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Small Variable Speed Hermetic Reciprocating Compressors for Domestic Refrigerators

B. D. Rasmussen
Technical University of Denmark

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B.D. Rasmussen

Department of Energy Engineering, Energy Systems Section
Technical University of Denmark

ABSTRACT

This paper contains both a theoretical and experimental investigation of some of the fundamental characteristics of a small variable speed hermetic reciprocating compressor intended for application in domestic refrigeration. The results of a previously published simulation model for variable speed compressors are compared with experimental results obtained in a compressor test bench. The influence of speed on compressor performance is discussed with focus on valve modelling and internal thermal phenomena. Further plans for development and validation of the model as well as experimental investigations are discussed.

NOMENCLATURE

C_C	Contraction coefficient	[-]	$T_{GAS,AVG}$	Avg. temperature of gas in shell	[°C]
COP	Coefficient Of Performance	[-]	T_{MOTOR}	Motor temperature (stator)	[°C]
d	Diameter of valve port	[m]	T_{OIL}	Temperature of oil in shell	[°C]
HRR	Heat Rejection Ratio	[-]	T_{SC}	Temperature of gas in suct. chamber	[°C]
N	Compressor speed	[rpm]	T_{SHELL}	Avg. temperature of shell surface	[°C]
N_{MAX}	Max compressor speed	[rpm]	X_{MAX}	Max valve lift	[m]
$\dot{m}$	Mass flow of refrigerant	[kg/s]	X_{MEAN}	Mean valve lift	[m]
MR	Mixing ratio	[-]	Δp	Pressure difference	[kPa]
P_{MOTOR}	Power consumption of motor	[W]	η_{IS}	Isentropic efficiency	[-]
$\dot{Q}_{CONV}$	Heat rejected from shell by convection	[W]	η_M	Motor efficiency	[-]
$\dot{Q}_E$	Refrigeration capacity	[W]	η_{VOL}	Volumetric efficiency	[-]
$\dot{Q}_{RAD}$	Heat rejected from shell by radiation	[W]	ζ	Loss coefficient for valves	[-]
$T_{C,INLET}$	Temperature of gas in suction stub	[°C]	ρ	Density of gas	
T_{DSCH}	Discharge gas temperature	[°C]	[kg/m^3]		

INTRODUCTION

Theoretical and practical investigations [1], [2] have shown that continuously controlled domestic refrigerators using variable speed compressors have a significantly lower energy consumption compared to the conventional refrigerators operating in on/off-mode. The major reason for the reduction in energy consumption is the significant improvement in the operating conditions of the refrigeration system (higher evaporation temperature and lower condensing temperature). In connection with the investigations mentioned other theoretical investigations [3] have shown that the performance of the compressor may also depend on the speed. To be able to design a continuously controlled refrigeration system for variable speed compressors it is therefore necessary to know how the speed of the compressor will influence it's performance in the refrigeration system.

Fundamental knowledge of the allowable speed range of the compressor during steady state operation and the general influence of speed on it's efficiency is necessary to be able to optimise the system.

The development of small hermetic reciprocating compressors with variable speed can be based on the fundamental design of the fixed speed compressors. Traditionally one-phase asynchronous induction motors are used due to their easy electrical connection to the mains, relatively low price and high robustness. The one-phase motors can also be used for variable speed applications, but due to their poor speed-efficiency characteristics they are often replaced by either their 3-phase counterparts or with more expensive high efficiency motors such as motors using permanent magnets. Another critical component of the compressor when switching to variable speed is the valves. They must be able to operate at both low and high frequencies with acceptable efficiency. Further more the characteristics of the lubrication system and other internal thermal phenomena with influence on compressor efficiency must be investigated.

In [3] a simple simulation model for small variable speed compressors is presented. The theoretical results presented showed that for constant operating conditions an optimal speed existed. Unfortunately a comparison of the theoretical results presented and experimental results could not be included in the paper due to lack of experimental data. Therefore a compressor test bench for small compressors have been designed and prototypes of variable speed compressors have been manufactured.

COMPRESSOR MODEL

The compressor model presented in [3] belongs to the category of dynamic compressor models, meaning that the description of phenomena inside the compressor are based on equations using the instantaneous values for the thermodynamic and -physical properties. Since the publication of [3] modifications have been made to the model. The number of control volumes have been increased so that the current model divides the compressor into 6 control volumes individually comprising the shell, suction muffler, suction chamber, cylinder, discharge chamber, discharge muffler and discharge tube. Please refer to [3] for a more detailed description of the model.

One of the most important areas in compressor modelling is the description of the flow through the valves. The choice of valve model detail level depends on the intended use of the compressor model. If investigation of the variations within a single revolution is the intended use for the compressor model then the valve model must include the dynamics of the reed. Models that are based on the description of the instantaneous lift of the reed, including the mechanical- and flow related parameters each depending on the lift, are available [4] and can be implemented into a compressor model. However, the knowledge of how the mechanical- and flow related parameters vary with the instantaneous lift and geometry of the valve is scarce. Further the implementation of these complex valve models into compressor models would significantly increase the time needed for simulation. If the intended use of the compressor model is to investigate the variations in overall performance more simple valve models may be used.

Knowledge of how speed influences compressor performance for small hermetic reciprocating compressors is very limited. Will the operation of the valves dominate the performance of the variable speed compressors and is it necessary to use complex models to describe the overall influence of valves on compressor performance? To gain knowledge about these subjects it was decided investigate just how good a simple valve model could be in describing the overall performance of the compressor.

The assumption behind the valve model presented in [3] is that existing valves designed for fixed speed operation will work properly in a speed interval only. It is further assumed that the averaged valve lift during the period when the valve is open increases with speed until a max. lift is reached (usually represented by the valve stop). Beyond this max. speed the valve may not function properly because the high operating frequency may cause serious instabilities in the reed making the valve float or otherwise malfunction.

The model for the flow through the valves is based on the following equation

$$\dot{m} = \zeta \rho \pi d X_{MEAN} C_C \sqrt{\frac{2\Delta p}{\rho}} \tag{1}$$

where $\dot{m}$ is the mass flow, ζ is an empirically determined loss coefficient, ρ is the density of the gas, d is the diameter of the valve port, C_C is the contraction coefficient for the flow through the valve and Δp is the pressure difference over the valve. Theoretically both ζ and C_C depend on the lift, but not knowing how ζ and C_C vary with the lift they are assumed to be constant. The model further assumes that the flow through the valve is incompressible.

The mean valve lift is defined as

$$X_{MEAN} \equiv X_{MAX}\sqrt{\frac{N}{N_{MAX}}} \tag{2}$$

where X_{MAX} is the max valve lift limited by the valve stop, N is the speed of the compressor and N_{MAX} is the max speed where proper valve function can be expected. The square-root profile for X_{MEAN} was selected among both linear and non-linear profiles because this profile resulted in pV-diagrams that in form agree with what is generally expected for small reciprocating compressors. N_{MAX} also depend on valve geometry and mechanical properties of the valve reed. For the type of compressors modelled a value of 5000 rpm was selected. With the profile and max speed selected the value of X_{MEAN} range from 60% to 95% of X_{MAX} for the speed interval investigated.

Apart from the parameters describing the general operating conditions the following parameters are needed as input for the simulation model:

- Temperature of the gas entering the suction muffler
- Viscosity of the oil/refrigerant mixture in the compressor
- Cylinder wall temperature

The high temperature of the gas entering the suction muffler is a result of the internal heating of the suction gas from the inlet stub to the suction muffler entrance. This phenomena can be expressed by a dimensionless number called mixing ratio defined by

$$MR \equiv \frac{\overline{T}_{SHELL} - T_{SC}}{\overline{T}_{SHELL} - T_{C,INLET}} \tag{3}$$

where MR is the mixing ratio, $\overline{T}_{SHELL}$ is the averaged temperature of the gas in the shell, T_{SC} is the temperature of the gas entering the first suction chamber and $T_{C,INLET}$ is the temperature of the gas entering the compressor shell through the suction stub. MR can be interpreted as the ratio between the mass flow entering the suction chamber directly from the shell and the total mass flow entering the suction chamber. The (1 - MR) part of the flow can be interpreted as the part coming directly from the suction stub without being mixed with the gas in the shell. MR is determined from measurements.

The viscosity of the oil-refrigerant mixture is determined from diagrams showing viscosity as a function of temperature with content of refrigerant (R600a) as parameter. The oil-refrigerant concentrations are determined from diagrams showing the concentration as a function of temperature and pressure. Both oil temperature and shell pressure are determined from measurements.

The evaluation of the cylinder wall temperature is based on measurements of the temperature distribution on the surface of the cylinder block.

EXPERIMENTAL SET-UP

The compressor used for the experiments is a prototype of a variable speed compressor using the compressor block, compression mechanism, valves, muffles and chambers from an existing compressor designed for fixed speed. Figure 1 shows the design of the compressor. The motor used is a new three-phase 4-pole asynchronous induction motor designed specially for this compressor when using R600a (isobutane) as refrigerant. The motor is operated by a PWM-type frequency converter (Pulse Width Modulation) also designed for this compressor and it's intended application in domestic refrigeration.

The compressor test bench used is designed to facilitate tests of very small variable speed hermetic reciprocating compressors with refrigeration capacities ranging from 20 to 500 W (68 to 1706 BTU/h) at evaporation temperatures ranging from -30 to +5 °C (-22 to 40 °F). When using R600a as refrigerant this is equivalent to testing compressors with stroke volumes ranging from 1.4 to 5.0 cm^3 (0.0854 to 0.3051 in^3) over a wide range of operating conditions. The mass flow of refrigerant can either be measured by a mass flow meter (Coriolis type) or be estimated from the heat balance of the evaporator.

Measuring the power consumed by a motor operated by an PWM-type frequency converter is non trivial compared to measuring the power consumed by a motor supplied with a sinusoidal voltage taken directly from the mains. The higher harmonics in the converter output creates electrical disturbances that exclude the use of conventional power transducers and analysers. Therefore the power to the compressor motor is measured using a Universal Power Analyser, manufactured by Voltech Ltd. type PM3000A, capable of handling the higher harmonics caused by the high switching frequency of the PWM-type converter used.

In the test bench the compressor is positioned in a calorimeter so that determination of the heat rejected from the shell by means of convection and radiation can be determined. Knowing the power consumed by the motor, the mass flow and enthalpy difference of the refrigerant through the compressor and the average temperatures of the shell and the radiation shields, the convective part of the heat rejected from the compressor can be determined. The shell surface temperature was averaged from 6 independent measurements and the shield temperatures from 3 independent measurements.

The compressor used was further instrumented internally with a total of 22 thermocouples measuring the temperatures of the various components. These measurements enables the determination of the parameters needed as input for the simulation model presented.

COMPARISON

The results from the simulations are compared to experimental data from a test series with constant operating conditions (evaporating- condensing, ambient and suction gas temperatures) and variable speed. The general operating conditions selected are close to the operating conditions measured for a continuously controlled refrigerator at ISO-7371 test conditions. Figure 2 compares the simulated capacity, power consumption and COP with experimentally obtained values. Both capacity and power consumption increase with speed but the COP decrease with speed. The max error between simulation and experimental values for capacity is 4.4 %, for the power consumption the max error is 3.4 %.

Figure 3 compares the simulated efficiencies with the experimentally obtained values. For the volumetric and isentropic efficiencies the model show the same tendency as the experimental results; the higher the speed the lower the efficiency. The max error between simulated and experimentally obtained values for the volumetric efficiency is less than 3.7 %, for the isentropic efficiency the max error is less than 1.6 %. The motor efficiency plotted is determined from measurements at various loads and speeds in separate motor test bench.

The experimentally obtained variations in both internal and external temperatures as a function of speed are shown on figure 4. The trend is that both internal and external temperatures increase with speed. A closer investigation of the results reveal that for low speeds the average gas temperature in the shell is higher than the oil temperature. For higher speeds the opposite occur. This change indicate that for low speeds the heat transfer from the components to the shell is dominated by the heat transfer capabilities/properties of the gas in the shell, whereas for high speeds the oil heat transfer capabilities/properties are dominating.

In figure 5 the heat rejected from the compressor shell by means of convection and conduction is shown. The Heat Rejection Ratio defined as

$$HRR = \frac{\dot{Q}_{RAD}}{\dot{Q}_{RAD} + \dot{Q}_{CONV}} \quad (4)$$

is almost constant, with only a slight dependency of speed. Figure 5 also show that the mixing ratio (MR) depend on speed. When speed increases the mixing ratio is reduced indicating that internal heating of gas is reduced.

CONCLUSIONS

The comparison of the simulation results with the experimental results show that the overall model accuracy is acceptable. No error between theoretical and experimental values presented are greater than 5 %. Further comparison with experimental values obtained for the same compressor and speed interval but with sliding operating conditions shows a similar accuracy. The presentation of this comparison is omitted due to limitations in the space available for this paper. It can therefore be argued that the valve model presented can be used in dynamic models for small reciprocating compressors with variable speed intended for describing the overall dependence of capacity and efficiency on speed within the speed interval investigated. Since the valve model doesn't include the phenomena necessary for describing the valve dynamics within the period when the valve is open, the model is not considered adequate for in depth investigations of the variations within a single revolution.

The results presented in [3] indicated that for fixed operating conditions an optimum speed existed. The results presented in this paper doesn't show this tendency. Instead the experimental results indicate that the lower the speed, the higher the efficiency. One of the reasons for this discrepancy could be that the efficiency-speed characteristics of the motor simulated in [3] are different than the experimentally obtained characteristics used in this paper.

FURTHER WORK

It is the intention to further investigate the influence of speed on compressor performance and validate the details in the simulation model concerning the cylinder and muffler processes. Based on the conclusions from this paper the valve model must then be replaced by a more detailed model capable of describing dynamic valve phenomena such as opening and closing delay, back flow and reed impact velocity ratios at the seat and stop respectively. For this investigation a prototype compressor with more intensive internal instrumentation is currently being prepared.

ACKNOWLEDGEMENTS

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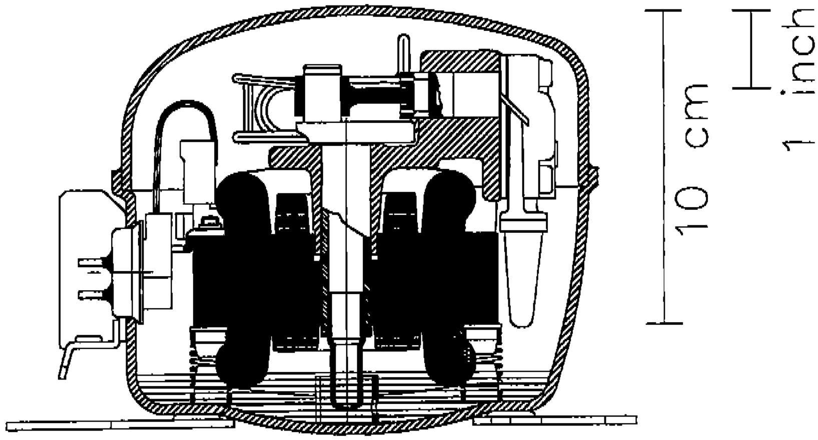
Figure 1: Sectional drawing of compressor



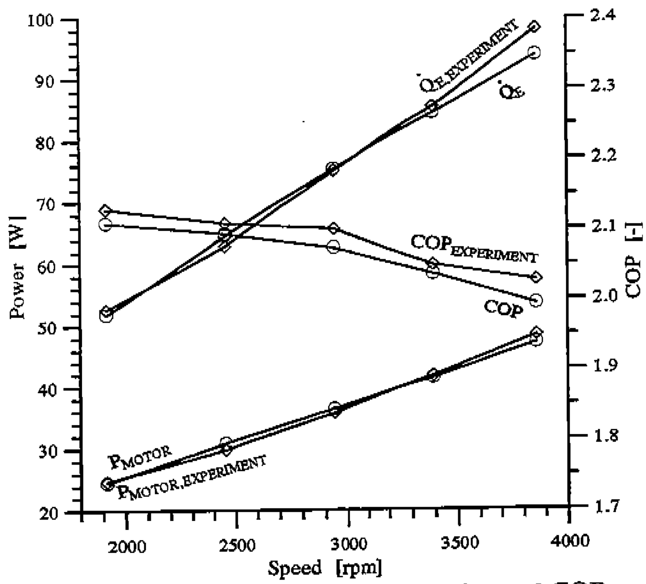


Figure 2: Capacity, power consumption and COP.

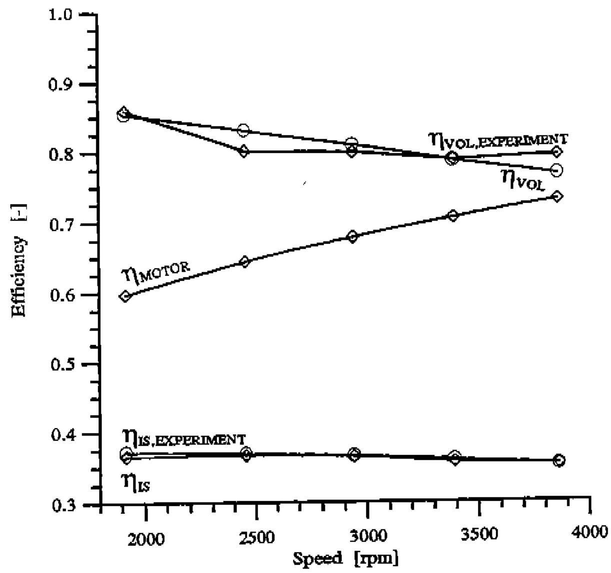


Figure 3: Efficiencies

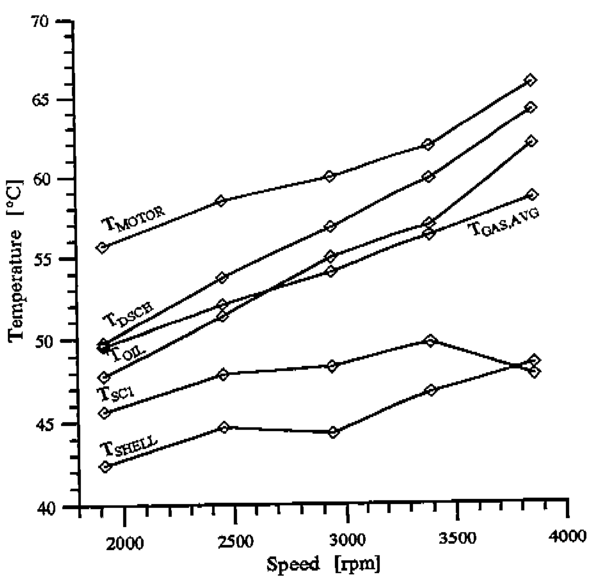
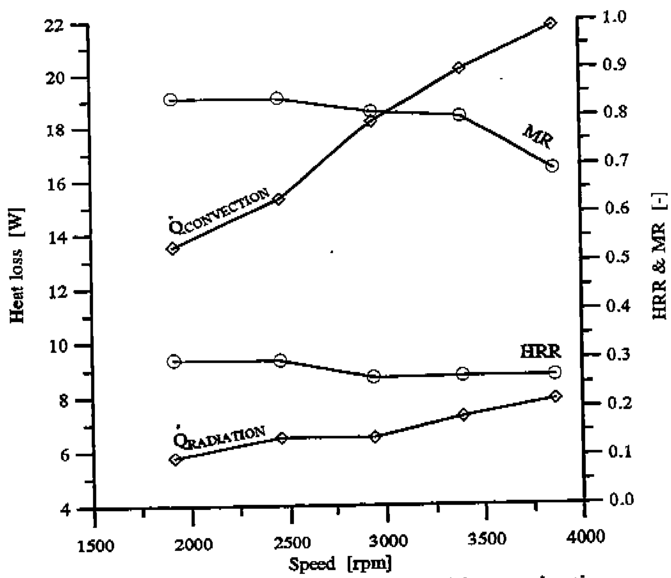


Figure 4: Internal temperatures



Figur 5: Mixing ratio and shell heat rejection