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Performance Simulations of Twin-Screw Compressors with Economizer

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PERFORMANCE SIMULATIONS OF TWIN-SCREW COMPRESSORS WITH ECONOMIZER

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ABSTRACT

In a compressor refrigeration plant with economizer system, the regular expansion valve is replaced by two valves and an intermediate pressure vessel. The refrigerant which is vapourized after the first valve is injected via the compressor economizer inlet to a thread under compression. The economizer arrangement increases the refrigeration capacity and improves the COP (Coefficient of Performance). In this paper a performance simulation computer program for twin-screw compressors with economizer arrangement is presented.

Comparisons of economizer performance for different arrangements are often difficult to carry out if real tests have not been run, since the performance is depending on the intermediate pressure and this pressure will not be the same for different arrangements. Such comparisons can be made with the simulation program. Examples are presented for an economizer arrangement combined with external liquid subcooling and for a two stage economizer arrangement.

1. INTRODUCTION

The purpose of performance simulations of compressors is to describe the thermodynamic process inside a compressor. The simulations then give an overall picture of the compression process and give a possibility to put different losses in quantitative relations to each other. Detailed presentations of this simulation program and the geometrical data program library have earlier been made in ref [1], [2], and [3]. Therefor a short overview is only given here.

The refrigeration system with economizer arrangement, shortly described in the abstract above and in section 4.1, can be designed in different ways which will affect performance. With the help of this simulation program, the performance of some different economizer alternatives are studied in this paper.

2. GENERAL OVERVIEW OF THE SIMULATION PROGRAM

One of the fundamental demands on simulation is that it must be possible to model the geometrical design in a mathematical form. The following geometrical parameters are calculated by computer programs and the results are inputs to the simulation program:

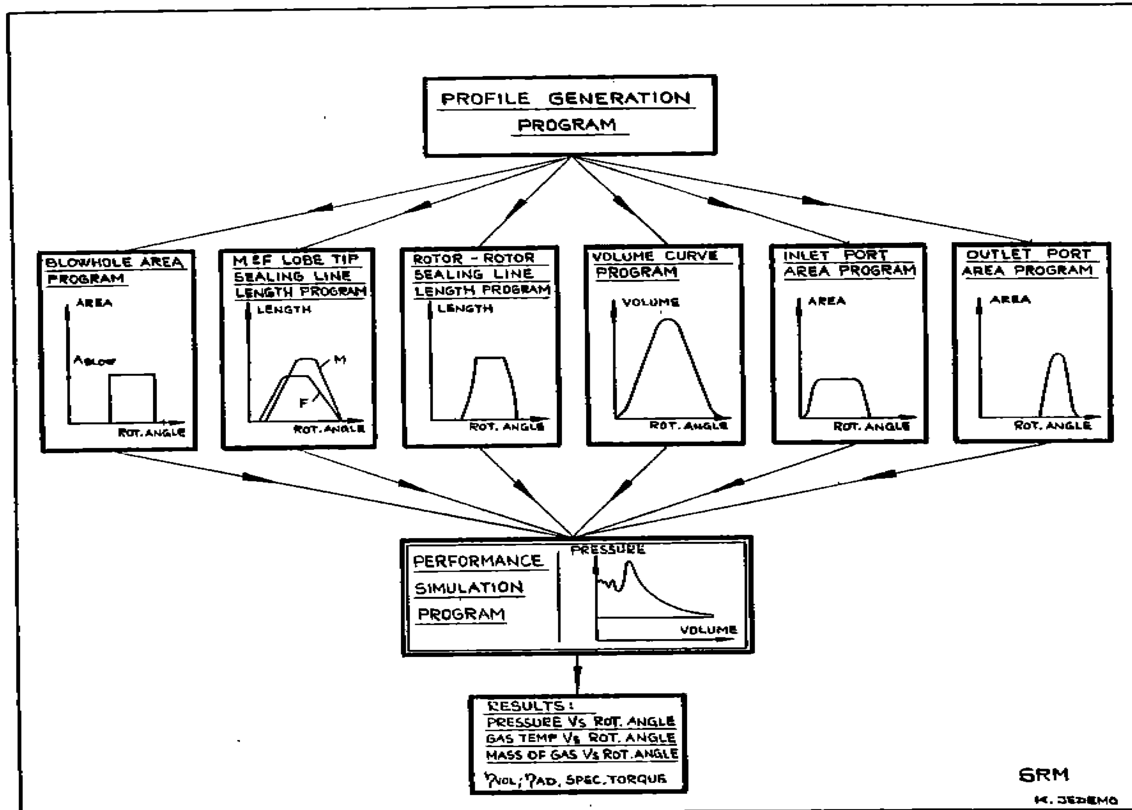

Figure 1.
Flow chart

- 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 and
- Blow-hole area.

Figure 1 shows the flow chart of the computer programs.

For calculation of part load performance one also needs:

- Slide valve by-pass port area vs. rotation angle.

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.

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.

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 and
- Heat transfer between gas and oil.

The effects of the 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 and
- 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.

As an example, the pressure profile (pressure versus rotation angle) from both a full-load and a part-load simulation is shown in figure 2. The condensing temperature is 50°C and the pressure ratio is equal to 3.0. On top of figure 2, a diagram showing the area curves for the inlet, the slide valve and the outlet ports is presented. This diagram shows that the outlet port at part load opens "later" than at full load, since the slide valve is pushed towards the discharge end plane at part load.

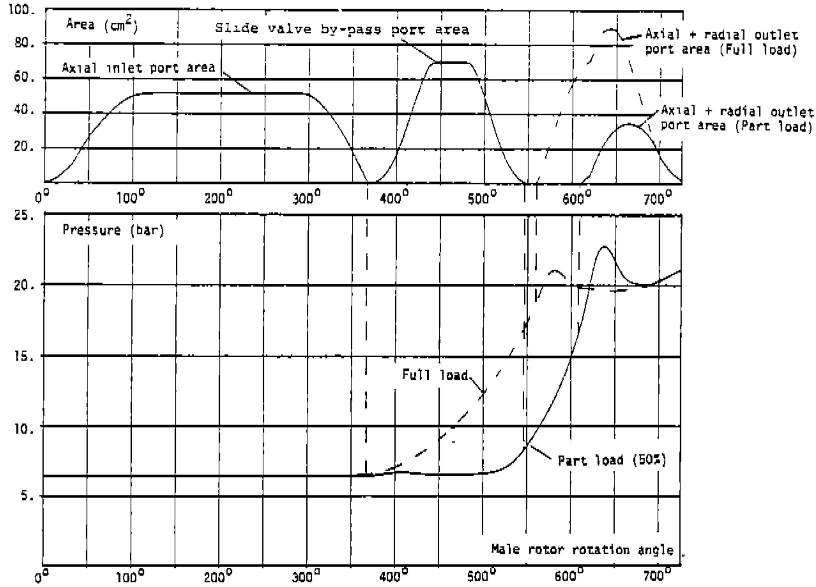

Figure 2. Pressure and port areas vs. rotation angle.
Full load and part load. R22. Cond temp = 50°C.
Pressure ratio = 3.0.

3. SIMULATION OF REFRIGERATION TWIN-SCREW COMPRESSORS

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 [5].

It is important for the geometrical modelling of refrigeration compressors to have a computer program which both calculates the area variation of the axial outlet port as well as the area variation of the radial outlet port of "slide valve" type.

For part load simulations, information about the slide valve by-pass area variation is needed, see the example in figure 2 above.

One difference from thermodynamic point of view between oil-flooded air compressors and machines compressing refrigerants is the solubility of refrigerant in oil. When the oil is injected into the compression chamber some of the refrigerant dissolved in the oil evaporates. This will affect the compressor performance in a negative way as a larger mass of gas has to be compressed. The modelling of this effect has been described in ref. [2].

Different combinations of oil type and refrigerant will dissolve the refrigerant to different extent. The number of experimental investigations in this field is unfortunately limited why only a few relationships are available. In ref. [4] the solubility 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.

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 both with and without refrigerant dissolved in the oil. By putting the mass flow of gas from the oil-refrigerant mixture equal to zero, it is possible to study a compressor running with an oil of the same viscosity, but free from dissolved refrigerant.

The simulations were run on R22 for a compressor with a theoretical capacity of 175 m^3/h at 3550 rpm (D_M = 113.4 mm).

The performance increase by the injection of "pure" oil can be studied in figure 3. The optimal V_i performance curve is here plotted and compared with the optimal V_i (within V_i = 2.5 to 5.0) 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 a pressure ratio of 4.0 and around 13 % at a pressure ratio of 12.0.

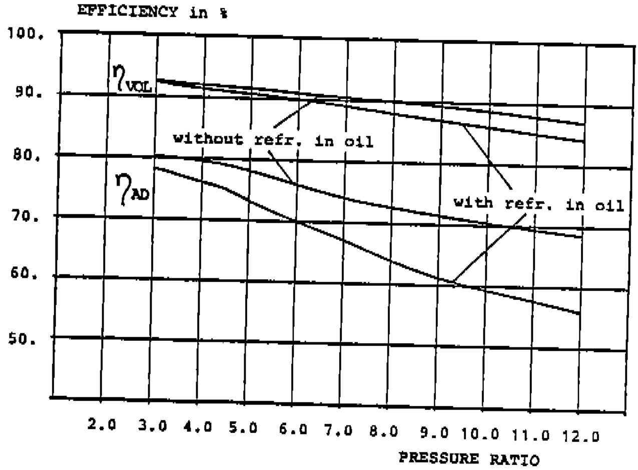

Figure 3. Computed performance vs. pressure ratio with and without dissolved refrigerant in the oil. R22. Cond temp = 35°C. n = 3550 rpm. Optimal V_i (V_i = 2.5 - 5.0).

A larger amount of gas is evaporating when the mixture is injected into a lower cavity pressure. 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.

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

4. ECONOMIZER

4.1 Theory

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). The refrigerant which is vapourized after the first valve is injected, via the economizer inlet, into a thread under compression, see figure 4. In the same figure the process is described in a Mollier diagram.

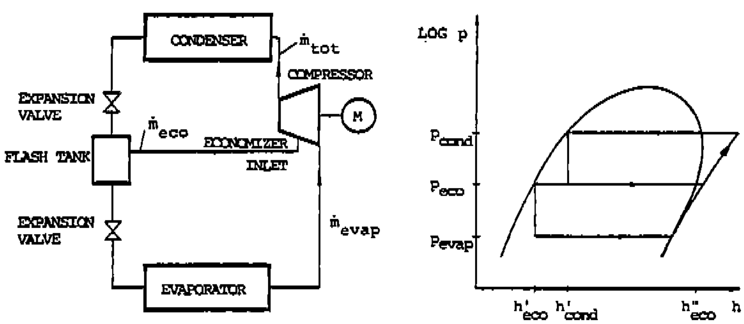

Figure 4. Principle of economizer refrigeration system.

Economizer arrangements improve the COP, since the gas evaporated in the upper (high pressure) expansion 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 of refrigerant liquid, which gives an increased cooling capacity.

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" has proved to describe the flow through the economizer port very well. The modelling of vapour injection has earlier been described in ref [2].

The variation of the exposed hole area, $A(\alpha)$ with rotation angle α 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 simulations, the economizer pressure is not known beforehand, 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 4. 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.

4.2 Comparison of Performance With and Without Economizer

All simulation examples in the following pertain to the same compressor type. The compressor has a theoretical capacity of 1220 m^3/h at 3000 rpm.

The rotors have outer diameters equal to 204 mm. The lobe combination is 4+6.

The vapour-injection hole in these simulations is a round hole located at a rotation angle close to the inlet port closing angle. The hole has the diameter 25 mm. A hole located as close to the inlet port closing angle as possible, but without connection between the economizer hole and the inlet, gives the lowest economizer pressure and thereby maximum cooling capacity.

In figure 5 simulation results are presented with COP versus pressure ratio for R22 and 40°C condensing temperature both with and without economizer arrangement. The computations were run with different V_i and the curves are presented for optimal V_i (V_i = 2.5 to 5.0).

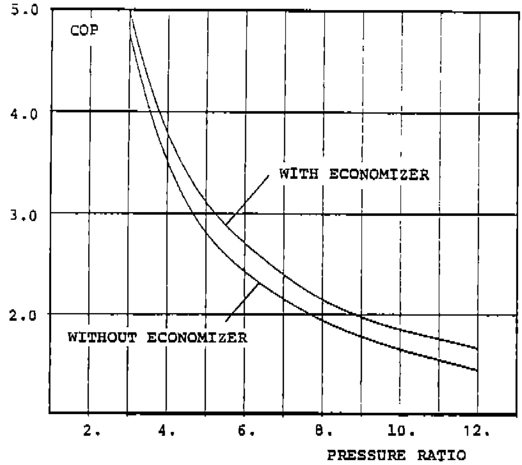

Figure 5. Computed COP vs. pressure ratio with and without economizer. R22. Cond temp = 40°C. n = 3000 rpm. Optimal V_i (2.5 to 5.0).

The curves show that the COP-improvement at low pressure ratios is very small, in the magnitude of a couple of percent, but becomes significant at high pressure ratios. At a pressure ratio of 12.0 the improvement amounts to 16 %.

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 a larger cavity volume due to the "super-filling" with economizer gas. The pressure increase in a compressor with economizer arrangement can be studied in figure 6. The figure shows p-V diagrams from both a simulation with economizer arrangement and a simulation for the same pressure ratio but without economizer.

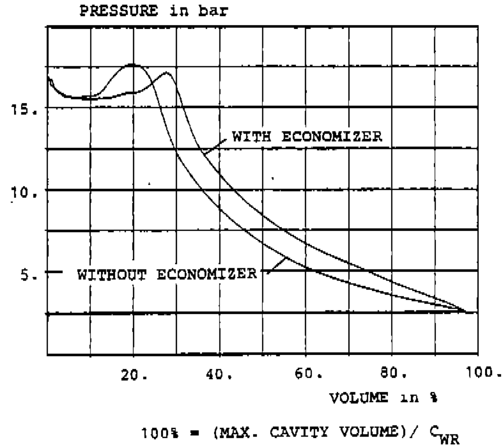

Figure 6. p-V diagram from simulations with and without economizer. R22. Cond temp = 40°C. n = 3000 rpm. Pressure ratio = 6.0.

4.3 Performance With Economizer and External Subcooling

By subcooling of liquid refrigerant, i.e. removing heat from the refrigeration system at the high pressure side between the condenser and the expansion valve, the percentage of liquid after the valve will increase and give larger cooling capacity as well as higher COP.

Sometime the use of subcooling is regarded as an alternative to a system with economizer arrangement, as both types of systems improve the performance by feeding the evaporator with a higher liquid percentage. When looking upon these systems as alternative to each other, one is not aware of the fact that the use of both subcooling and economizer arrangement will improve the performance even more and that they do not counteract from performance point of view. The subcooling will lead to a lower economizer pressure. That means higher percentage of liquid to the evaporator, as can be seen in the enthalpy-diagram in figure 7. In this diagram the system with subcooling is shown with dashed lines.

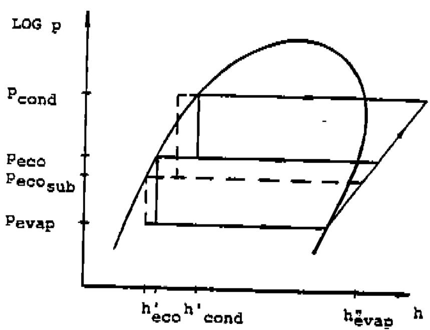

Figure 7. Enthalpy diagram for economizer refrigeration system. External subcooling shown with dashed lines.

The simulation program can be used, to get a comparison between economizer performance with and without subcooling. The result of one such comparison is presented in figure 8 for R22 and a condensing temperature of 40°C with COP versus pressure ratio. To make a correct comparison, the results are presented for optimal V_i. The curve for no subcooling is the same as the economizer curve in figure 5. The results show that the relative increase in COP is 0.6 % per degree Celsius of subcooling. This means, for example, that the COP will increase 6 % for 10°C of subcooling.

In figure 9 the relative increase in capacity with subcooling for an economizer system is presented.

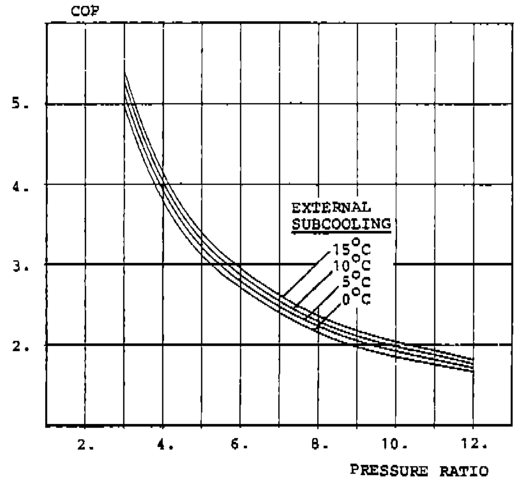

Figure 8. COP vs. pressure ratio. Economizer. R22.
Cond temp = 40°C. 0° to 15°C external subcooling.
Optimal V_i (2.5 to 5.0).

Since a COP-increase is always achieved with economizer arrangement the conclusion of this analysis must be that subcooling is a good complement to an economizer arrangement and should not be regarded as an alternative to the economizer arrangement.

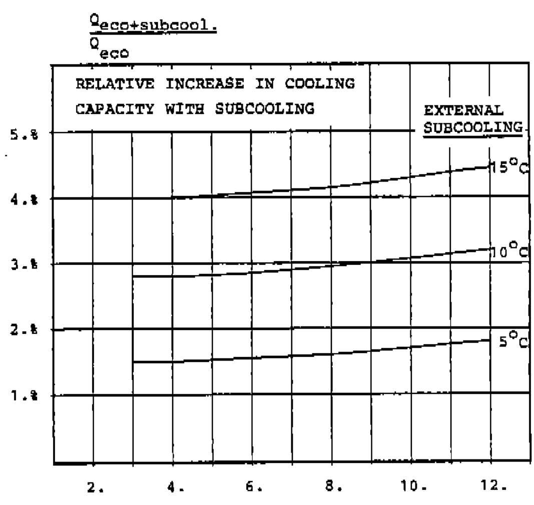

Figure 9. Relative increase in cooling capacity with
external subcooling vs. pressure ratio.
Economizer refrigeration system. R22.
Cond temp = 40°C.

4.4 Performance With Two-Stage Economizer Arrangement

A one-stage economizer arrangement improves the COP and increases cooling capacity, as described in section 4.2. A logical question is now if, and in that case how much, the performance will be increased by a second economizer stage. A sketch of a two-stage economizer system with two flash tanks and three expansion valves is shown in figure 10, together with the corresponding enthalpy diagram.

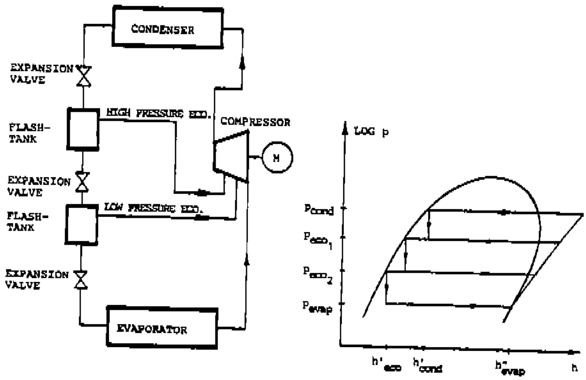

Figure 10. Principle of two-stage economizer refrigeration system.

One problem from a practical point of view, is that the two economizer inlet holes must be placed at such rotation angles that neither communication between the holes, nor communication to the inlet or outlet may occur. At the same time the "superfeeding" of gas is large and this decreases the optimum V_i. The optimum V_i at low pressure ratios is therefor lower than what can be used without achieving communication between the high stage economizer and the outlet port.

From simulation point of view the calculations are time consuming to carry out. Modelling of the two-stage vapour injection is in itself not more complicated compared with one-stage economizer modelling. However, one has to find a solution for two unknown economizer pressures in this case. The convergence of two unknown variables to a correct solution involves a complicated mathematical procedure with many individual simulations for different economizer pressures.

In figures 11 and 12 the results from two-stage economizer simulations are presented with COP and relative increase of cooling capacity versus pressure ratio for 40°C condensing temperature. For comparison the results from section 4.2 are also presented.

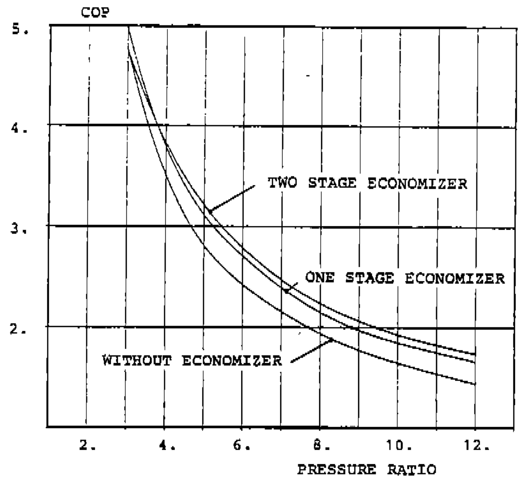

Figure 11. Computed COP vs. pressure ratio for one- and two-stage economizer system. COP for a system without economizer is shown for comparison. R22. Cond temp = 40°C. n = 3000 rpm. Optimal V_i (2.5 - 5.0).

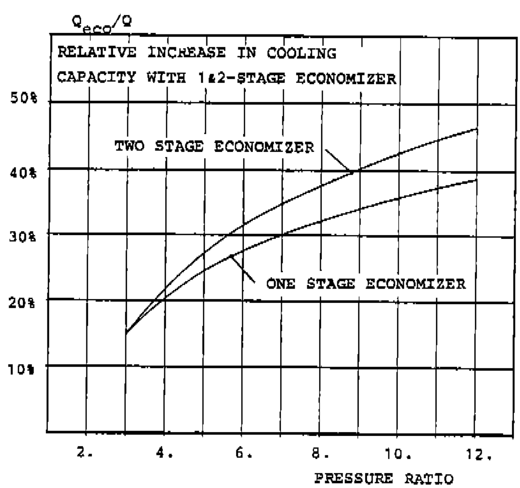

Figure 12. Relative increase in cooling capacity with one- and two-stage economizer vs. pressure ratio. R22. Cond temp = 40°C.

The results show that a COP improvement is obtained at higher pressure ratio. The increase, compared with one stage economizer, is not as large as the COP increase that was obtained with a one stage economizer, compared with a system without economizer. At a pressure ratio of 12.0 the improvement amounts to about 5 % compared with a one stage economizer system. At low pressure ratios, a COP reduction is obtained due to the fact that a too high V_i has to be used as mentioned above. The performance can be improved somewhat at low pressure ratios by using rotors with more lobes than the 4+6 combination used in these simulations. With more lobes there is more "space" for the economizer holes and a lower V_i can be used.

The low- and high-stage economizer pressures are plotted in figure 13. The one stage economizer pressure is also plotted for comparison. As can be seen in the diagram the low-stage pressure for the two stage economizer system is lower than the pressure for the one stage economizer resulting in larger capacity for the two-stage economizer.

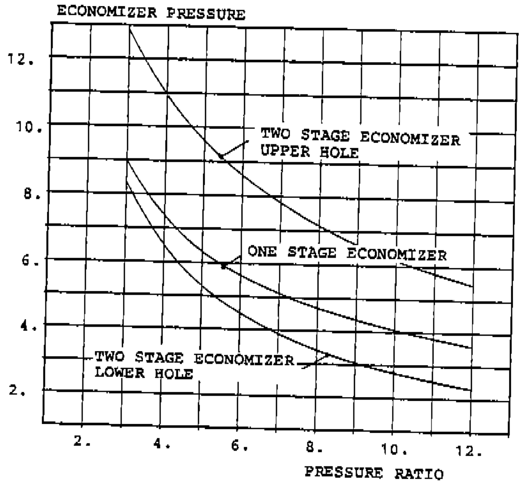

Figure 13. Economizer pressure vs. pressure ratio. One- and two-stage economizer. R22. Cond temp = 40°C.

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