Performance simulations of twin-screw compressors for refrigeration purpose

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SUMMARY

The aim of this paper is to present and give a brief description of a computer program for simulation of twin-screw compressors for refrigeration purpose.

The effect of evaporation of refrigerant, dissolved in the injected oil, is considered in the computations. A simulation example, with R22 as working medium, shows an increase in adiabatic efficiency of 3 - 13 %, depending on pressure ratio, by using oil without any dissolved refrigerant.

Simulations of compressors with economizer arrangements are also described and exemplified.

Zusammenfassung

In der vorliegenden Arbeit wird ein Rechenprogramm beschrieben, das für die Simulation von dem Verdichtungsverlauf in Kältemittelschraubenverdichter abgesehen ist.

Der Einfluss von Verdampfung des in dem Öl eingemischten Kältemittels ist dabei berücksigtigt. Ein Simulationsbeispiel mit dem Kältemittel R22 zeigt eine Verbesserung des adiabatischen Wirkungsgrades mit 3 - 13 %, abhängig vom Druckverhältnis, bei Öl ohne Kältemitteleinmischung im Vergleich mit eingemischtem Kältemittel in dem Öl.

Die Simulation von Verdichtern mit Economizersystem ist auch behandelt und mit einem Beispiel beschrieben.

1. INTRODUCTION

The purpose of performance simulations of oil-flooded screw compressors is to describe the thermodynamic process inside a screw compressor in mathematical terms. The simulations then give an overall picture of the compression process and are therefore of considerable use. At the same time they give one a possibility to put different losses in quantitative relations to each other. The calculations are very extensive and can, in practice, only be performed with the aid of a computer.

Together with computer programs which describes the rotor geometry, the simulations have successfully been used for performance predictions, optimizations of rotor geometry and analysis of thermodynamic effects.

2. PERFORMANCE SIMULATIONS

2.1 Calculation of geometrical data

One of the fundamental demands on simulation is that it must be possible to model the geometrical design in a mathematical form.

A detailed presentation of a comprehensive program library in this field has earlier been presented in ref (1). The following geometrical parameters are calculated by computer programs and the results are input to the simulation program, see also figure 1.

- Volume curve
- Inlet port area vs. rotation angle
- Outlet port area vs. rotation angle
- Rotor-rotor sealing line length vs. rotation angle
- Male and female lobe tip sealing line length vs. rotation angle
- Blow-hole area

It is important to have programs with as general applicability as possible. All the programs have a set of profile coordinates as input and they do not deal with analytical profile definitions. This means that they can be used for all types of profiles which can presumably be developed in the future as long as coordinates can be generated, and this is, of course, fundamental. An advanced coordinate generation program, with large options in input parameters, is therefore of considerable help for development in this field.

Apart from these geometrical inputs, average clearances and the lengths of leakage paths, for the leakage calculations, are also inputs. The options in geometrical input are so large that all cases which can conceivably appear in practice can be dealt with.

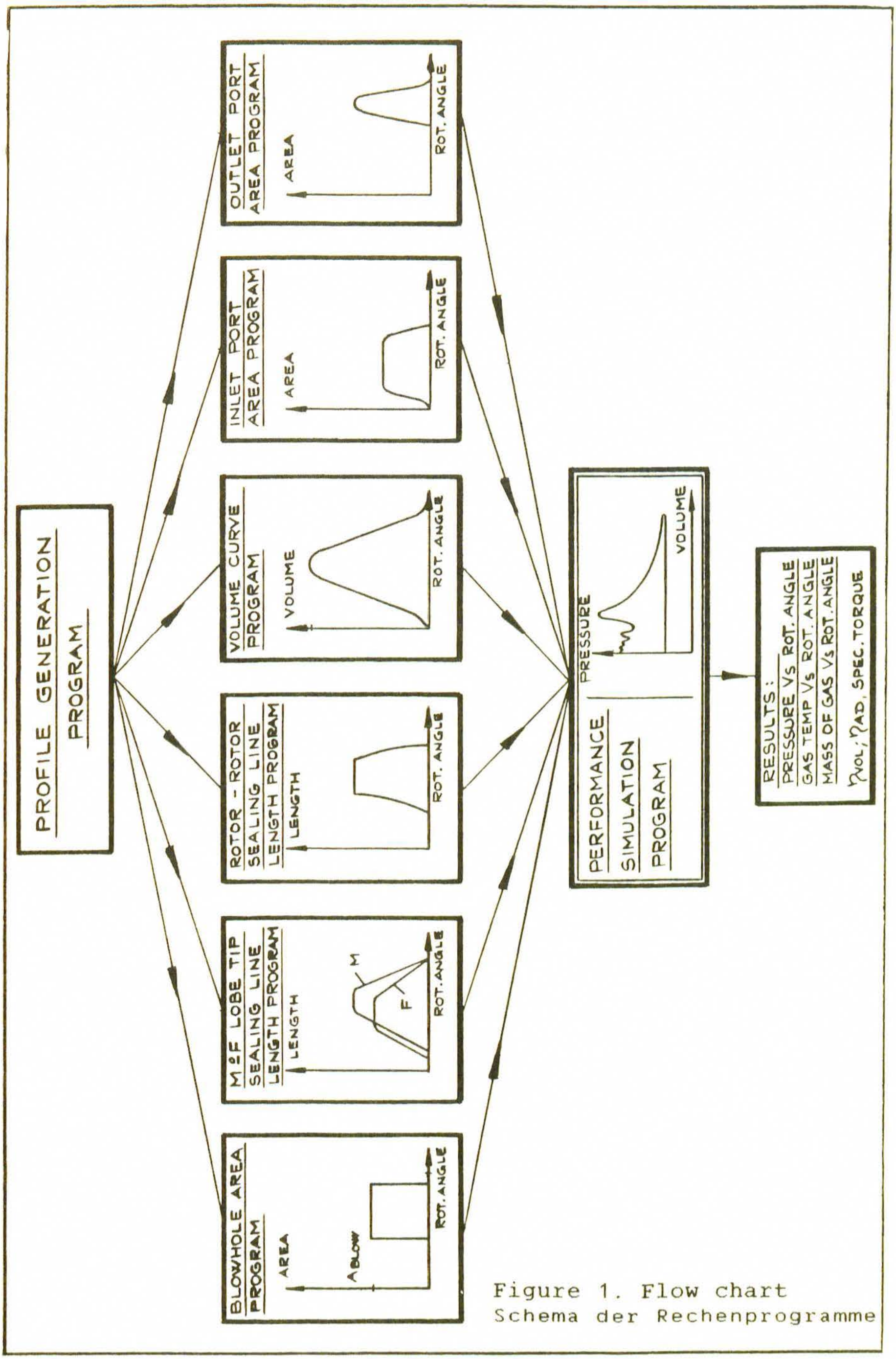

Figure 1. Flow chart
Schema der Rechenprogramme

2.2 **Some general points about the simulation program**

The simulation program itself considers the effects of

- Internal leakage through all types of clearances and through the blow-hole.
- Inlet and outlet port throttling losses.
- Gas pulsations in inlet and outlet ports.
- Viscous losses.
- Heat transfer between gas and oil.

The effects of solubility of refrigerant in oil also have to be considered when simulating compressors with halocarbons (freons) as working medium.

A condition for this type of program is also that one must be able to use a wide variety of operational conditions. The following parameters are used

- Working medium
- Inlet gas temperature and pressure
- Outlet pressure
- Rotor speed
- Oil-injection rate, oil temperature, oil viscosity

Together with the instantaneous values of the mass of gas, gas temperature and pressure, the program also calculates volumetric and adiabatic efficiencies, specific torque and discharge temperature. Figure 2 shows an example of a calculated pressure profile (versus rotation angle) for an oil-flooded air compressor running with a pressure ratio of 8.0. The instantaneous values of the gas torque for this pressure distribution is plotted in figure 3. For a male rotor with four lobes the torque characteristic is repeated at every 90° of male rotor rotation.

2.3 **Theory**

The basic assumptions of the thermodynamic model will be presented in this chapter. The equations for this theory are presented in reference (1). A similar theory has earlier been presented in a paper written by Bo Sangfors, ref (2).

The control volume consists of the cavity formed by the cooperating male and female rotor threads. It is followed through the whole cycle from the suction phase, when communication to the inlet occurs, to the discharge phase, when communication to the outlet occurs. For this control volume the program computes

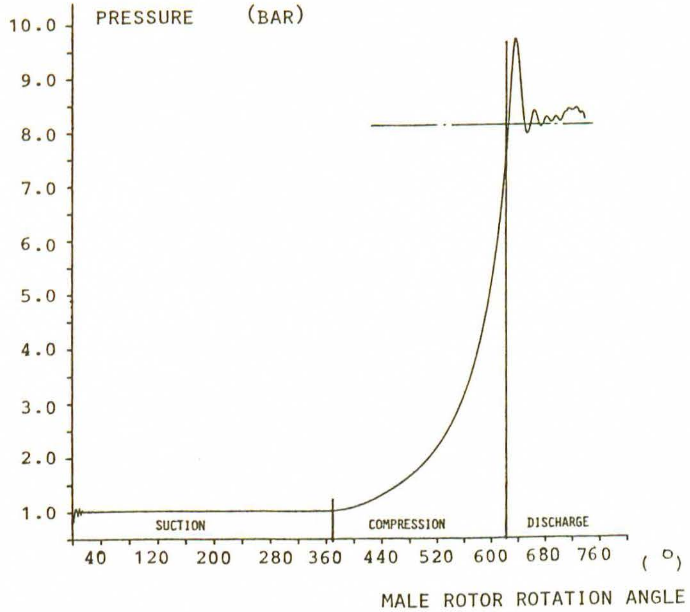

Figure 2. Pressure vs. male rotor rotation angle
Druck als Funktion des Drehwinkels

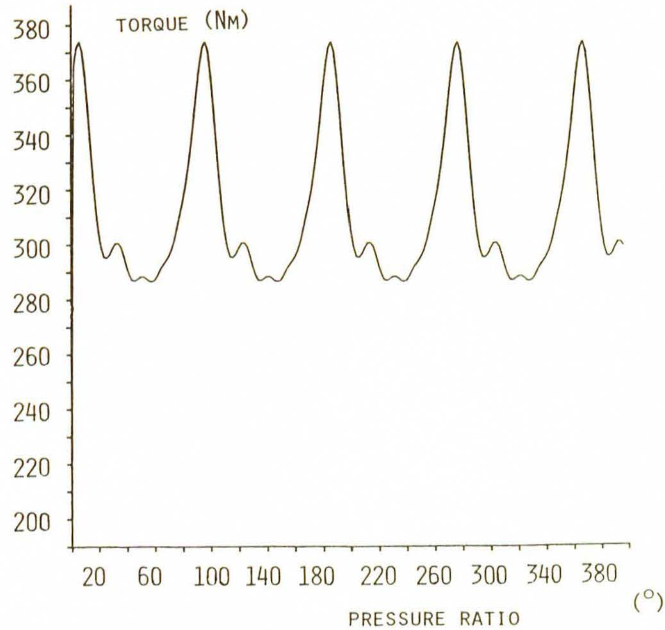

Figure 3. Gas torque vs. male rotor rotation angle
Drehmoment als Funktion des Drehwinkels

pressure, gas temperature, oil temperature and mass of gas and oil in the control volume for a large number of rotation angles by a step-by-step procedure. The equations used for this are heat- and massbalances for gas and oil, equations describing the leakage rate of mass flows as well as the inlet and outlet rate of mass flows and the equation of state for the gas. This gives a system of linear differential equations which is solved in the computer program by use of the Runge-Kutta method from a numerical algorithm library routine.

The leakage to the cavity under consideration (the control volume) comes from cavities at rotation angles further on in the cycle, the thermodynamic gas properties of which have not yet been calculated. This introduces complications, as the leakage rate depends on the conditions of the incoming gas and oil. An iterative procedure is therefore used and an isentropic pressure and temperature process is first assumed for the leakage calculations under the first iteration. Thereafter the new conditions are stored and used in the next iteration. The iteration proceeds until the pressure distribution varies less than a given value. Normally 3-4 iterations are enough for convergence.

The following basic assumptions have been used in the program:

1) The gas is assumed to be a perfect gas.

2) The gas and oil mixture in all leakage paths is homogeneous and the mixture ratios in the paths are the same as the mixture ratio in the cavity from where the leakage comes. Apart from the gas-oil mixture ratio, the theoretical model used for calculations of leakage takes into consideration also the effects of oil viscosity, leakage path length and rotor rotation.

3) It is very difficult to get a full understanding of the heat transfer in the compressor. The following assumptions seem to be appropriate:

a) The model does not include heat transfer to the housing but takes into account heat transfer between gas and oil. Instead of using a polytropic process, a heat transfer term is used in the energy equation. The effect of this heat transfer term is in most applications small.

b) Oil and gas leaking over the rotor tips and through the blow-hole mix in the leakage path and come to the same temperature. Due to the centrifugal forces, it is assumed that the oil that has leaked into the control volume forms a layer against the housing. The heat-transfer between the gas and this oil-layer is neglected.

c) Because of the high pressure difference and rotor rotation, it is assumed that the oil leaking through the mesh and the discharge end plane into the control volume is atomized. This results in fully developed heat transfer between the gas in the control volume and the incoming oil leakage.

d) Most of the heat transfer takes place after the gas and oil mixture has passed the outlet port and for calculation of the discharge temperature fully developed heat transfer is assumed.

4) When modelling the suction and discharge process with regard taken to gas pulsations, the momentum equation is used as described in ref (1). The flow through the outlet port is assumed to take place with constant enthalpy.

5) Viscous shear losses in the compressor come from the narrow gaps with large relative velocities between the two surfaces. The gap between rotor tips and housing has been shown to give the largest contribution to these losses. The shear stresses in the rotor tip - housing gap is assumed to be proportional to $\mu \cdot \frac{u}{h}$, where μ is dynamic viscosity, u is tip speed and h is the average clearance.

3. MODELING OF TWIN-SCREW COMPRESSORS FOR REFRIGERATION PURPOSE

The refrigeration twin-screw compressors differ in design compared with air compressors in the way that most of them have means for capacity control. Means for adjustable built-in volume ratio are also becoming more and more used. A survey of these means is made in reference (3).

It is important for the geometrical modeling of refrigeration compressors to have a computer program which both calculate the area variation of the axial outlet port as well as the area variation of a radial outlet port of "slide valve" type. This type of radial port is shown by the unwrapped view of the twin-screw rotor bores in figure 4.

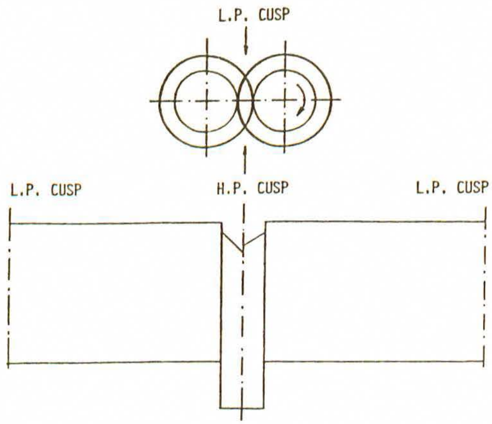

Figure 4. Unwrapped view of rotor bores and a V_i-slide valve.
Ausgebreitete Ansicht der Bohrungen und ein V_i-Steuerungsschieber

From thermodynamic point of view there are two large differences between oil-flooded air compressors and machines compressing refrigerants. One difference is that the gas in refrigeration compressors cannot unquestionable be assumed to be a perfect gas. Today, computer programs for gas properties of commonly used refrigerants are not unusual and a correct way to do the modeling would be to use gas properties from this sort of computer routines. It was found, though, that the equations for a perfect gas are possible to use also for refrigeration compressors if mean values are used for the specific heat and the gas constant for the isentropic process in question. The error from this simplification is small especially if the results are considered in relative units, i.e. volumetric and adiabatic efficiency.

The second difference is the solubility of refrigerant in the oil. When the oil is injected into the compression chamber some of the refrigerant dissolved in the oil evaporates and this will affect the compressor performance in a negative way as a larger mass of gas has to be compressed. The modeling of this effect will be described below in part 3.1.

To improve the capacity as well as the COP in refrigeration plants with twin-screw compressors, economizer arrangements are becoming more and more used. A compressor refrigeration plant with economizer system is accomplished by replacing the regular expansion valve by two valves and an intermediate pressure vessel (flash tank). Refrigerant vapourized after the first valve is injected, via the economizer inlet, to a thread under compression. See figure 5. In the same figure the process is described in a Mollier diagram. Simulations of compressors with vapour injection (economizer) will be described in part 3.2.

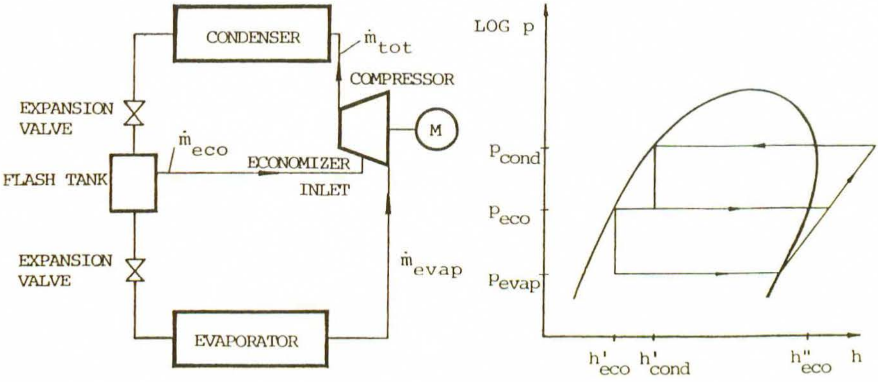

Figure 5. Principle of economizer refrigeration system
Principschema für ein Kälteanlage mit Economizer

3.1 Solubility of refrigerant in oil

As mentioned above, refrigerants are dissolved to different extent in the oil. The essence of all the refrigerant-oil mixture property calculations is the fundamental relationship between the pressure, temperature and the strength of the liquid refrigerant-oil solution. The solution strength is expressed as the fraction ξ of refrigerant in the mixture solution.

Different combinations of oil type and refrigerant will dissolve the refrigerant to different extent. The solubility relationship is therefore of the form

$$\xi = f\ (p,\ T,\ \text{refrigerant},\ \text{oil type})$$

where p and T are the pressure and the temperature. The number of experimental investigations in this field is unfortunately limited why only a few relationships are available. In ref (4) such relationships $\xi(p,T)$ are presented for the combinations of mineral oils and some of the more commonly used refrigerants, e.g. R12, R22 and R114.

The fraction of refrigerant dissolved in the oil injected in the compressor will in practice be dependent on the design of the oil system in the refrigeration plant. To make the simulation program independent of the plant design, it is assumed that the pressure and temperature in the oil separator is equal to the compressor discharge pressure and temperature. The fraction of refrigerant in the mixture will then be $\xi(p_{dis},\ T_{dis})$ when leaving the oil separator.

The time needed for evaporation of refrigerant due to a decrease in pressure is very short and here the process is assumed to be instantaneous.

The fraction of refrigerant set free by evaporation when injecting the oil can now be calculated, if one knows the temperature of the mixture at the moment of injection and the pressure in the cavity.

In the simulation program this pressure is taken as the average pressure, $p_{oil\ inj}$, in the cavity during the part of compression when oil is injected. The true temperature of the mixture will depend on the design of the oil system. In a system with an external oil-cooler, as in figure 6, the oil will be cooled to a fixed temperature t_{oil}. It is also here plausible to make an assumption regarding the temperature of the mixture which makes this theory independent of the system and it is assumed that the temperature of the oil-refrigerant mixture as well as the temperature of the evaporated refrigerant will be equal to the decided oil temperature t_{oil}.

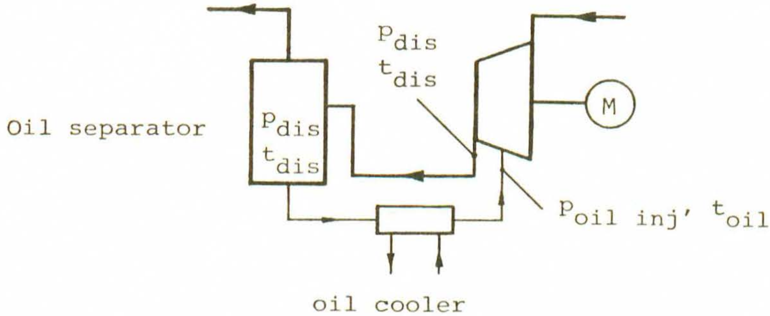

Figure 6. Schematic diagram of compressor and oil system
Principschema für einen Verdichter mit Öleinspritzungsystem

The rate of mass flow of evaporated refrigerant $\dot{m}_{evap}$ can now be calculated as

$$\dot{m}_{evap} = \dot{m}_{oil} \cdot (\xi\ (p_{dis},\ t_{dis}) - \xi\ (p_{oil\ inj},\ t_{oil}))$$

where $\dot{m}_{oil}$ = rate of mass flow of oil-refrigerant mixture.

This quantity is added to the mass of the gas in the cavity (control volume) during the part of the process when oil is injected to the cavity and it will lead to an increased power consumption as will be made clear in the simulation example below.

The discharge temperature and the average pressure $p_{oil\ inj}$ are not known in beforehand and these values have to be guessed before the system of differential equations is solved for the first time. As mentioned in section 2.3 an iterative process is used and the discharge temperature as well as the pressure $p_{oil\ inj}$ are corrected before each new iteration.

3.2 Economizer arrangements

Economizer arrangements, as the one shown in figure 5, improve the COP, since the gas evaporated in the upper (high pressure) throttle valve is not compressed from the evaporation pressure, but from a higher pressure (i.e. the intermediate economizer pressure) and at the same time the evaporator is fed with a larger percentage refrigerant liquid, which gives an increased cooling capacity.

Simulations of compressors with vapour injection (economizer arrangement) are important, since tests in test rigs with economizer arrangements can be rather complicated. The simulations also make it possible to study how the COP is affected by changing size, shape and location of the vapour injection hole in the compressor. Thereby, it is possible to optimize the performance for a screw compressor with economizer arrangement.

The vapour injection is modelled in the simulation program by introduction of one more "rate of mass flow" equation to the set of differential equations and by adding one more term in the heat balance equation (energy equation).

The equation for isentropic "nozzle flow" have proved to describe the flow through the hole very well.

The instantaneous rate of mass flow, $\dot{m}$, is given by

$$\dot{m}\ (\alpha) = A\ (\alpha) \cdot c \cdot \psi\ (\alpha) \cdot \sqrt{\frac{p_{eco}}{v_{eco}}}$$

Where

$$\psi = \sqrt{\frac{2 \cdot \kappa}{\kappa - 1} \cdot \left(\frac{p\ (\alpha)}{p_{eco}}\right)^{2/\kappa} \cdot \left(1 - \left(\frac{p\ (\alpha)}{p_{eco}}\right)^{\frac{\kappa-1}{\kappa}}\right)}$$

if $\frac{p\ (\alpha)}{p_{eco}} \geq \left(\frac{2}{\kappa + 1}\right)^{\frac{\kappa}{\kappa-1}}$ and

$$\psi = \sqrt{\kappa \cdot \left(\frac{2}{\kappa + 1}\right)^{\frac{\kappa+1}{\kappa-1}}} \quad \text{if} \quad \frac{p\ (\alpha)}{p_{eco}} < \left(\frac{2}{\kappa + 1}\right)^{\frac{\kappa}{\kappa-1}}$$

α = rotation angle
$A\ (\alpha)$ = hole area
c = empirical coefficient
κ = isentropic exponent
p_{eco} = economizer pressure
v_{eco} = specific volume at hole inlet
$p\ (\alpha)$ = pressure in the cavity (control volume) at actual rotation angle

For a given economizer pressure and temperature this equation is solved for a large number of rotation angles together with the rest of the equations in the system. When a solution is obtained for all variables, the instantaneous rates of mass flow can be summed to get the total economizer mass flow to one cavity.

The variation of the exposed hole area, $A(\alpha)$ with rotation angle must also be modelled. In many cases, a round hole located in the housing is chosen.

In a real economizer refrigeration system, the intermediate pressure will automatically be adjusted to a value corresponding to the actual economizer arrangement. In the modelling above, the economizer pressure is assumed to be known in beforehand. That is in fact not the case, as the economizer pressure will depend on the ratio of the mass flow through the economizer hole and the mass flow through the compressor inlet. The mass flow through the economizer inlet and the compressor inlet must respectively be the same as the total mass flow from the flash tank, see figure 5. This is only possible for a certain economizer pressure. The simulation program has therefore to be executed several times with different economizer pressures to determine the correct pressure.

A program, which uses the simulation program as a program routine has been developed to determine this pressure. A mass and heat balance over the flash tank gives the necessary equation, see figures 5 and 7.

We get

$$\dot{m}_{tot} = \dot{m}_{evap} + \dot{m}_{eco}$$

$$\dot{m}_{tot} \cdot h'_{cond} = \dot{m}_{evap} \cdot h'_{eco} + \dot{m}_{eco} \cdot h''_{eco}$$

and finally

$$F\,(p_{eco}) = \frac{\dot{m}_{eco}}{\dot{m}_{evap}} - \frac{h'_{cond} - h'_{eco}}{h''_{eco} - h'_{cond}}$$

The secant method is used to solve the equation $F(p_{eco}) = 0$. The mass flows $\dot{m}_{eco}$ and $\dot{m}_{evap}$ are calculated in the simulation program for a given economizer pressure and the saturation enthalpies h'_{eco} and h''_{eco} are also determined by the economizer pressure.

Finally, when the right economizer pressure is found the program calculates pressure profiles as well as power consumption, cooling capacities and COP.

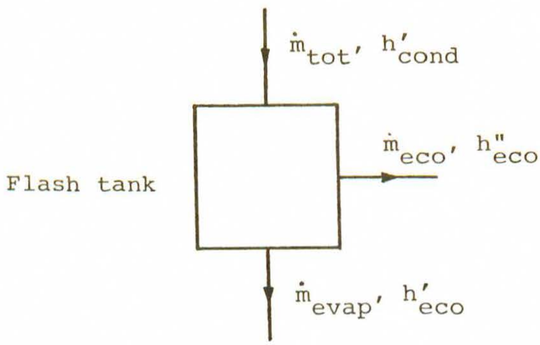

Figure 7. Flash tank. Flow rates and enthalpies.
Mitteldruckbehälter. Massenströmen und Enthalpien.

4. SIMULATION EXAMPLES

All simulation examples in this chapter pertain to the same compressor type. The compressor has a theoretical capacity of 175 m^3/h at 3550 rpm. Prototypes of this compressor have been manufactured.

The rotors have the following data:

Male outer diameter:	113.4 mm
Female outer diameter:	95.8 mm
Length:	150.0 mm
Wrap angle:	300°
Lobe combination:	5+6
Profile:	SRM D-profile

A plot of the end plane profile of the rotors is shown in figure 8.

The prototypes are designed with two lift valves for capacity control, a separate V_i-slide valve and holes for vapour injection.

Much testing has been carried out with the prototypes and the simulations are in good agreement with the test results.

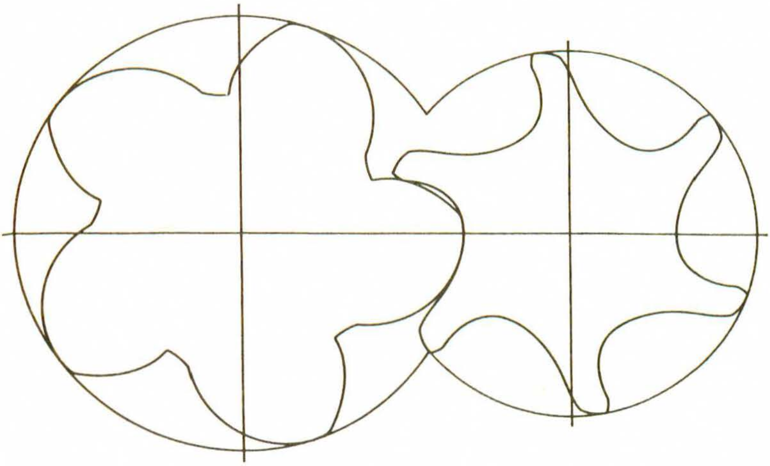

Figure 8. SRM D-profile
SRM D-Profil

4.1 Dependence of V_i

As mentioned in chapter 3 it is suitable to have a program for calculation of "slide valve" outlet port areas at different built-in volume ratios (V_i). How the performance is dependent of the V_i can be studied in figure 9, which shows graphs of computed performance versus pressure ratio for R22 and with V_i as parameter. The V_i is varied from 2.5 to 5.0, which corresponds to outlet port opening angles from 97° to 38°.

The curves show the importance of a right decision of the V_i. Especially at low pressure ratios a too high V_i can lead to a performance loss of many percent. Since many compressors are designed with a fixed V_i at full load, design studies by help of performance simulations give valuable information regarding the V_i for the application in question.

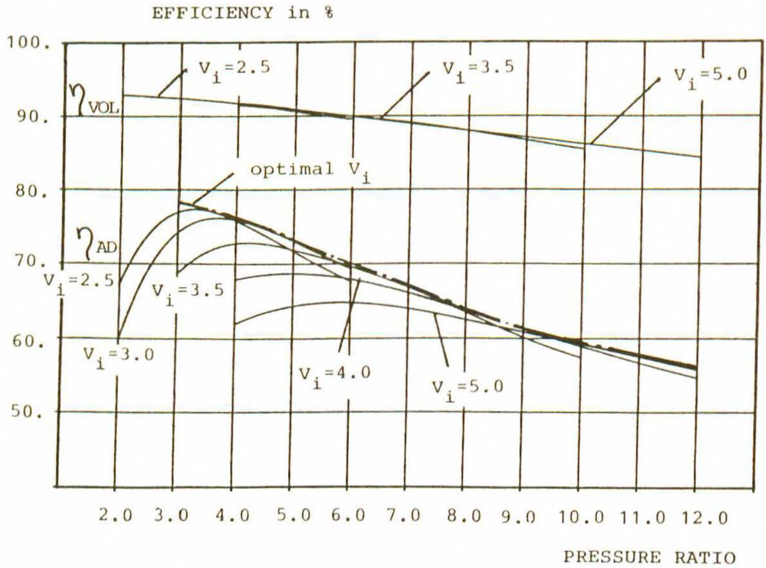

Figure 9. Computed performance vs. pressure ratio. R22, Cond. temp = 35°C, n = 3550 rpm
Gerechneter Wirkungsgrad in Abhängigkeit vom Betriebsdruckverhältnis

The envelope to this set of curves forms the curve for optimal performance at all pressure ratios. This optimal curve is plotted with short dashes.

The variation in volumetric efficiency with V_i is rather small, why only three curves of volumetric efficiencies have been plotted in figure 9.

4.2 Influence of refrigerant in oil

An interesting question is how much the refrigerant dissolved in the oil affects the performance when it evaporates. To get an understanding of this, the simulation program was executed with the same conditions as the simulations discussed in part 4.1, but the mass flow of gas from the oil-refrigerant mixture was put equal to zero. By this it is possible to study a compressor running with an oil of the same viscosity, but free from dissolved refrigerant.

Figure 10 shows the p-V diagrams from two simulations of the same pressure ratio ($\pi = 6$). The difference between them is that one simulation was made with injection of "pure" oil and the other with an oil-refrigerant mixture. It is seen in the diagram that the pressure increase is more rapid for the curve with dissolved refrigerant and consequently this results in a larger work requirement. The outlet port has also to be opened earlier to reach optimal performance for the simulation with dissolved refrigerant in oil.

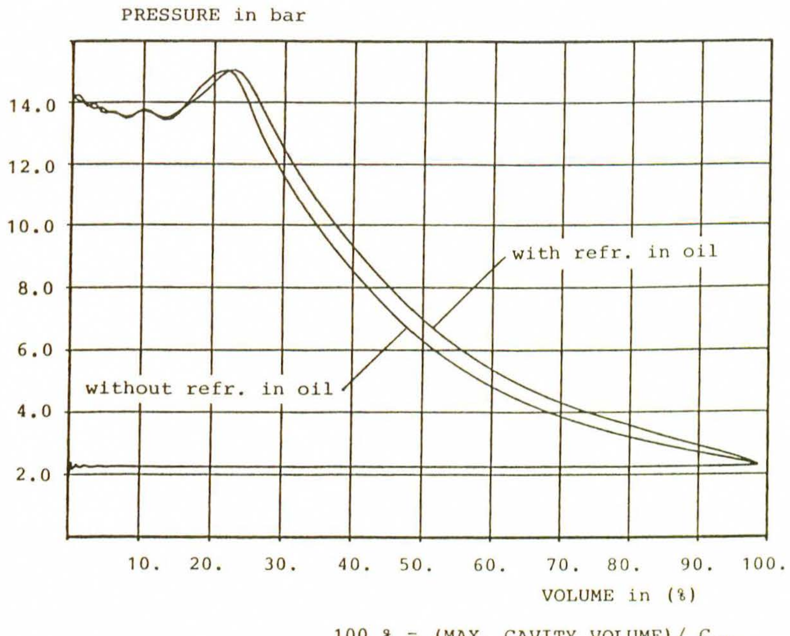

Figure 10. p-V diagram from simulations with and without refrigerant in oil. R22, Cond. temp = 35°C, n = 3550 rpm
Gerechnetes p-V Diagramm mit und ohne gemischtes Kältemittel in dem Öl.

The performance increase by injection of "pure" oil can be studied in figure 11. The optimal performance curve from figure 9 is here plotted and compared with the optimal performance curve from simulations without dissolved refrigerant in the oil. The two curves are diverging with increasing pressure ratio. The results show an increase in adiabatic efficiency of around 3 % at pressure ratio of 4.0 and around 13 % at pressure ratio of 12.0.

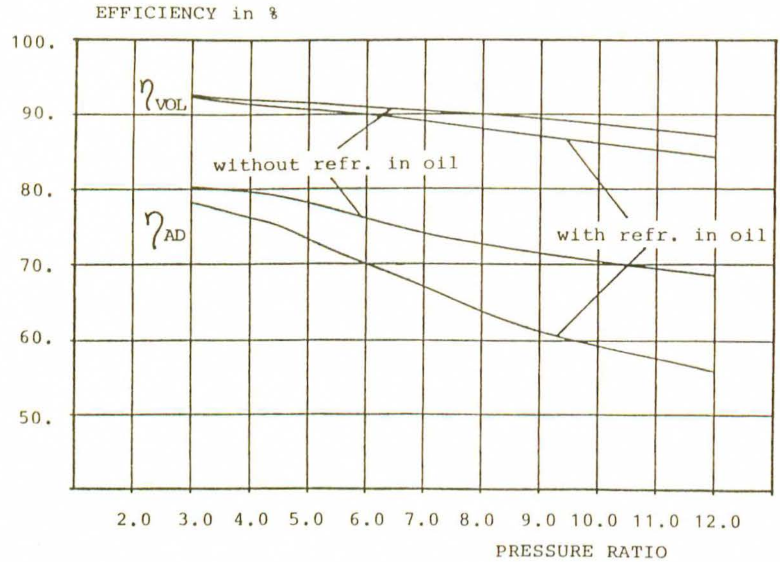

Figure 11. Computed performance vs. pressure ratio with and without refrigerant in oil. R22, Cond. temp = 35°C, n = 3550 rpm, Optimal V_i
Gerechneter Wirkungsgrad in Abhängigkeit vom Betriebsdruckverhältnis mit und ohne gemischtes Kältemittel in dem Öl.

A larger amount of gas is evaporating when the mixture is injected to a lower cavity pressure and this is the reason for the larger influence at high pressure ratios.

A small increase in volumetric efficiency for "pure" oil is also shown in the diagram. This is a result of the "lower" pressure level in the cavities under compression in the case of "pure" oil, as shown in figure 10.

When the oil-refrigerant mixture is leaking back to the inlet, an additional amount of gas will evaporate due to the pressure decrease, but this effect has not been included in these computations.

4.3 Economizer arrangement

Economizer simulations for the same condensing temperature (35°C) as in the other simulations have been carried out and the results are shown in figure 12 with COP versus pressure ratio. For comparison the simulation results from section 4.1 without economizer arrangement are also plotted. The curves show that the COP-improvement at the low pressure ratios is very small, in the magnitude of a couple of percent, but becomes significant at the high pressure ratios. At a pressure ratio of 12.0 the improvement amounts to 19 %.

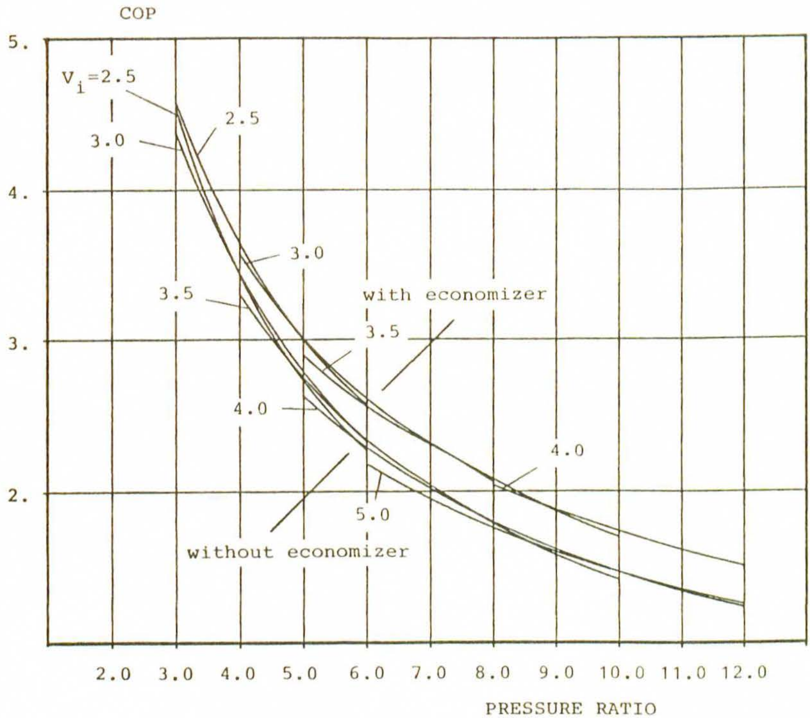

Figure 12. Computed COP vs. pressure ratio. With and without economizer. R22. Cond. temp = 35°C, n = 3550 rpm.
Gerechneter COP in Abhängigkeit vom Betriebsdruckverhältnis mit und ohne Economizer

It can also be observed that the optimal V_i is lower for the economizer simulations, since the cavity will reach the discharge pressure at an earlier rotation angle due to the "super-filling" with economizer gas. The pressure increase in a compressor with economizer arrangement can be studied in figure 13. The figure shows p-V diagrams from both a simulation with economizer arrangement and a simulation for the same pressure ratio but without economizer.

The vapour injection hole in this simulation is a round hole, located at a rotation angle close to the inlet port closing angle.

Finally, in figure 14 the economizer pressures are plotted versus pressure ratios. The economizer pressure is more or less independent of the V_i, why only two curves with different V_i are presented.

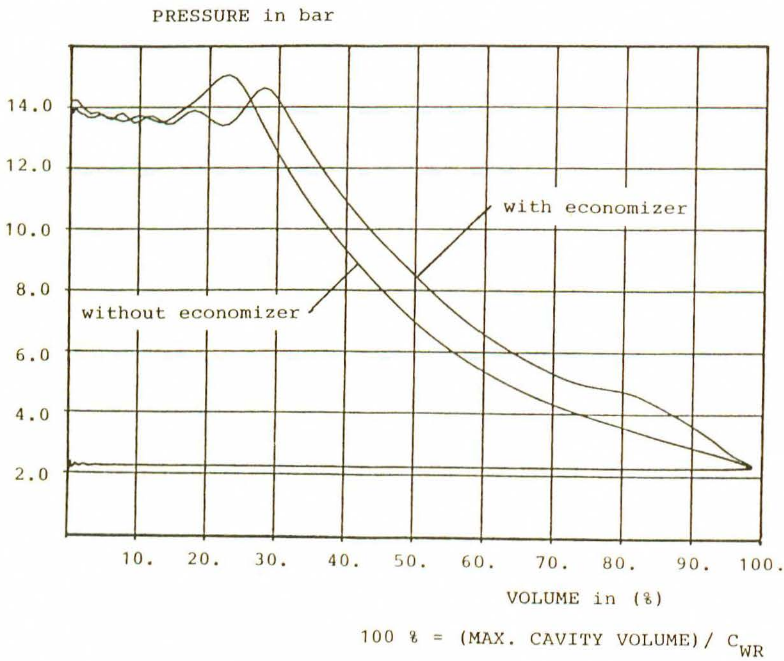

Figure 13. p-V diagram from simulations with and without economizer. R22. Cond. temp = 35°C, n = 3550 rpm

Gerechnetes p-V Diagramm mit und ohne Economizer

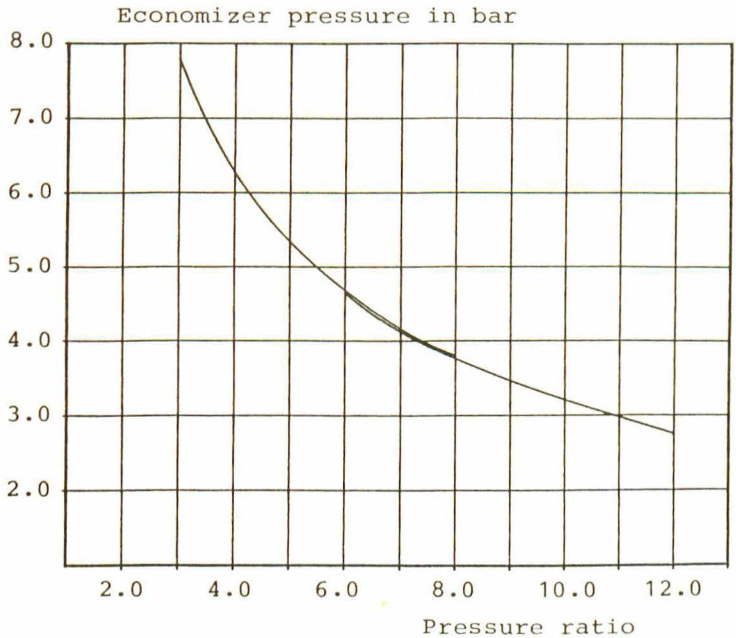

Figure 14. Economizer pressure vs. pressure ratio. R22, Cond. temp = 35°C, n = 3550 rpm.
Economizerdruck in Abhängigkeit vom Betriebsdruckverhältnis

REFERENCES

1. S. Jonsson, Computer calculations for design and analysis of screw compressors, Royal Institute of Technology. Department of Thermal Engineering, Stockholm, Sweden, Trita åtk 860907.

2. B. Sångfors, Computer simulation of the oil-injected twin-screw compressor, 1984 International Compressor Engineering Conference, Purdue University, West Lafayette, U.S.A.

3. L. Sjöholm, variable volume-ratio and capacity control in twin-screw compressors, 1986 International Compressor Engineering Conference proceedings, Purdue University, West Lafayette, U.S.A.

4. H.P. Jaeger, Kältetechnik - Klimatisierung, 24, p. 35 - 52, 1973.