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HERMETIC COMPRESSOR MODELS DETERMINATION OF PARAMETERS FROM A MINIMUM NUMBER OF TESTS

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ABSTRACT

The model described in the present paper simulates hermetically sealed scroll, reciprocating and rotary compressors operating over a wide range of conditions in air-conditioning systems for low-capacity machines (4 to 10 kW). It is a part of the overall model developed for simulating air-conditioning equipment under steady conditions. Only a small number of testing points (three) is required to determine the parameters of the model and to predict output conditions from input variables. Tests were carried out on six different types of compressor to determine these parameters. The results give maximum average quadratic errors of less than 5 K on the discharge temperature and less than 5 and 6% respectively on the refrigerant mass flow rate and the electrical power consumption.

INTRODUCTION

When designing an air-conditioner, the performance of each individual component must be known over a wide range of operating conditions to choose the best components, study their interactions within the machine and determine the behaviour of the complete system. Determining performance through experiments can be costly and time-consuming. It is therefore preferable to use simulation : a sufficiently accurate, and experimentally proven, model is capable of accurately predicting the system response. To this end, we have developed a program built as a set of modules implementing the models of individual components [1].

The models published in the literature vary widely in complexity. The most complex models imply the computation of a set of differential equations for mass, momentum and energy balances with appropriate initial and boundary conditions. This approach has been applied to reciprocating compressors [2] and scroll compressors [3, 4, 5]. The disadvantage is that the model requires a large number of parameters concerning the compressor internal geometry (height of scrolls, excentricity, ...) which are not always known. These models are only used for detailed design of compressors and are difficult to apply to a complete unit. Another approach, which has been widely applied to reciprocating compressors [6, 7], is to follow the fluid evolution as it passes through the compressor. However, these models also require a large number of parameters, difficult to quantify.

We selected a more overall approach to model the behaviour of scroll [8], reciprocating [9] and rotary [10] compressors. The result is a simple, fast model whose parameters can be determined from a minimum number of tests. The model is applicable to all types of small (4 to 10kW) hermetic compressors mentioned above.

THE MODEL

The model output variables (discharge temperature T_d, refrigerant mass flow rate $\dot{m}_f$ and electrical power consumption $\dot{E}$) are deduced from the input variables (suction temperature T_s, suction pressure P_s and discharge pressure P_d). Both the main input and output variables were selected to be module interfaces of other components in the system [1]. The relationships used to calculate the output variables were determined from the refrigerant thermodynamic cycle and considerations on the compressors operation. The following assumptions are made on the refrigerant cycle :

- constant suction pressure and temperature
- a polytropic compression. In reciprocating and rotary compressors, the maximum internal pressure is equal to the discharge pressure, while, in scroll compressors, the maximum internal pressure P_2 is determined from the "built-in" pressure ratio,
- constant discharge pressure and temperature,
- if the scroll compressor maximum internal pressure P_2 is different to the discharge pressure, a certain amount of fluid is adiabatically injected or expelled,
- expansion of the clearance volume in reciprocating compressors.

Reciprocating compressor

Knowing the polytropic compression coefficient, it is possible to calculate the specific volume at discharge from the equation :

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

The discharge temperature is calculated from the gas state equation giving the real properties of the refrigerant (R22). To calculate the refrigerant mass flow rate $\dot{m}_f$, it is assumed that the polytropic expansion coefficient of the clearance volume is the same as for the compression :

$$\dot{m}_f = \rho_s V N\left(1 + \tau\left(1 - \delta^{1/k}\right)\right) \tag{2}$$

V is the geometric displacement of the compressor, ρ_s the specific mass of the refrigerant at inlet conditions, N the compressor speed, τ the effective clearance factor and $\delta = P_d/P_s$ the pressure ratio. The electrical power consumption $\dot{E}$ is calculated from the compressor enthalpy balance :

$$\dot{E} = \dot{m}_f\left(h_d - h_s\right) - \dot{Q} \tag{3}$$

where h_s and h_d are the specific enthalpies before and after compression and $\dot{Q}$ is the thermal losses dissipated to the compressor environment.

Scroll compressor

The maximum internal pressure P_2, which depends on the "built-in" pressure ratio, and the specific volume v_2 are calculated from the following equations :

$$P_2 = P_s \varepsilon^k \qquad \text{et} \qquad v_2 = \frac{v_s}{\varepsilon} \tag{4}$$

ε is the volumetric ratio.

The refrigerant supply and exhaust phases are very short and, therefore, assumed to be adiabatic :

$$v_d = v_2 \left(\frac{P_2}{P_d}\right)^{1/\gamma} \tag{5}$$

where γ is the mean isentropic coefficient. As previously, T_d is calculated from the real characteristics of R22. If the gas leakage through the tips and flanks of the scrolls parts is $\dot{m}_l$, the refrigerant mass flow rate $\dot{m}_f$ can be estimated as :

$$\dot{m}_f = \rho_s V N - \dot{m}_l \tag{6}$$

The electrical power input is calculated from equation (3).

Rotary compressor

The compression phase is identical to that in a reciprocating compressor and, consequently, equation (1) is used to calculate the discharge specific volume. However, the refrigerant mass flow rate is determined by equation (6) to take into account the internal leakage. Finally, the electrical power is determined using equation (3).

DETERMINATION OF THE MODEL PARAMETERS

Some of the parameters used in the models are known geometrical or technical characteristics :
- for scroll compressors : the volumetric ratio, geometric displacement and speed,
- for other compressors : the geometric displacement and speed.

Other parameters (leakage flow rate, effective clearance factor, polytropic coefficient, heat losses with the environment) must be determined by tests.

Three reciprocating compressors (A, B and C), two rotary compressors (D and E) and a scroll compressor (F) were tested to calculate the model parameters using the above set of equations. Table 1 gives the characteristics of these compressors. A large number of tests were carried out under varying conditions to determine how the parameter laws varied with the input variables. The scroll compressor was tested at variable speed.

Compressor	Reciprocating			Rotary		Scroll
	A	B	C	D	E	F
Geometric displacement (cm^3)	34.45	82.20	63.94	20.75	24.70	45.00
Refrigerating capacity* (W)	4170	9260	8100	3140	4070	7520

Table 1 : Characteristics of compressors (* ASHRAE conditions)

The polytropic coefficient is an overall coefficient which depends on the heat exchange, i.e. on the mass flow rate of refrigerant and pressure ratio. The polytropic coefficient for the scroll compressor obeys the law :

$$k = a_k \dot{m}_k + b_k \delta + c_k \tag{7}$$

For reciprocating and rotary compressors, it obeys an exponential law of the type :

$$k = (b_k P_s + c_k)\delta^{a_k} \tag{8}$$

The equation for the leakage flow rate in scroll and rotary compressors, generally calculated from isentropic expansion laws, is :

$$\dot{m}_l = a_m \rho_s V N + b_m \sqrt{\delta} + c_m \tag{9}$$

The effective clearance factor in reciprocating compressors, which reflects all imperfections in their flow characteristics, is calculated from the following equation [10] :

$$\tau = a_m \delta^{b_m} \tag{10}$$

The same law is used to describe heat-exchange with the environment, regardless of the type of compressor :

$$\dot{Q} = a_q \dot{m}_f + b_q \delta + c_q \tag{11}$$

RESULTS

Initially, tests were carried out over a wide range of operating conditions ($3<P_s<8$ bar, $10<P_d<22$ bar) and coefficients a_i, b_i and c_i were determined by regression on all the results. With these values of the coefficients, the maximum average quadratic errors in the output variables were 4 K on the discharge temperature, 4% on the refrigerant mass flow rate and 5% on the power consumption. This applied to all the compressors tested.

Three tests are sufficient to determine the nine coefficients used in the model. However, the tests must be carefully selected. All the tests carried out were systematically investigated to determine combinations of three tests which gave the minimum average quadratic error. It was found that the three tests can be selected as follows :

- two tests must use similar inlet conditions but with the largest possible difference in the discharge pressure,
- the third must use a different set of inlet conditions but the discharge pressure must be as close as possible to one of the previous two tests.

Table 2 gives the values of the coefficients found during these tests for the six compressors. Figures 1, 2 and 3 give the values (discharge temperature, refrigerant mass flow rate and power consumption) calculated with each model. The maximum root mean square errors are less than 5 K on discharge temperature, 5% on mass flow rate and 6% on electrical power consumption for all types of compressor : these results are almost as accurate as when the coefficients were determined from the complete set of tests.

Coefficient / Compressor	a_m	b_m	c_m	a_k	b_k x 10^6	c_k	a_q	b_q	c_q
A	0.416	-0.757		-0.071	-0.0037	1.348	17320	131	-353
B	0.357	-0.425		-0.060	-0.0170	1.466	6780	159	-666
C	0.273	-0.252		-0.082	-0.0260	1.542	10840	185	-732
D	0.120	0.003	-0.005	-0.100	-0.0280	1.545	5959	59	-58.7
E	0.347	0.007	-0.022	0,262	-0.0006	1.171	19810	79	-610
F	0.059	0.003	-0.006	0.468	0.0630	1.001	7224	181	-235

Table 2 : Model coefficient values for different types of compressor (SI units)

CONCLUSION

The models developed for the hermetic reciprocating, scroll and rotary compressors accurately simulate the behaviour of the compressors under various operating conditions. These models are based on

generally accepted assumptions and require a minimum number of parameters (polytropic coefficient, leakage flow rate or effective clearance factor and heat losses with the environment). The laws representing variations in these parameters are based on tests covering a wide range of operating conditions ($3<P_s<8$bar, $10<P_d<22$ bar). These laws were determined for several compressors and use a minimum number of coefficients whose values can be identified from three tests. However, these tests must be carefully selected to give different operating points. The results obtained from six compressors of different types give average quadratic errors less than 5 K on the discharge temperature, less than 5% on flow rate and less than 6% on electrical power consumption.

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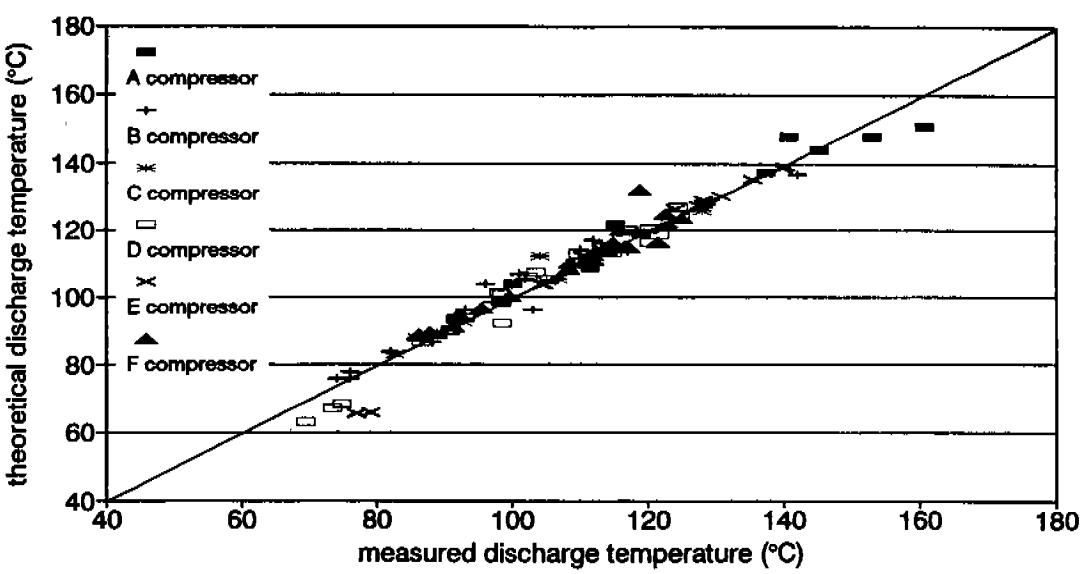

Figure 1 : Calculated vs measured discharge temperature

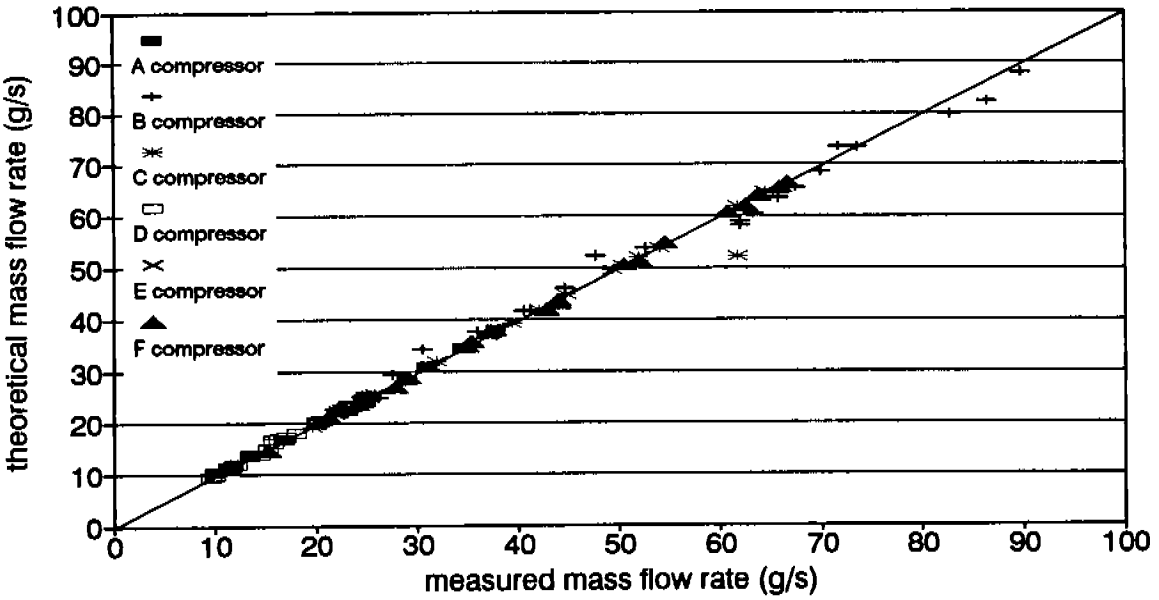

Figure 2 : Calculated vs measured refrigerant mass flow rate

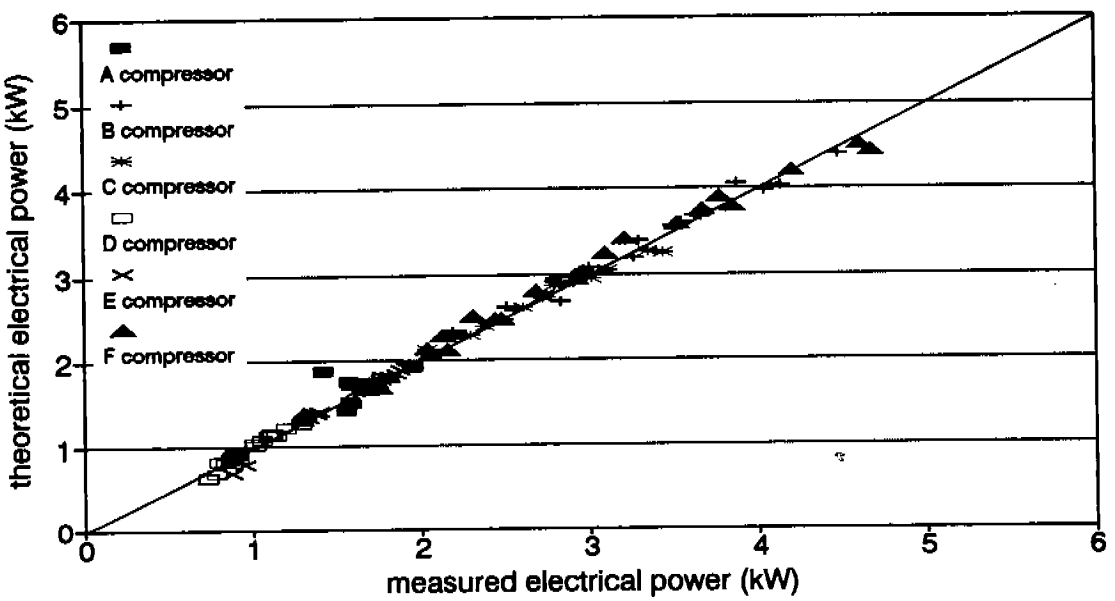

Figure 3 : Calculated vs measured electrical power