Power [kw]	Cotrolled by:
Mine Water	KSB ETA 250/50	0.41	300	200	Hot dam level
Filter pumps - A	Sulzer AZS 200/250	0.25	75	15	Cool dam Level
Filter pumps - B	Allis-Chalmer	0.3	120	30	Cool dam Level
York Evaporator	Salweir SDB 200/250	0.32	114	55	Chill 1 dam Level
York Condensor	Salweir SDB 10/12	0.36	280	132	Chill 1 dam Level
Howden Evap	Salweir SDB 350/450	0.415	420	160	Chill 2&3 dam Level
Howden Cond	Salweir SDB 350/450	0.475	420	185	Chill 2&3 dam Level

Table C.2: Summary of pump specifications

C.1.3 Cooling Towers

There are basically two sets of cooling towers. Firstly is the pre-cooling tower used to cool the water from the hot water dam. Secondly, there are the cooling towers on the condenser sides of the chillers. The two types are almost the same, therefore only one tower, the pre-cooling tower, will be discussed. Table C.3 shows all the data of the pre-cooling tower.

Performance Data	
Inlet Water Temp. (°C)	28
Outlet Water Temp. (°C)	18.5
Inlet WB Air Temp. (°C)	16
Outlet WB Air Temp. (°C)	23.3
Pressure (kPa)	85
Losses (%)	1.63
Water Flow (m^3/h)	1728
Per Cell (m^3/h)	864
Air flow/cell (m^3/s)	525.8
Static Pressure drop (Pa)	136.8
Heat Exchanged (MW)	19.06

Fan Details	
Diameter (m)	7.315
RPM	161
Number of blades	6
Pitch of blades (°)	16.3
Manufacturer	Howden
Type	ENF

Motor Details	
Power per motor (kW)	132
RPM	1480
Absorbed power (kW)	117.4
Manufacturer	ZEST/WEG

Construction Details	
Number of cells	2
Total Plot area (m^2)	14.85x28.75
Fill Area (m^2)	14.34x14
Induced draught	
Air opening (m^2)	80
Pitch of fill (m)	0.25
Fill Depth (m)	3.25
Height to bottom of fill (m)	3

Table C.3: Specifications for Pre-cooling Tower

C.1.4 Chillers

The specifications for the York chillers are shown in Table C.4 and for the Howden in Table C.5.

Design Details

Voltage (V)	11000
Amps (A)	85.5
Bearing Temp. - DE (°C)	60
Bearing Temp. - NDE (°C)	60
Stator Temperature - Phase1	100
Stator Temperature - Phase2	100
Stator Temperature - Phase3	100
Barometric Pressure (kPa)	84
% Vane opening	100
Oil Level	Halftop
Oil Temp. (°C)	60
Oil Press. (kPa)	450
Differantial oil Press. (kPa)	210
Suction Temp. (°C)	2
Suction Press. - Gauge (kPa)	240
Suction Press. - Absolute (kPa)	324
Corresponding Temp. (°C)	1.62
Suction Superheat (°C)	0.4
Discharge Temp. (°C)	50.5
Condenser Press.-Gauge (kPa)	670
Condenser Press.-Absolute(kPa)	754
Corresponding Temp. (°C)	30.52
Discharge superheat (°C)	20.5
High Pressure Liquid Temp. (°C)	30
Degrees system Air	0.5

Chilled water inlet Temp. (°C)	14.5
Chilled water outlet Temp. (°C)	4
Chilled water Delta T (°C)	10.5
Chilled water flow (l/s)	114
Evaporator duty (kW)	5012
Condenser water outlet Temp.(°C)	27.5
Condenser water inlet Temp. (°C)	22
Condenser water Delta T (°C)	5.5
Condenser water flow (l/s)	270
Condenser duty (kW)	6217
Compressor shaft power (kW)	1264
Coeffiesient of performance	3.96
Carnot COP	9.5
Cycle efficiency (%)	41.71
Power to cooling Ratio	0.25
LMTD Condenser	4.73
LMTD Evaporator	6.21

Table C.4: Specifications for York Chillers

Design Details - Normal running

Voltage (V)	11000	Evap. Ammonia Inlet Temp. (°C)	1
Amps (A)	160	Evap. Ammonia Outlet Temp. (°C)	10
Bearing Temp. - DE (°C)	45	Evap. Water Inlet Temp. (°C)	11
Bearing Temp. - NDE (°C)	34	Evap. Water Outlet Temp. (°C)	5
Stator Temperature - Phase1	59	Chilled water Delta T (°C)	5.5
Stator Temperature - Phase2	63	Chilled water flow (l/s)	420
Stator Temperature - Phase3	60	Evaporator duty (kW)	8300
Barometric Pressure (kPa)	84	Cond. Ammonia Inlet Temp. (°C)	40
% Vane opening	100	Cond.Ammonia Outlet Temp.(°C)	28
Oil Level	Halftop	Cond. water outlet Temp.(°C)	30
Oil Temp. in Seperator (°C)	40	Cond. water inlet Temp. (°C)	21
Oil Temp. in Manifold (°C)	40	Cond. water Delta T (°C)	9
Oil Filter Diff. Press. (kPa)	10	Cond. water flow (l/s)	350
Differantial oil Press. (kPa)	440	Cond. duty (kW)	11340
Suction Temp. (°C)	3	Compressor shaft power (kW)	1462
Surge drum Press. (kPa)	417	Coeffiesient of performance	5.7
Surge drum Level (%)	7	Carnot COP	13.434
Discharge Temp. (°C)	40-50	Cycle efficiency (%)	42.3
Discharge Press. (kPa)	1230	Power to cooling Ratio	0.1761

Table C.5: Specifications for Howden Chiller

C.2. VERIFICATION OF TEMPERATURES, LEVELS AND FLOWS

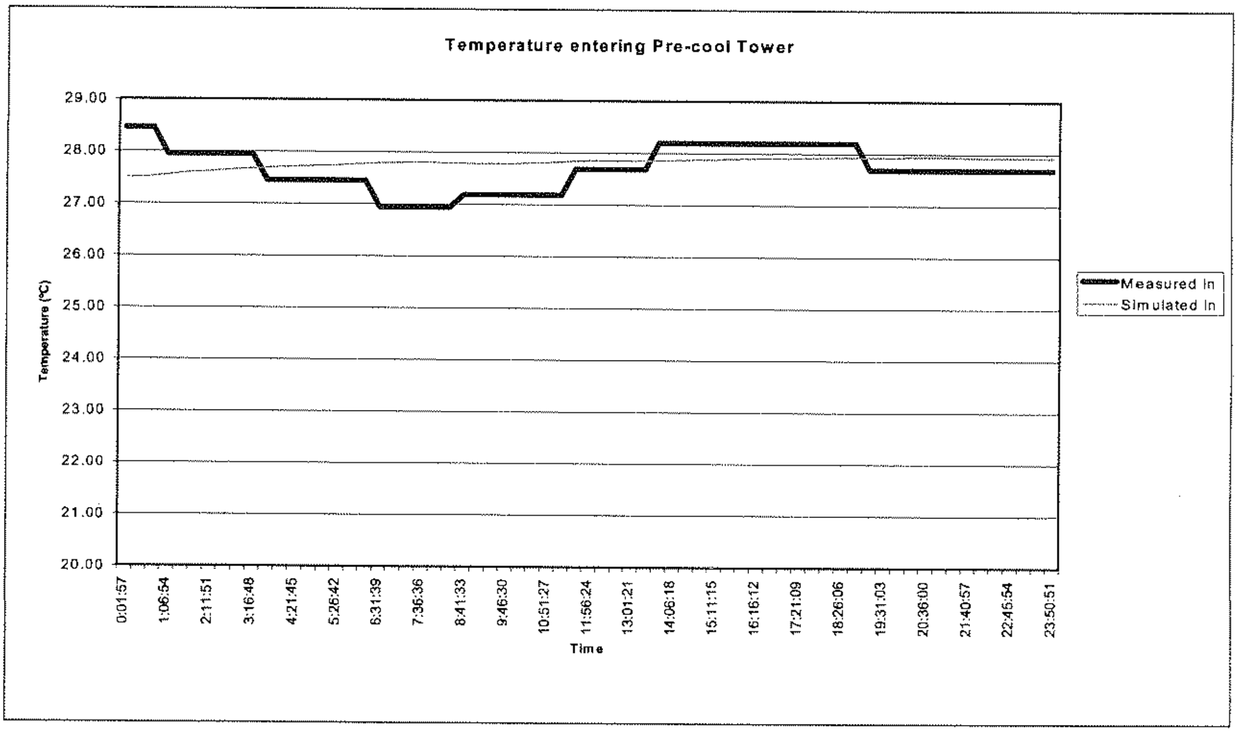

Figure C.3: Pre-cool towers inlet temperature verification

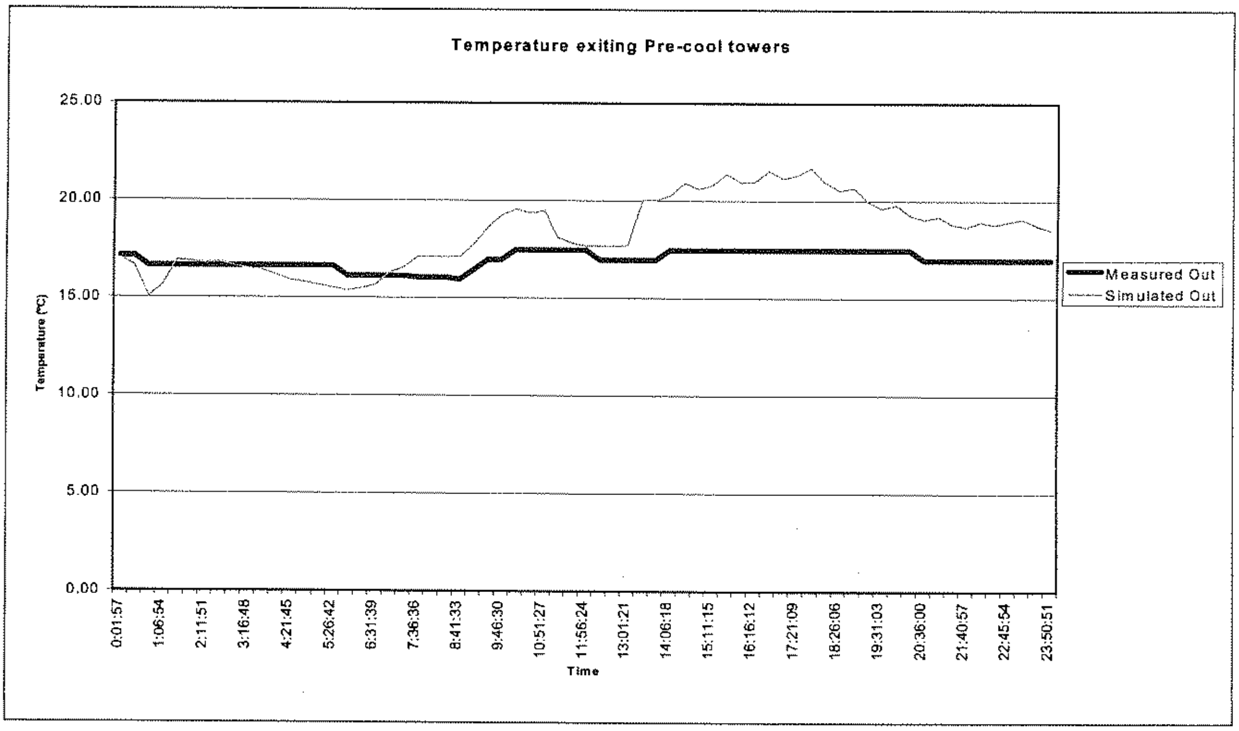

Figure C.4: Pre-cool tower exiting water temperature verification

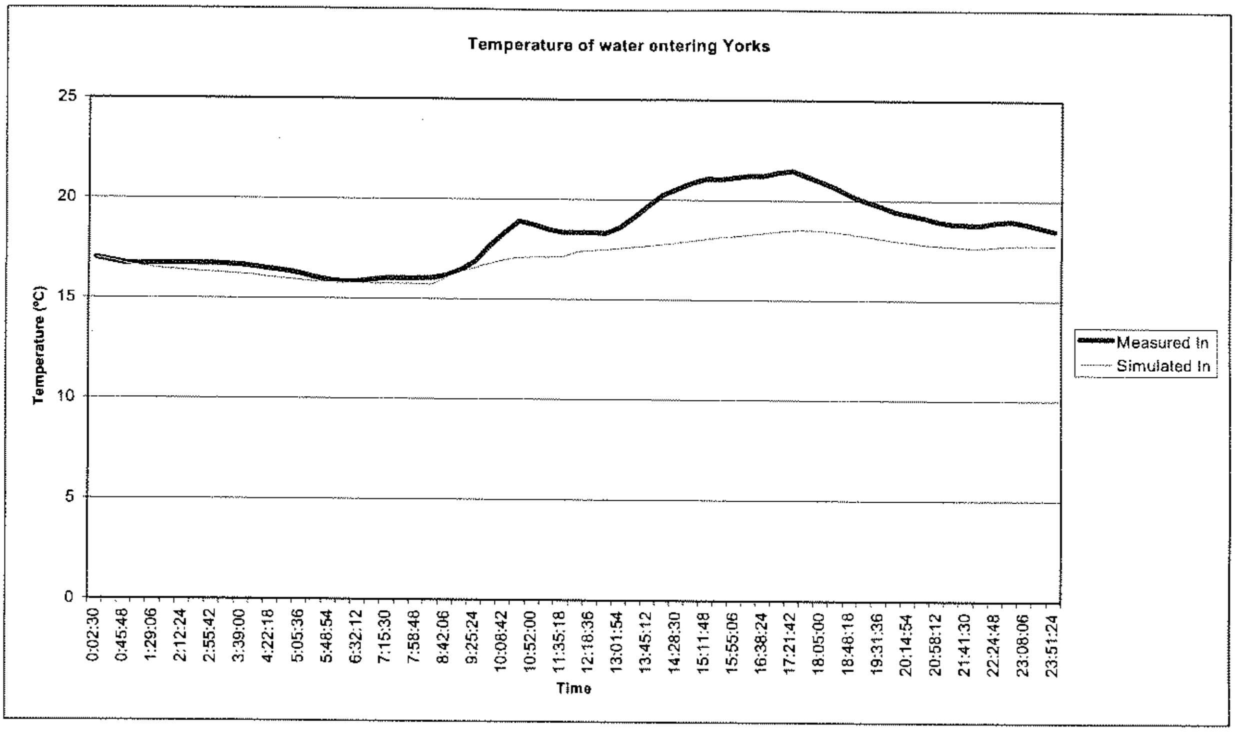

Figure C.5: Temperature of water entering the York chillers

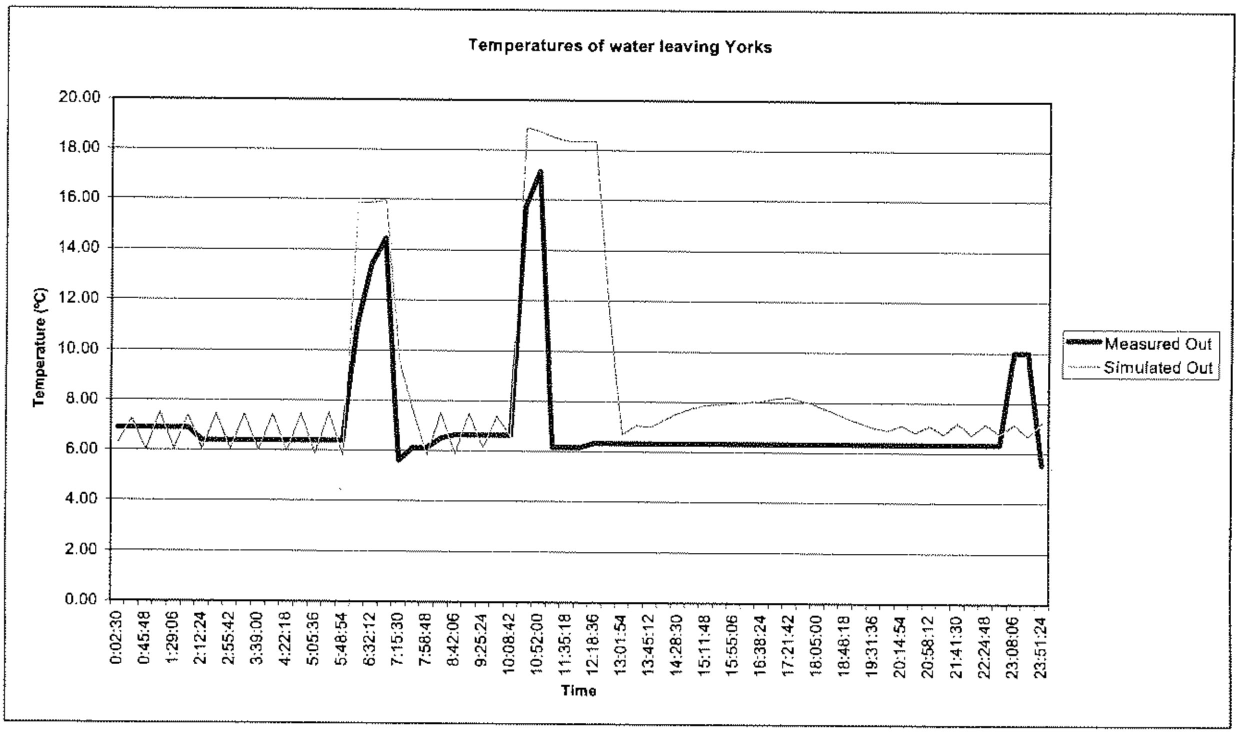

Figure C.6: Temperature of water exiting York chillers

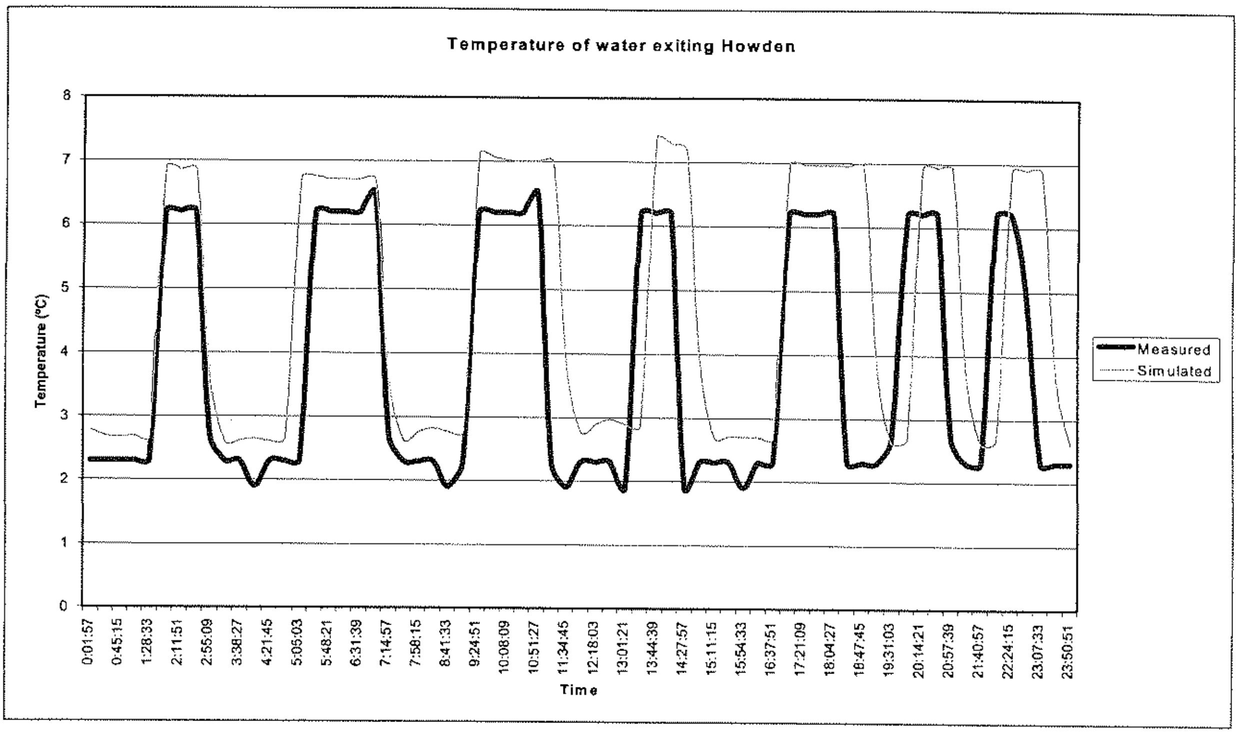

Figure C.7: Temperature of water exiting Howden chiller

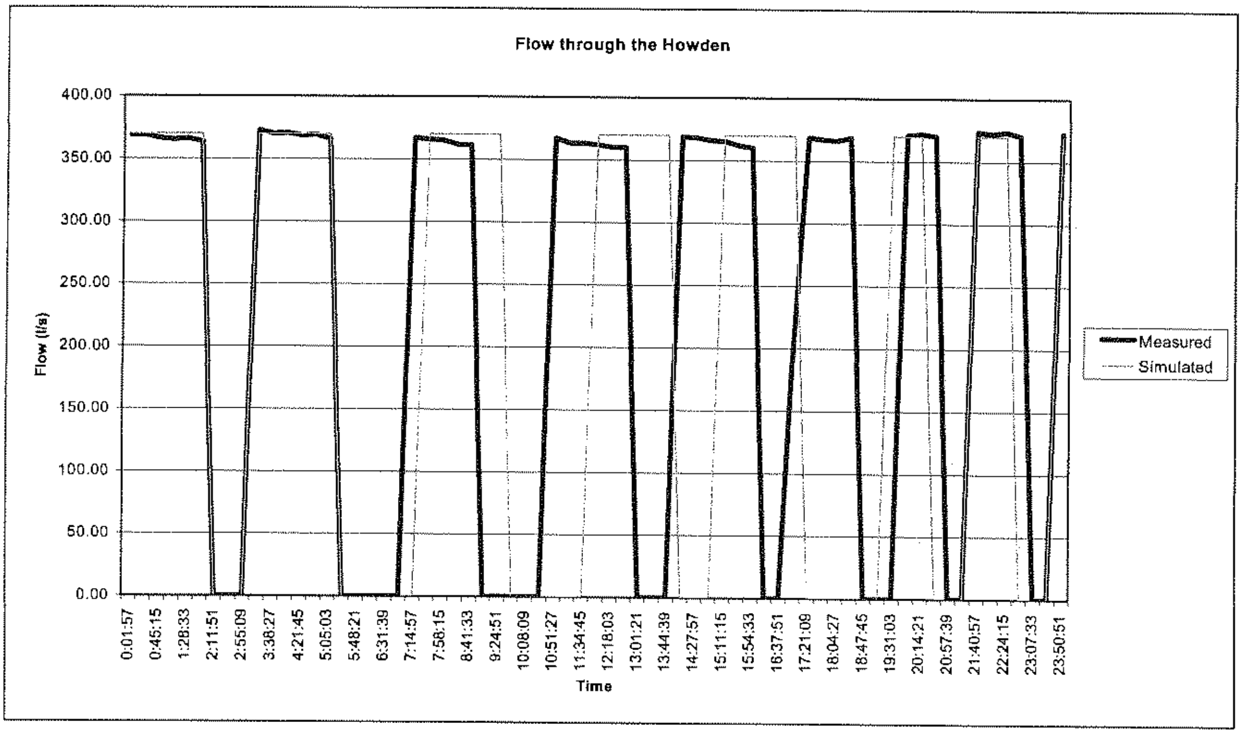

Figure C.8: Flow through the Howden Chiller

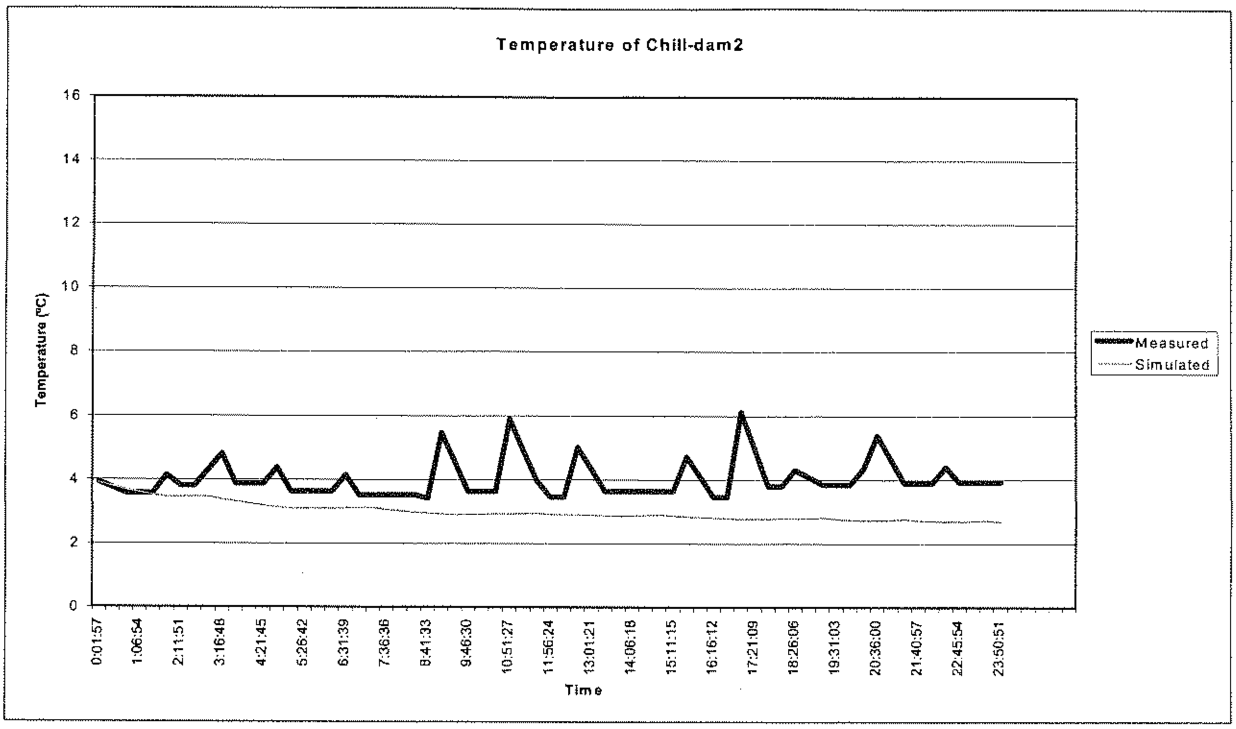

Figure C.9: Temperature of Chill-dam2

C.3. VERIFICATION OF POWER CONSUMPTION

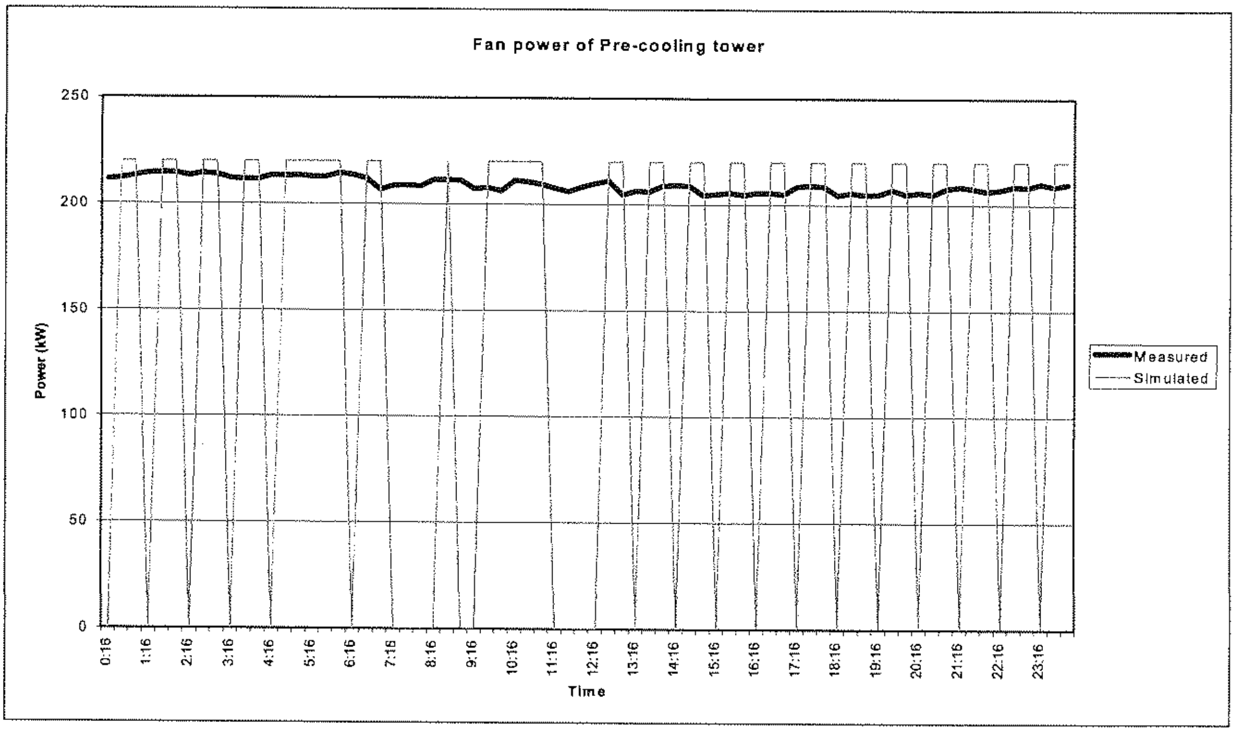

Figure C.10: Pre-cooling tower fan power verification result

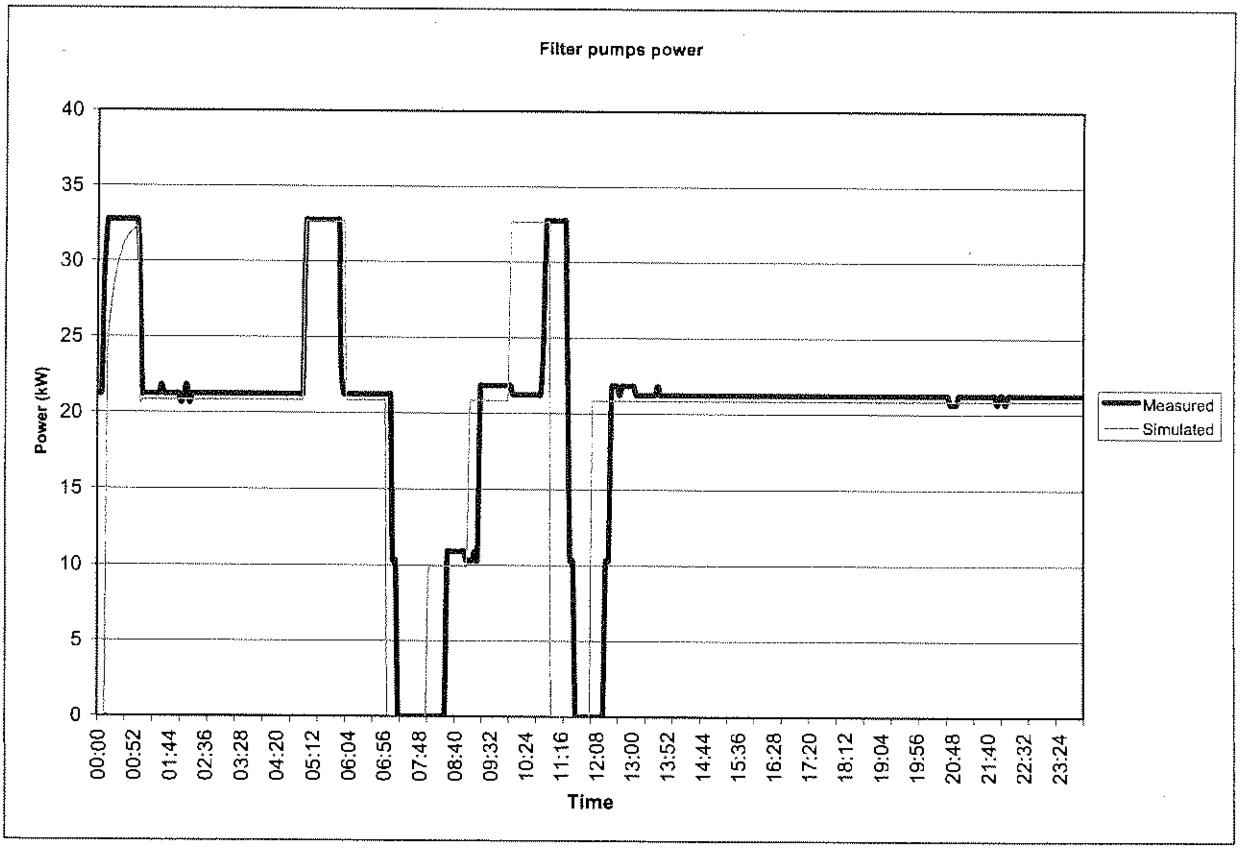

Figure C.11: Filter pumps power verification results

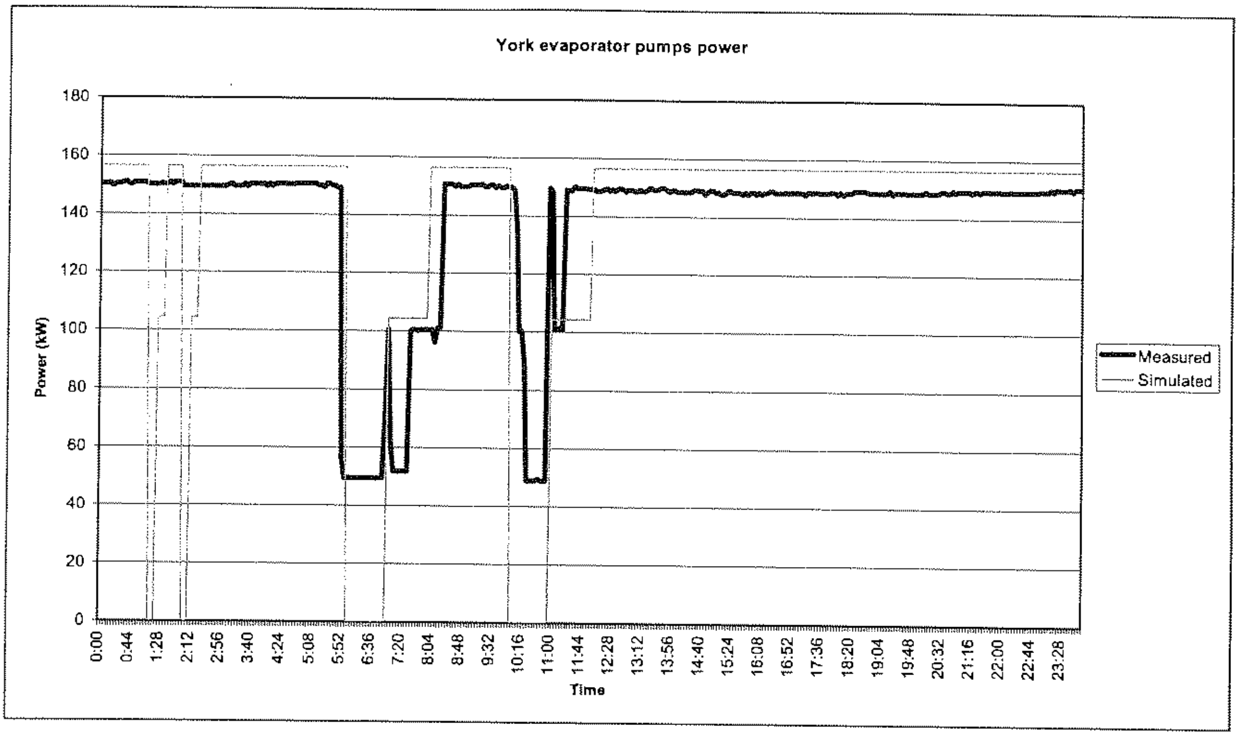

Figure C.12: York evaporator pumps power verification results

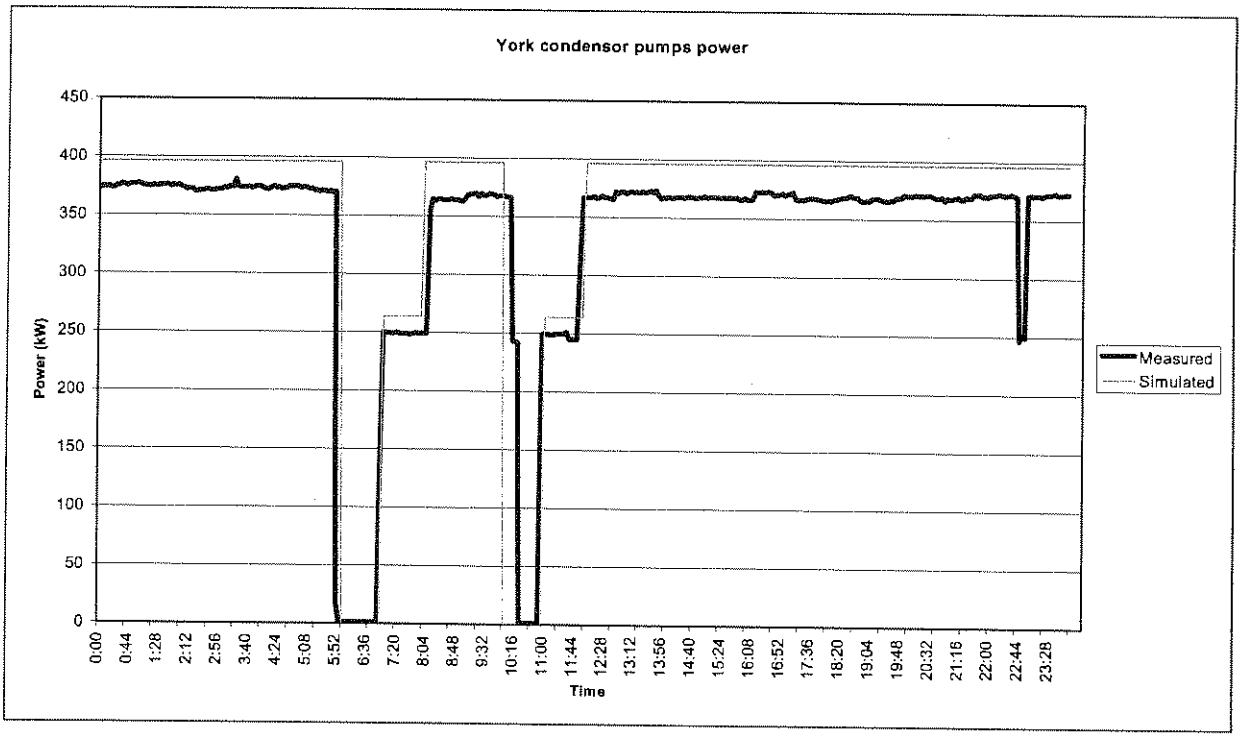

Figure C.13: York condenser pumps power verification results

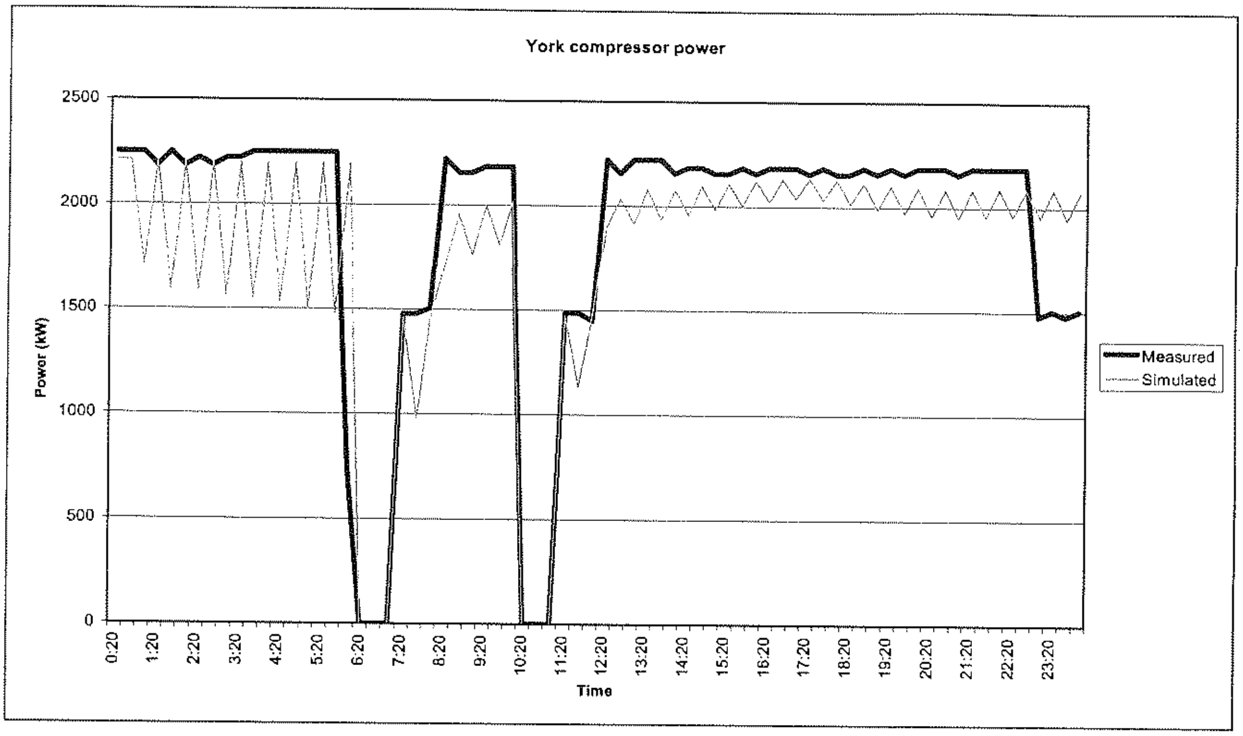

Figure C.14: York chiller compressor power verification results

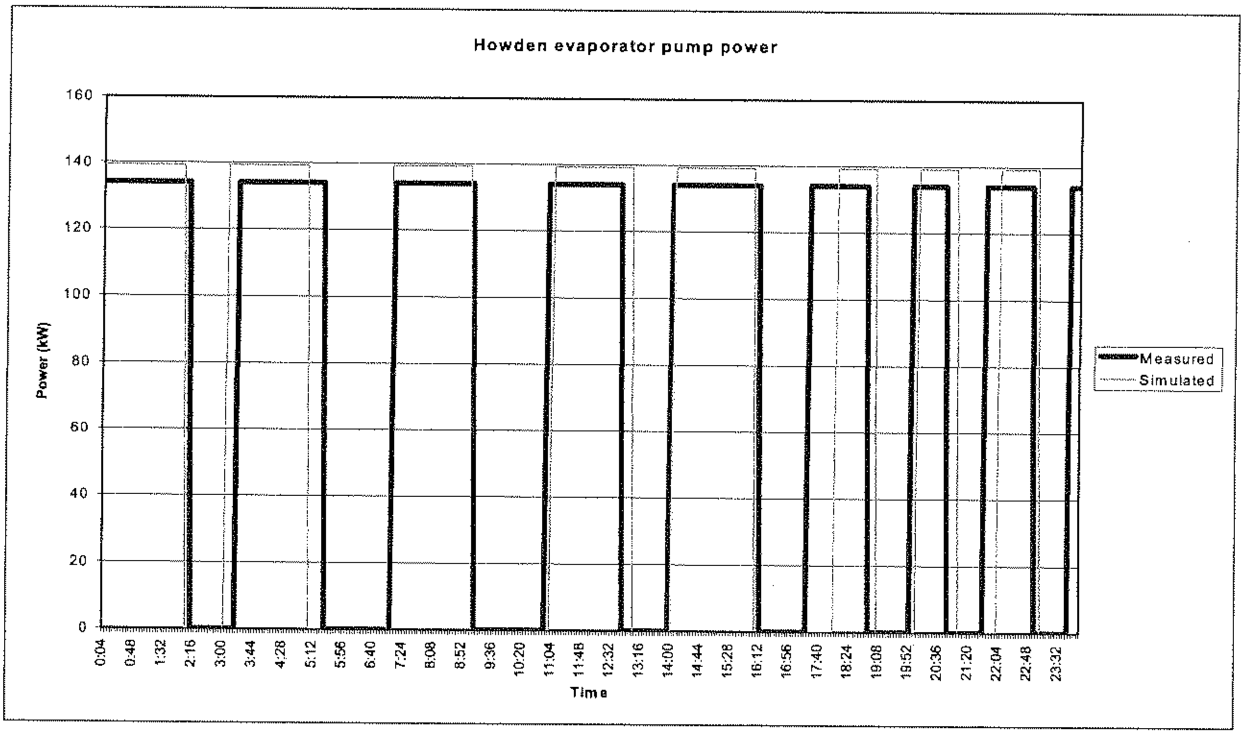

Figure C.15: Howden evaporator pump power verification results

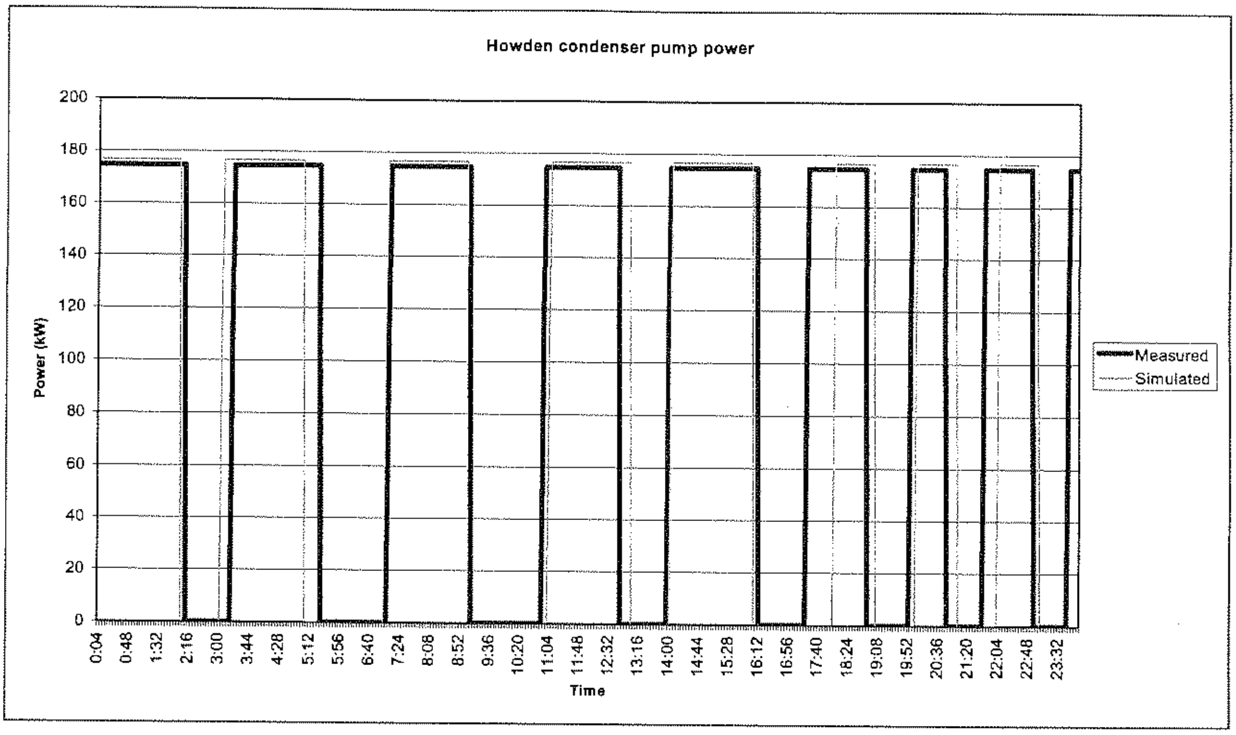

Figure C.16: Howden condenser pump power verification results

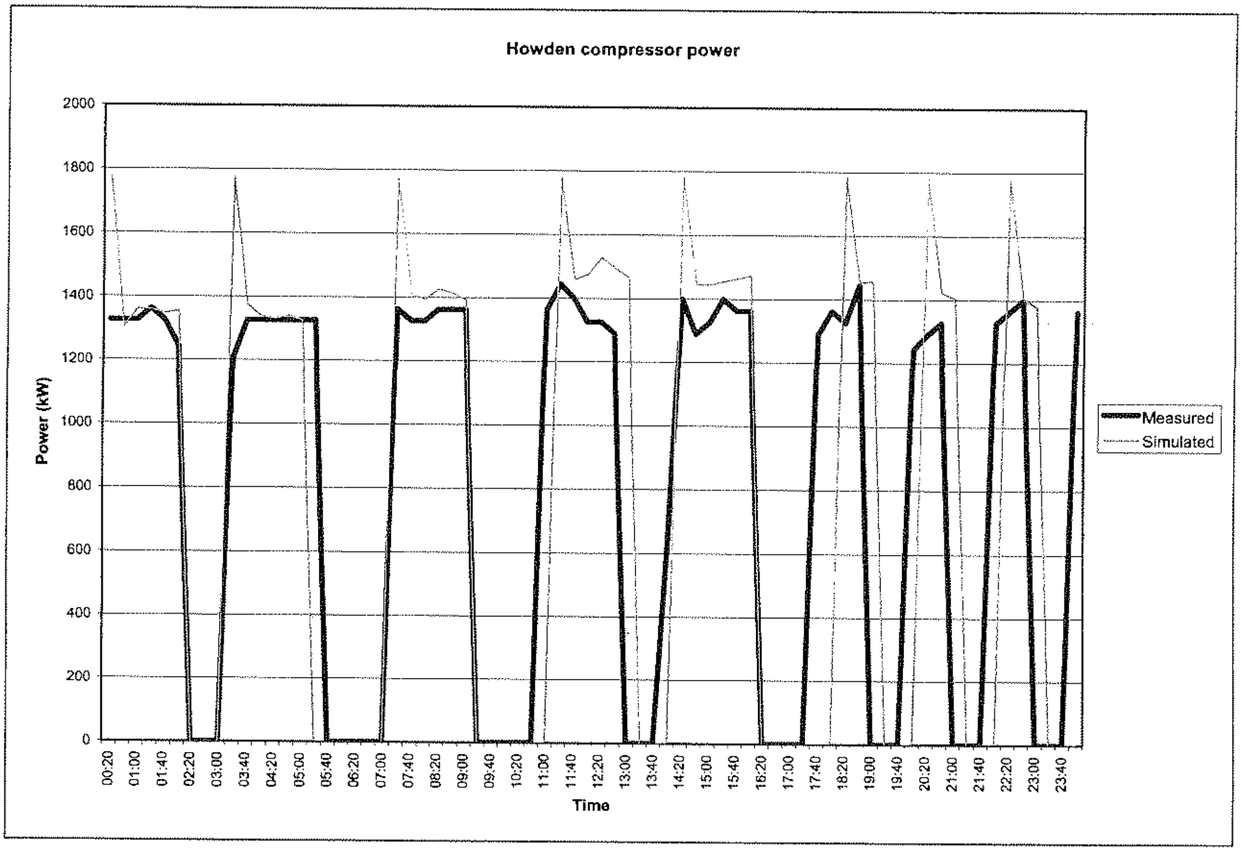

Figure C.17: Howden compressor power verification results

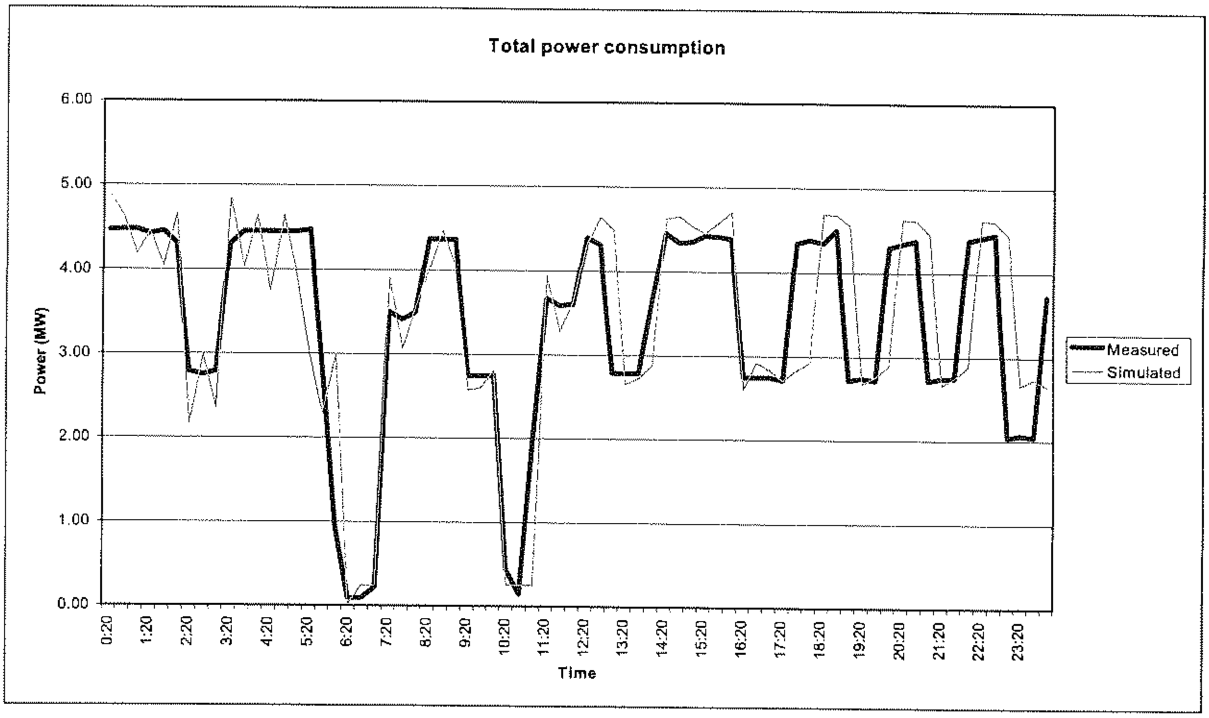

Figure C.18: Total power consumption verification results

C.4. LOAD SHIFTING POTENTIAL FOR RTP

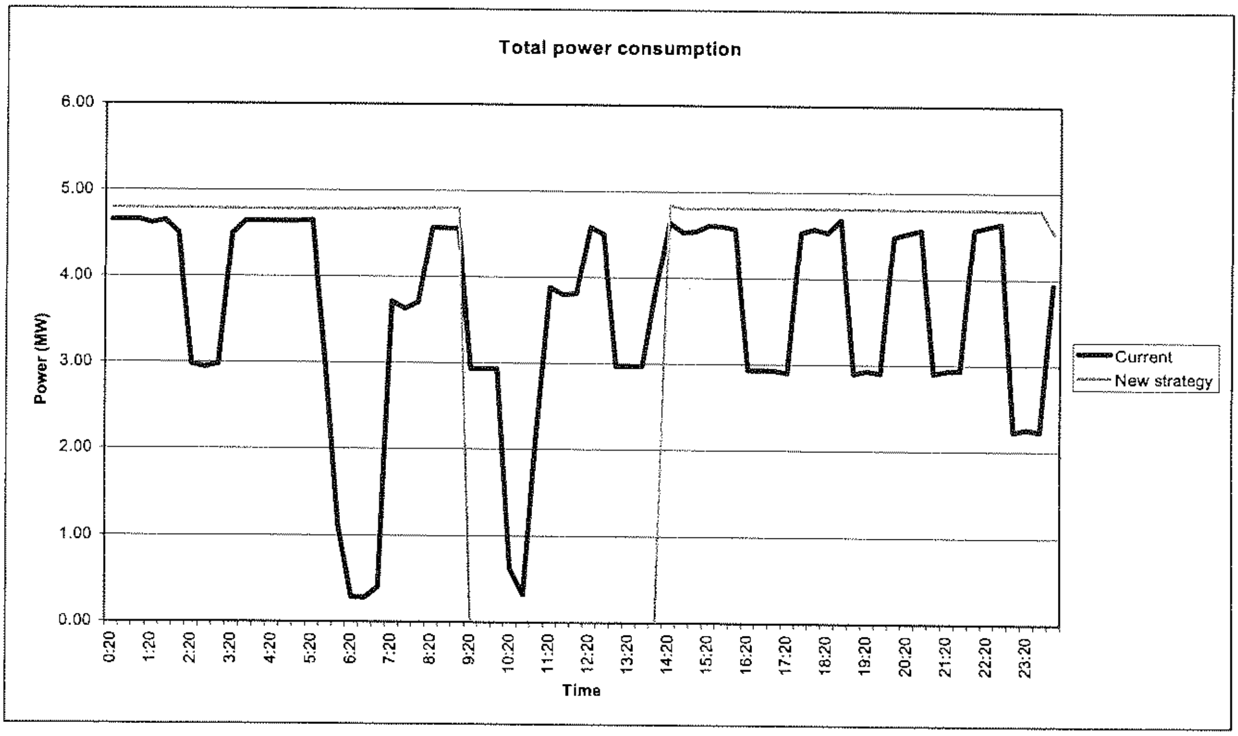

Figure C.19: The potential load shift of surface plant

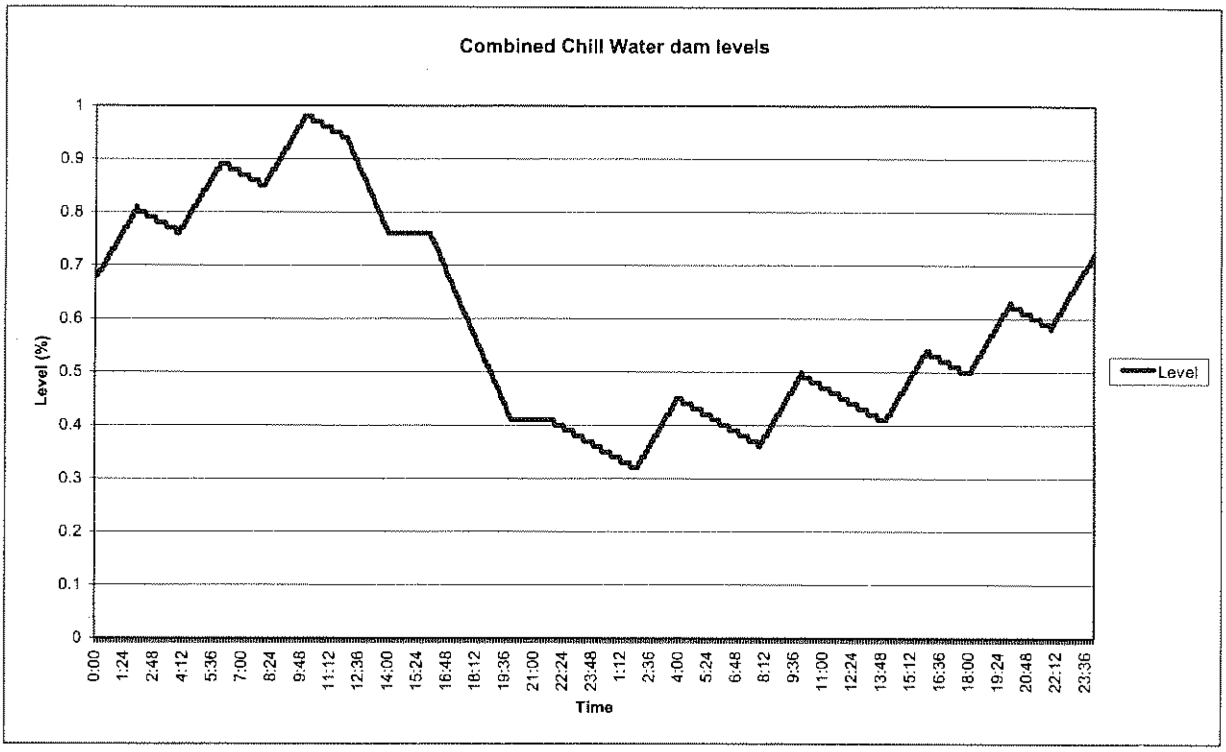

Figure C.20: The combined dam levels for the Chilled Water dams

APPENDIX D

POTENTIAL FOR DSM ON MINE PUMPING SYSTEMS

This appendix presents all the equipment specifications and base year verification results of chapter 9.

D.1. EQUIPMENT SPECIFICATIONS

The schematic layout of the Underground Pumping Stations can be seen in Figure D-1. This shows the pumps' mass flow capacity and the dams' storage capacity.

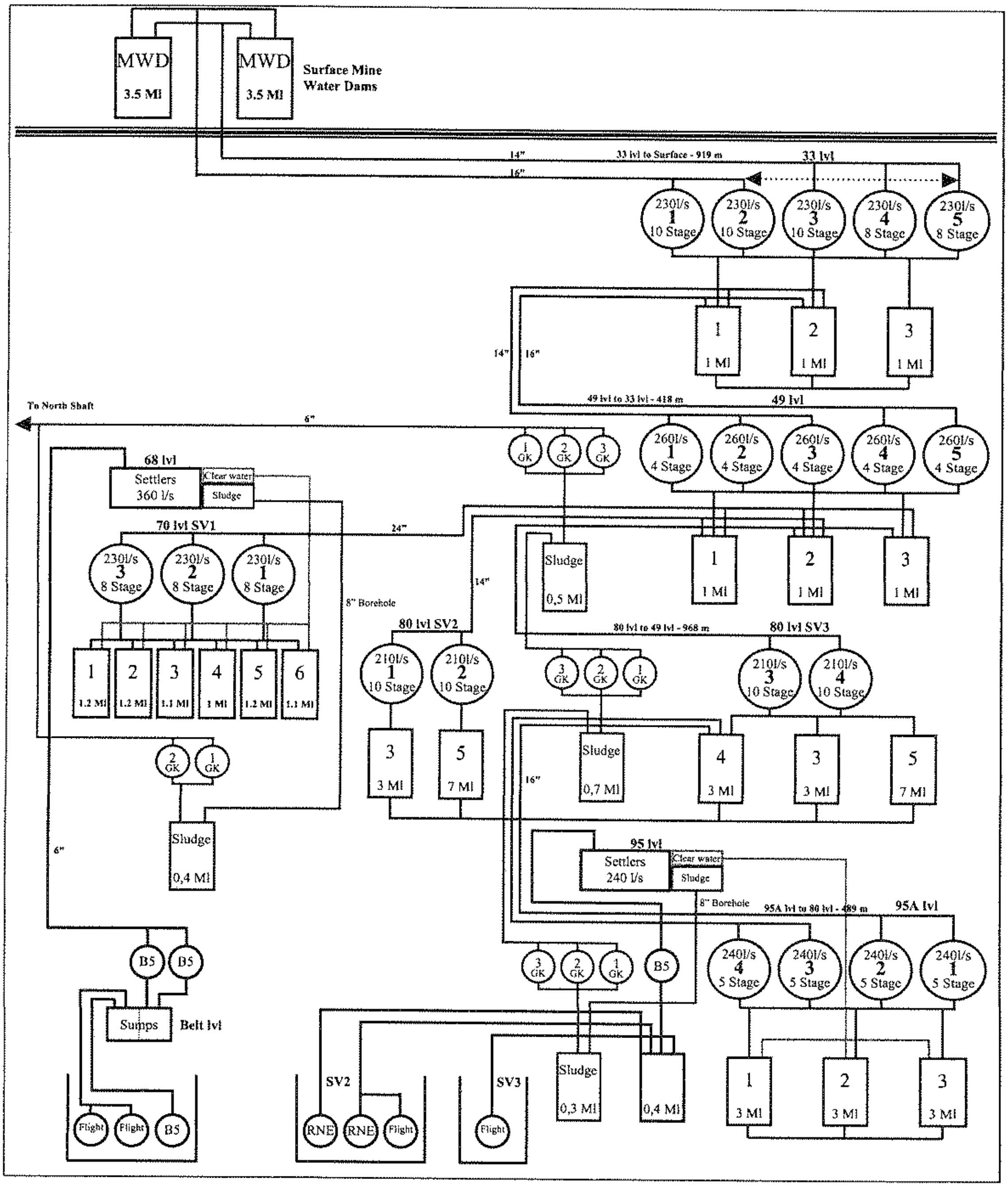

Figure D.1: Schematic layout of Underground Pumping Stations

D.1.1 Dams

Level	Depth [m]	Number of Dams	Total Capacity [m^3]
Surface	0	2	7000
33	919	3	3000
49	1337	3	3000
70	2000	6	6800
80	2305	5	23000
95A	2794	3	9000

Table D.1: Specifications of Underground Dams

D.1.2 Pumps

Level	Pumps details	Quantity	Flow [l/s]	Power [kW]	Controlled by:
33	GRIFO 10-Stage, GHP 53-29	5	230	3000	Dam level
49	GRIFO 4-Stage, GHP 58-29	5	260	1160	Dam level
70	GRIFO 8-Stage, GHP 53-29	3	230	2200	Dam level
80	GRIFO 10-Stage, GHP 53-29	4	210	3000	Dam level
95A	GRIFO 5-Stage, GHP 58-29	4	240	1600	Dam level

Table D.2: Specifications of Underground Pumps

D.2. BASE YEAR VERIFICATION

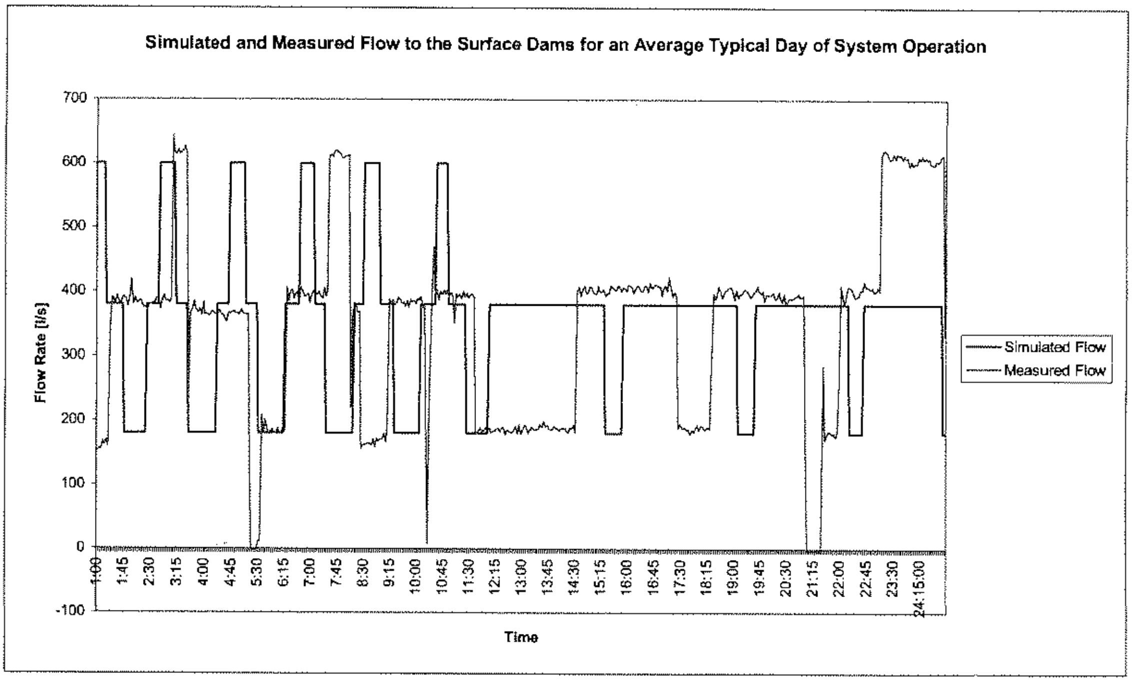

Figure D.2: Graph showing the base year verification results of the predicted flow rate from the underground workings to the surface mine water dams for an average typical day of system operation.

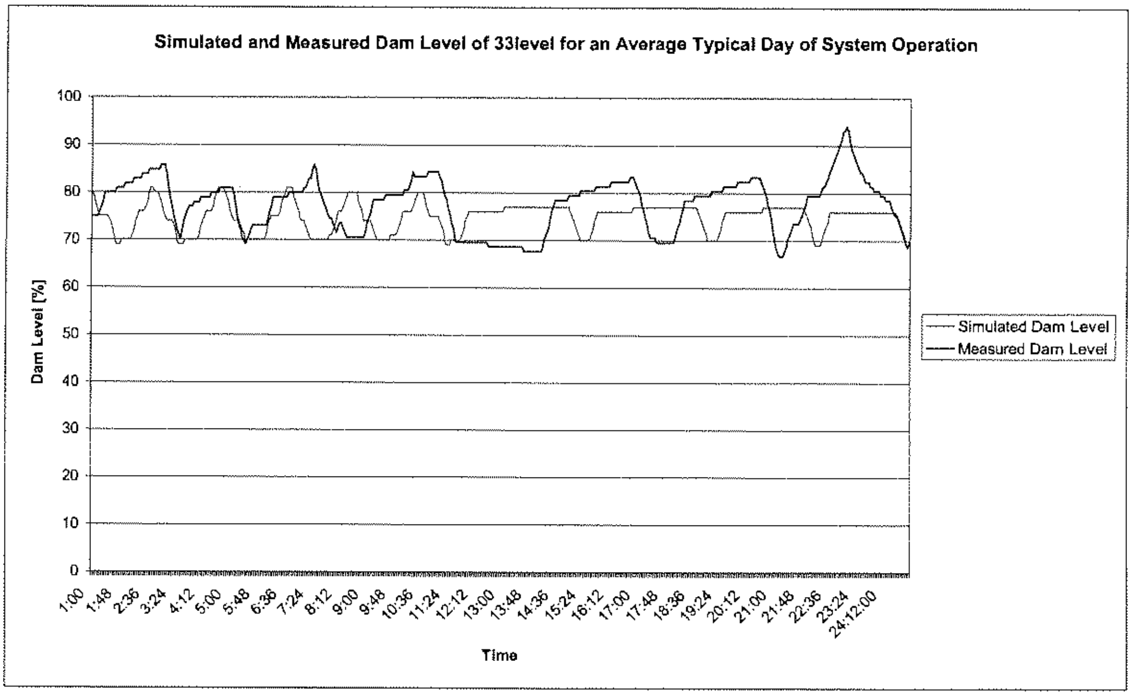

Figure D.3: Graph showing the base year verification of the predicted dam level of 33level against the actual measured dam level for an average typical day of system operation.

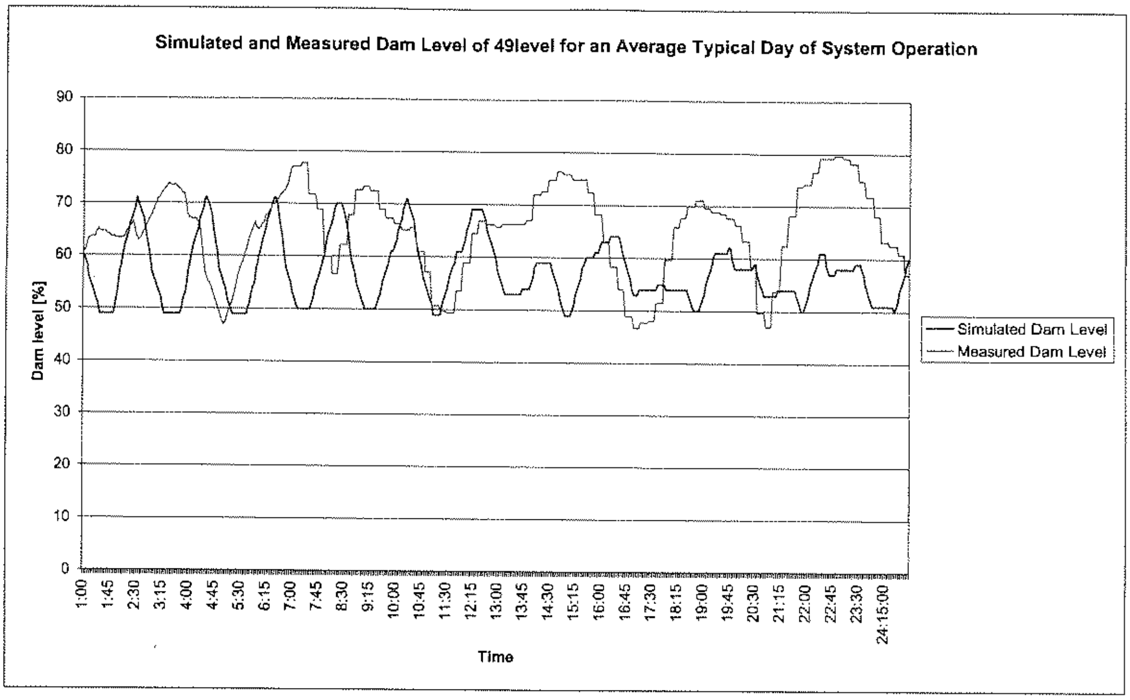

Figure D.4: Graph showing the base year verification of the predicted dam water level of 49level against the actual measured dam level for an average typical day of system operation.

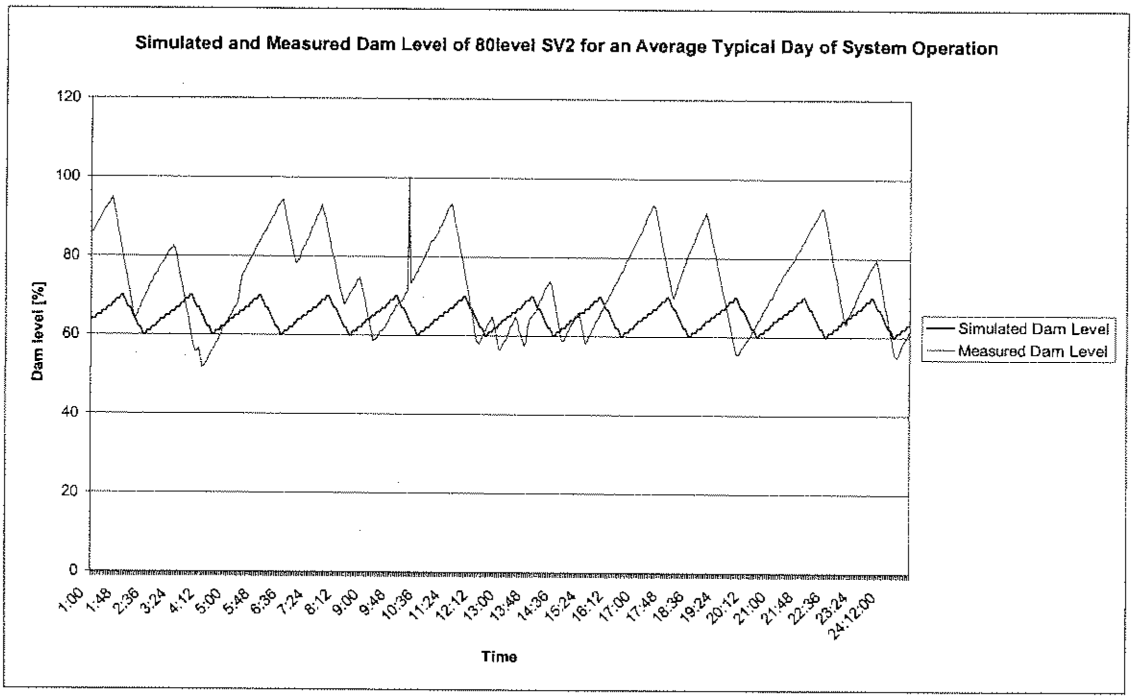

Figure D.5: Graph showing the base year verification of the predicted dam water level of 80level SV2 against the actual measured dam level for an average typical day of system operation.

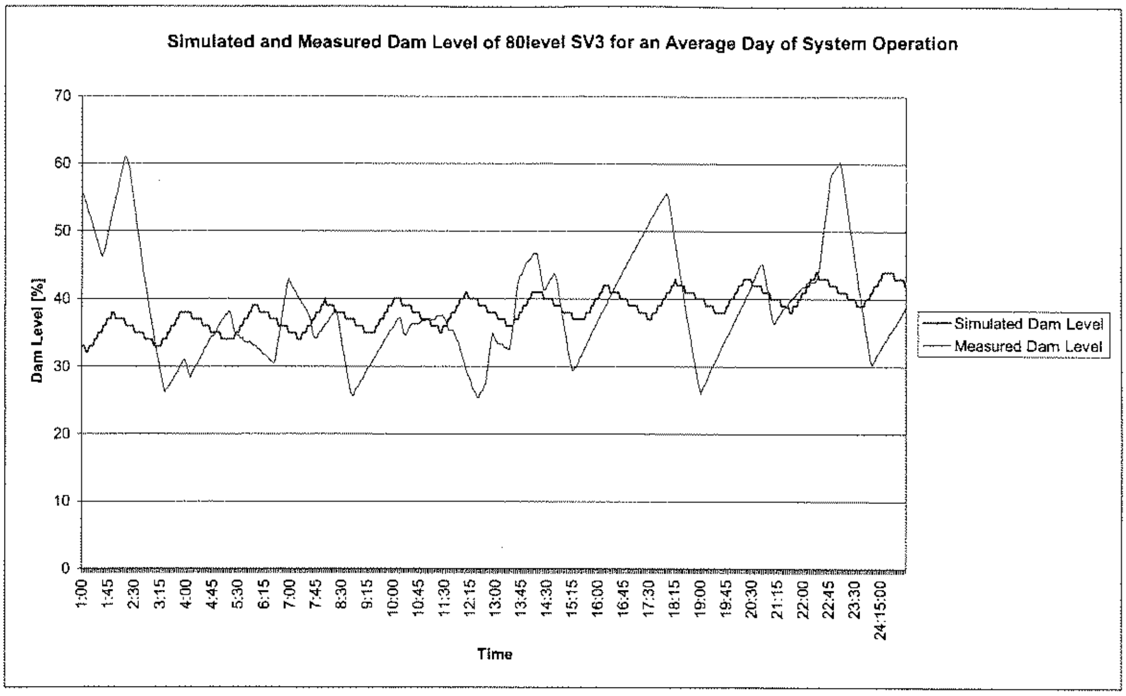

Figure D.6: Graph showing the base year verification of the dam water level of 80level SV3 against the actual measured dam level for an average typical day of system operation.

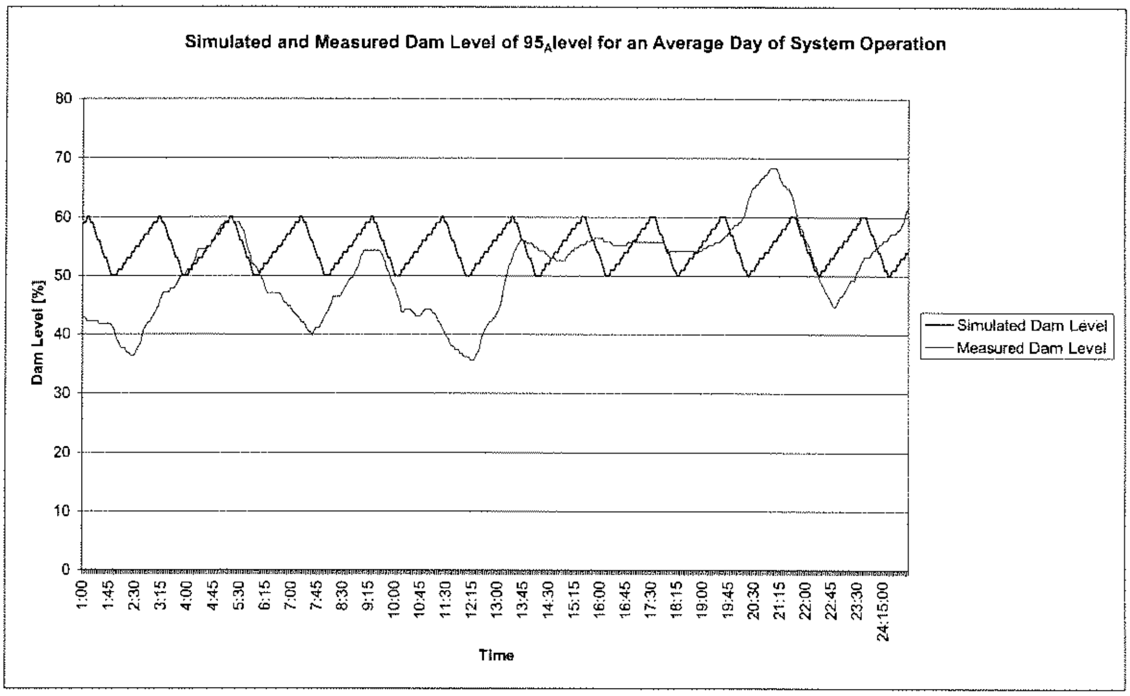

Figure D.7: Graph showing the base year verification of the predicted dam water level of 95_Alevel against the actual measured dam level for an average typical day of system operation.

APPENDIX E

ESCO PROTOCOL EVALUATION

In this appendix two ESCO protocols were evaluated. The one protocol is based on the building and HVAC system energy simulation software VisualDOE and the other one on QUICKcontrol.

ESCO PROTOCOL EVALUATION

Parties involved:

ESKOM

TEMM International

Persons involved:

S F van Geems

DC Arndt

E.1. INTRODUCTION

In this short report two ESCO (Energy Service Company) protocols were evaluated. The one protocol is based on the building and HVAC system energy simulation software VisualDOE, and the other one on QUICK*control*.

The evaluation was conducted during a period of one week and was based on the requirements of the two evaluators. In this investigation a point out of five was given by the two evaluators for each step of a typical ESCO study. Five means that the tool satisfies the required specification of the evaluators completely. The evaluators' requirements were based on the following criteria from most important to least important:

- Time to perform each step
- Level of knowledge and needed information to perform each step
- Shortcomings to perform each step
- Accuracy of assumptions made during each step

E.2. SUMMARY OF RESULTS

Total points from evaluation questionnaire out of 125:

VisualDOE: 83/125 (66%)

QUICK*control*: 107/125 (86%)

VisualDOE shortcomings:

- It is time consuming to construct the building model.
- It is only possible to configure pre-constructed typical HVAC system types.

- It is only possible to implement the control options available in the program.
- It is time consuming to create a new weather file via DOE in the USA.
- It is almost impossible to verify the simulation model without the correct climate data.
- The program does not include a report writer to complete the final report in a short period of time.
- Software support is only available in the USA or via email.
- No new software extensions can be made or initiated by the user.
- Error messages can appear regularly due to the quantity of information processed.
- The model size is limited in terms of the zones per block that can be created.
- If the programme is not used on a regular basis it requires some time to orientate oneself within the package.
- Typical equipment used in South Africa must be created.

QUICK*control* shortcomings:

- The system must be configured on a component basis. There are no typical system types to select from.
- Detailed control parameters are needed for input. There are no typical control strategies to select from, only typical controllers (PID and Step controllers).
- Typical system components (fuel boiler, air/air heat exchangers) still need to be implemented into the software.
- The simulation currently runs in excess of 20 minutes for a year simulation. The software is currently being revised which will also reduce the time required for simulations.
- The programme relies on the inputs given by the operator in terms of correctness.

E.3. EVALUATION QUESTIONNAIRE

1. General	**Point**
1.1 Simulation engine	
VisualDOE: Use the transfer function method to calculate hourly heating and cooling loads. From there it is a hand–down method from component to component to calculate the energy consumption of the system. The influence of the system control parameters can therefore not be seen in the indoor conditions. The dynamic response of the system cannot be simulated and real life control simulation is not possible.	2
QUICK*control:* Is a fully integrated building, system and control simulation program. The response of the system and building can be simulated on a minute basis to predict the effect control has on the indoor comfort. Any change in a component effect therefore the entire integrated system and building network. This makes real life control simulation possible.	5
1.2 Simulation time	
VisualDOE: A 19-zone year simulation takes about 2 minutes on a Pentium II 233 MHz PC.	5
QUICK*control:* A 10-zone year simulation takes about 9 minutes on a Pentium II 233 MHz PC.	2
1.3 Support	
VisualDOE: Support is only available from the USA via email and one local SA user.	2
QUICK*control*: Support is locally available via telephone, email and personal contact.	5
1.4 Future extensions	
VisualDOE: The program cannot be extended to satisfy new requirements of the user.	1
QUICK*control:* The program can be extended to satisfy any new requirements of the user. The program can also be extended to make provision for new areas, like mine ventilation and cooling energy simulation.	4

2. BUILDING INPUT DATA	**Point**
2.1 Obtain Structure information	
VisualDOE: Detail building structure data is needed for input, which involves detail building drawings. Typical structure data for South Africa is not available in the library and material properties and structure thicknesses must be obtained.	2
QUICK*control:* All needed input data can be gathered on site. No drawings are needed. A complete structure database is available for typical South African constructions. Sheets are available to gather all the relevant information.	5
2.2 Obtain building dimensions	
VisualDOE: The dimensions of each surface (including glazing) must be obtained from building drawings or measured in the building. The model will therefore reflect accurate building detail.	2
QUICK*control:* Only the internal floor area, north wall length, wall height and % glazing of each zone is needed for input. All this information can be gathered by walking through the building. Sheets are available to gather all the relevant information.	5
2.3 Obtain or measure zone air flows	
VisualDOE: Zone air is specified in terms of the supply air options: Let programme size Total flow Flow/area Air changes/hour This is again linked with options, which specify the outside air. Thermostat types together with a throttling range are specified for control purposes. Typical min & max zone temps are displayed with max supply air and outside air quantities.	4
QUICK*control:* Supply and fresh airflow rates must be obtained or measured for each zone. Sheets are available to gather all the relevant information.	3
2.4 Obtain ventilation schedules	
VisualDOE: Yearly ventilation schedules must be obtained for each	3

zone.	
QUICK*control:* Yearly ventilation schedules must be obtained for each zone. Sheets are available to gather all the relevant information.	4
2.5 Obtain lighting and other internal loads	
VisualDOE: Lighting and internal heat generation fixtures and schedules must be obtained through a walk through audit. Custom data sheet used.	4
QUICK*control:* Lighting and internal heat generation fixtures and schedules must be obtained through a walk through audit. Sheets are available to gather all the relevant information.	4
2.6 Obtain occupancy	
VisualDOE: Occupancy numbers and schedules must be obtained through a walk through audit. Custom data sheet used.	3
QUICK*control:* Occupancy numbers and schedules must be obtained through a walk through audit. Sheets are available to gather all the relevant information.	3
3. HVAC SYSTEM INPUT DATA	**Point**
3.1 Air distribution system	
VisualDOE: Component performance data, flow rates, pressures and the configuration of system must be obtained. Sheets are available to gather all the relevant information.	3
QUICK*control:* Component performance data, flow rates, pressures, the configuration of system and the control parameters must be obtained. Sheets are available to gather all the relevant information.	4
3.2 Water distribution system	
VisualDOE: Component performance data, flow rates, pressures and the configuration of system must be obtained. Sheets are available to gather all the relevant information.	3
QUICK*control:* Component performance data, flow rates, pressures, the configuration of system and the control parameters must be obtained. Sheets are available to gather all the relevant information.	4
4. VERIFICATION MEASUREMENTS	**Point**

VisualDOE: A number of conditions and the consumption of component power must be monitored to identify problem areas and to calibrate the simulation model.	3
QUICK*control:* A number of conditions and the consumption of component power must be monitored to identify problem areas and to calibrate the simulation model. Sheets are available to ensure the correct measurements are taken and for automatic data processing.	5
5. VERIFICATION SIMULATIONS	**Point**
5.1 Structure input	
VisualDOE: Each surface's dimensions and construction with its properties must be read into the program in detail. No South Africa pre-constructed library is available.	3
QUICK*control:* Only the internal floor area, north wall length and the wall height need to be read into the program. South African pre-constructed structure database is available for selection. This input can be completed on site while doing the walk through audit.	5
5.2 Occupancy input	
VisualDOE: Occupancy numbers and scheduling must be read into the program for each zone type.	3
QUICK*control:* Occupancy numbers and scheduling must be read into the program for each day type. This input can be completed on site while doing the walk through audit.	3
5.3 Internal load input	
VisualDOE: The equipment power density (w/m^2) of fixtures and the lighting power density of lights (w/m^2) and other equipment must be read into the program for each zone type.	4
QUICK*control:* The wattage fixtures and number of lights and other equipment must be read into the program for each day type. This input can be completed on site while doing the walk through audit.	4
5.4 Weather input	
VisualDOE: Weather data can be collected form the weather station or measured on site for the verification day. All new weather data including typical year data must first be sent to DOE to be transformed	2

into BIN data. This was done by Eskom.	
QUICK*control*: Weather data can be collected form the weather station or measured on site for the verification day. All new weather data including typical year data can directly be entered into the program's climate database. Typical year data is already available for all the climate regions of South Africa.	5
5.5 System input	
VisualDOE: Only typical HVAC system configurations and control options are available for selection. The user must have the necessary knowledge of system types to select the closest one to the real one. The components of the systems need not to be configured or linked.	3
QUICK*control:* Any existing system type or new type can be constructed. The system can be built exactly the same in the model as in real life. Any control logic can be modelled. Each component of the system, including the control, must be configured and linked correctly.	5
5.6 Calibrate simulation model and compare to measurements	
VisualDOE: Only average hourly verification and calibration on the main component types is possible. It is difficult to obtain the correct weather data for the verification period. Without the correct weather data it is impossible to verify.	3
QUICK*control:* Real life minute verification and calibration on component basis is possible. This implies that each component can be calibrated to ensure that the existing system and building is modelled correctly. The correct dynamics of the system can therefore be incorporated into the simulations. A logic and simple method is available to perform the calibration.	4

6. RETROFIT SIMULATIONS	**Point**
6.1 Base year simulation	
VisualDOE: Hourly year simulations can be executed which include weekday, Saturday, Sunday and public holiday schedules. Peak demand and energy consumption are therefore based on average hourly values.	3
QUICK*control:* Minute year simulations can be executed which include weekday, Saturday and Sunday schedules. Peak demand and energy consumption are therefore based on integrated hourly values as used in South Africa.	4
6.2 Retrofit simulations: Structure changes	
VisualDOE: Structural changes (insulation, shading devices) can be implemented into the simulation model with ease.	4
QUICK*control:* Structural changes (insulation, shading devices) can be implemented into the simulation model with ease.	4
6.3 Retrofit simulations: Equipment scheduling	
VisualDOE: The dynamic effect of equipment scheduling can only be investigated on an hourly basis. It is therefore difficult to ensure indoor comfort for optimal scheduling.	3
QUICK*control:* The dynamic effect of equipment scheduling can be investigated on a minute basis. It is therefore possible to ensure indoor comfort for optimal scheduling.	5
6.4 Retrofit simulations: New control strategies	
VisualDOE: The user has the option to select from a menu typical energy efficient control strategies. However the user is not always sure of the strategy implemented by the software. The user cannot implement new customised inventions. The effect of new control setpoints is displayed in terms of min & max temps.	3
QUICK*control:* Energy efficient control must be implemented with the typical controllers available in the software. This implies that any new control inventions can be implemented into the software and the user will be sure of the logic involved. The effect new control has on indoor comfort can be predicted by the software.	4

6.5 Retrofit simulations: New tariff structure	
VisualDOE: The effect of new tariff structures can be investigated.	4
QUICK*control:* The effect of new tariff structures can be investigated. Relative easy to create new structures. Certain assumptions need to be made in terms of kVAr charges.	2
6.6 Retrofit simulations: Plant upgrades	
VisualDOE: Plant changes can be implemented by initiating standard futures. i.e. variable speed, inlet vanes, discharge dampers.	5
QUICK*control:* Plant changes must physically be constructed which might be time consuming.	2
6.7 Retrofit simulations: Combination of options	
VisualDOE: This can easily be achieved after the verification stage of the process.	4
QUICK*control:* It requires the construction of a new layout. From a time point of view it will take longer to construct. Standard models can however be generated on request.	3
7 REPORT WRITING	**Point**
VisualDOE: The package does not include any type of report writer. The user must therefore build his own report templates in MS Word. This can take up to a week to complete the report.	2
QUICK*control:* This protocol provides a report writer, which is based on a master and standard template database structure. The master database can be updated after each project and pulled into the template database of each new project. Simulation results can be linked to the database in graph or table format for import. The report writer automatically generates the report in MS Word. It will only take 2 days to finalise the report.	4

NOTES TO THE READER:

The writers' opinions within this ESCO protocol evaluation are in no way a final judgement on any of the two packages. Both packages offer unique features, which makes the application thereof acceptable.

The opinions offered in this document do not necessarily reflect the opinion of Eskom, and therefore Eskom does not accept responsibility for the comments by the authors.