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# Port Optimization of a Voorhees Modified $CO_2$ Compressor Using Indicator Diagram Analysis

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## ABSTRACT

The economizer principle is a well-known method to reduce the expansion losses of a refrigeration system. The alternative refrigerant $CO_2$ has a relatively large isentropic exponent. Therefore it has a high potential for raising efficiency by application of this principle. Reciprocating piston compressors are favored for $CO_2$ refrigeration applications due to their small sealing length and therefore low internal leakage rate. The economizer principle can be applied to reciprocating compressors by use of the "multiple effect" principle patented by Voorhees in 1905.

This paper investigates the influence of the admission port design on the admission process. The admission port is one of the main compressor modifications to realize the "multiple effect" principle. Its design has a large impact on the system efficiency and therefore needs to be optimized. For this purpose, an optimization process has been established. It is based on indicator diagram analysis. To exemplify, the measurements of three admission ports were compared at constant suction pressure (1.2 MPa) and varying admission pressure (1,4 MPa – 2,6 MPa).

## 1. INTRODUCTION

Since the early nineties, the use of the natural refrigerant $CO_2$ has gained acceptance, especially in refrigeration and air conditioning applications. On the one hand, this can be explained by its environmental friendliness; on the other hand, it can be attributed to its positive heat transfer behaviour, which is especially good under supercritical conditions. The latter leads to compensation of the $CO_2$ specific increased expansion losses and increased power consumption of the compressor due to the relatively high isentropic exponent.

The use of $CO_2$ leads to encouraging COPs in cooling applications at evaporating temperatures around –5 °C. This sparks investigations into the possible use of $CO_2$ in low temperature applications at evaporating temperatures around –35 °C. However, with decreasing evaporation pressure, the isentropic efficiency's negative influence increases. That is why alternative $CO_2$ freezing application system designs have been investigated recently, mainly focused on the reduction of expansion losses.

One approach to reduce these losses is the economizer principle which can be applied as a compressor modification invented by Voorhees. He called this modified compressor a "multiple effect" compressor (Voorhees, 1905). There is a theoretical potential of COP improvement of approx. 17 % at an ambient temperature of 32 °C, as previous investigations show (Lambers *et a.*, 2006). With first measurements, an enhancement of 10 % could be achieved.

To get closer to the theoretical limit, optimization of the admission port is necessary. The admission port is one of the main compressor modifications to realize the "multiple effect" principle. Its design has a major impact on the system efficiency.

# 2. THE "MULTIPE EFFECT" PRINCIPLE

## 2.1 The "Multiple Effect" Compressor Modification to Realize the Economizer Principle

The efficiency of a system can be increased by the well known economizer principle. Here, flash gas is seperated at the different pressure levels of a multi-stage throttling cycle. This flash gas is introduced at the corresponding pressure levels of the multi-stage compression. This can be done more economically by implementating a scoll or screw compressor. In that case, only one compressor is necessary. The flash gas is introduced into the compression process at the specific pressure level.

The additional expenses due to this advanced system design results in four beneficial effects: the COP increases, there is a rise in refrigeration capacity, the mass-flow through the evaporator decreases, which implies reduced dimensioning and the discharge temperature decreases which is especially important for refrigerants with larger isentropic exponents as is the case with $CO_2$.

The principle of a reciprocating piston compressor suits the thermodynamical demands of the refrigerant $CO_2$ best (Süss, 1998). Nevertheless, to apply the advantageous economizer principle to a $CO_2$ refrigeration system, the "multiple effect" compressor modification of Voorhees seems to be an option.

The "multiple effect" modification is based on a conventional reciprocating piston compressor. The flash gas which is seperated in the two-stage expansion and which is called auxiliary gas from now on, is injected into the compression chamber when the piston reaches the bottom dead center. The auxiliary gas hereby compresses the suction gas from suction pressure to admission pressure.

Admittetly, the admission comprises unavoidable throtteling losses due to the discontinuity of the admission process. These losses don't occur when applying the economizer principle to the compressor types mentioned before.

Also, the "multiple effect" modification allows the application of several admission pressure levels, as already suggested by Voorhees. This contributes to reducing the discussed throttle losses. This is not a part of the present research because of its increased system complexity.

## 2.2 The Design of the Compressor Modification

Figure 1 shows schematically the functional principle of the compressor modification suggested by Voorhees. The additional bore at the bottom dead center of the piston allows additional gas admission in the end of the suction phase.

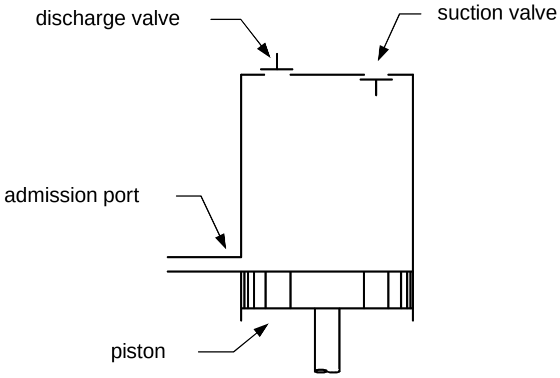


**Figure 1: The "multiple effect" principle by Voorhees**

There is no need for an additional valve, because the piston blocks the admission port while it moves backwards. Close to the dead bottom center it finally opens the port to start the admission phase. Hereby, the piston velocity is very low and it becomes zero while the piston changes direction of movement. That's why there is a relatively long period for the admission process to take place.

The distance, the piston covers form the bottom dead center to the closing of the orifice is called admission clearance in the following. It can be scaled based on the stroke of the piston.

## 2.3 The Design of the Admission Port

The design of the admission port has a major influence on the efficiency of the modified compressor. Three aspects are of particular importance:

The orifice should be large enough to allow largest possible admission. Additionally, the admission clearance should be as small as possible, not to reduce unnecessarily the volume that is filled by the suction gas. A too large admission clearance also causes backflow through the admission port, when the piston leaves the bottom dead center. As a third demand heat transfer from the hot cylinder material to the auxiliary gas should be avoided. These three design criteria are contradictory leading to a demand for optimization.

The design of the admission port has four major aspects: The geometry of the orifice in the cylinder wall, the geometry of the bore or chamber in the cylinder material, the number of orifices in the cylinder wall and the position

of the orifice in relation to the bottom dead center; to simplify production, it can make sense to move the orifice partly below the bottom dead center.
In the first approach, calculations based on fluid dynamics and gas dynamics can help to make a first estimate of the necessary dimensions. For further optimization, an experimental procedure is necessary.

## 2.4 Characteristics of the p,V–Diagram of the Compression with Admission

Caused by the admission, the characteristics of the p,V-diagram of the compression changes compared to the p,V-diagram of a conventional reciprocating piston compressor. As a part of the research project, calculations of the compression process with admission were carried out. In the related model, the volume of the admitted gas is taken into the balance the moment the admission process starts. One of the main assumptions the model is based on, is that the unavoidable throttling of the auxiliary gas occurs in the plane of the admission orifice. The ideal admission is quasi-stationary. The temperature distribution in the compression chamber is homogeneous during the period of admission. The friction of the flow as well as the inertia of the gas is neglected. For the described ideal case, the admission clearance approaches zero. Consequently, the crank angle approaches the bottom dead center when the port is opened and the admission pressure is achieved. Thus, there is no reduction of the effective suction volume. That makes backflow through the admission port impossible.
Figure 2 displays the resulting p,V-diagram with the described assumption of an ideal admission. On the abscissa, the clearance volume, cylinder volume and the total balance volume are drawn. The latter is the sum of clearance volume, stroke volume and the volume of the admission gas under admission pressure just before admission. The suction pressure, admission pressure and discharge pressure are drawn on the ordinate. The compression curves starting at the cylinder volume represent the compression line of the gas fraction that entered the compression chamber through the suction valve. The dotted line shows the course assuming there is no heat transfer, which implies no blending of the two gas fractions. This line represents the compression line of a conventional compression process. Suction and admission gas enter the compressor with the same temperature. As a part of the quasi-stationary admission, entropy equalization takes place including unavoidable entropy generation. Consequently, the entropy of the suction gas fraction decreases. That’s why the actual compression line of this mass fraction, which is displayed as a solid line, is drawn further to the left.
The compression line that starts at the total balanced volume is the compression line of the total gas mass. An imaginary plane that separates the admitted gas of one stroke and the remaining gas in the admission line, is pushed to the plane of the admission orifice by the admission pressure. Thus the volume decreases and the pressure in the compression chamber increases. The thicker solid line represents the pressure course of the compression chamber, regardless of the volume of the admitted gas.

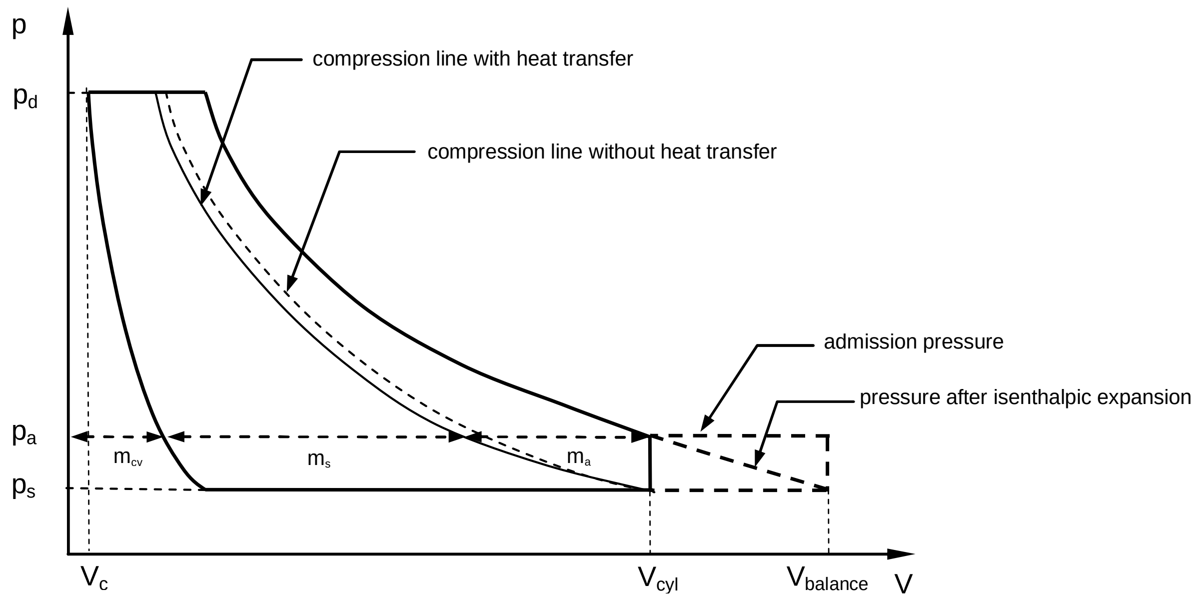


**Figure 2: p,V-digram of the modified compressor with ideal admission**

# 3. RAPID PRESSURE MEASUREMENTS IN THE COMPRESSION CHAMBER

## 3.1 The Test Rig

The test rig is designed to run a gas cycle process without phase changes under normal operating conditions. A discharge pressure controlling expansion device is followed by a gas container, compensating the different charge needs. A low pressure controlling expansion valve follows to adjust the admission pressure. A third expansion valve throttles from admission pressure to suction pressure. Two thermal baths allow adjusting the temperatures of suction and admission gas flow within the relevant range. Both mass-flows are measured by Coriolis mass-flow meters. The system is equipped with two data acquisition systems. One logs stationary system data as mass-flows, temperatures, pressures and the power consumption of the compressor. The second takes the rapid pressure measurements, necessary to perform the indicator diagram analysis.

The low duty heat exchanger, which only rejects the energy taken by the compressor, is designed based on a single copper tub. In comparison to a real gas cooler, it shows hardly any pressure pulsation damping effect. Therefore, an additional volume was added to the discharge line. It decreases the amplitude of the pulsation and increases its frequency. This pressure pulsation superimposes the compression chamber pressure during the discharge phase, as illustrated in chapter 4. In general, this pulsation has no impact on the interpretation of the measurements because it has no impact on the analyzed admission process.

## 3.2 Modifications of the Compressors

A large variety of admission port geometries is imaginable, as described in chapter 2.3. Figure 3 displays the two manufactured geometries for testing. The radial drill hole, as shown to the left, seems to be easiest for both production and reproduction. That's why it is taken as a baseline. The scheme to the right shows a tangential drill hole. The interpenetration of the bore and the drill hole results in an elliptically shaped orifice. A better flow profile was expected due to the enlarged orifice even if the advantages are assumed to be slightly compensated by increased turbulence. Both geometries were manufactured with a scaled admission clearance of 0.125. An additional compressor is equipped with a radial admission port that has a scaled admission clearance of 0.094.

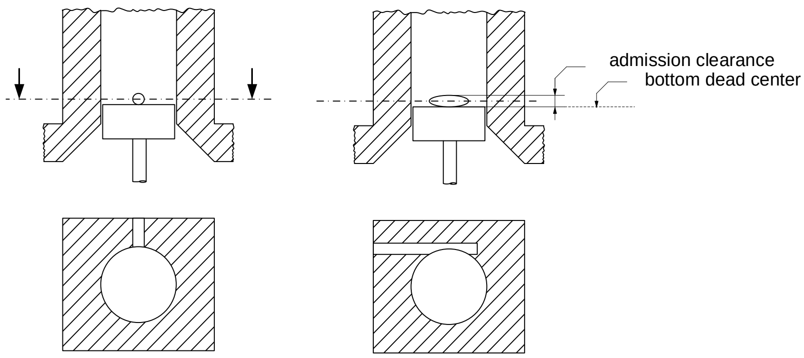


**Figure 3:**

**Geometries of the manufactured admission ports**

Three pressure sensors were placed in one plenum chamber assembly that could be exchanged between the three compressors. These sensors were able to measure the pressure of the suction chamber, the discharge chamber and the compression chamber.

## 3.3 Experimental Conditions

The freezing application is of increased interest, running at a suction pressure of 1.2 MPa (corresponding to approx. -35 °C) and an ambient temperature between 20 °C and 50 °C. Preliminary theoretical studies have shown that this results in an admission pressure between 1.5 MPa and 2 MPa (Lambers *et al*.: 2006). In these investigations, the heat exchangers of the system were assumed to behave ideally. Due to the non-ideal behavior of the heat exchangers, the content of enthalpy of the mass-flow entering the first expansion device is enlarged compared to the ideal case. Therefore, the optimal admission pressure of the real application is expected to be slightly higher than the prediction of the preliminary studies. Measurements revealed that the flow losses don't influence the ratio of auxiliary gas mass-flow to suction gas mass-flow to a larger extent. Thus the impact of the admission losses on the admission pressure is low as well. That's why the test range of the admission pressure was set between 1.4 MPa and 2.6 MPa.

# 4. DATA ANALYSIS

## 4.1 Pressure Course against Crank Angle

The first step in the data analysis procedure is to set the pressure curve and the crank angle in correlation. The resulting cylinder pressure course, as shown in Figure 4, can be divided into five main phases of the compression process with admission. The phases of re-expansion, suction, compression and discharge are known from the conventional compression. The fifth additional phase is caused by the admission of the flash gas, taking place when the piston opens the admission port.

The diagram shows that the pressure losses are small at the suction and discharge valve. The pressure pulsations in the discharge line, which are described in chapter 3.1, are distinctly visible. The plotted pressure course belongs to the compressor that is modified with a radial admission port with a scaled admission clearance of 0.094. The admission pressure is drawn as a horizontal dotted line. It can be seen easily, that the pressure equilibrium of the admission port and the compression chamber is not reached the moment the admission orifice becomes completely obstructed by the piston. This will be discussed below.

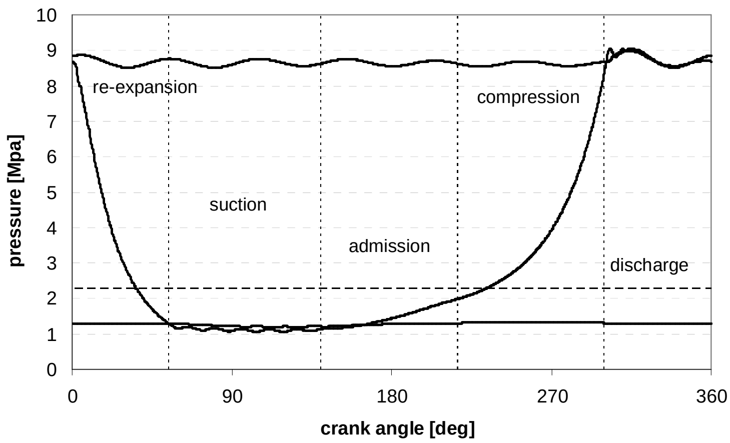


**Figure 4:**

**Pressures of suction chamber, discharge chamber and compression chamber against the crank angle**

## 4.2 Variation of the Admission Pressure

Figure 5 shows five superimposed pressure courses of the compression chamber plotted against the crank angle. The admission pressure was varied between 1.4 MPa and 2.6 MPa. The temperature of suction and discharge gas was 32 °C. These pressure courses belong to the baseline compressor modified with a radial admission port of a scaled admission clearance of 0.125.

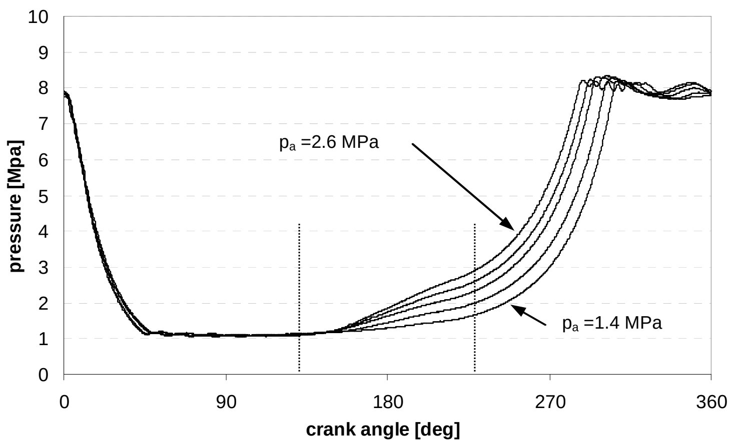


**Figure 5:**

**Pressure courses with admission pressures between 1.4 MPa and 2.6 MPa**

It is obvious that the admission increases with increasing admission pressure. This causes a higher pressure during the compression phase. Due to the increased charge, the discharge phase starts earlier.
The temperature of the mixture decreases with increasing admission fraction because of the lower specific entropy of the additional gas. That's why the re-expansion takes a steeper course at higher admission pressure.

Figure 6 shows a close-up of the actual admission phase of the compressor with a scaled admission clearance of 0.125. There is now remarkable reaction of the pressure courses at the beginning of the admission port's opening. At this moment, the orifice is still small and the piston moves backwards relatively fast. Further to the center of the displayed picture, the piston's velocity decreases continuously until it becomes zero in the center. There, the port is completely open. In this period, when the piston is close to the bottom dead center, the admission is largest. Further to the right, the piston starts accelerating forward. Therefore, the orifice of the admission port becomes increasingly obstructed again. The plot shows that the compression of the gas to pressures above admission pressure starts before the admission port is closed completely. This causes backflow through the admission port, as discussed below.

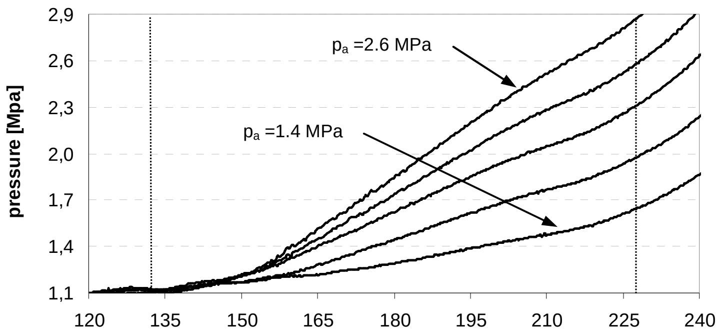


**Figure 6:**
**Admission process at different admission pressures**

## 4.3 Indicator Diagram

Figure 7 shows two sets of curves with admission pressures between 1.4 MPa and 2.6 MPa. The compressor's admission clearance of the left hand diagram is 0.125, the one of the right hand diagram is 0.094. The pressure courses are plotted against the scaled piston position. This type of diagram is called indicator diagram. The scaled admission clearance is drawn as a dotted line.

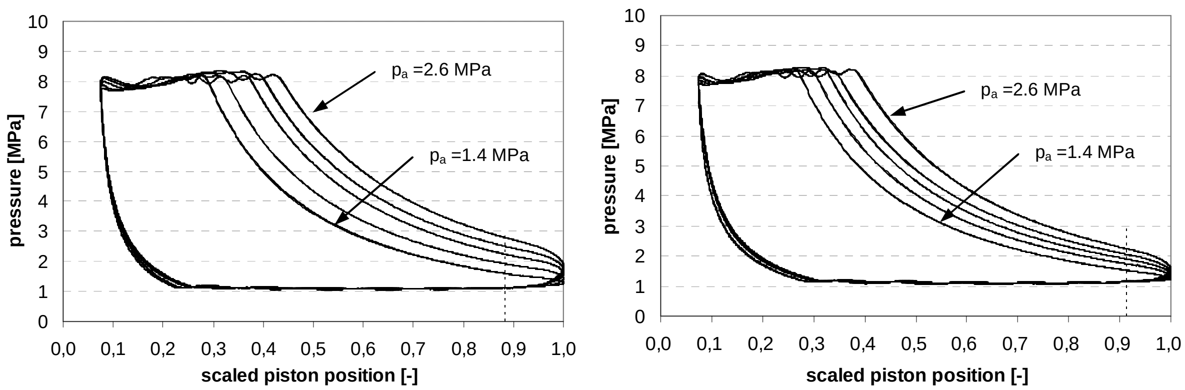


**Figure 7: Superimposed indicator diagrams with admission pressures between 1.4 MPa and 2.6 MPa, the scaled admission clearance is 0.125 left and 0.094 right**

The indicated courses are heavily curved in contrast to the ideal course as illustrated in Figure 2. This can be attributed to the flow losses of the admission process. The curves with high admission pressures of the compressor with a scaled admission clearance of 0.094 are remarkably jolted compared to the corresponding curves of the compressor with a clearance of 0.125. This can be explained by the flow resistance of the almost 50 % smaller flow area of the admission port. Additionally, the reduced opening period has a negative impact.

The indicator diagrams of all three modified compressors are compared in Figure 8. The left diagram shows the pressure curves at an admission pressure of 1.4 MPa. The curves are almost completely superimposed at low admission pressure. That changes with increasing admission pressure as visible in the right diagram which shows the same constellation with an admission pressure of 2.6 MPa. The lines of the two compressors with a scaled admission clearance of 1.25 indicate a larger admission as the one with a clearance of 0.094. The one with the tangential drill hole seems to be slightly superior to the one with the radial hole.
The fan-out of the three pressure courses at the end of the re-expansion can probably be attributed to differences in clearance volume of the three prototype compressors. However, the diagrams have been plotted with the same clearance volume, which is extended due to the hole for the pressure sensor. This was done to make a visual comparison possible.

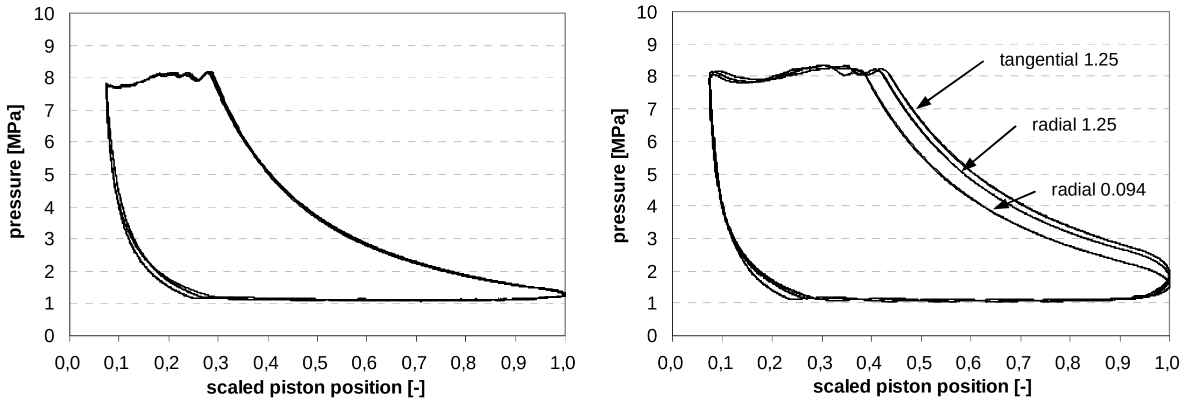


**Figure 8: Comparison of the indicator diagrams of the three modified compressors to the left: admission pressure = 1.4 MPa, to the right: admission pressure = 2.6 MPa**

### 4.4 Application Based Optimization of the Admission Port

This leads to the conclusion that each operating condition has its own optimum geometry. The operation condition depends on the evaporation temperature and on the ambient temperature. These temperatures as well as the system's characteristics define the suction and the admission pressure.

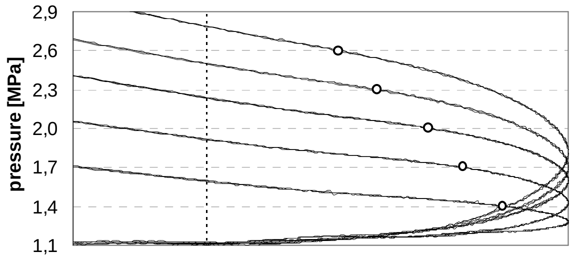


**Figure 9:**

**Piston positions when particular admission pressures are reached**

Figure 9 shows the relevant section of the indicated pressure courses of the baseline compressor. As before, each course corresponds to one admission pressure between 1.4 MPa and 2.6 MPa. The intersection of the admission pressure lines and the related pressure course is marked with a dot. Consequently, each dot shows the piston position were the admission pressure is reached in the compression chamber, i.e. the admission process is completed. Further piston movement with an open admission port causes backflow. In the ideal case, the dot is exactly on the dotted line indicating the upper edge of the admission orifice. The admission characteristics of one modified compressor for one suction pressure can be described by a line connecting the dots. The intersection of this line with the dotted line represents the admission pressure at which the analyzed modified compressor runs optimally for the given suction pressure. For the given compressor and a suction pressure of 1.2 MPa, the optimal operating admission pressure can be estimated to 3.6 MPa by extrapolation.

The pressure difference across the admission port can be calculated by subtracting the pressure measured in the compression chamber when the admission port closes from the admission pressure that is measured at the outside

connector of the admission port. This differential pressure can be drawn against the admission pressure as shown in Figure 10. A positive differential pressure indicates a too large admission clearance. This means that the effective suction volume of the compressor is reduced unnecessarily. Additionally, it causes backflow.
A too small admission clearance and therefore a too high flow resistance results in a negative pressure difference. Thus, the potential of the economizer principle can not be used to full capacity. Consequently, the pressure difference should be close to zero at operation conditions.

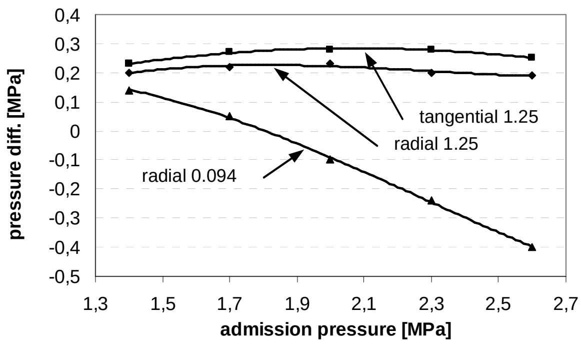


**Figure 10:**
**Readouts of differential pressure sacross the admission port**

The tangential admission bore hole seems to be slightly superior to the radial admission bore hole. The differential pressure of the tangential version is higher at the same admission clearance and the same admission pressures. This allows the assumption that the tangential version could perform equally to the radial version at a reduced admission clearance. However, the compressor specimens are prototypes with some variations in manufacturing. Furthermore, the indeterminate oil circulation rate influences the measurement results. That's why the difference is expected to be within the magnitude of the uncertainty of the measurements. For a final judgment, the heat transfer behavior of the admission port needs to be considered as well. The measured mass-flows are used to evaluate the overall performance of the compressor modification.
The differential pressures of the two compressors with a modification with a scaled admission clearance of 0.125 are positive in the entire relevant range. This leads to the conclusion that the diameter of the bore should be reduced for application at the targeted operation conditions. Based on the assumption of high quality heat exchangers and low ambient temperature, the scaled admission clearance of 0.094 seems to be appropriate. In the case of heat exchangers with a lower capacity and higher ambient temperatures, a scaled admission clearance seems to be optimal in the range of 0.1 – 0.125. It is expected that the admission clearance can be reduced by enhanced port geometries.

## CONCLUSION

The method of admission port evaluation by indicator diagram analysis is applicable as measurements have shown. It can be used to evaluate the suitability of a modified compressor to a certain application. The geometry can be improved by iteration of prototyping and measurements. It can also be used to map the characteristics of a particular admission port geometry. By means of a mapping described by the admission and suction pressure, a geometry and its scaling can be chosen for a certain application.

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