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An Investigation on Reciprocating and Rotary Refrigeration Compressors

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ABSTRACT

Besides reciprocating compressors which are the traditional working machines in refrigeration cycles a growing number of different types of rotary compressors is used especially for air conditioning purposes.

In an experimental investigation various reciprocating and rotary compressors were analysed concerning their thermodynamic and mechanical losses. The compressors investigated are all nearly of the same size and designed for automotive air conditioning systems.

They have been chosen for this comparative investigation since nearly all systems of positive displacement compressors are manufactored today for automotive air conditioning purposes.

Besides reciprocating compressor as 2-cylinder-in-line-compressor and two types of swash-plate-compressors also different types of rotary compressors as stationary vane, rotary-vane, Wankelcompressor, screw compressor etc. have been tested experimentally. In this paper the test results will be presented and compared with simplified computer models for the various designs.

INTRODUCTION

Recent developments in the area of refrigerant compressors have been characterized by constant improvements in the reciprocating piston compressor and by a number of newly developed rotary compressors. Since the early 1970s these trends increasingly have been affected by rising energy costs and the resulting growth in the market for heat pumps.

The market for open, small-size compressors for mobile use in automotive air conditioning systems is especially varied. In recent years Japanese compressors manufacturers have brought an increasing number of new compressor types onto the market.

This large number of new compressor concepts and extensive optimization work done on a rotary vane compressor served as the point of departure for a comparative experimental study.

The industry's need for convenient, sufficiently accurate compressor simulation programs for determining performance characteristics in advance led us to develop simple computational programs for reciprocating piston, rotary vane, stationary vane, and Wankel compressors based on the analysis of efficiency losses determined from actual measurements.

These simulation programs are based on simple mathematical expressions for thermodynamic and mechanical processes involving a few easily measured characteristic values.

DESCRIPTION OF THE COMPRESSORS WHICH WERE STUDIED

Eight compressors of varying designs were compared in a test utilizing idential operating conditions and variations in compressor speed. Since it was necessary to measure the effective power requirements of the compressors, only open compressors were used.

The compressors which were selected are refrigerant-type compressors used in automotive air conditioning systems. They are designed for relatively high-speed operation.

Because of design considerations, reciprocating piston types have larger displacements, while rotary piston type compressors generally have smaller displacments. The following compressore were used:

- a 2-cylinder reciprocating piston compressor (in-line design, displacement: 168 cm^3)
- a 6-cylinder swash plate compressor (3 double pistons, displacement: 166 cm^3)
- a 5-cylinder swash plate compressor (5 piston, displacement: 136 cm^3)
- a rotary vane compressor (double flow design, 4 vanes, displacement: 140 cm^3)
- a Wankel compressor (Si 2:3, displacement: 151 cm^3)
- a stationary vane compressor (displacement: 80 cm^3)
- a screw-type compressor (displacement: 109 cm^3, 2 rotors, no valves)
- a Scroll compressor (displacement: 70 cm^3)

INDICATION OF COMPRESSORS AND EVALUATIONS OF P,V-DIAGRAMS

The compressors were bench tested using a secondary flow calorimeter (see also /1/).

In making the measurements for plotting pressure-to-volume relationships, it is necessary to correlate the pressure signal with the crankshaft angle. This relationship has a great effect on the performance curves and resulting variables and was therefore determined with particular care.

In the reciprocating piston compressors the proximity of the piston to top dead center was measured using an inductive position sensor mounted in the valve plate, while with rotational piston compressors the piston position was measured by determining the proximity of suitable rotating components (rotor, shaft) or oscillating components (vane on the stationary vane compressor) to inductive sensors. An additional method was always used to verify the angular position of the crankshaft (photoelectric signal).

Absolute pressure sensors were used to measure internal pressure. In reciprocating piston compressors these sensors were installed in the valve plate in such a way that the plate was able to close flush with the cylinder chamber and that no additional clearance space was created.

The main problem generally encountered in evaluating rotational piston compressors is that the compression chamber rotates along with the angular position of the shaft.

Since the goal was to record pressure over as much of the shaft's angular position as possible, several pressure sensors had to be used on the rotational piston compressors.

Although Wankel and stationary vane compressors could only be evaluated from the "outside" - since the piston and the drive shaft are not solidly connected to one another - the rotary vane compressor was evaluated using two sensors rotating in the compressor rotor /1/.

The test data obtained in the evaluation work were transmitted via a multiprogrammer (HP 6940 B) to a small computer (HP 9845).

COMPARISON OF COMPRESSORS

The test data in the graphs are plotted as a function of speed for a constant pressure ratio. Only relative values were utilized, since the different displacement sizes have to be taken into account in comparing the absolute values.

Refrigerating output is directly proportional to refrigerant mass flow, while in turn is dependent on volumetric efficiency and on the displacement over time of the various compressors. This means that one can compensate for poor volumetric efficiency by increasing volumetric displacement. It is for this reason that the overall volumetric efficiency (see Figure 1) is the proper parameter to use in comparing the refrigerating output of the various compressors. In general, volumetric efficiency drops as speed increases due to increasing throttling and reexpansion losses. The low efficiency of stationary vane and screw-type compressors is noticeable at lower speeds. Internal leakage is the primary factor here. As speeds increases (peripohernal speed), the absolut leakage losses remain constant per unit time; on the other hand, the throughput per unit time increases, which eventually leads to achieving a maximum efficiency level. In the screw compressor the lack of valves and reexpansion also is reflected in comparatively high efficiency.

These problems do not occur to such a degree in rotary vane compressor. Good seal at low speeds is adequately assured by the action of centrifugal force, which forces the vanes against the walls of the chamber.
An indicator-diagram can be used to break efficiency down into clearance volumetric efficiency (see Figure 2) and thermometric efficiency (see Figure 3).

With clearance volumetric efficiency the losses are attributable to reexpansion and throttling losses in the suction valve. Stationary vane and rotary vane compressors have the highest values, since they lack suction valves and, thus, have a minimum amount of cylinder clearance. With reciprocating piston compressors, a clear drop in efficiency can be observed beginning at ca. 3000 rpm - primarily due to throttling losses in the inlet valve cross section. The lower efficiency of the Wankel compressor is due to its relatively large amount of clearance volume - which is not compensated for, since the unit also has no inlet valves.

By definition, thermometric efficiency includes those losses produced by the heating of the refrigerant in the inlet passage and in the cylinder, as well as losses caused by leaks at the valves and pistons.

The mixing of fresh gas with the reexpanding gas remaining in the cylinder must also be included in thermometric efficiency. Here the reciprocating piston types which have piston rings are at an advantage. The 5-cylinder swash plate compressor is particularly low in leakage losses and inlet gas heating.

The Wankel compressor also has sealing elements, but here inlet gas overheating is more severe. In addition, gas already compressed in a special overflow passage flows into the following chamber to improve volumetric efficiency. The charging effect causes an additional increase in temperature, which has an adverse effect on thermal efficiency. Real improvements cannot be achieved here since this gas was already compressed and is now merely being pushed back and forth. In the rotary vane and stationary vane compressors the losses were caused by the mixing of reexpanding gas with fresh gas - as described earlier - and by heating losses. These heating losses are due to the fact that the compressors are surrounded by a compressor housing and thereby subjected to higher temperatures.

The effective power required to drive the compressors is comprised of the frictional forces needed to overcome mechanical losses and of the indicated power, which represents the power transferred from the piston or rotor to the gas.

Furthermore, drive power is directly proportional to mass flow (volumetric), and thus to geometric displacement per unit time. For the purpose of comparison it is therefore better to use the coefficient of performance (COP), which expresses compressor cooling output as a function of drive power.

The coefficient of performance (COP) output is shown in Figure 4. Here the reciprocating piston clearly has the best values. Stationary vane and screw-type compressors exhibit similar performance. The have a maximum at a certain speed and then fall off in a fearly linear fashion at higher speeds. Compared with the rotary vane compressor, the stationary vane compressor exhibits an equally large coefficient of cooling efficiency beginning at approximately 3000 rpm. However, under 3000 rpm the rotary vane compressor is clearly superior because of lower leakage losses. Because of the 6-cylinder swash plate design's high drive power relative to refrigerating efficiency, the compressor has the lowest coefficient of refrigerating capacity.

The effective isentropic efficiency expresses the relationship between the theoretically required power and that actually used (see Figure 5).

In general, the trend is for this value to fall at higher speeds due to increases in the indicated and mechanical losses. The curves for reciprocating piston, stationary vane, and screw-type compressors have a maximum at this evaporation temperature.

The rise in the curves is due to the indicated losses for the compressor at low speeds and gas densities, which is also reflected in the indicated isentropic efficiency factor. Less than optimal valve performance (late opening) at the low gas densities causes the curves to once again fall off at higher speeds.

The indicated output is the output which the compressor delivers to the gas. It is comprised of the non-isentropic compression work, the additional work required for intake and discharge, and the refrigerant mass flow.

Since the volume of compressors also plays a role here, the indicated isentropic efficiency factor can also be used as a comparison value (see Figure 6). This value expresses the ratio of the theoretical to the actual work required for compression, which is determined using the area under the indicator diagram.

Rotary vane and reciprocating piston compressors exhibit a relatively high indicated isentropic efficiency factor with a low speed dependency. The curves reveal a clearly defined maximum. The drop in the curves at lower speeds points to heat losses and valve performance (late opening); and the drop in the curves at higher speeds is due to increasing throttling losses in the valves. The poor indicated isentropic efficiency factor at low speeds is noticeable in the stationary vane compressor, which does not achieve its best values until about 3000 rpm. However, the values for this compressor are far lower than those for reciprocating piston and rotary vane compressors. This is due to internal leakage losses, which relative become lower at greater speeds. A similar low dependency on speed is observed with the Wankel compressor. Because of undesirable charging with already compressed hot gas and because of excessive reexpansion losses, the efficiency factor is at a similarly low level. On the other hand, the efficiency factor curves for the two swash plate compressors fall off more rapidly with increasing speed, an effect which is particularly due to increasing valve losses.

The relatively inlet and discharge valve losses discussed earlier are shown in Figures 7 and 8. These losses increase with speed as a result of increasing throttling losses. As expected, valveless (suction) compressors have the lowest losses, with the exception of the Wankel compressor, in which inlet passage design does not appear to be optimal. The lower inlet valve losses with the reciprocating piston compressor stand out. These losses do not climb sharply until around 3500 1/min.

The discharge valve losses are also lowest with the rotational piston compressors. Although the rotary vane compressor has an optimized discharge valve, the rotary piston and rolling piston compressors have specially designed valves (Omega valves or curved valve plates), which offer advantages here, given their larger cross-sectional areas.

The power required to overcome friction in the compressors includes that required to overcome the friction in bearings, pistons, slides, sealing elements, bearing seals, etc. as well as the power required to drive the oil pump in the 6-cylinder swash plate compressor. Because of increasing friction losses, the power required to overcome the mechanical efficiency (see Figure 9). The 6-cylinder swash plate compressor and the rotary vane compressor have high friction losses. In the rotary vane compressor the main source of friction is that produced by the vanes due to the running speed. With swash plate compressors the main cause of friction is the number of cylinders and the swash-plate mechanism with drive ball and shoe disc. Also the compressor has an oil-pump. The sliding mechanism on the 5-cylinder swash plate compressor, which uses roller bearings, produces significantly better values. The Wankel compressor and the stationary vane compressors also have low friction losses. Relatively low running speeds are present in the stationary vane compressor.

THEORETICAL WORK

The measurements described above were the starting point for the compressor simulation programs described below. Beginning with extensive computational models describing the compressor in all its details, simpler expressions were used to develop simpler and more easy-to-use compressor programs.

In addition, all the less relevant variables were eliminated by comparing the test data. This makes it possible to use simple computational expressions for thermodynamic and mechanical processes in predicting actual compressor performance. These computational models were developed for the four most important compressors, which utilize the principle of positive displacement: namely, the reciprocating piston compressor, the stationary vane compressor, the rotary vane compressor, and the Wankel compressor.

These simple computational models are based on a simulation model by Soedel /2/. Soedel's model operates with an ideal gas and has two second-order differential equations for dynamic valve motion.

In general the fourth-order Runge-Kutta procedure is used to numerically solve differential equations in initial value problems. However, this procedure can also be used to solve first-order differential equations. Since two differential equations are already present - one for the mass in the control volume and one for the mass forced out through the discharge valve - the required computing time is primarily needed for solving the differential equations.

Based on the need to model real-world conditions as accurately as possible - in spite of simplification -

we continued to consider the valve as a spring system, however without the speed and acceleration terms. Thus, the motion equation is only dependent on the engaged valve force F_v and the spring rate c_v, which is easily measured:

$$h_v = \frac{F_v}{c_v} \qquad (1)$$

The equation offers the advantage of reproducing the opening and closing operations similar to the dynamic motion equation, of describing the valve opening proportional to the engaged valve force, and of thereby taking the effective area of flow into account.

SUPPORTING THE COMPUTATIONAL EXPRESSIONS WITH TEST DATA

As already noted, computational models of this type are of a semi-empirical nature. By using free variables in computation - variables which can be adjusted to mach test data - a good correlation between computations and measurements can be achieved.

Because of the thermodynamic incompleteness of the model (such inadequacies are all expressed by the polytropic exponent n), some differences between computed and observed values are present, especially for indicated output and refrigerant mass flow. However, since it is not possible to quantify these losses, the program was modified to permit the computation to be corrected with the aid of experimental values.

Particularly with the rotary compressors one finds that the internal losses are not completely reflected by the model. The reasons for this area:

- evaporation of refrigerant from the oil which is drawn in
- heat transfer between the refrigerant drawn in and a cylinder wall
- heat transfer between the refrigerant drawn in and the hot oil which gains access to the working chamber through the sealing and side gaps
- evaporation of refrigerant from the hot refrigerant which is almost at compressor pressure
- leakage of refrigerant through the compression chamber seal gap into the inlet chamber
- leakage from the working chamber located ahead of the inlet chamber; the refrigerant is already compressed in this working chamber.

On the other hand, the internal output can be calculated better for the reciprocating piston compressor. Here the program is able to completely account for reexpansion. Assuming that a maximum deviation in indicated output of approximately 10% is permissible relative to the experimental value, no correction would be needed.

The correction is performed in such a way that the computational program is based on a few easily measured parameters. Here one could use the temperatures and pressures on the inlet and outlet side of the compressor, as well as the refrigerating output and compressor speed. If the compressor displacement is taken into account, then the volumetric efficiency can be determined as a function of pressure ratio and speed.

Here it is quite sufficient to construct the program on only four observed data points on the compressor performance curve. If these basic points are selected such that they reproduce the compressor's volumetric characteristics, the program can be corrected over the entire range of computations by performing linear interpolations on the efficiency. With the relationship of measured and calculated volumetric efficiency the inner performance can be corrected.

The enthalpies, entropies, and specific volumes required to compute the refrigerant-specific values, such as volumetric efficiency, refrigeration output, etc. are not calculated here with an ideal gas, but rather with refrigerant equations. The elaborate Rombusch and Giesen state equations were replaced by approximation equations. These quations require very little storage capacity and - compared with the original program - they are reproducable on an average to better than 0,5% /3/.

One significant advantage of the polynomial representation is that one can explicitly calculate temperature as a function of enthalpy, respectively entropy and pressure.

MECHANICAL LOSSES

For an adequate theoretical description of the entire compressor, mechanical losses must be taken into account in addition to thermodynamic losses.

Assuming that the friction surfaces are always adequately coated with oil, an expressions for frictional force based on Newton's shear force hypothesis was selected. The friction force F_r can be calculated to be:

$$dF_r = \eta \cdot \frac{u}{h} \cdot dA \qquad (2)$$

In this equation 'dA' is the oil-covered enveloping surface area and 'u' is the speed of the sliding elements relative to one another; 'h' is the thickness of the lubricant film with complete oil coating, and 'η' is the viscosity of the lubricant, here the oil-refrigerant mixture. In this case, viscosity is not only dependent on temperature but also on the refrigerant content of the oil. Based on actual operation conditions involving friction which previously were not taken into account and which related to the refrigerant-oil content and temperatures, the viscosity was varied in such a way that the power required to overcome friction was calculated in the most accurate way possible.

In this way, for example, the friction due to pistons, the piston ring on the reciprocating compressor, and the sealing strips on the Wankel compressor was calculated. For the rotary vane compressor an expression by Platts /4/ was used,and for the stationary vane compressor one by Pandeya and Soedel /5/.

COMPARISON OF SIMULATION CALULATIONS WITH TEST RESULTS

In order to further explain the results which can simplified computational model provides, we shall now compare the test results with the computer results.

For the reciprocating piston and rotary vane compressor it was possible to compare program versions with and without dynamic valve performance. For both versions of the program the optinal parameters describing the valve (damping constant, natural frequencies, spring rate, etc.) were measured for the actual valve or calculated from actual data. The variables characterizing the internal performance of the compressor, the indicated power and the indicated pressure p_i, are compared in Table 1 for the reciprocating piston compressor and for the rotary vane compressor.

The values calculated in the reciprocating piston model were not corrected with the specified volumetric efficiency. It was found that if the free variables are selected carefully, the test data can be duplicated with sufficient accuracy.

A comparison of the programs with and without dynamic spring motion reveals that for determining losses simple valve motion equations can reproduce the internal losses just as well as dynamic oscillation equations, which require much more computing time. Using these simplifications, the programs can be performed on a small computer (HP 9836) with acceptable computing times.

Computations involving dynamic valve motion and simple valve motion also do not lead to large differences for the rotary vane model.

The difference between the calculations performed with the simplified model and the actual test data were within ± 4% based on the test data. Figures 10 and 11 offer a comparison of the simulation calculations (without valve dynamics) and experimental data. The experimental data are indicated here by symbols (circles, boxes). The solid lines indicate the computed values.

With all the compressors there was excellent correlation between the computed and observed data: nearly always with a difference of less than ± 4%. Only with frictional power is this accuracy not achieved at some speeds due to the simple mathematical expressions used. However, because the power required to overcome friction is relatively low in comparison with the power required to drive the compressor, these greater differences did not have a large impact on the final outcome.

By use of the above mentioned model, p-V diagrams were calculated for reciprocating compressors working with refrigerant mixtures. The thermodynamic and caloric data in this special case were obtained by use of the Redlich-Kwong-Soave equation of state.

By calculations carried out for the mixture R 12/ R 114 was shown that an opimum of the indicated isentropic efficiency can be reached. This optimum depends on the pressure ratio and on the concentration of the lower boiling component and was found in the range of 60% to 80% R 12 (see Figure 12). +

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LEGEND

○ Reciprocating Compressor
✳ Rotary Vane Compressor
⋕ Swash Plate Compressor (6-Cyl.)
+ Swash Plate Compressor (5-Cyl.)
▫ Wankel Compressor
ⓔ Screw Compressor
× Stationary Vane Compressor
△ Scroll Compressor

Refrigerant R 12
Pressure Ratio $P_{V2} / P_{V1} = 13.5 / 1.82$ (bar / bar)
Suction Gas Temperature = $T_0 + 10$ K

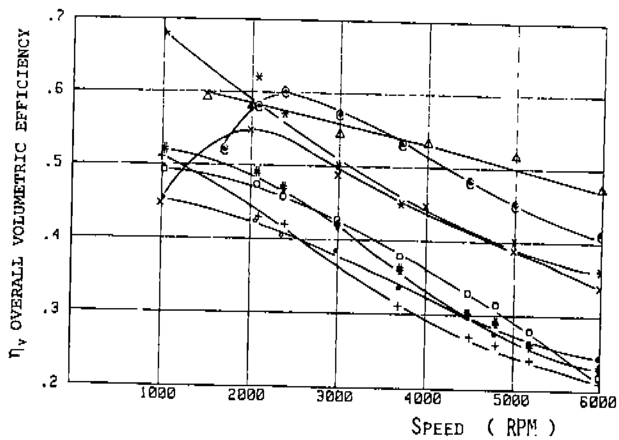


Figure 1

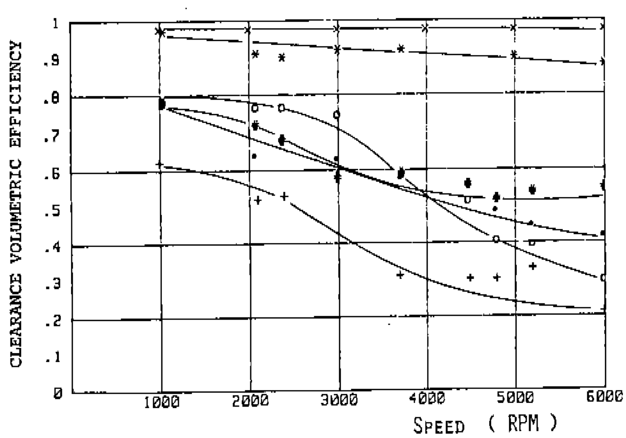


Figure 2

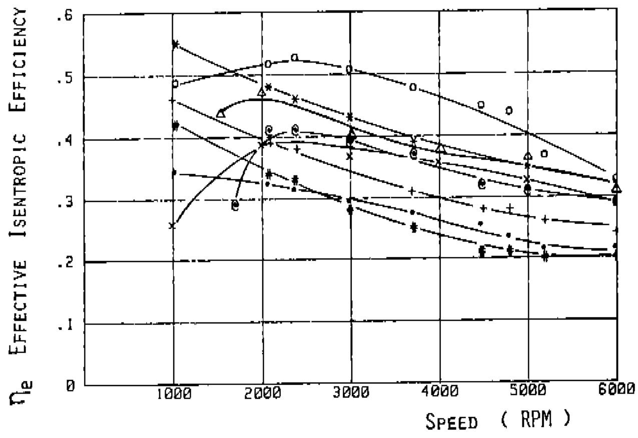
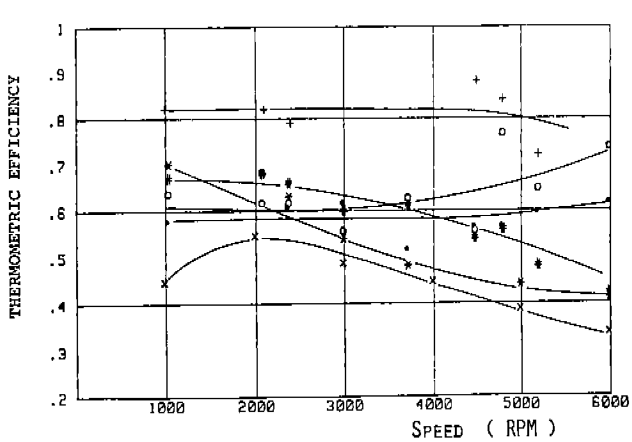


Figure 5



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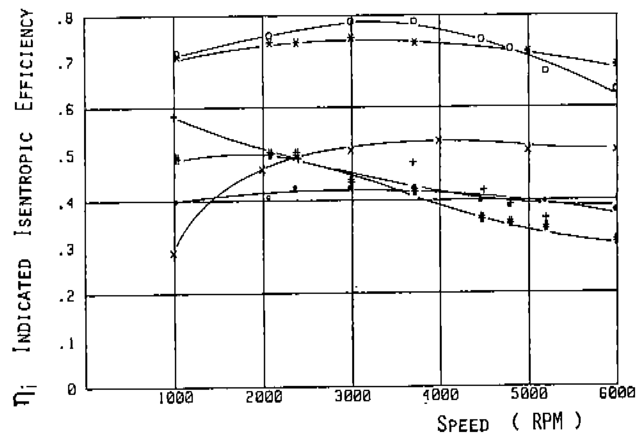

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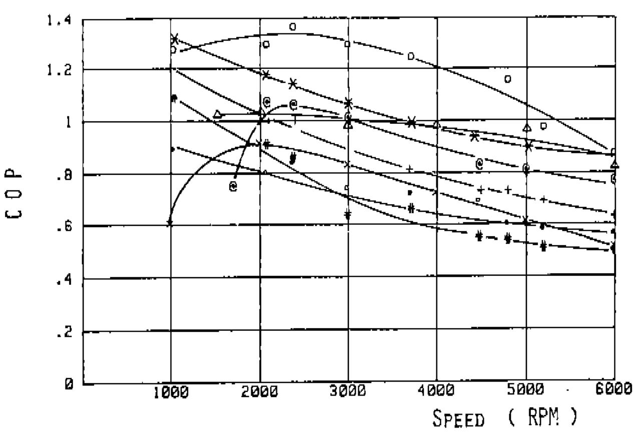


Figure 4

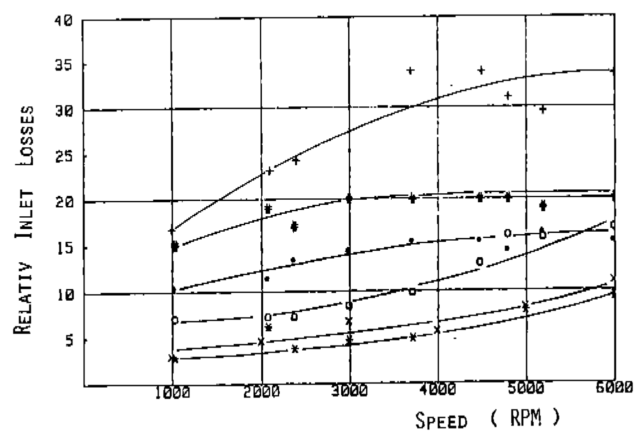


Figure 7

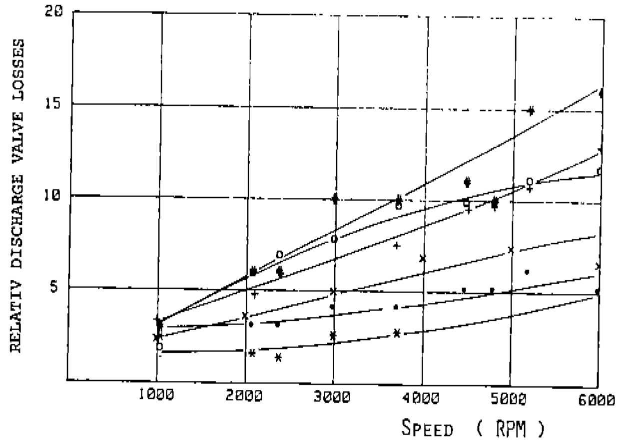


Figure 8

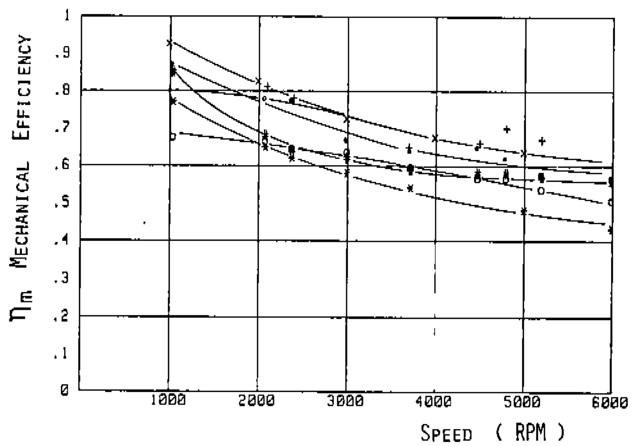
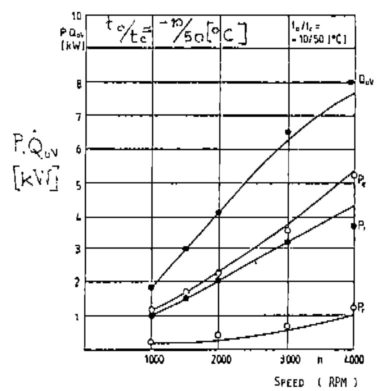


Figure 9



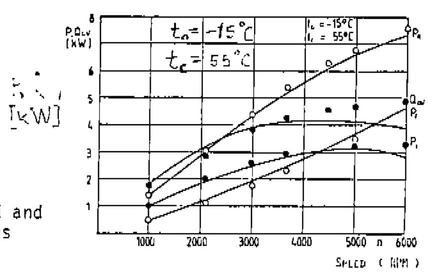


Figure 11

Comparison of calculated and measured (symbols) values

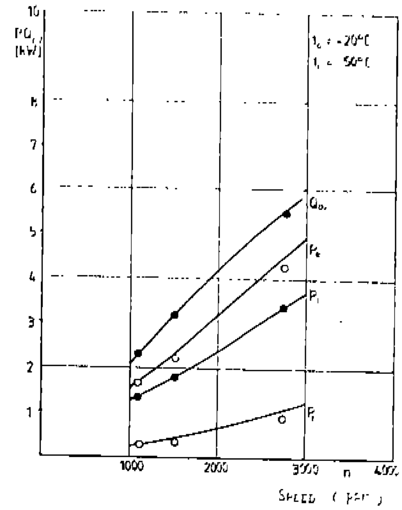


Figure 10

Comparison of calculated and measured (symbols) values of the reciprocating compressor

Mean Effective Pressure p_i (bar)

(Deviation of the Measured)

t_o/t_c °C/°C	speed rpm	measured value	with valve dynamic	without valve dynamic
Reciprocating Compressor				
-10/40	1100	3,6	3,75 (4,2%)	3,6 (<1%)
	1500	3,7	3,9 (5,4%)	3,7 (<1%)
	2000	3,8	4,02 (5,9%)	3,8 (<1%)
	3000	3,8	4,05 (6,4%)	4,1 (7,9%)
Rotary Vane Compressor				
-15/55	1000	4,25	4,27 (<1%)	4,03 (5,2%)
	3000	3,61	3,74 (3,6%)	3,48 (3,6%)
	5000	2,71	2,86 (5,5%)	2,64 (2,6%)

Table 1. Comparison of Calculated Values (With and Without Valve Dynamic) and Measured Values

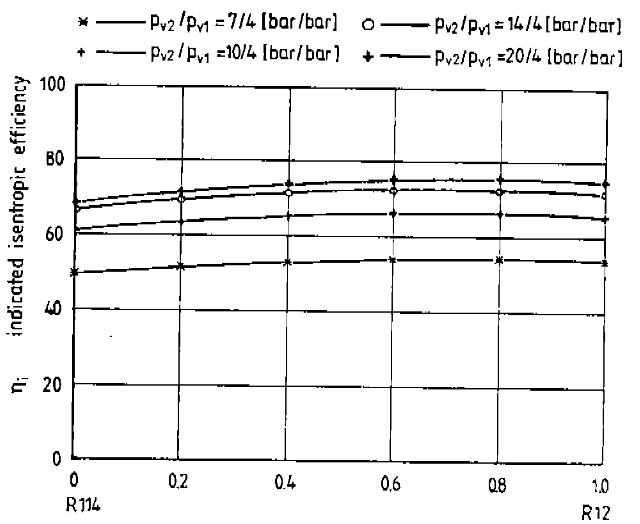


Figure 12