Design and simulation of heat pumps and A/C equipment using pure and mixed refrigerants with MoMo (Modular Modelling)

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SUMMARY

An approach to the steady-state simulation of heat pumps and refrigerating equipment is presented. Novel features of the proposed model are the strategy and formalism adopted in the programme development. Individual mathematical models for component design are linked to enable the simulation of a basic vapour compression cycle. This paper introduces some of the component modules, which can be used for pure as well as mixed refrigerants. Because numerous models can be used for a specific piece of equipment making up the whole refrigerating/heating unit, the Modular Modelling approach leads to a flexible and evolutive structure. A machine can actually be built-up from component libraries that include several types of heat exchangers, compressors and expansion devices and the effects of operating conditions or component geometrical parameters on cycle performance can be examined. Because actual test data are used to "calibrate" the component modules, very accurate results are obtained. The proposed model is validated with experimental results from an air-to-air split unit using R22 and a water chiller using R22 and R407C. Discrepancies smaller than 5% are obtained on the cooling duty and COP of the split unit and the chiller components.

INTRODUCTION

The advent of cheap computing power has lead to the development of numerous models for the simulation of A/C and refrigeration systems over the last two decades. These models are used to predict system performance depending on operating parameters and cycle components. For equipment designers and manufacturers, this means drastically reduced time-consuming field experiments and development costs.

Currently available models differ in degree of complexity, of which the number of cycle operating parameters to specify (i.e. degrees of subcooling, superheating, refrigerant charge in the system, etc.) is a good indication. Less sophisticated models (very often solely based on thermodynamics) are not equipment size-sensitive, hence requiring the least number of equipment geometrical parameters. On the other hand they require more cycle working parameters as input data. Typically, condensing and evaporating pressures (or temperatures) must be supplied, along with an indicator of compressor efficiency. Values of superheat at compressor suction and subcooling at condenser exit must also be fixed. Refrigerant flow rate is assumed to be independent of the expansion device and the compressor size. Since these models do not include any rate relationship, change of performance with varying heat exchanger size cannot be computed either. Consequently, these models can only be used to draw comparisons between different overall design alternatives, such as working pressure levels or screening of different refrigerants. Other trends can be formulated from these models if overall heat transfer is given consideration, for example in terms of "UA" values. However, these models are still very limited; a value of cooling/heating duty must be supplied - which is precisely what the designer is looking for when knowing which pieces of equipment are used at some specified external conditions.

Degrees of complexity can be added to simulation models by introducing heat transfer and momentum rate equations, mass balances and refrigerant flow rate dependency relationships (for expansion devices and compressors). Although it requires more data input, this approach is preferable because, as mentioned above, less cycle parameters are required, with performance becoming equipment-dependant. For a thorough literature review on the subject, see Conde (1992), who conducted a detailed review of different types of heat pump simulation models and presented his own model (ENHANCED). This program uses detailed models for all cycle components and performs a refrigerant charge inventory. However, it typically required two to fifteen hours to simulate an air-to-water heat pump on a 386-based computer.

Damasceno et al.(1992) also conducted an in-depth comparison of three air-to-air heat pump models. These were:

- MARK III , which was developed at the ORNL by Fisher and Rice (1983);
- HPSIM, developed at NBS by Domanski and Didion (1983) and which was later updated to become HPSIM 3;
- HN, developed at Purdue and further updated by Damasceno et. al, as cited in Damasceno et al.(1992).

All three programs are very elaborate ones, with detailed or semi-detailed models for all cycle components. It is noteworthy that only the HPSIM model accounts for the effect of refrigerant charge in the system. Damasceno et al.(1992) analysed the ability of the three models to predict the performance of a sample heat pump, using experimental results. Their major conclusion was that "*computer simulations by themselves at best predict trends and approximate levels*". A more useful result of their comparison was to reveal the need for extensive test data to validate and/or "calibrate" computer models.

The model introduced herein is similar in many aspects to the above-mentioned models. Common features include general convergence procedure, which consists in several nested iteration loops, with each loop calling a routine that represent a physical model (also called 'module') of a cycle component. Novel features of the proposed model are the strategy and formalism adopted in the programme development. These lead to a flexible and evolutive structure because, for a specific piece of equipment making up the whole refrigerating/heating unit, numerous models can be used. Thus a user can actually 'build-up' a machine from component libraries that include several types of heat exchangers, compressors and expansion devices. The effects of operating conditions or component geometrical parameters on cycle performance can be examined. Additional features include the possibility of simulating the performance of separated components, such as heat exchangers or compressors, and the identification of correlation constants from a few experiments performed on these components. This identification provides a feedback for component library build-up and therefore leads to better overall cycle computations. This flexibility is due to the programming strategy, and advantage is taken of the working environment, i.e. PC WINDOWS, and its Dynamic Link Library procedures.

PROGRAMME GENERAL STRUCTURE

Thermodynamic Cycle and input parameters

The developed model simulates the steady-state performance of A/C and refrigerating systems based on a single-stage vapour compression cycle. Figure 1 shows a representation of such a cycle on a P-H diagram.

Refrigerant exiting the evaporator at point 1 enters the compressor suction and is discharged at point 2. After desuperheating and condensation (points 2 to 3), it is throttled through an expansion device and exits at point 4. The two-phase refrigerant is then evaporated to complete the cycle at point 1.

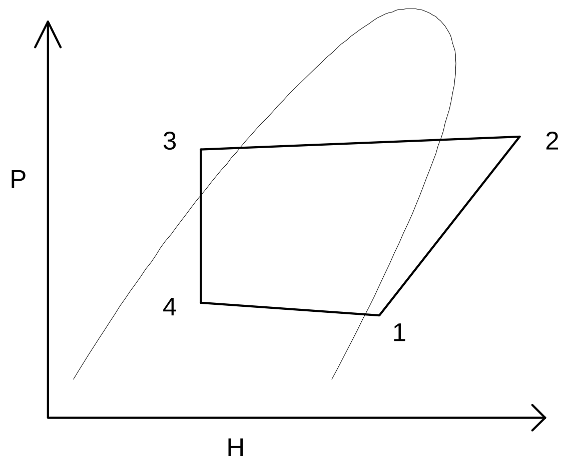

Figure 1: Vapour compression cycle P-H diagram

As shown by the non-horizontal lines joining points 2 and 3 and points 4 and 1, the pressure drops occurring during the heat transfer processes are accounted for. The effect of refrigerant lines is neglected, so pressure/temperature changes between two consecutive components are not taken into account.

At its present state of development, the model is insensitive to refrigerant mass in the system. Refrigerant subcooling is user-defined. Superheat at evaporator exit must only be specified if an ideal expansion device is used. The input variables are of two types:

- external fluid operating parameters, e.g. for an air-to-air heat pump, inlet air temperatures, flow rates and humidity to both heat exchangers must be specified;
- geometrical dimensions and correlation parameters for cycle components: e.g. for a fin-tube HEX, specify number of rows, number of tubes per row, number of circuits, tube diameter, thickness, fin pitch, etc., and if they are not assumed as the default values, coefficients for air-side and refrigerant-side heat transfer correlations must be supplied (see later the discussion on HEXs).

Simulation Programme Flowchart

Figure 2 shows the simulation programme flowchart. When all input operating parameters for the external fluids and cycle components geometrical parameters have been specified, the simulation process starts with guessed values of refrigerant condensing and evaporating temperatures and superheat at exit of the evaporator. Starting pressure levels are first calculated.

The compressor module computes a refrigerant flow rate, which is subsequently used for the condenser module calculations. A value of refrigerant subcooling at condenser exit is determined. This value should coincide with that specified as an input parameter. At this stage, a first loop iterates on compressor discharge pressure to satisfy the subcooling criterion.

With refrigerant state at point 3 being known (temperature and pressure), the refrigerant vapour quality is computed on the basis of an isenthalpic expansion process. A second loop iterates on evaporator inlet pressure to satisfy equality between the values of refrigerant quality at exit of the expansion device and inlet to the evaporator. The evaporator inlet pressure can be back-calculated from the exit pressure (which is equal to the compressor suction pressure) and the computed pressure drop through the HEX. When the value of the evaporator inlet pressure converges, the evaporator module yields a value for the refrigerant flow rate. This newly-computed value is compared to that obtained from the first iteration loop using the compressor and condenser modules. A third iteration loop, embedding the two aforementioned loops, iterates on evaporator exit pressure to satisfy the equality criterion between the two values of refrigerant flow rate. An updated value of compressor suction pressure is thus sent back to the compressor module to close the loop. Upon convergence of this loop, the expansion device module is put to use to calculate a refrigerant flow rate based on the device characteristics and operating conditions. This flow rate is again used in the last, outermost iteration loop to check the assumption made on the value of superheat at compressor suction. The current value of superheat is then adjusted depending on the difference between this flow rate and the value given upon convergence of the inner loops. The simulation stops when both values of the refrigerant flow rate coincide within a specified tolerance.

Upon convergence, the program yields outlet conditions for the external fluids, condenser and evaporator duties, compressor power, refrigerant flow rate, superheat, evaporating and condensing pressures and temperatures, and all parameters relevant to the calculation scheme.

MODELLING OF CYCLE COMPONENTS

Compressor Modules

The compressor modules simulate hermetically sealed reciprocating, scroll and rotary compressors. The detailed models were described by Haberschill et. al (1994) and assume a polytropic compression, i.e.:

$$Pv^k = \text{constant} \qquad (1)$$

where P is pressure, v the molar volume and k the polytropic exponent.

The output variables (refrigerant discharge temperature T_d, mass flow rate M_r and electric power input W_c) are computed from the model input variables (suction temperature T_s and pressure P_s and discharge pressure P_d).

Reciprocating compressor

The maximum internal pressure is equal to the discharge pressure. The refrigerant specific volume at discharge is expressed as:

$$v_d = v_s\left(\frac{P_s}{P_d}\right)^{1/k} \qquad (2)$$

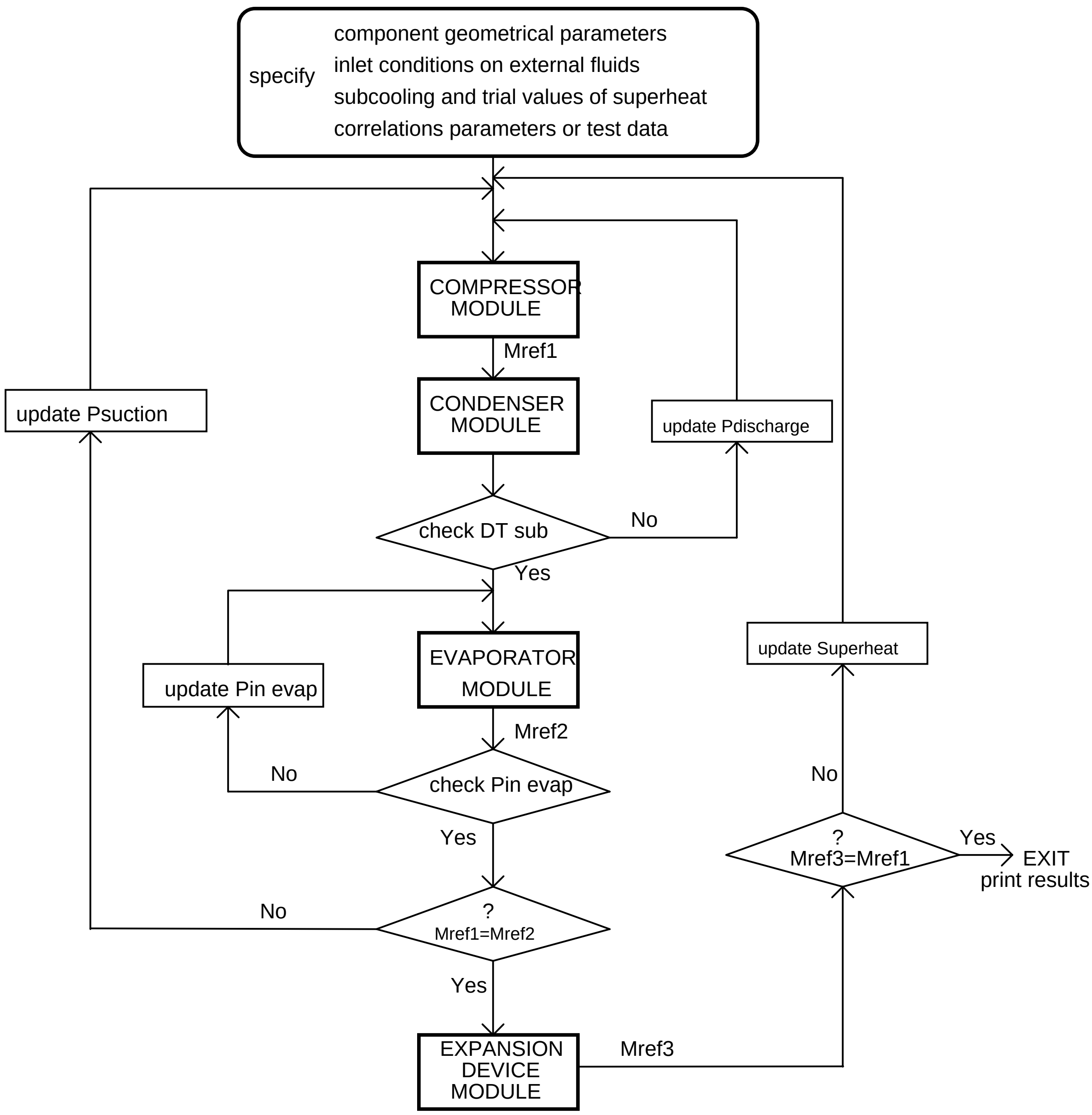

Figure 2 : Simulation programme flowchart

A discharge temperature T_d can then be calculated from an equation of state using the values of pressure and specific volume. Assuming a polytropic expansion coefficient, the mass flow rate is computed from:

$$M_r = \rho_s VN\left(1 - \tau\left(\delta^{1/k} - 1\right)\right) \qquad (3)$$

where ρ_s is the fluid density, V the compressor swept volume per revolution, N the compressor speed, τ the clearance volume factor, δ the pressure ratio and k the polytropic factor. Compressor electrical power input W_e is expressed as:

$$W_e = M_r\left(H_d - H_s\right) - Q_{lost} \qquad (4)$$

where H_d and H_s are refrigerant suction and discharge enthalpies, respectively, and Q_{lost} accounts for heat losses to ambient.

Scroll Compressor

For scroll compressors, the maximum internal pressure P_{mip} and the specific volume v_{mip} are calculated from the suction conditions and the volumetric ratio ε:

$$P_{\mathrm{mip}} = P_s \varepsilon^k \quad \text{and} \quad v_{mip} = \frac{v_s}{\varepsilon} \qquad (5), (6)$$

The discharge specific volume is computed assuming an adiabatic compression:

$$v_d = v_{mip}\left(\frac{P_{mip}}{P_d}\right)^{\gamma} \qquad (7)$$

where γ is the mean isentropic coefficient. The discharge temperature can be computed from an equation of state, based on the values of discharge pressure and specific volume. The mass flow rate is then determined by subtracting gas leakage m_{leak} from overall displaced volume as follows:

$$M_r = \rho_s VN - m_{leak} \qquad (8)$$

Compressor electrical power input is calculated using Eq. 4.

Rotary Compressor

Discharge specific volume is computed using Eq.2. Internal leakage is accounted for by using Eq. 8 and compressor power input is determined using Eq. 4.

Required parameters

The parameters used in Eqs 1 to 8 need to be supplied. Some of these are geometrical and are known (displacement volume and speed for all models and volumetric ratio for scroll type). Other parameters (leakage flow rate, effective clearance factor, polytropic coefficient, heat losses) are not readily available. They must therefore be determined from experimental data. The identification routine available in MoMo allows the calculation of these parameters from the following empirical laws:

for scroll compressors: $$k = a_k m + b_k \delta + c_k \qquad (9)$$

polytropic exponent

for reciprocating and rotary compressors $$k = \left(b_k P_s + c_k\right)\delta^{a_k} \qquad (10)$$

Leakage flow rate for scroll and rotary compressors is expressed as:

$$m_{leak} = a_l \rho_l VN + b_l \sqrt{\delta} + c_l \qquad (11)$$

Effective clearance factor in reciprocating compressors is determined from:

$$\tau = a_m \delta^{b_m} \qquad (12)$$

Regardless of compressor type, energy losses to the ambient are expressed as:

$$Q_{lost} = a_q m_r + b_q \delta + c_q \qquad (13)$$

Correlation constants a_k, b_k, c_k, a_l, b_l,c_l, a_m, b_m, a_q, b_q and c_q are determined from few experimental results using a minimisation procedure.

Validation of compressor modules for pure and mixed refrigerants

By performing tests on three reciprocating, a scroll and two rotary compressors, Haberschill et. al (1994) showed that the above correlations lead to very satisfactory results. For a wide range of operating conditions (suction pressures between 3 and 8 bars, and discharge pressures between 10 and 22 bars), they obtained average errors of 4% on the refrigerant mass flow rate, 5% on compressor electrical power input and 4K on discharge temperatures.

Two compressors (one scroll and one reciprocating) were tested at CETIAT using R407C and R22. By using results from several tests, correlation constants were identified. Simulation was then performed using the operating parameters. Resulting average errors were less than 1K on the discharge temperature, about 0.5% on the refrigerant flow rate and less than 1% on electrical power. The simulation results for the scroll compressor are included in the section devoted to the validation of a water chiller components (see later).

Heat Exchangers Modules

Presently available models include air-to-refrigerant finned coils evaporators and condensers and shell-and-tube water-to-refrigerant condensers. Additional models have been developed and are currently under validation (shell-and-tube evaporators and plate HEXs). Only models relevant to finned tubes will be briefly reviewed here.

The approach for heat transfer and pressure drop calculations is the same for both modules. It is based on an Effectiveness-NTU (number of transfer units) computation scheme and was implemented in a very popular in-house made software, CANUT (Armand and Molle, 1991).

Air side heat transfer

For both modules, the air heat transfer coefficient based on the overall external surface area is computed from the j Colburn factor, which is expressed as:

$$j = A \,\mathrm{Re}^n \qquad (14)$$

where A and n are parameters which depend on the fin surface characteristics (material, geometrical factors, etc) and Re the air Reynolds number based on the minimum flow area. A few experimental results on coils equipped with the same fin surface and with water as the internal fluid are required to determine coefficients A and n. Typically, errors of less than 5% on the HEX duty are obtained following this method. When moisture condensation occurs on the fins, experiments become even more necessary to yield accurate results. If no data is available, default values of A and n can be taken for the fin surface under consideration. In this case, the condensation of moisture on the fins is accounted for following the methods outlined in Threlkeld (1970).

Evaporator module

Refrigerant side input parameters are inlet vapour quality, pressure and degrees of superheat at outlet (or vapour fraction if two-phase outlet). On the air side, flow rate, temperature and humidity at inlet are required. The module yields refrigerant flow rate, inlet pressure, air outlet conditions and evaporator duty.

The coil is divided into two zones corresponding to refrigerant boiling and superheating. For single-phase flow heat transfer, use of the well-known Dittus-Boelter correlation is made. For boiling, the Bo Pierre relationship (as cited in Armand and Molle, 1991), is used. The Nusselt number is then expressed as:

$$Nu = B\left(\mathrm{Re}^2 . K_f\right)^C \qquad (15)$$

where, again, coefficients B and C can be identified through experiments and preceded by the determination of the air coefficients (A and n). The default values for B and C are 0.017 and 0.372 respectively.

For refrigerant mixtures, a boiling heat transfer coefficient is first calculated as described above by using the properties of the mixture. A corrected HTC is then calculated following the method outlined by Bivens and Yokozeki (1994):

$$\frac{1}{h_m} = \frac{1}{h_{pure}} + \frac{0.175\left(T_d - T_b\right)}{Q}\left\{1 - \exp\left(-\frac{Q}{1.3x10^{-4}\rho_l H_{vp}}\right)\right\} \tag{16}$$

where h_{pure} is calculated from Eq. 15 with the properties of the mixture, T_d and T_b are the dew and bubble temperatures, respectively. Q and H_{vap} are the heat flux and latent heat of vaporisation, respectively.

Condenser module

Refrigerant side input parameters are inlet pressure, temperature and flow rate. On the air side, flow rate and temperature at inlet are required. The module yields refrigerant and air outlet conditions and condenser duty.

In the condenser, three zones, corresponding to refrigerant desuperheating, condensation and subcooling, are considered. Single-phase flow is dealt with by the use of the Dittus-Boelter correlation. The condensation heat transfer coefficient is calculated based on the Shah's correlation (Shah, 1979), at an average vapour quality. Shah expresses the condensation Nusselt number as:

$$Nu_c = f\left(x, p_r\right).Nu_{liq} \tag{17}$$

and

$$Nu_{liq} = M\,\mathrm{Re}^e\,\mathrm{Pr}^{0.4} \tag{18}$$

Where Nu_{liq} is based on the liquid film HTC and f is a function of vapour quality and reduced pressure p_r. Coefficients M and e can be identified through experiments. The default values for M and e are 0.0389 and 0.7244 respectively. It is noteworthy that this method of correlating heat transfer coefficients can be successfully applied to smooth as well as to enhanced tubes, because proper correlation constants are back-calculated from experimental data. Subsequent coil performance simulations usually lead to errors of less than 5% on the overall duty, as checked for pure refrigerants by Armand and Molle (1991) and from numerous experiments conducted in our Laboratory.

For refrigerant mixtures, a condensation HTC is first calculated as described above by using the properties of the mixture. A corrected HTC is then calculated following the method recommended by Bell and Ghally (1971):

$$\frac{1}{h_m} = \frac{1}{h_c} + \frac{\Delta T C_p}{\Delta Q h_{vap}} \tag{19}$$

where ΔT is the temperature glide, Cp the vapour specific heat, h_{vap} the heat transfer coefficient based on the vapour properties and ΔQ the enthalpy change of the mixture.

A water chiller with a three-row condenser was tested with R407C and R22. Following the identification of parameters from experimental results, simulations were carried out to compare calculated and measured condenser duties. Figure 3 shows the comparison results. Although duties are slightly underpredicted for the simulations with R407C, they always fall within the ±5% region.

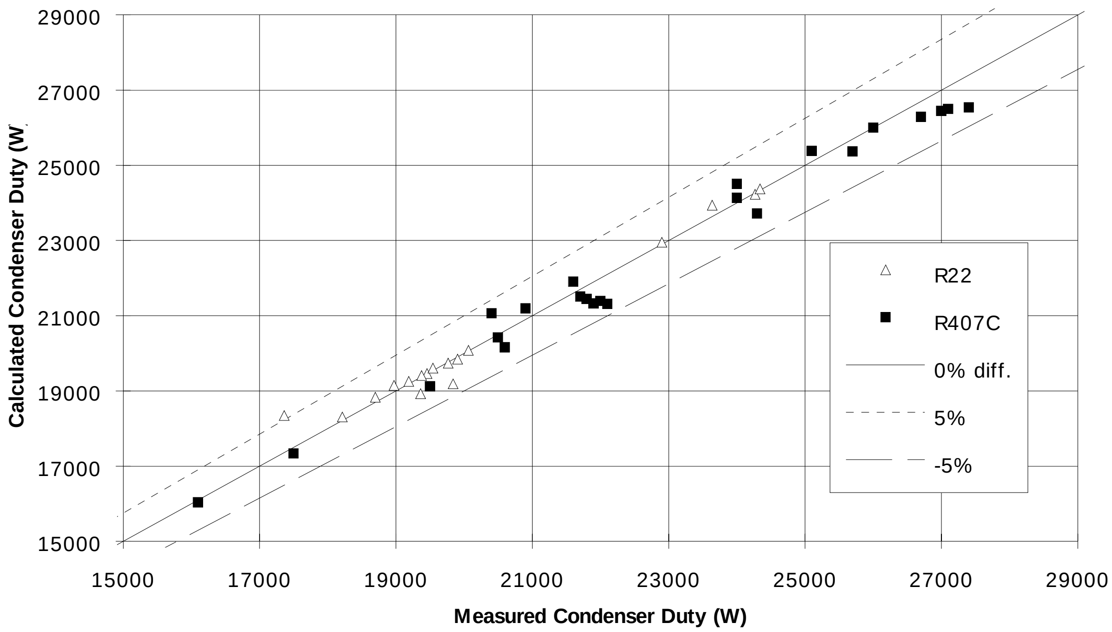

Figure 3: Comparison of calculated and measured duties
from condenser tests using R407C and R22

Expansion device modules

Currently available expansion device modules can simulate the behaviour of ideal (electronic), thermostatic valves, and adiabatic capillary tubes. The model output parameter is refrigerant mass flow rate. The ideal device module is the simplest model, for which the refrigerant flow rate is not calculated; hence it needs not be developed herein.

Thermostatic expansion valves (TXV)

The module is based on work originally developed by Tamainot-Telto et. al (1994), and uses slightly simpler models. Valves with external as well as internal equalisation are dealt with. Operating input variables are the bulb temperature T_B (assumed to be the refrigerant temperature prevailing at evaporator exit), the evaporating pressure P_E, the condensing pressure P_C, the refrigerant enthalpy at valve inlet and the value of superheat at evaporator exit. Mass flow rate through the valve is given by:

$$M_r = K\Phi\sqrt{2\rho_{vl}\Delta P} \tag{20}$$

where K is the flow coefficient, ρ_{vl} is the liquid density at entry to the valve, ΔP the pressure drop across the valve and Φ an empirical function. In the original work of Tamainot-Telto et. al (1994), flow coefficient K was assumed to be a power function of the flow rate of the following form:

$$K = q\left(\frac{M_r}{\rho_l}\right)^s \tag{21}$$

where s and q are constants to be deduced from experimental data. However, it has been found that assuming K as a constant yielded good results as well. By performing a balance of the forces controlling the pin displacement, Tamainot-Telto et al. obtained function Φ as:

$$\Phi = C_1 + C_2 T_B + C_3 T_B^2 + C_4 P_E + C_5 \Delta P \tag{22}$$

Equation 22 assumes that the valve port area is a linear function of pin displacement and that refrigerant vapour pressure in the bulb is a quadratic function of temperature.

Coefficients C_1 to C_5 and flow coefficient K need therefore be determined from experiments, which is accomplished using a minimisation procedure.

Adiabatic Capillary Tubes

An adiabatic capillary tube model has been developed by Grodent (1995) for pure refrigerants. The model assumes steady adiabatic flow in a straight horizontal tube with constant internal diameter, homogeneous two-phase flow (no slip between phases) in thermodynamic equilibrium and no effect of oil. Required input parameters are capillary tube inside diameter, length, roughness and operating conditions (inlet and discharge pressure, degrees of subcooling or vapour quality at inlet). The outlet parameter is refrigerant flow rate. If critical mass flow rate is obtained at the outlet section, the program yields critical pressure and flow rate.

The model takes into account refrigerant metastable flow by using the correlation developed by Chen et al. (1990). Grodent (1995) validated his model by conducting a comparison using results from 10 sources, totally around 100 test points. Reported tests used R134a, R12 and R22, with capillary lengths ranging from 0.6 to 5 m and diameters from 0.6 to 2 mm. It is noteworthy that at its present state of development, the model does not require any adjustable parameter.

Very satisfactory results were generally obtained, with errors below 10% on the predicted flow rates. Grodent further observed that when metastable flow was not taken into account, more accurate results were obtained for coiled capillary tubes.

Expansion device calculation modules are currently under modification to handle refrigerant mixtures with moderate temperature glides.

APPLICATION TO A SPLIT A/C UNIT

Results from tests conducted on an air-to-air split-A/C unit are used to illustrate the applicability of the programme. The unit was equipped with a reciprocating hermetic compressor, a 2-row plate-fin-and-tube condenser, a 4-row plate-fin-and-tube evaporator and a capillary tube (1.4 m long and 2 mm internal diameter). Details of construction can be found in Vadon (1992). Air temperatures, humidity and flow rates were measured at inlet and outlet of both HEXs. Refrigerant temperatures and pressures were also measured at inlet and outlet of each component. A refrigerant mass flowmeter was mounted downstream of the condenser. Heat balances were carried out on the air and refrigerant sides and showed differences of about 5%, but duties for component simulation were based on the refrigerant side, as this was thought to yield better results.

Determination of correlation constants for component modules

Tests results were first used to determine the correlations constants required in the calculation modules, following minimisation or regression techniques.

For the heat exchangers, no separate tests with air and water could be conducted. Hence, there were no means of determining the adjustable constants corresponding to air and refrigerant heat transfer coefficients (Eqs. 14, 15 and 18). For the refrigerant side, the constants were taken equal to their default values, i.e. the original heat transfer correlations were used. The tests results were then used to characterise the heat transfer coefficients on the air side for both HEXs. The derived constants (A and n for both HEXs) were then used to recalculate the duty using the operating parameters set during the tests. For both HEXs, errors smaller than 5% on the duty were found.

The constants required for the compressor module were determined. The subsequent simulation yielded errors smaller than 2% between calculated and measured refrigerant flow rates, and errors smaller than 4% between calculated and measured electrical power input. The expansion device module (capillary tube) did not require adjustable parameters.

Split A/C unit simulation

Using the newly-determined module parameters, the A/C unit operation was simulated. The evaporator duty calculation results are shown in Figure 4. All calculated duties lie well within ± 5% of the measured duties. A comparison of calculated and measured condenser duties is shown in Figure 5. All calculated and measured duties agree within ±5%, but for one point. The compressor power input calculations (Figure 6) show a little more scatter, with all but 3 points lying within the ±5% error region. COP calculations (Figure 7) lead to discrepancies of the same order (5%).

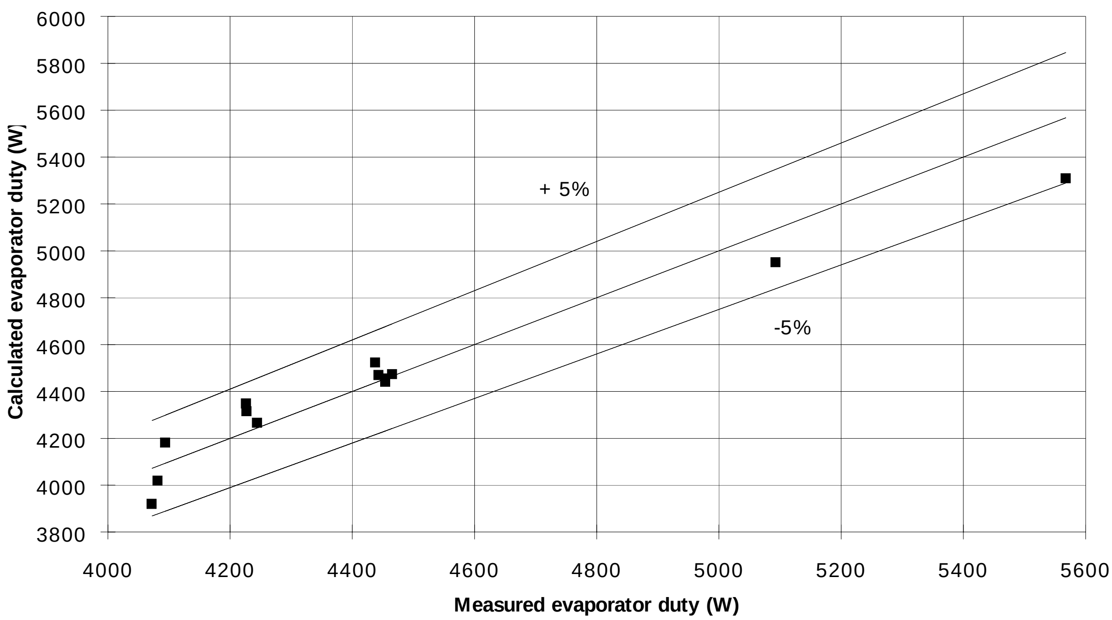

Figure 4: Comparison of calculated and measured evaporator duties

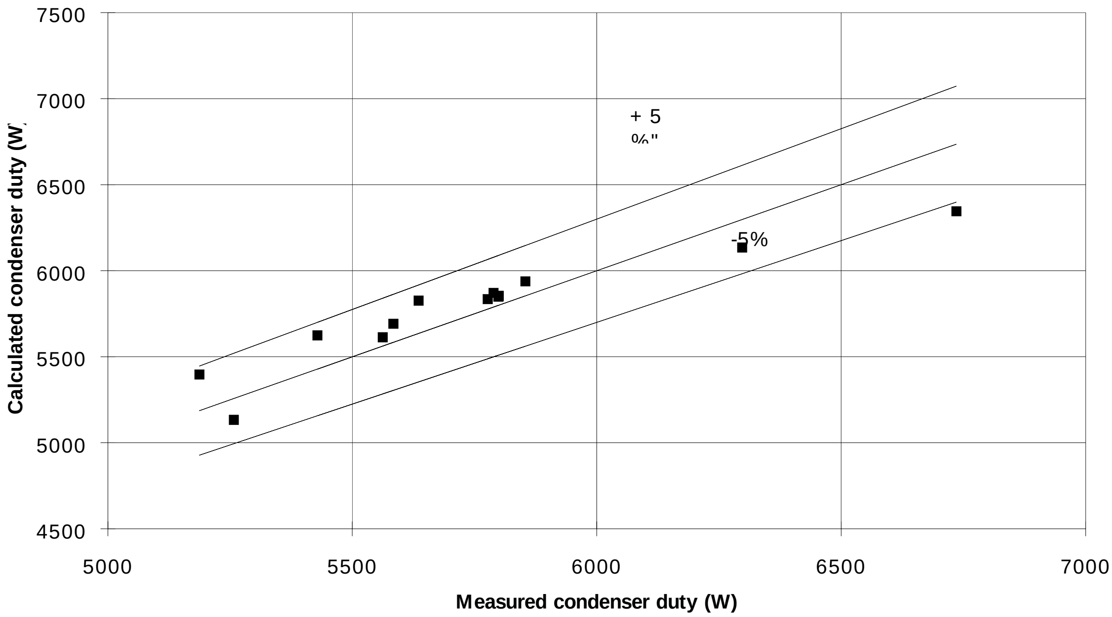

Figure 5: Comparison of calculated and measured condenser duties

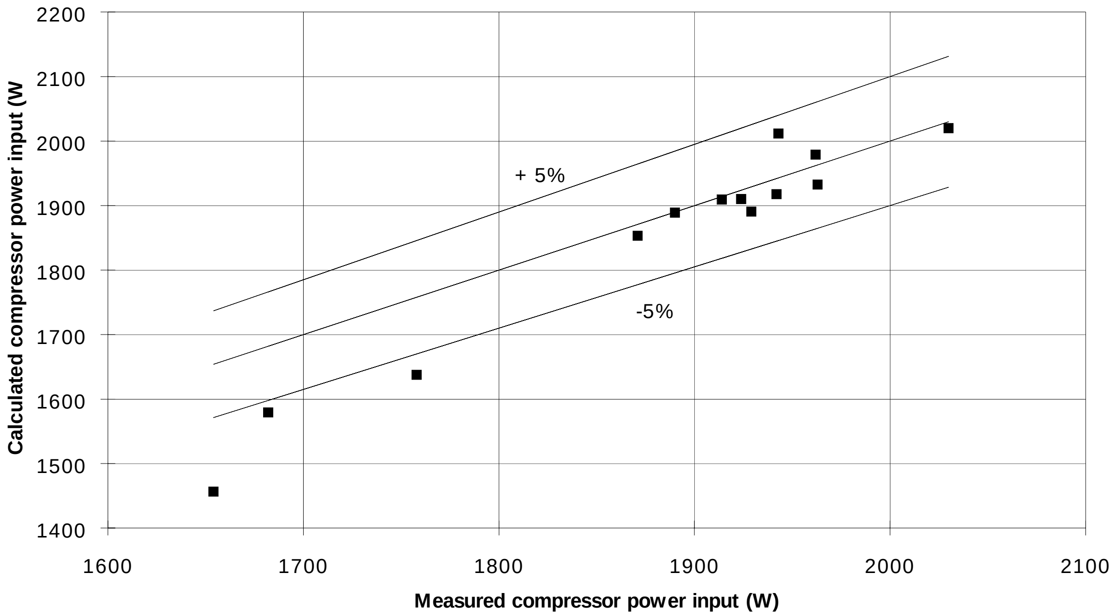

Figure 6: Comparison of calculated and measured compressor power input

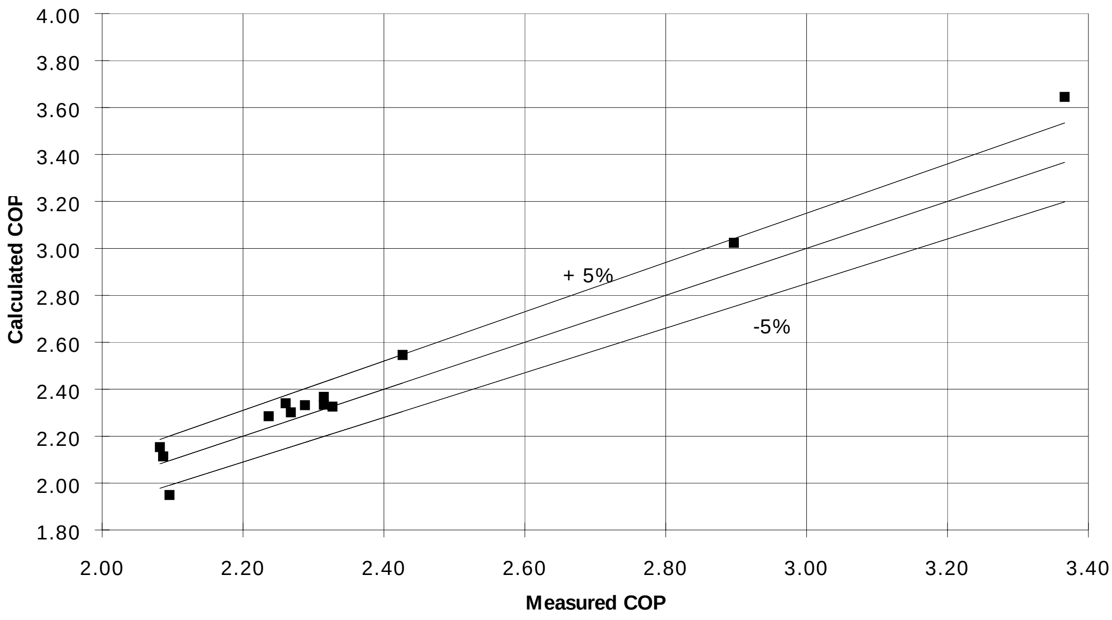

Figure 7: Comparison of calculated and measured COP

APPLICATION TO COMPONENTS OF A WATER CHILLER USING R22 AND R407C

A 15 kW water chiller was tested in the cooling mode. The chiller consisted of a single circuit comprising a counter-current plate evaporator, a scroll compressor, a thermostatic expansion valve and a 3-row plate-fin-and-tube air-cooled condenser.

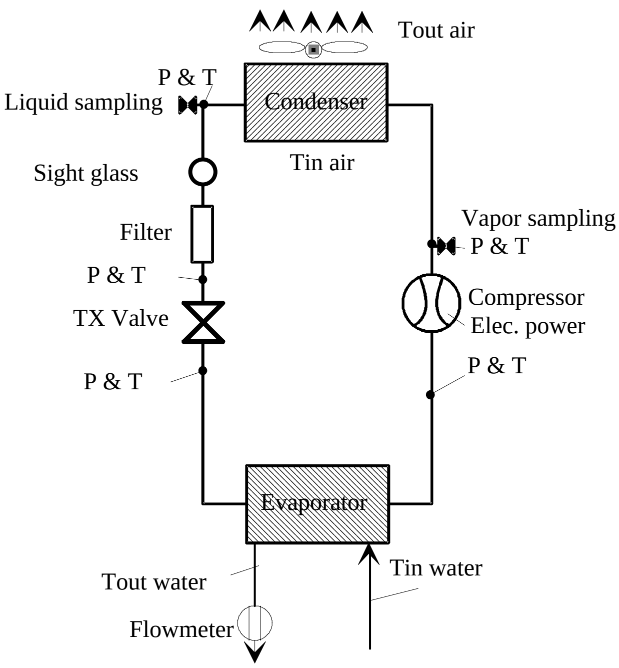

Figure 8: Water chiller process flow diagram and instrumentation

Simulation of components

Following the same procedure used for the split A/C unit, correlation constants required in the calculation modules were obtained from tests results. Using these newly-determined module parameters, the condenser, compressor and thermostatic expansion valve were simulated.

The compressor power input calculations (Figure 9) show very good results, with all the points lying well within the ±5% error region. Good correlations were also obtained for the mass flow rate (save one point with the R407C fluid, see Figure 10). As for the discharge temperature (Figure 11), the calculation results were also accurate, with all the points lying within the ±3K region.

A comparison of calculated and measured condenser duties was shown in Figure 3. All calculated and measured duties agree within ±5%, but for one point.

Finally, the expansion device module calculations have lead to good results too, as it can be seen from Figure 12.

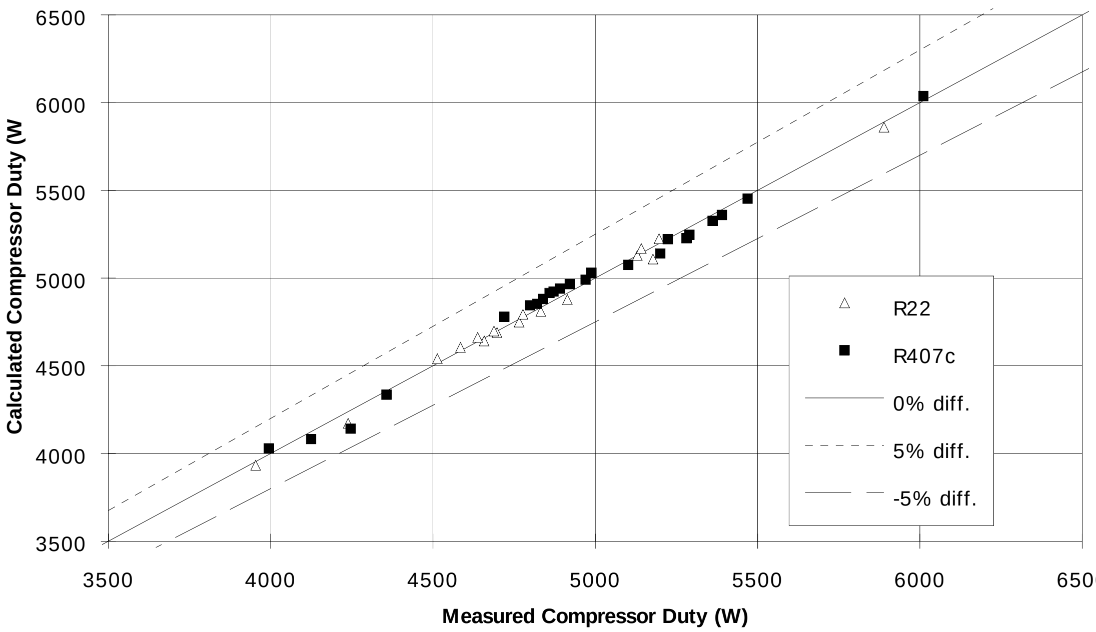

Figure 9: Comparison of calculated and measured compressor work

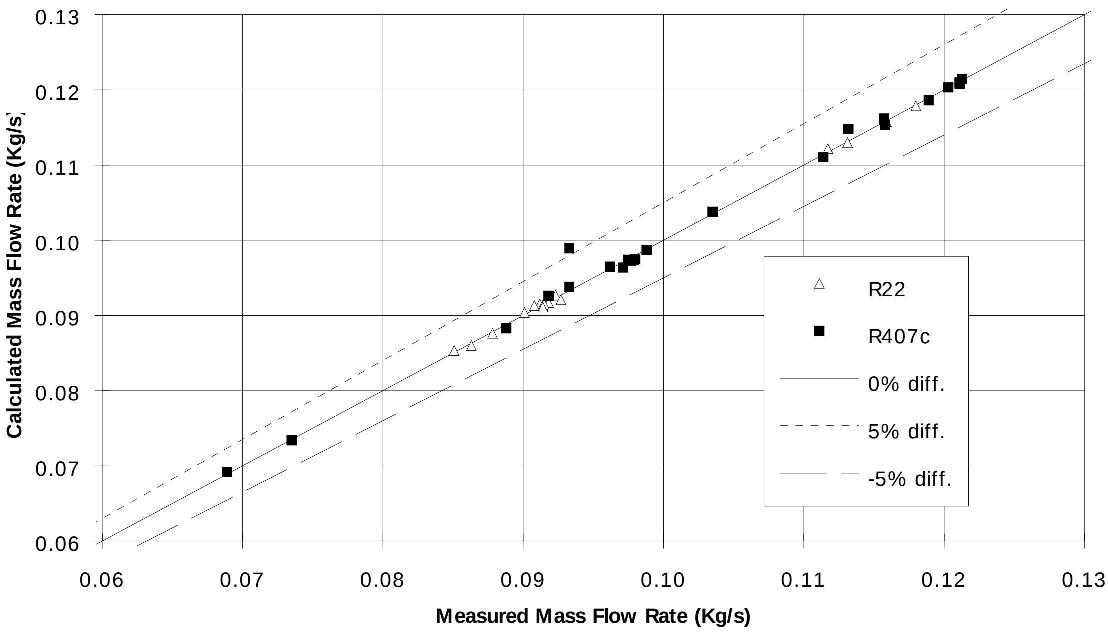

Figure 10: Comparison of calculated and measured compressor mass flow rate

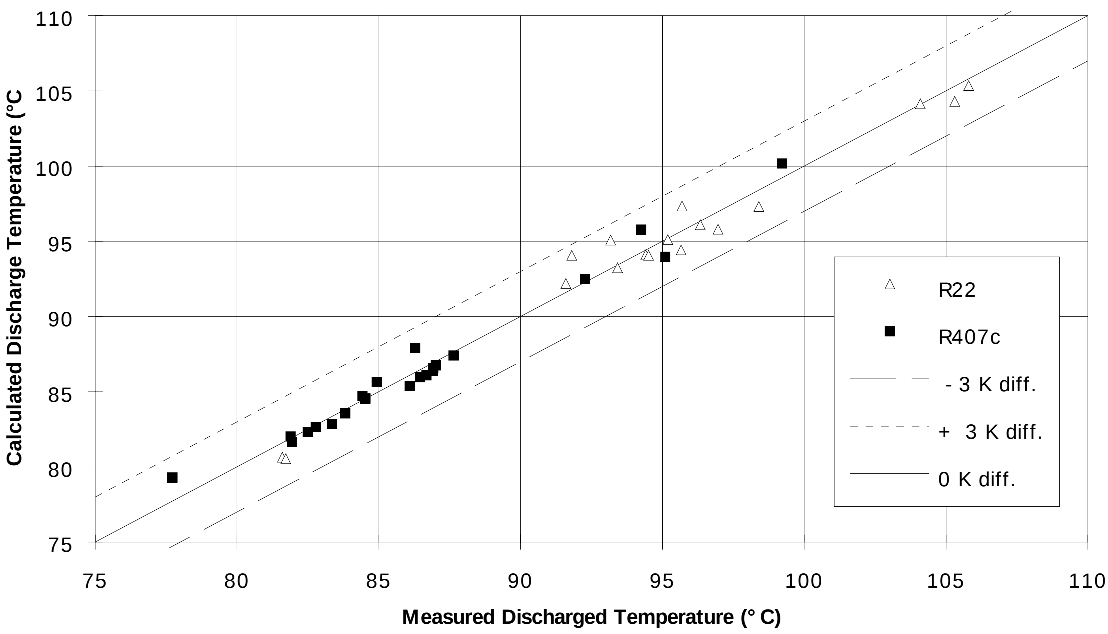

Figure 11: Comparison of calculated and measured compressor discharge temperature

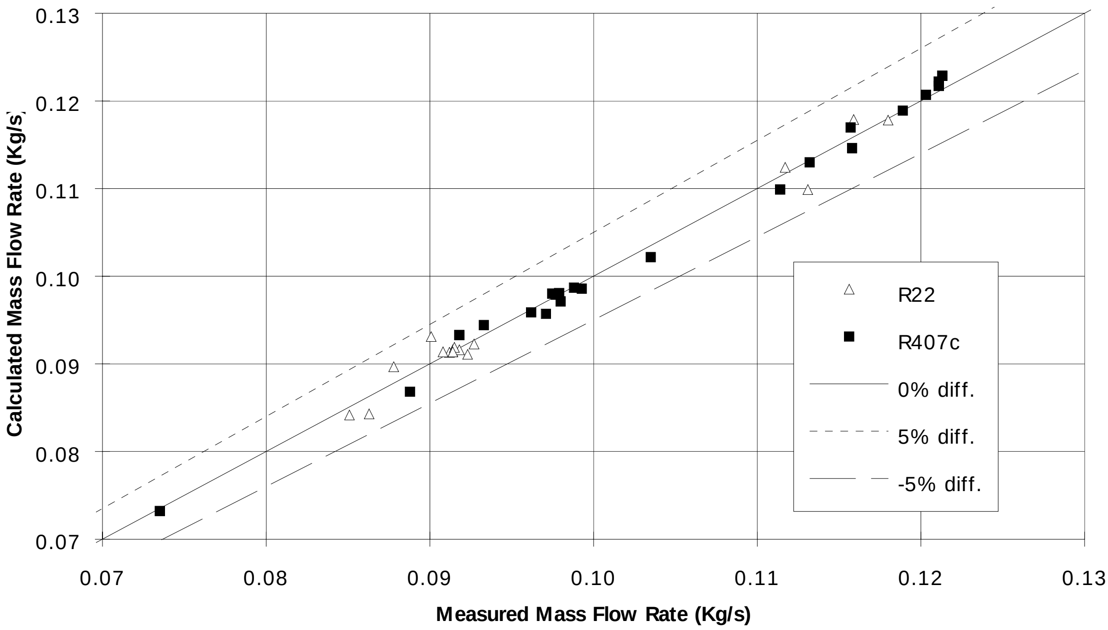

Figure 12: Comparison of calculated and measured mass flow rate through the expansion valve

CONCLUSIONS

A model for the steady-state simulation of heat pumps and refrigerating equipment has been presented. Because the component modules are "calibrated" via actual test data, very accurate results can be obtained. The numerous individual mathematical models can be dynamically linked to form a flexible programme and enable the simulation of a variety of different machine configurations with both pure and mixed refrigerants. This structure is evolutive because new configurations can be simulated as new component modules are developed. The proposed model was validated with experimental results from an air-to-air split unit equipped with a capillary tube and finned-tube heat exchangers over a range of external conditions. Discrepancies smaller than 5% are obtained on the cooling duty and COP. Current programme developments will enable the simulation of more types of heat exchangers (e.g. plate HEX) and to take into account the effect of refrigerant mass in the system.

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