Comparison of thermodynamic efficiency between constant, dual and multiple lead rotors for an industrial air screw compressor

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Abstract. The goal of the European Union funded project “MOTOR” (Multi-ObjecTive design Optimization of fluid eneRgy machines) is the optimisation of fluid energy machines by using isogeometric analysis. This paper deals with the optimisation of the geometry of screw compressors. Screw compressors are usually built using helical rotors with constant lead. However, variable rotor lead holds the promise of higher efficiency due to an optimised progression of chamber volume, clearance, and discharge areas. This is especially true for high pressure applications and for working fluids with low isentropic exponent. This paper describes the optimisation of a variable lead rotor pair for an industrial air screw compressor. The rotors are divided into a discrete number of segments of constant lead, named multiple lead. Using a high number of rotor segments allows a design that approaches a uniformly varying lead. Thermodynamic simulation of the machines is performed using multi-chamber simulation with the simulation tool “KaSim”. The thermodynamic simulation is coupled with a Nelder Mead algorithm, which evaluates the efficiency of the machines and optimises the progression of rotor lead. The achievable efficiency of this optimised multiple lead machine is compared with constant and dual lead machine configurations. Results of this study show that the highest efficiency can be achieved by using dual lead rotors.

1. Introduction
The shape-optimisation of fluid energy machines, e.g. turbines and compressors, is a challenging task due to the complexities in geometry and numerical simulation involved. The industrial design process is usually performed using CAD systems, whereas the numerical simulation of fluids and solids is performed using CAE tools. The independent development of the respective research fields led to different data formats, requiring a conversion between them, which is always linked to an approximation and therefore to a loss in accuracy. Therefore, the vision of the Horizon 2020 project MOTOR is to reduce the approximation error resulting from the conversion steps and, at best, to eliminate it completely by linking all CAE tools involved in the design and development pipeline to a common and highly accurate master geometry model. To achieve this vision, MOTOR makes use of novel CAx technologies like isogeometric Analysis (IgA) [1], which tries to adopt the same mathematical approach (B-Splines or NURBS) to describe the geometry in the design and the simulation tools. This paper
focusses on the optimisation process of the rotor wrap angle for an industrial air screw compressor. The optimisation approach is shown in figure 1. The rotor profile is described by a two-dimensional point cloud, which is used to generate a zero-dimensional chamber model. This chamber model is used to perform a thermodynamic simulation for user-specified boundary conditions. The results are input to an optimiser, which adjusts the geometric parameters in order to increase efficiency. The simulation and optimisation loop is repeated until the objective of maximum efficiency is realised, a converged case, or no such result is found within a specified number of iterations, a failed case. A meaningful chamber model simulation should provide reasonably precise calculation of mass flow exchanges in clearance and port areas. The determination of these flows is pictured on the right side of figure 1, where the geometry of the screw machine is used to perform more detailed flow simulation of the flow paths. The full IgA-based design and analysis of screw machines is not yet possible with the available CAE tools, but essential concepts from the IgA approach have been integrated into the workflow to realise the concept of a master geometry model from which both geometry parameterisations for IgA simulations and highly accurate computational meshes for classical CFD simulations can be easily generated. The construction of the master geometry model by the IgA approach is addressed in Hinz et al. [2] and the generation of computational meshes is presented by Möller et al. [3]. The utilisation of these meshes in CFD simulation is performed by Utri et al. and presented in [4]. Shamanskiy and Simeon present a coupling to the solid parts of the machine in order to estimate thermal expansion [5]. This paper focusses on the description of the left side of the presented optimisation approach, where the chamber model based optimisation of the screw machine geometry is performed.

**Figure 1.** Approach for the optimisation of screw machines

**Figure 2.** Constant (left), dual (middle) and variable (right) lead

Industrial screw compressors are usually built with constant rotor lead (figure 2 left). However, recent experience reveals that use of a non-constant rotor lead has a positive impact on compressor efficiency. Probably the easiest way for a practical realisation of this new kind of machine is the combination of two different lead segments on one rotor (so called “dual-lead”, figure 2 middle), a configuration
investigated by Fost [6] and by the authors [7, 8]. Other contributions [9, 10] examine a continuous decrease in lead between the low and high pressure ends of the rotor (figure 2 right). Both approaches show promising results in terms of efficiency. In this paper, an optimisation of the rotor lead distribution is performed for an industrial air compressor in order to reveal which concept is most effective and to give practical advice for screw compressor design.

2. Concept for variable rotor lead
The lead of a screw machine rotor has great impact on important geometric properties (clearances and discharge port area) and should be matched to the intended compression task. High rotor lead (low wrap angle) usually implies a rapid compression, lower clearance areas and maximum possible chamber volume. However, it is associated with a small discharge port area. In contrast, low rotor lead (high wrap angle) leads to a longer duration of the working cycle, larger clearance and a beneficially large discharge port area. The wrap angle, \( \phi \), is defined as the angle of twist between front and rear ends of the rotor. The concept of variable rotor lead presented in this paper is to define a rotor using discrete segments with different leads in order to realise an appropriate balance of advantages of high and low lead configurations. An example of a multiple lead rotor is shown in figure 3. This rotor consists of three lead segments of equal length. The overall wrap angle is defined as the sum of the wrap angles of the separate rotor segments; the wrap angle associated with a rotor pair is usually that of the male rotor:

\[
\phi = \sum_{i=1}^{n} \phi_i
\]

The following investigations are limited to rotors of constant overall length with individual lead segments of equal length, as shown in figure 3. A multiple lead rotor with a large number of segments is almost equivalent to a rotor with a continuous change in rotor lead. The number of rotor segments and the overall wrap angle are varied in this study.

3. Chamber model simulation
Thermodynamic simulation is performed using the multi-chamber simulation tool “KaSim” [11]. The simulation requires a chamber model which contains all relevant geometric properties of the screw compressor under examination. This requires a detailed analysis of the geometry of the screw machine, this being performed in a series of two-dimensional cross sections which are then combined to describe

![Figure 3: Screw machine with three lead segments (n = 3)](image-url)
the three-dimensional machine under investigation. It is the rotation angle-dependent progression of volumes and areas that are provided to the thermodynamic analysis.

Chamber model simulation is based on mass and energy conservation with the assumptions of an adiabatic process with a homogeneous distribution of pressure and temperature inside the working chambers. The simulation used here considers all working chambers simultaneously and has been used and validated for several use cases. The simulation process is divided into two consecutive steps: a change of chamber volume and an exchange of mass in the form of leakage and port flows. Mass flow is determined assuming a theoretical isentropic, frictionless flow \( \dot{m}_{th} \). This flow is then adjusted using a flow coefficient, \( \alpha \), afterwards to approximate real flow conditions, equation (2):

\[
\dot{m} = \alpha \cdot \dot{m}_{th}
\]  

(2)

The flow coefficient depends on the geometry of the flow path and on the thermodynamic states within the connected volumes. Determination of the flow coefficient for a certain geometry is discussed in more detail in [4].

The results of the chamber model simulation are, amongst others, delivered mass flow, \( \dot{m} \), and indicated power, \( P_i \). These can be used together with the isentropic power, \( P_s \), which would be necessary for an isentropic compression, to evaluate the indicated isentropic efficiency of the compressor:

\[
\eta_{is} = \frac{P_s}{P_i} = \frac{\dot{m} \cdot \Delta h_s}{P_i}
\]  

(3)

This efficiency is used by the optimiser to adjust the wrap angle parameters of all segments.

4. Optimisation principle

The geometry of the screw machine is optimised for application specific, user specified boundary conditions. In addition to the wrap angle parameters, the compressor’s internal volume ratio, defined in equation (4), is used as a factor in the process.

\[
v_i = \frac{V_{\text{max}}}{V_{\text{compr,end}}}
\]  

(4)

The internal volume ratio affects the duration of the compression and discharge phases as well as the size of the discharge port and needs to be selected based on screw machine geometry and application conditions. For a given set of rotor parameters, the optimisation of \( v_i \) is an easy search for a global maximum in efficiency. An example of this is shown in [8]. In contrast, optimisation of wrap angle is less trivial. The free wrap angle parameters of all segments lead to thousands of possible rotor geometries. An optimisation loop is used in order to reduce the necessary number of simulations to find the best rotor geometry. The optimisation algorithm used is the downhill simplex algorithm [12], also called the Nelder Mead algorithm. This gradient free method can optimise problems with multiple input parameters.

5. Industrial air compressor configuration and boundary conditions

The industrial compressor under investigation is a single-stage wet-running air compressor in the medium power range. Oil is injected into the compressor and serves to seal clearances. The compressor under examination can operate at a maximum outlet pressure of 16·10^5 Pa and runs at a circumferential speed of up to about 50 m/s. The boundary conditions, which remain constant for all simulations, are shown in table 1. Since the machine is wet-running, the heat of compression is partly absorbed by the injected fluid. Instead of an additional simulation of this fluid, the isentropic exponent of the homogeneous representation of the actual fluid conditions used in the simulation is adjusted so that the outlet temperature matches the temperature measured in experiments. Atmospheric conditions prevail at the inlet. The operating parameters selected for the study are a compression ratio of 10:1 with the compressor operating at its maximal circumferential speed. The rotors have an L/D-ratio of 1.6 and use a profile with four male and six female rotor lobes.
Table 1. Constant boundary conditions

| Parameter [dimension]               | Value |
|------------------------------------|-------|
| Fluid parameters:                  |       |
| - Specific gas constant [J/kg/K]   | 287   |
| - Isentropic exponent [-]          | 1.1   |
| - Compressor inlet temperature [K] | 293   |
| - Compressor inlet pressure [Pa]   | $10^5$ |
| - Compressor outlet pressure [Pa]  | $10^6$ |
| Machine parameters:                |       |
| - Male rotor diameter [mm]         | 150   |
| - Rotor length [mm]                | 238   |
| Operating parameters:              |       |
| - Circumferential speed [m/s]      | 50    |

Table 2 shows parameters varied in this study. The number of lead segments is varied between 1 (which is equivalent to constant rotor lead) and 15. The wrap angle values of each segment are determined by the optimiser. The overall wrapping of the rotors is also varied and restricts the lead distribution in accordance with equation (1). The optimisation process is performed for different clearance heights and flow coefficients, which affect mass flow through clearances and ports. The flow coefficients are assumed to be the same for all clearances and port areas and do not vary with geometry or operating conditions. Two different clearance heights are investigated; these are applied to the inter-lobe, housing and front clearance. The blowhole area is determined based on the profile geometry and rotor lead.

| Parameter [dimension]               | Minimum / maximum value |
|------------------------------------|-------------------------|
| Number of segments n [-]           | 1 / 15                  |
| Overall wrap angle $\phi$ [°]      | 200 / 350 (variable and dual lead) | 200 / 500 (constant lead) |
| Clearance height $h$ [mm]          | 0.025 / 0.075           |
| Flow coefficient $\alpha$ [-]      | 0.5 / 0.7               |

6. Optimisation results

6.1. Optimisation of multiple lead - variation of the number of segments

Optimisation is performed by varying the number of segments used to define the multiple lead rotor set in order to show the global optimal geometry. Dual lead rotors are treated in a later section and are not comparable to multiple lead rotors with two segments since the segment length is different. Figure 4 shows optimisation results where indicated isentropic efficiency is maximised with the parameters being overall wrap angles and number of segments. This study is further defined by use of the smallest values of clearance heights and flow coefficient. Figure 5 shows the related rotor geometries for an exemplary wrap angle of $\phi = 275^\circ$; figure 6 shows the corresponding progressions of volume and optimised discharge area. The related indicator diagrams are shown in figure 7.

For a single rotor segment (which represents constant rotor lead), wrap angle has a large impact on the efficiency of the compressor. In general, outlet area is small for low wrap angles, leading to significant discharge throttling. For greater wrap angles, discharge throttling is reduced; the increased clearance areas do not affect efficiency significantly since clearance height and flow coefficients are set to small values in all simulations. A significant increase in efficiency is possible by splitting the rotor into two equal length segments ($n = 2$). The corresponding optimised rotor geometry, shown in figure 5, provides 80 percent of the overall wrap angle on the high pressure side. As shown in figure 6, the maximum chamber volume is independent of the wrap angle progression; rather, it is only influenced by the overall wrap angle [8]. When comparing constant lead with two lead segments, the two segments...
configuration leads to a strong impact on volume curve and discharge area. The reduction in chamber volume during compression is much greater for the optimised variable lead screw machine due to the low wrap angle of the low pressure side segment. As a result, the chamber volume when discharge begins is reached at earlier rotation angles, leading to an increased discharge area and longer discharge duration, even though the value of the optimised internal volume ratio is increased compared to constant lead. Together with the flat progression of the volume curve during discharge, this leads to a lower discharge volume flow rate and reduces the risk of throttling. This relation can be explained by referring to the indicator diagrams, shown in figure 7, which are used to calculate the indicated compressor power. The diagram shows that the highest pressure inside the screw machine is found with constant rotor lead due to discharge throttling. The optimised internal volume ratio of this machine opens the working chamber to the high pressure port long before reaching the high pressure conditions. The resulting under-compression is less detrimental in terms of efficiency than the discharge throttling which would occur for a higher internal volume ratio.

![Graph showing indicated isentropic efficiency vs number of segments for different wrap angles](image)

**Figure 4:** Indicated isentropic efficiency of the optimised screw compressors for a variation of segment numbers and overall wrap angle

![Optimised screw machines with different segment numbers](image)

**Figure 5:** Optimised screw machines with $\phi = 275^\circ$ for different segment numbers (top: high pressure side, bottom: low pressure side)

For three rotor segments efficiency almost reached its maximum value. For $\phi = 275^\circ$ the rotor consists of a segment with high wrap angle on the high pressure side, a medium wrap angle on the low pressure side and a low wrap angle in between. In comparison with the rotor consisting of two segments, the medium wrap angle on the low pressure side leads to a decrease in throttling during suction. The low wrap angle of the middle segment maintains a fast compression and the high wrap angle on the high
pressure side leads to a gentle volume curve progression, so that the internal volume ratio can be increased without causing throttling during discharge.

![Figure 6: Chamber volume and discharge area for a fixed overall wrap angle of $\phi = 275^\circ$ for the optimised screw machines for different numbers of segment](image)

![Figure 7: Indicator diagrams for the optimised screw machines (including optimised $v_i$) for different numbers of segments for a fixed overall wrap angle of $\phi = 275^\circ$](image)

A further increase of the number of segments ($n = 5$ and $n = 15$) leads to almost the same results and reveals a characteristic shape of the optimised rotors. Although a continuous change in rotor lead (as shown in figure 2) from high to low pressure side would be possible as a result, the optimised rotors
consist mainly of two wrap angle parts, a high wrapping on the high and a low wrapping on the low pressure side. The results emphasise that it is sufficient in terms of efficiency to build a rotor consisting of two rotor parts with different lengths. The shape represents the “dual lead” rotors, which the authors previously examined for different fields of application [7, 8].

6.2. Optimisation of dual lead - variation of flow coefficient and clearance height

An examination of multiple lead in the last section revealed that properly selected dual lead rotors can provide optimum performance. Since two rotor parts with different lengths are enough to optimise efficiency, it is not necessary to optimise the screw machine with completely variable lead; that reduces the number of influencing parameters and simplifies the optimisation process. With respect to this finding, further results for non-constant lead are shown for dual lead setups only (two parts on one rotor with different lengths).

The optimal wrap angle and the resulting efficiency of screw compressors are strongly influenced by the clearance heights which can be realised. Practical decisions must be made considering the clearance heights which can actually be realised in practise. Furthermore, the flow coefficient is constant in this study, so that the influence of the flow coefficient should be examined. Clearance height and flow coefficient are therefore varied in order to show the possible enhancement for other boundary conditions.

Figure 8: Indicated isentropic efficiency of the optimised screw compressor as a function of rotor wrap angle for two flow coefficients for constant and dual lead (clearance height h = 0.025 mm, \( \alpha \) optimised for each point)

Figure 8 shows results for different flow coefficients with fixed small clearance heights. With higher flow coefficients port and clearance mass flows are increased. Due to higher port mass flow, the compressor is less at risk of discharge throttling, for which reason the optimal wrap angle decreases from 400 to 350 degrees for constant lead and from 300 to 275 degrees for dual lead. Due to higher clearance mass flow, maximal achievable efficiency is generally lower. By using the dual lead setup, discharge throttling can be reduced for lower overall wrap angles, for which reason the dual lead setup increases efficiency by three percentage points for the lower flow coefficient and by two percentage points for the higher flow coefficient. In addition to the higher efficiency, the dual lead setup increases the mass flow rate due to the lower overall wrap angle; this relation is explained in more detail in [8].

Figure 9 shows corresponding results for a higher clearance height, which again decreases overall efficiency. In comparison to the results from figure 8, the change in rotor lead of the dual lead rotors is
less intense. The enhancement in efficiency which can be achieved by using dual lead rotors is reduced and, as explained before, the optimal wrap angle values are reduced. The diagram emphasises that using dual lead rotors is especially beneficial where low clearance heights can be achieved e.g. in the manufacturing process or by using fluids with favourable properties, since the enhancement is reduced for higher flow coefficients.

![Diagram](image)

**Figure 9:** Indicated isentropic efficiency of the optimised screw compressor as a function of rotor wrap angle for two flow coefficients for constant and dual lead (clearance height \( h = 0.075 \) mm, \( \alpha \) optimised for each point)

7. **Manufacturing of prototype and outlook**

The enhancement in efficiency suggested in this paper will be validated by experiment. Therefore, we have taken into account the optimisation results concerning wrap angles and lengths of the rotor parts in design of a dual lead prototype. We plan to build and test an industrial air compressor with dual lead rotors, where the casing remains unchanged. Manufacturing dual lead rotors is a challenging task since they cannot be manufactured by grinding in one piece. Instead, the two lead parts need to be

![Image of prototype](image)

**Figure 10:** Dual lead male rotor before finishing with a two millimeter milling cutter on a five-axis milling machine (left) and finished dual lead rotor pair prototype made from aluminium (right)
manufactured separately and joined together afterwards. This is realised by grinding one part on the shaft and the other part on a ring. Another possibility to manufacture dual lead rotors is by milling on a five-axis milling machine, figure 10 left. We have used this approach in the workshops of TU Dortmund University to produce dual lead rotor prototypes. Manufacturing was first performed on aluminium rotors in order to investigate the possible dimensional accuracy. The prototype rotor pair (figure 10 right) shows promising results for the manufacturing of steel rotors, a step which is currently under way. The steel rotors will be investigated in a test rig at the facilities of GHH Rand in the near future and compared to the simulation results presented in this paper.

8. Conclusion
The present study examines the efficiency improvement that can be achieved in an industrial wet-running air compressor by using non-constant rotor lead. The optimisation is performed by chamber model simulation, coupled with a Nelder Mead optimisation algorithm. To realise variable rotor lead, the rotors are sectioned into discrete rotor segments. Each segment has a constant lead, so that an almost continuous progression of wrap angle distribution is possible by simulating a large number of rotor segments (multiple lead). The results reveal that dual lead rotors represent the optimised shape of variable lead. These rotors consist of two parts with different rotor lead and length. The increase in efficiency in comparison to constant rotor lead is up to three percentage points for the compressor under examination. However, this improvement needs to be opposed with the higher manufacturing costs of the dual lead rotors. The improvement depends to a great extent on the clearance heights which can be realised in manufacturing. A first dual lead rotor pair prototype made from aluminium has been realised. A steel rotor pair is currently manufactured and will be investigated in experiments in the near future.

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Nomenclature

- $h$: Clearance height (m)
- $\Delta h$: Specific enthalpy difference (J/kg)
- $\dot{m}$: Mass flow (kg/s)
- $n$: Number of segments (-)
- $P$: Power (W)
- $V$: Volume (m³)
- $v_i$: Internal volume ratio (-)
- $\alpha$: Flow coefficient (-)
- $\eta$: Efficiency (-)
- $\phi$: Overall wrap angle (°)
- $\phi_i$: Wrap angle of segment $i$ (°)

Subscript

- compr,end: End of compression
- i: Indicated
- max: Maximal
- $s$: Isentropic
- th: Theoretical

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