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Variable Automotive CO2 Compressor

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ABSTRACT

The inclination angle of the wobble plate of a variable automotive CO_2 air-conditioning compressor is initially calculated using a simple static model and then multi body dynamics. Finally, optimization of the swept volume mechanism is performed using the static model and multi body dynamics.

INTRODUCTION

The background of the development of a completely new automotive air-conditioning system for refrigerant CO_2

The effect of chemical refrigerant on the atmosphere is a subject of current discussion. If a chemical refrigerant is replaced by the natural refrigerant, CO_2, heating of the atmosphere caused by leakage from direct-drive compressors is reduced. Based on this knowledge, the RACE project (*Refrigeration and Automotive Climate under Environmental Aspects, Contract No. BRE2-CT94-0555*) was started under the auspices of the EU on 1 June 1994. During the three years this project lasted, a transcritical CO_2 air-conditioning system was developed as a possible substitute for current R134a systems. Two cars were equipped with the new system and underwent wind tunnel and road tests. These experiments show that a system based on CO_2 as the refrigerant complies with general standards as far as cooling capacity, efficiency, and comfort are concerned. The transcritical CO_2 system consists of a gas cooler, an evaporator, an internal heat exchanger, a receiver, an expansion valve, and a compressor - the heart of the system and the subject of this paper.

The reason why single-acting axial piston compressors are used for CO_2

Prior to the design process a survey was carried out to establish which compression principles are most suitable for use with V-belt drive in connection with CO_2. Turbo compressors were excluded owing to the high density of CO_2 and because of the assumed excessive internal leakage. In the whole range of positive displacement compressors, we found that only reciprocating units would be able to keep the internal leakage rate at an acceptable low level. Scrolls, screws and rotary compressors must from a first-hand view be considered as unsuitable for operating with transcritical CO_2. If the possibility of being able to control refrigeration power at constant V-belt speed is desirable, then an open-type single-acting axial piston compressor would appear to be the most natural choice.

Why the same capacity control mechanism for CO_2 as for high-end R134a compressors?

The prevailing opinion among the RACE partners was that the CO_2 compressor should be of the variable type. The reasons for this point of view were partly that variable compressors give greater passenger comfort, partly that CO_2 systems reach saturation of heat exchanger performance at higher motor speeds than state-of-the-art R134a systems - resulting in unacceptably high power consumption in the compressor. Danfoss delegated the task of finding the most energy-efficient means of controlling the cooling capacity of a single-acting axial CO_2 compressor to FKW (*Forschungs-Zentrum für Kältetechnik und Wärmepumpen GmbH*) in Hannover. In order to be able to meet the tight schedules drawn up for the project, the areas of investigation were limited to the following six possibilities:

1. Suction valve delay
2. Opening of compression chamber to the suction side at the beginning of the stroke by moving a slide.
3. Switching a clearance pocket on and off at a given cylinder pressure threshold value.
4. Keeping the stroke constant and varying the clearance volume by moving the entire cylinder block.

5. Varying the stroke and clearance volume simply by changing the wobble plate angle.
6. Varying the stroke - leaving the clearance volume constant.

On the basis of purely theoretical considerations which included computer calculations of the selected criteria - indicated efficiency together with friction and leakage loss - FKW concluded:

- Options 1 and 2, which both control by delaying compression start, are reasonably good alternatives. However, both solutions appear to be expensive to put into practice for the actual application.
- Options 3, 4, and 5, which use re-expansion of gas from a clearance volume, are excellent with regard to indicated efficiency, but the very high mean pressure connected with the process leads to large leak rates and substantial mechanical friction loss.
- Of all the solutions, option 6, today's high-end design within R134a automotive A/C, is the one that will most satisfactorily be able to meet what will inevitably be demanded of CO_2 automotive air-conditioning compressors of the future.

COMPRESSOR CONCEPT

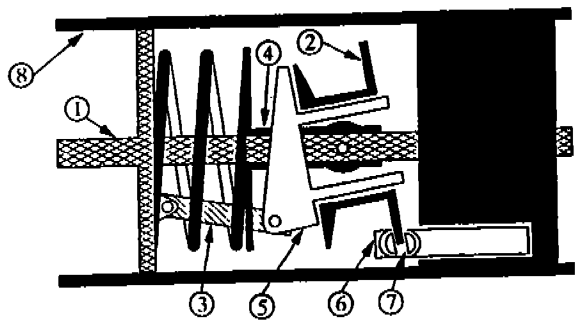

Figure 1. 2D picture of the compressor.

The wobble plate compressor concept shown in fig. 1 was chosen for the RACE project:

1. *Shaft*	2. *Wobble plate*	3. *Fish joint*	4. *Spring carrier*
5. *Slant plate*	6. *One of 7 pistons*	7. *Spherical joint*	8. *Housing*

NOMENCLATURE

Variables and parameters:

D	Cylinder bore
d_{pre}	Pre-tension of the spring expressed as displacement
k	Spring stiffness
l	Distance between the two fish joints
m	Piston mass
n	Number of pistons (*here 7*)
P_{crank}	Pressure in crankcase *(Dependent variable)*
P_{bot}	Pressure in suction plenum chamber
P_{top}	Pressure in discharge plenum chamber
R	Distance from shaft centreline to piston centreline
x_R	Distance from shaft centreline to left-hand fish joint
z_R	Distance from shaft bottom plate to left-hand fish joint (*given as a negative quantity in the formulas*)
x_S	Distance from right-hand fish joint to the revolute joint in the spring carrier projected to the wobble plate
z_S	Distance from right-hand fish joint to the mid-plane of the wobble plate
α	Actual inclination angle of the wobble plate
α_{max}	Maximum inclination angle of the wobble plate

$\alpha_{\min}$ Minimum inclination angle of the wobble plate
γ Ratio of specific heats
ε Relative clearance volume at maximum stroke
θ Angle of rotation of the shaft (*Independent variable*)

STATIC MODEL

If the shaft speed of rotation is constant $\ddot{\theta} = 0$ and the revolute joint of the spring carrier lies in the mid-plane of the wobble plate, the following applies:

Distance d from shaft bottom plate to wobble-plate intersection point with the shaft is

$$d = -\Psi + \sqrt{l^2 - \Phi^2}\text{, where } \Phi = x_S\cos\alpha - z_S\sin\alpha - x_R \text{ and } \Psi = x_S\sin\alpha + z_S\cos\alpha - z_R$$

Substituting $\alpha_{\max}$ in these expressions gives $d_{\max}$ (*correspondingly for* $d_{\min}$), and extreme positions z_{tdc} and z_{bdc} for all seven pistons can now be determined.

$$z_{tdc} = R\left(\tan\alpha_{\max} - \tan\alpha\right) - \Delta d \text{ and } z_{bdc} = R\left(\tan\alpha_{\max} + \tan\alpha\right) - \Delta d\text{, where } \Delta d = d - d_{\max}$$

We convert the clearance volume at maximum stroke to an equivalent piston travel z_c

$$z_c = \varepsilon\, s_{\max}\text{, where } s_{\max} = 2R\tan\alpha_{\max}$$

set up the equation of motion for piston no. i as a function of θ

$$\beta_i = \theta + 2\pi(i-1)/n$$

$$z_i = R\left(\tan\alpha_{\max} - \cos\beta_i\tan\alpha\right) - \Delta d$$

$$\ddot{z}_i = R\omega^2\tan\alpha\cos\beta_i\text{, where } \omega = \dot{\theta}$$

find the pressure over piston i, either if $\beta_i \le \pi$ (*isentropic ideal gas expansion*)

$$P_i = P_{top}\left(\frac{z_{tdc} + z_c}{z_i + z_c}\right)^{\gamma}, \quad P_i \le P_{bot} \Rightarrow P_i = P_{bot} \quad \textit{(logic suction valve)}$$

or if $\beta_i \ge \pi$ (*isentropic ideal gas compression*)

$$P_i = P_{bot}\left(\frac{z_{bdc} + z_c}{z_i + z_c}\right)^{\gamma}, \quad P_i \ge P_{top} \Rightarrow P_i = P_{top} \quad \textit{(logic discharge valve)}$$

and can thus determine the sum of all the piston forces

$$F_{sum} = \sum_{i=1}^{n}\left[A\left(P_i - P_{crank}\right) - m\ddot{z}_i\right]\text{, where } A = \frac{\pi}{4}D^2$$

Note! The d'Alambert forces from piston masses are included in the calculation. If the sum of all the forces applied to the wobble plate are now set at zero for all θ (*condition for equilibrium*) the truss force from the fish joint is

$$L = l\frac{F_{sum} - S}{\Psi + d}\text{, where the spring force } S = k\left(d_{\min} + d_{pre} - d\right)$$

The torque relative to the revolute joint in the spring carrier - of the truss force and all the n piston forces - is thus

$$M = \frac{L}{l}\left[\Phi\left(z_R - d\right) - x_R(\Psi + d)\right] + R\sec^2\alpha\sum_{i=1}^{n}\left[A\left(P_i - P_{crank}\right) - m\ddot{z}_i\right]\cos\beta_i$$

The pressure in the crank case P_{crank} is finally adjusted numerically to suit a pre-selected value for α so that torque M integrated over the revolution angle interval $0 \le \theta \le 2\pi/n$ gives zero (*static equilibrium in mean of one revolution*), thus resolving the system of equations for the required inclination angle of the wobble plate α.

USE OF THE STATIC MODEL

General observations

Parameter studies with the static model show:

1. The higher the mean pressure in the cylinder chambers, the greater the inclination angle of the wobble plate.
2. The greater the spring stiffness (*the greater the pre-tension*), the smaller the inclination angle of the wobble plate.
3. The higher the pressure in the crankcase, the smaller the inclination angle of the wobble plate. In practice, the stroke volume of the compressor is controlled by changing the crank case pressure.
4. The greater the piston masses or the higher the speed of the compressor, the greater the inclination angle of the wobble plate and thus the stroke volume. This is a malfunction of the compressor which can only be compensated for by increasing the crankcase pressure (*There is not sufficient space to build a counterweight inside the compressor housing*).

Sizing a CO_2 compressor

If a designer not experienced in automotive air-conditioning uses the static model for a few days to size a variable CO_2 compressor, the following parameters might appear:

$R = 30$ mm, $D = 16$ mm, $m = 64.4$ g, $\varepsilon = 0.07$, $\gamma = 1.31$, $n = 7$,

$x_R = 20$ mm, $z_R = 7$ mm, $x_S = 22.9$ mm, $z_S = -27$ mm, $l = 24$ mm, $k = 200$ kN/m,

$\alpha_{max} = 18°$, $\alpha_{min} = 4°$, $d_{pre} = 0$.

If these figures are used, the characteristics below might result:

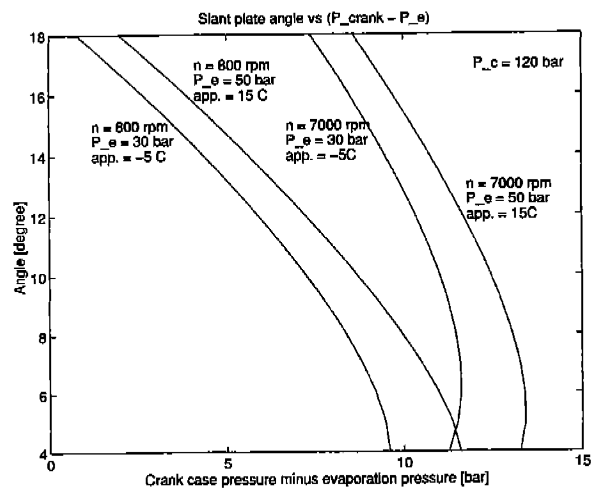

Figure 2

The figure deals with two suction pressures "P_e" at 30 bar and 50 bar respectively plus two different speeds of rotation "n" of 800 rpm and 7000 rpm: This gives a total of four characteristics. The discharge pressure "P_c" is 120 bar overall. It can be seen that the compressor gains a steeper characteristic when

1. the speed increases
2. the inclination angle of the wobble plate falls

and that it becomes impossible to control when "1" and "2" occur simultaneously (*The program is able to show that a characteristic must have a negative derivative to be stable*). If the static model is realistic, the calculated compressor will be unable to operate at high speed and small stroke volumes at the same time. Therefore if this operating range were to be insisted upon, it would be necessary to consider operation in cycling clutch mode to meet the refrigeration power demand.

MULTI BODY DYNAMIC MODEL

There are two conditions for the equilibrium of a rigid body: The law that governs the motion of the centre of mass (Newton's second law) and Euler's equations, which describe the rate of change of the instantaneous axis of rotation relative to the body-centered reference system of the principal axes. All the rigid bodies which are part of this mechanism - the mechanism that controls the swept volume of the compressor, with their respective masses and moments of inertia - are included in the multi body dynamic model.

Comparison with the static model at low speed of rotation

Using the condition $P_{top} = 120$ bar, $P_{crank} = 48$ bar, $P_{bot} = 40$ bar, and $\dot{\theta} / 2\pi = 800$ rpm together with the previously named parameter set, provides a simulation of the inclination angle α for the revolutions shown below:

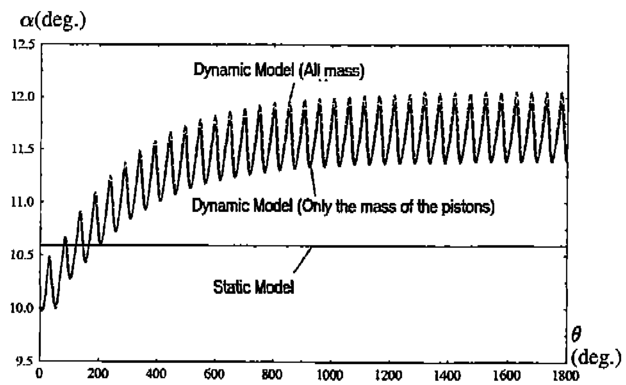

Figure 3

In addition to the seven oscillations per revolution there is significant variation between the result from the static model and the mean equilibrium angle given by the dynamic model. The main reason is that the given piston-excited rigid body oscillations of the wobble plate interfere the sinusoidal motion of the pistons and thereby the pressure build up in the cylinder chambers. The validity of this explanation is shown in the following two ways

1. Substituting the cylinder pressure cycle from the dynamic model in the static model gives an inclination angle around the mean value of the oscillation of the dynamic model.
2. With the dynamic model, adding substantial damping in the revolute joint of the spring carrier gives almost the same result as with the static model (*No rigid vibrations of the wobble plate occur*).

As can be seen in the figure, including the mass from the remaining mechanism (*in addition to the mass of the pistons*) has no significant influence at 800 rpm.

Comparison with the static model at high speed of rotations

All Parameters are unchanged except $\dot{\theta} / 2\pi = 7000$ rpm. In addition to the rigid body oscillations of the wobble plate now becoming incredibly small, $\alpha_{dynamic} = 21.3°$ as against $\alpha_{static} = 16.1°$.

1. Damping is introduced again in the dynamic model, but this time without result: $\alpha_{dynamic,damping} \approx 21.3°$.
   (*No wobble plate oscillations* $\Rightarrow$ *No deviation of pressure in the cylinder chambers*)
2. The retention of the mass of the pistons only, gives $\alpha_{dynamic,piston\_mass} \approx 16.1°$.

It is thus necessary to perform the dynamic simulation with the complete mechanism mass at 7000 rpm!

OPTIMIZATION WITH RESPECT TO PERFORMANCE

As previously mentioned, the characteristics found manually with the static model are not satisfactory. Therefore the first task is the optimization of the static model with analytical sensitivities. Optimization changes the following parameters:

$R = 30.089$ mm, $m = 50.0$ g, $x_R = 15.987$ mm,

$x_S = 28.976$ mm, $z_S = -13.296$ mm, $l = 31.329$ mm,

The improvement of the characteristics is obvious when a comparison is made with the previous figure 2:

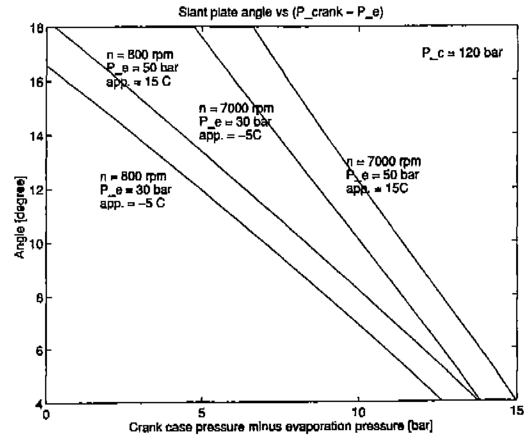

Figure 4

Finally, optimization is performed with multi body dynamics with numerical sensitivities. The data set changes accordingly:

$R = 31.148$ mm, $m = 50.0$ g, $x_R = 12.847$ mm,

$x_S = 25.159$ mm, $z_S = -18.262$ mm, $l = 24.274$ mm,

The optimized characteristics are shown below together with those originally dimensioned manually in the condition: $P_{top} = 120$ bar, $P_{bot} = 40$ bar.

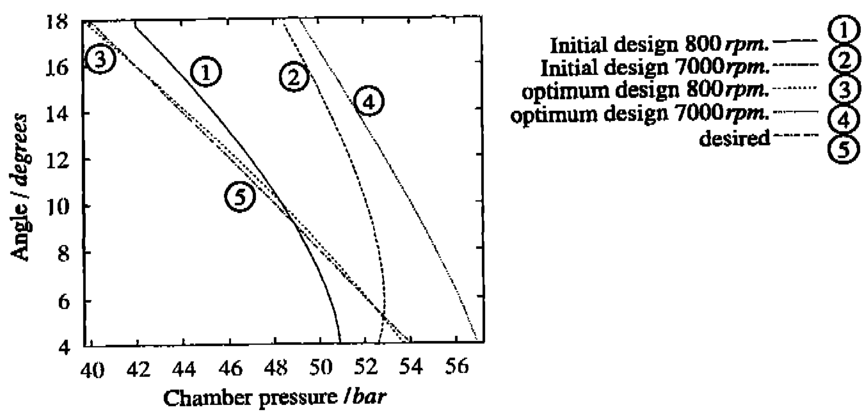

Figure 5

CONCLUSIONS

The simple static model is well evaluated, qualitatively, but quantitatively it is nowhere near as good as the dynamic model. If clutchless automotive CO_2 air-conditioning compressors are to be produced they should - to be able to operate with small inclination angles at high speeds - be optimized by means of multi body dynamics.

ACKNOWLEDGMENTS

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