Implementation of scroll compressors into the Cordier diagram

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Abstract. Selecting the most suitable machine type for a specific compression task is an essential step to ensure a reliable and economical operation of the overall system. For this, the eligibility of a compression principle can be estimated by the enhanced Cordier diagram by Grabow based on only a few input parameters (specific isentropic work, flow rate, speed and characteristic diameter). The working principle of scroll machines is known for more than one century. However, only the improvements in manufacturing processes during the last decades allowed their utilisation and a significantly increasing interest in scrolls in more and more fields of applications can be seen. Unfortunately, the scroll machine has not yet been introduced to the Cordier diagram.

Therefore, this paper shows a method for determining the characteristic key figures for different given scroll geometries. Based on this, it is shown how to implement scroll compressors into the Cordier diagram, i.e. different designs are analysed and the resulting data points are introduced to the diagram.

1. Machine conception supported by empirical data

During the process of designing fluid machines empirical data gives the first hints for the selection of the principle. For the field of compressors this data is listed in the diagrams according to Bláha, Cordier [1] or the extended Cordier diagram from Grabow.

Starting with the Bláha diagram [2] the most suitable operating principle can be figured out with the help of characteristic values, see Figure 1. On the y-axis of the diagram, the specific work per stage \( Y \) can be found and on the x-axis the product of the volume flow \( \dot{V} \) and the rotational speed \( n \) are listed. The sections for pumps and compressors can easily be detected and a first approach for the design of new machines can be taken.
The Cordier diagram [1] is another tool for the conception of fluid machines. It provides the relation between the specific speed $\sigma^*$ and a specific geometric value which is called specific diameter $\delta^*$. (1) and (2) show the dependencies of the specific values from the volume flow $V$, the specific work per stage $Y$, the rotational speed $n$ and a characteristic diameter $D$. For continuous-flow machines $D$ is the rotor diameter.

\begin{align}
\sigma^* &= 2 \cdot \sqrt{\pi} \cdot V^{0.5} \cdot (2 \cdot Y)^{-0.75} \cdot n \\
\delta^* &= \frac{\sqrt{\pi}}{2} \cdot (2 \cdot Y)^{0.25} \cdot V^{-0.5} \cdot D
\end{align}

Until the year 1993 the Cordier diagram was only limited to continuous-flow machines. To enable an overview over all compression and pumping methods, an extension was necessary. Based on the work of Grabow [4] the aim of this research work is the first approach to implement the scroll compressor into the extended Cordier diagram.

2. Reasons for the implementation of scroll machines into the extended Cordier diagram

2.1. The Extended Cordier diagram

Grabow extended the known Cordier diagram for turbomachines by side-channel and displacer machines as well as combustion engines [4]. This offered the opportunity to combine the most important classification numbers of a large number of operating machines for a further development. Favourable technical and economical solutions can be found in less time with the background of the collected empirical data.
Figure 2. Extended Cordier diagram by Grabow [4].

The presentation of the main known energy machines in the same diagram gives the opportunity to achieve a systematic evaluation of the conditions during the energy transmission process. Figure 2 shows the extended Cordier diagram with an almost linear behaviour (dashed pink line) of the positive displacement machines.

For each displacement machine the specific characteristic value has to be determined. For piston compressors the piston diameter \( d_p \) is required as characteristic diameter \( D \) for (2). For other principles the derivation of \( D \) is described in [4].
Nevertheless, not all machines are implemented in this diagram so far.

2.2. Scroll compressors gain in importance

The principle of scroll compressors is known since the beginning of the 20th century. The geometrical basis for almost every scroll compressor is the Archimedean spiral. The basic equations will be explained in section 2.3. First patents appeared e.g. in 1886 by G. Pelizzola [5] or the most famous one in 1905 by L. Creux [6]. Due to the challenging shape of the scroll set and the necessary manufacturing accuracy to achieve the required tolerances, the concept was put aside first. Only due to the improvement of manufacturing technology during the last decades, it is possible to produce scroll compressors in an economically reasonable way and to achieve efficiencies that are comparable to so far more familiar compression principles. Nowadays, scroll compressors are used for both refrigeration and process gas compression systems (see [7], [8]).

The question why the scroll compressor has to be implemented in the Grabow diagram can be answered very quickly evaluating the development potential of scroll machines over the last 10 years. Figure 3 shows the development of worldwide approved patents for the different compressor options with the mentioned names of the principles either in the title of the patent or in the abstract. It can be seen that the scroll is the principle of interest. This behaviour can also be found for other search criteria like publications per year in different conferences or online libraries.

![Figure 3](image)

In stationary as well as in transport refrigeration systems the scroll compressor gained more and more influence.
Figure 4. Worldwide sales of refrigeration compressor types 2017, according to [10].

Figure 4 shows an overview of sold compressor principles published in 2018. Regarding these values, it is reasonable to provide the opportunity to compare the scroll compressor to other principles.

2.3. Scroll compressors – principle and design

The principle of the scroll compressor can be described as follows: Two spiral profiles are interlaced while one spiral is fixed and the second one describes a circular orbit. The gas is sucked in at the outer circumference of the scroll set at maximum volume. The gas is compressed in between the fixed and the orbiting scroll. At maximum pressure the gas is discharged at the centre of the scroll set, see Figure 5.

The scroll compressor provides a number of advantages:
- continuous and steady compression
- no valves are required due to the edge-controlled compression principle
- only a small amount of moving components
- favourable mass balance
- comparably insensitive against liquids
- favourable efficiencies at high rotational speeds
- lower relative speed compared to piston compressors

Figure 5. Compression principle of scroll compressor.

On the other hand, several challenges also need to be addressed, like the internal leakage as dominant loss component, the built-in volume ratio and the high manufacturing tolerances. Due to the position of the discharge port being in the centre of the geometry, the heat dissipation of the hot compressed gas and the resulting thermal expansion of the compressor material is a problem. For these reasons several experimental and numerical investigations (e.g. [11]) were conducted.
To increase the thermal efficiency and the internal sealing, oil can be injected in high quantities. The resulting effects in the compressor and at specific critical positions have to be examined [12]. Generally, the modern manufacturing technologies as well as certain concepts for a pre-outlet or part-load operation of scroll compressors can deal with the mentioned challenges.

The geometry of the scroll compressor derives from the calculation of the involute.

\[ r = r(\varphi) = K \cdot \varphi^m + r_b \]  

In (3) \( r \) represents the actual radius depending on the angle \( \varphi \), \( K \) is a linear factor, \( m \) is the spiral exponent and \( r_b \) is the radius of the base circle, see Figure 6.

For a detailed description of the necessary equations for the construction of scroll compressors please refer to the literature [13].

3. Definition of Characteristic Dimension

3.1. Characteristic Diameter

Based on Grabow's statements [4], the pressure figure \( \Psi \) and the flow rate \( \varphi^* \) form the basis for the comparison of individual machine types. These two characteristic values result in a relationship to the so-called characteristic diameter \( D \) for turbomachines. Based on these three variables, the similarity coefficients of diameter and speed are defined as \( \sigma^* \) and \( \delta^* \) (see equations (1), (2)).

These characteristic values are now transferred to the field of the displacement machines and the respective characteristic diameters are derived. Depending on the geometry of the respective machine, different characteristic values result, which can be taken from the literature [4]. This procedure has not yet been carried out on the scroll principle and is now explained in this paper.

A detailed calculation tool for the design of scroll compressors was developed at the Technische Universität Dresden including e.g. different geometry options for the spirals. Next steps are the consideration of leakage losses, heat flows to the walls and varying wall thicknesses.
The maximum enclosed area between the two scroll spirals (see Figure 7) is equated to the area $A_{\text{ref}}$ of a circle with a diameter which corresponds to the characteristic diameter $D$. Since the height $h$ of the enclosed volume $V_{\text{ref}}$ is always constant, the correlation can be described according to (4).

$$\frac{\pi}{4} \cdot D^2 = A_{\text{ref}} = \frac{V_{\text{ref}}}{h}$$

In the case of given suction volume and height of the scroll spirals, the characteristic diameter can be calculated. To create the first basic values for the implementation of the scroll compressor in the extended Cordier diagram, several compressors of different sizes have been investigated and measured. The four relevant geometrical values, wall thickness $t_W$, wall distance $D_W$, height of scrolls $h$ and circumferential diameter $D_C$ have to be determined (see Figure 8). This paper shows the results of six examples of different manufacturers and sizes.

![Figure 8. Geometrical values of scroll compressors.](image)

At this point the derivation for the geometrical design of scroll spirals is renounced and it is referred to the literature, e.g. [13].

![Figure 9. Geometric values of scroll geometry.](image)

Based on the values of the base circle diameter $D_b$, the inner base angle $\varphi_{\text{fib}}$, the orbit diameter $D_0$ and the wrap-around angle $\varphi_e$ (see Figure 9), the area which is enclosed by two symmetrical spirals can be calculated according to
\[ A_{\text{ref}} = \frac{\pi}{4} D_b \cdot D_o \cdot (2 \cdot \varphi_e - 3 \cdot \pi - \varphi_{\text{fib}} - \varphi_{\text{ob}}) \]  \hspace{1cm} (5)

Combining (4) and (5), the characteristic diameter for scroll compressors can be calculated:

\[ D = \sqrt{\frac{4 \cdot A_{\text{ref}}}{\pi}} = \sqrt{D_b \cdot D_o \cdot (2 \cdot \varphi_e - 3 \cdot \pi - \varphi_{\text{fib}} - \varphi_{\text{ob}})} \]  \hspace{1cm} (6)

\( D \) represents the input for the specific diameter for the implementation of scroll compressors in the extended Cordier diagram according to (2). It should be noted that this formula only applies to scroll compressors with constant wall thickness and constant wall height. In other cases, the characteristic diameter must be calculated taking these additional factors into account.

3.2. Specific speed

So far, no measurements of operation points can be provided from the mentioned scroll compressors. Based on the data sheets several assumptions were made for the calculation of the specific speed. For the solution of (1) an isentropic enthalpy difference of 30 kJ/kg, a speed of 50 Hz and a volumetric efficiency of 0.9 was assumed.

3.3. Input values

Table 1 shows the main geometric values as well as the results for the specific speed and the specific diameters. It has to be mentioned that the Models 5 and 6 have a variable wall thickness. The values are mean values and the applicability of (6) needs to be investigated further.

**Table 1. Input parameters and specific parameters.**

| Scroll profile            | Model 1 | Model 2 | Model 3 | Model 4 | Model 5 | Model 6 |
|---------------------------|---------|---------|---------|---------|---------|---------|
| Wall thickness \( h \) [mm] | 4.1     | 4.6     | 2.8     | 3.6     | ~4.5    | ~3      |
| Wall distance \( D_W \) [mm] | 14.0    | 15.5    | 8.9     | 12.4    | ~14.0   | ~13.7   |
| Profile height \( t_p \) [mm] | 32.5    | 33.4    | 24.9    | 36.6    | 33.2    | 20.9    |
| Circumferential diameter \( D_C \) [mm] | 103.2   | 96.0    | 75.8    | 90.8    | 98.8    | 71.0    |

Specific speed \( \sigma^* \) [1]  
1.9E-3  1.97E-3  1.18E-3  1.31E-3  1.90E-3  1.86E-3

Specific diameter \( \delta^* \) [1]  
1.29E+1  1.28E+1  1.48E+1  1.61E+1  1.28E+1  1.22E+1

4. Introduction of Scroll Compressors into the Cordier diagram

At the Technische Universität Dresden the first steps of the implementation of the scroll compressor into the extended Cordier diagram have been taken. As shown in Figure 2 the examples for positive displacement machines are almost perfectly aligned. The scroll compressor is also a member of the positive displacement machines. Hence it is assumed that also the scroll machine can be embedded in this behaviour. The results are shown in Figure 10. For a better orientation the trend line for the so far implemented positive displacement machines is highlighted again as a dashed red line – comparable with Figure 2.

It can clearly be seen that the investigated compressors fit to the behavior of positive displacement machines in general. The results can shift depending on the change of speed and the isentropic enthalpy difference. Also, the assumed volumetric efficiency has an impact on the position of the scroll compressors in the extended Cordier diagram. These changes would result in a shift of the results in parallel to the trend line of positive displacement machines.

Hence, the next steps in this context are the experimental analysis of the compressors at different boundary conditions and an ongoing literature survey.
Figure 10. Extended Cordier diagram with analysed scroll compressors.

5. Conclusion and Outlook
To enable an updated overview over the commonly used compression principles, the implementation of scroll compressors in the already extended Cordier diagram is necessary. This enables the possibility to compare the different principles concerning an evaluation of the transfer of energy. Based on the known similarity numbers for continuous-flow machines defined by Cordier, the characteristic diameter for scroll compressors could be defined. With this approach an easier access in the choice of a reasonable compressor principle and the first step for the design of an appropriate scroll geometry is provided. The next steps are further experimental investigations on other scroll compressors of different sizes to proof the calculation principle shown in this paper.

At this point manufacturers of scroll compressors are more than welcome to contribute data for the improvement of the extension of the Cordier diagram considering scroll compressors.
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