A study on the kinematics of a new Schukey-type rotary compressor

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Abstract. A new type of rotary compressor, called “rotary-chamber compressor”, consists of two interlocking rotors with 4 wings each, that perform non-uniform rotary movements. Both rotors have the same direction of rotation, while one rotor is accelerating, the other rotor is retarding. After surpassing a specific mark, the sequence changes and the leading rotor begins to retard and vice versa. Due to the resulting relative phase difference, the volume between the two wings is changing periodically, which allows pulsating working chambers. The technology was first introduced by its founder Jürgen Schukey in 1987. Since then, no further development on this machine is known to us except our own. In this contribution, a study on the kinematics of the rotary-chamber-compressor is presented. Initial studies have shown that changes in the kinematics of the rotors will have a direct influence on the thermodynamical variables, which, if optimized, can lead to an increased performance of the machine. Therefore, a mathematical model has been developed to obtain the performance parameters from different kinematic concepts by using numerical CFD analysis. Furthermore, additional optimization possibilities will be listed and discussed.

1. Introduction

In recent years, several novel compressor types are of interest with the focus to achieve the same or better performance levels compared to existing technologies at reduced costs, noise and vibration. They are typically rated on mass flow rate (MFR) and energy performance, given by the volumetric efficiency \(\lambda\) and isentropic or polytropic efficiencies [1]. Therefore, a variety of studies on novel rotary compressors [2–7], mostly on rolling piston types, have been carried out.

A new rotary type compressor called “rotary-chamber compressor” was first introduced by Jürgen Schukey [8]. He designed the machine to be operated as a compression or expansion engine for refrigeration and heat recovery applications. Since then, the interest of the machine was fading until in recent years, it has been gaining more attention to be used as a compressor technology for refrigeration cycles. The main components of the examined compressor consist of two interlocking rotors, each with four wings, which form the 8 moving chambers within the housing. The rotational movement of each rotor is specified through its own gearbox with special toothed gears, which allows the change of rotational speeds relative to each other. Both rotors rotate in the same direction, with one rotor accelerating and the other rotor retarding vice versa while the main shaft rotates constantly. As a result,
the chambers fluctuate in volume, which leads to its suction, compression and discharge movement. Depending on the position of the inlet and outlet ports, the movement allows the compression of the working fluid. Furthermore, the machine is working without suction and discharge valves, since the suction and discharge process is controlled by the position of the rotor wings and ports. Figure 1 shows the main components of the rotary-chamber compressor in an exploded view.

**Figure 1.** Exploded view of the rotary-chamber compressor.

The compression cycle is illustrated in Figure 2 with its three steps. Each cycle begins with the filling of the working chamber \( \gamma \) with the fluid through the inlet port (3). During the suction process, the leading rotor (1) is in its accelerating phase of its movement pattern while the complementary rotor (2) is retarding. In that case, the working chamber increases in volume and the fluid is sucked in. This step ends, when rotor blade (2) closes the inlet port during the counterclockwise rotation. In that moment, the chamber volume is at its maximum. After closing the inlet, rotor blade (2) starts accelerating while rotor blade (1) starts retarding and thus the chamber volume begins to decrease. Since the working chamber is not connected to any inlet- or outlet port, the compression takes place between rotor blade (1) and (2). After reaching its minimum volume, rotor blade (1) passes the outlet port (4) and the fluid will be discharged. From then on, rotor blade (1) starts to accelerate while rotor blade (2) starts to retard again. The discharge process ends when rotor blade (2) passes and closes the outlet port. From then on, the compression cycle starts again. The cycle takes places in all chambers simultaneously. Therefore, one revolution of the main shaft results in 32 compression cycles (8 chambers x 4 sections), which distinguishes the machine with a high volume turnover.

**Figure 2.** Compression cycle in three steps.

Preliminary studies have investigated the impact of different port positions and rotational speeds of the engine [9], showing that the focus should lie on reducing the minimum clearance volume to improve the volumetric efficiency. One way to do so, is to optimize the current kinematics of the rotors. Beforehand, the impact of compressor kinematics for reciprocating and rolling piston type compressors was examined by Pandeya [10], who concludes that discharge flow losses are significantly different by different compressor kinematics due to higher flow velocity and mass density during the discharge.
process. Yet, further motion analysis only was done for gerotor- [11], Wankel- [12] or rolling piston [13]-compressor types.

In this paper, different concepts of the rotor kinematics will be developed and presented, starting with the current function of this machine which was obtained through incremental measurements. Afterwards, continuous and discontinuous functions are developed and examined, also with regard to whether both rotors should have the same or whether each rotor should have its own individual kinematic function. In order to take effects of the momentum of the gas into account, each kinematic function is tested in the commercial CFD-Solver ANSYS CFX to determine the changes in mass flow rate MFR, volumetric efficiency $\lambda$, internal pressure ratio $\Pi$ and indicated work $W_I$. The goal of this contribution is the quantitative determination of the effect of various kinematic functions for this rotary piston-type compressor.

2. Mathematical Model /Methods

2.1. Current kinematic function and settings

The oscillating specific movement pattern for each rotor is defined by its own special toothed gears, which are installed with an offset of 45° to each other and connected to the main shaft. The main shaft itself is driven by an electric motor and rotates at constant speed. In contrary, the angular movement of the rotors are not constant, but oscillating and rotating faster and slower relative to the main shaft, which results in the accelerating and retarding movement of the rotor blades.

To determine the current kinematic function, incremental measurements of the angular velocity at the rotor blades were taken. Subsequently, the kinematic was determined by using a non-linear regression with the Levenberg-Marquardt-Algorithm to determine the coefficients $a$, $b$, $c$ and $d$ of eq. (1) and (2).

$$\omega_{R1}(\theta) = \left\{- a \cdot \left[1 - \cos\left(\frac{\pi}{45^\circ} \cdot \theta + b\right) + \frac{c}{2} \cdot \sin^2\left(\frac{\pi}{45^\circ} \cdot \theta + b\right)\right] + d\right\} \cdot k_b \cdot \frac{\omega_\theta}{\omega_N} + \omega_\theta$$

$$\omega_{R2}(\theta) = \left\{- a \cdot \left[1 - \cos\left(\frac{\pi}{45^\circ} \cdot \theta + b + \frac{\pi}{2}\right) + \frac{c}{2} \cdot \sin^2\left(\frac{\pi}{45^\circ} \cdot \theta + b + \frac{\pi}{2}\right)\right] + d\right\} \cdot k_b \cdot \frac{\omega_\theta}{\omega_N} + \omega_\theta$$

With $\omega_\theta$ as the (constant) angular velocity of the main shaft, $\omega_N$ being the normal angular velocity of the measurement and $\theta$ as the crank angle of the main shaft, the relative changes in angular velocity can be plotted shown in Figure 3, where each rotor passes four minima and four maxima during one revolution of 360° of the main shaft crank angle. As a result, the chamber angle and volume function can be derived (see eq. (3) and (4)), showing a congruent sine-like behavior in Figure 3. The maxima or minima of the chamber volume occurs during the intersection of the angular velocity curves. This is when the compression or the suction process ends. In addition, the geometrical dimensions of the machine are represented in Table 1.

Chamber Angle function:

$$\gamma(t) = \gamma(t - \Delta t) + \int (\omega_{R1,2} - \omega_{R2,1}) \, dt$$

Volume function:

$$V_c(\gamma) = l \cdot \frac{\pi}{360^\circ} \cdot \gamma (R_0^2 - R_1^2)$$
Figure 3. Function of (a) angular velocity $\omega$ of rotors and main shaft at $\omega_0 = \omega_N = 0.717 \text{ s}^{-1} = \text{const.}$, (b) chamber angle $\gamma$ with $\gamma_{\text{max}} = 32.82^\circ$ and $\gamma_{\text{min}} = 9.18^\circ$.

Table 1. Geometry parameters and dimensions of the present compressor.

| Parameter                             | Value  |
|---------------------------------------|--------|
| Min. chamber angle $\gamma_{\text{min}}$ | $9.18^\circ$ |
| Max. chamber angle $\gamma_{\text{max}}$ | $32.82^\circ$ |
| Min. chamber volume $V_{\text{min}}$  | 28.46 cm$^3$ |
| Max. chamber volume $V_{\text{max}}$  | 101.82 cm$^3$ |
| Piston displacement $V_H$             | 73.36 cm$^3$ |
| Total vol. turnover per revolution    | 2350 cm$^3$ |

2.2. Kinematic function approaches

The current kinematic function is showing a minimum chamber angle of $\gamma_{\text{min}} = 9.18^\circ$ which is equal to the minimum clearance angle and equivalent to the minimum clearance volume. This is due to the fact that the rotor width and rotor kinematics do not fit together perfectly. However, for a better clearance of the fluid and a possibly better performance of the compressor in general, the goal of developing new rotor kinematics is the reduction of $\gamma_{\text{min}}$ while maintaining a high volumetric efficiency with low backflow losses. Of course, it would be possible to set the minimum chamber angle to approximately 0$^\circ$ over a larger rotor width. As a result, however, the volume throughput itself would also decrease.

Therefore, innumerable kinematics can be developed, assuming every rotor motion can be realized. This leads to the question whether a continuous or discontinuous function is better in terms of performance parameters. It is also of interest, if both rotors should rotate with the same function but phase-shifted with periodical behavior so that every chamber fulfills the same movement, or if each rotor should have its own individual function. For each constellation, some functions have been developed, shown and categorized with their fictional name in Figure 4.
Same function for each rotor, phase-shifted | Individual function for each rotor
---|---
**Continuous function** | Amplitude, Flattened | Fourier
**Discontinuous function** | Linear, MixLinear | Bestfit

**Figure 4.** Types and names of the developed kinematic functions.

For the purpose of reducing the minimum clearance angle from $\gamma_{\text{min}} = 9.18^\circ$, all of the developed kinematic functions should indicate a minimum clearance angle of $\gamma_{\text{min}} = 2^\circ$. This is a freely selected minimum clearance angle, since an angle of $0^\circ$ is avoided due to safety factors, where the rotors could possibly hit each other.

Consequently, thus also increases the maximum chamber angle to $\gamma_{\text{max}} = 40^\circ$, which on the other hand results in a bigger piston displacement $V_p$ and an increase of the mass flow rate. For a better understanding on how some of the following functions were determined, the compression cycle is subdivided into the significant angular positions, shown in Figure 5. Both rotors start in the neutral rotor position, defining chamber 1 and 2 with the left and right chamber wall positions $\phi_{C1/2,L}$ and $\phi_{C1/2,R}$. By taking a look at the sketch in Figure 5, following associations can be derived

$$\phi_{C1,L} = \text{Rotor2}_R$$  \hspace{1cm} (5)
$$\phi_{C1,R} = \text{Rotor1}_L$$  \hspace{1cm} (6)
$$\phi_{C2,L} = \text{Rotor1}_R$$  \hspace{1cm} (7)
$$\phi_{C2,R} = \text{Rotor2}_L.$$  \hspace{1cm} (8)

The suction begins when the left chamber wall $\phi_{C1/2,L}$ passes the right inlet wall $\phi_{\text{In},R}$ and opens the chamber for the filling. The suction process ends when the right chamber wall $\phi_{C1/2,R}$ passes the left inlet wall $\phi_{\text{In},L}$ so that the chamber is not being connected with the inlet port. The discharge process on contrary begins when the left chamber wall has $\phi_{C1/2,L}$ reaches the right outlet wall $\phi_{\text{Out},R}$, allowing the fluid inside the chamber to leave through the outlet. Likewise, to the suction process, the discharge ends when the right chamber wall $\phi_{C1/2,R}$ passes the left outlet wall $\phi_{\text{Out},L}$, closing the chamber for the fluid to exit. With this in mind, several continuous and discontinuous functions can be determined and optimized.

**Figure 5.** Position of each rotor, inlet and outlet walls in its neutral position for the development of new kinematic functions.
2.2.1. Continuous functions
Continuous functions have the property to be defined at every point and undergo no interruptions, jumps, or breaks and can be easily described with mathematical equations. To find and adapt functions for our optimization problem, it is vital to determine the correct coefficients. The easiest way of determining a continuous function for our problem is to take the current angular displacement function and increase the amplitude, which results in a larger amplitude of the volume fluctuation and a smaller minimum chamber angle. In the following, the first developed function is called ‘Amplitude’ since only the amplitude $u_4$ for rotor 1 and $v_4$ for rotor 2 was increased, compared to the current function. The amplitudes were iteratively increased in a numerical scheme until the minimum chamber equals 2\(^\circ\). The angular displacement functions $f_1$ and $f_2$ for each chamber wall are shown in eq. 9 and 10.

$$f_1(\theta) = u_4 \cdot \left\{ \atan\left( \frac{\sin\left( \frac{\pi}{45^\circ} \cdot \theta + u_4 \right)}{u_3 - \sin\left( \frac{\pi}{45^\circ} \cdot \theta + u_4 \right)} \right) + \atan\left( \frac{\sin\left( \frac{\pi}{45^\circ} \cdot \theta + u_4 \right)}{u_3 - \sin\left( \frac{\pi}{45^\circ} \cdot \theta + u_4 \right)} \right) \right\}$$

$$f_2(\theta) = v_4 \cdot \left\{ \atan\left( \frac{\sin\left( \frac{\pi}{45^\circ} \cdot \theta + v_4 \right)}{v_3 - \sin\left( \frac{\pi}{45^\circ} \cdot \theta + v_4 \right)} \right) + \atan\left( \frac{\sin\left( \frac{\pi}{45^\circ} \cdot \theta + v_4 \right)}{v_3 - \sin\left( \frac{\pi}{45^\circ} \cdot \theta + v_4 \right)} \right) \right\}$$

Furthermore, another continuous function was developed by the superimposition of various functions showing some flattening behaviour and thus called ‘Flattened’, where both rotors also have the same kinematic but phase-shifted (see eq. 11 and 12).

$$f_1(\theta) = z_1 \cdot \atan (z_2 \cdot \sin(\frac{\pi}{45^\circ} \cdot \theta))$$

$$f_2(\theta) = -z_1 \cdot \atan (z_2 \cdot \sin(\frac{\pi}{45^\circ} \cdot \theta))$$

One continuous function, where each rotor has his own individual kinematic function, has been developed and is named ‘Fourier’, inspired by the Fourier series.

$$f_1(\theta) = w_1 \cdot \sin\left( \frac{\pi}{45^\circ} \cdot \theta \right) + w_2 \cdot \sin\left( 2 \cdot \frac{\pi}{45^\circ} \cdot \theta \right)$$

$$f_2(\theta) = -w_1 \cdot \sin\left( \frac{\pi}{45^\circ} \cdot \theta \right) - w_2 \cdot \sin\left( 2 \cdot \frac{\pi}{45^\circ} \cdot \theta \right)$$

The coefficients $z_1$ and $z_2$ and $w_1$ and $w_2$ were also determined iteratively by a numerical scheme, where both values were iterated to match the minimum clearance angle of 2\(^\circ\). The only difference between them both is, that during the determination of the coefficients for the Fourier function only one chamber was considered, and the other chamber was being neglected according to the suction and discharge process shown in Figure 5.

2.2.2. Discontinuous function
Discontinuous functions indicate breaks, jumps and abrupt changes in their behaviour, mostly being described with mathematical equations for a given definition range. In this study, three discontinuous functions have been developed which indicate linear behaviour and holding phases during the rotor movement. The functions are called ‘Linear’, ‘MixLinear’ and ‘BestFit’ and were determined according to Figure 6. They represent the case that the transmission gears are able to perform the discontinuous functions.

At first, a best-guess function is used as the input for the optimization. From that, each timestep is obtained, where the left and right inlet and outlet walls are passed by each chamber wall. Subsequently, those are the characteristic timemarks for which the discontinuous functions will be fitted in. If the right chamber wall passed the right outlet wall, the leading rotor with the left chamber wall starts to stop in order to reach the minimum chamber angle as soon as possible. If the minimum chamber angle is reached, a holding phase occurs where the chamber angle is kept constant and is not rising until the right chamber wall has passed the left outlet wall. This is due to suppress backflows back into the chamber. If the discharge process is passed, the suction can continue. The same analogy can be adopted for the suction step, where a holding phase can be established when the maximum chamber angle is reached.
The function ‘BestFit’ only tries to optimize one chamber movement and neglects the other one, which results in two individual kinematic functions for each rotor. ‘Linear’ on the other hand tries to match both holding phases, where it assures that both rotors have the same kinematics, but their rotating phase is shifted. Due to the abrupt changes in the ‘Linear’ and ‘BestFit’ function, ‘MixLinear’ tries to smooth the curvature course, by using a smoother acceleration movement of both rotors at the start of the suction and compression phase.

Figure 6. Flow diagram on how the discontinuous function were determined.

Figure 7. Chamber angle course of the developed functions during one working cycle.

Figure 7 shows the resulting chamber angles from the continuous and discontinuous functions. As shown, each function beside the current function is matching the criteria of a minimum chamber angle of $\gamma_{\text{min}} = 2^\circ$ and a maximum chamber angle of $\gamma_{\text{max}} = 40^\circ$. All of these functions show a quite similar behavior compared to each other except for the current function.

3. Solution Procedure

As mentioned, each function will be tested in a CFD parametric study to consider effects on the momentum of the gas. To achieve reliable numerical results, a fine mesh and time resolution is needed side by side with an adequate setup of boundary conditions. In order to calculate one full revolution of the main shaft, 2000 timesteps for each function are necessary to fulfil the convergence criteria of RMS 1E-4. The volume fluctuation and change at each timestep require a fine resolution and are modelled with the moving mesh model in CFX. Due to the limited computing time, a hexa-mesh with 4,704,593 nodes and 7,934,527 elements is used, which was proven to be accurate enough in a preliminary mesh study. The leakage flow through the annular gap between the rotor blades and the housing is also taken into account with an annular gap size of 0.1mm.

The chosen boundary conditions are derived from the application as a compressor in an air refrigeration cycle on a laboratory scale, shown in Table 2. The SST-turbulence model is chosen, since it provides accurate results for a wide range of applications and delivers a robust scheme. The characteristic movement of each rotor is described with specified angular displacement functions (presented in section 2.2) of the left and right chamber wall by using the CFX expression language (CEL).

Table 2: Boundary conditions used in simulation.

| Boundary condition | Inlet        | Outlet       |
|--------------------|--------------|--------------|
| Temperature Inlet   | 300.15 K     | 308.15 K     |
| Pressure Inlet      | 1.0 bar      | 1.8 bar      |
| Fluid               | Air, ideal gas |              |
| Rotational Speed    | 1200 rev/min |              |
4. Results and Discussion

The results of the kinematic study are shown in Figure 8 and Table 3. The comparison of the volumetric efficiency \( \lambda \) and mass flow rate MFR between each function is presented in Figure 8 (a) and (b). Both graphs intend to show, that the current function has the lowest volumetric efficiency and mass flow rate, having an efficiency of only \( \lambda = 85.3\% \) and a mass flow rate of 0.047 kg/s. The mass flow rate of every examined function was increased to 0.08-0.085 kg/s, while the volumetric efficiency increases up to 5-10% compared to the current function. The main reason for the increase in the mass flow rate is due to the larger displacement volume. Furthermore, ‘Amplitude’, ‘Linear’ and ‘MixLinear’ function indicate the biggest increase of the volumetric efficiency up to 94%, followed by the ‘Flattened’ and ‘BestFit’ function with 91% and 90%.

Figure 8 (c) shows the pressure-volume diagram, where deviations can be identified between the examined functions. Starting with the current function, the area is significantly smaller compared to the examined functions due to the lower displacement volume. However, ‘Bestfit’ is showing a very small area compared to the other functions which indicate that this function is probably not ideal at all. ‘Amplitude’ and ‘Linear’ is showing the largest area due to the large increase in pressure. Figure 8 (d) also shows the pressure behaviour of one chamber during the first revolution. Compared to the current function, ‘BestFit’ only indicates a slightly higher maximum pressure up to 4.5 bar and an internal pressure ratio of \( \Pi = 3.68 \). ‘Flattened’ and ‘MixLinear’ on the contrary show a quite higher maximum pressure between 7.5-8.5 bar with a pressure ratio of about \( \Pi = 6.4 \). ‘Amplitude’ and ‘Linear’ show a similar behaviour with a maximum pressure at 10 bar and a pressure ratio of over \( \Pi = 7 \) seen in Table 3. However, for a delivery pressure of 1.8bar, an internal pressure increased to nearly 10 bar results in a very low adiabatic efficiency. Therefore, port timing and design needs to be improved as well as this machine consists of no valves.

It should be mentioned that the results from the Fourier function were disabled due to convergence problems, showing huge deviations between both chamber pressures and temperatures. The results from the ‘BestFit’ function also indicates, that individual functions for each rotor do not show any significant improvements compared to the equal functions for both rotors. Therefore, it can be concluded that due to the periodical design of the machine, each rotor should have the same function, but phase shifted.

All of the examined functions do show a significant improvement of the mass flow rate due to the reduction of the minimum clearance angle and also led to a better volumetric efficiency with the same displacement volumes. If we just take a look on the mass flow rate, ‘Linear’ and ‘MixLinear’ show the best result, closely followed by ‘Amplitude’. However, in terms of pressure ratio, ‘Amplitude’ is the leading kinematic slightly followed by ‘Linear’. Therefore, it can be concluded, that either one of these functions should be considered for the upcoming optimization. Which and how both of these functions are realized depends on the gear design. For cylindrical gears, the ‘Amplitude’ function is probably the one to choose, where stepping gears could be more suitable for discontinuous functions.

| Performance Parameters       | Current  | Amplitude | BestFit | Linear | MixLinear | Flattened |
|------------------------------|----------|-----------|---------|--------|-----------|-----------|
| Internal Pressure ratio \( \Pi \) [-] | 2.72     | 7.64      | 3.68    | 7.13   | 6.44      | 6.42      |
| Mass flow rate MFR [kg/s]     | 0.0473   | 0.0839    | 0.0796  | 0.0842 | 0.0841    | 0.0811    |
| Volumetric Efficiency \( \lambda \) [-] | 0.853    | 0.941     | 0.896   | 0.944  | 0.939     | 0.910     |
| Indicated Work \( W_{id} \) [J] | 9.015    | 40.087    | 23.340  | 43.368 | 38.638    | 34.108    |

5. Conclusions and future work

A new rotary type compressor named rotary-chamber compressor has been introduced. The working principle is based on two phase-shifted rotors, which are realized by special toothed gears. The construction and working principle are relatively simple and the size easily scalable. Compared to other rotary compressors, this novel compressor has a great potential for the application in refrigeration
systems because of its high volume turnover. Due to the operation without oil and active valves systems, the compressor comes with low maintenance, making it sustainable.

![Graphs showing volumetric efficiency, mean mass flow rate, pressure-volume diagram, and pressure behaviour of the examined kinematics.](image)

**Figure 8.** Results of the study at 1200 rev/min. (a) volumetric efficiency, (b) mean mass flow rate, (c) pressure-volume diagram, (d) pressure behaviour of the examined kinematics.

Different kinematic functions have been developed and tested with a mathematical model using the CFD-solver ANSYS CFX to take effects on the momentum of the gas into account. It was shown that the current kinematic function, obtained by measurements, has potential for improvements. By increasing the amplitude of the function its volumetric efficiency of $\lambda = 85.3\%$ can rise to a value of $\lambda = 94\%$. Also, by reducing the minimum clearance angle in general, all kinematics have shown an efficiency between 91% and 94% while the mass flow rate could nearly be increased to 90%. The highest volumetric efficiency was determined for the ‘Amplitude’ function slightly followed by the ‘Linear’ function. However, it is also in question which kinematic can be realized in the future since each function requires a change of the gear. Possible gear-types could be stepping or cylindrical gears. It can also be concluded that each rotor should have the same kinematic function but should be phase shifted due to the periodical design of the machine, which also can be seen in the performance parameters.

Additionally, it is not clear if changing one parameter in the ‘Amplitude’ is the optimum solution or if additional coefficients need to be changed as well to achieve a higher volumetric efficiency. Therefore, the next work is focusing on optimizing further parameters of the ‘Amplitude’ kinematic. Since the machine is still in its development phase, additional optimization possibilities do exist like determining the optimum rotor blade geometry or improving the sealing. Even though the results show great potential in the use for air refrigeration processes, the experimental validation of the presented method needs to be done before further analysis is carried out.
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Nomenclature
| Term   | Description                              |
|--------|------------------------------------------|
| CEL    | CFX Expression Language                  |
| MFR    | Mass flow rate                           |
| SST    | Shear Stress Transport                   |
| λ      | Volumetric Efficiency                    |
| ω      | Angular velocity                         |
| θ      | Main shaft rotation                      |
| γ      | Chamber angle                            |
| Π      | Internal pressure ratio                  |
| φ      | Angular position                         |
| f      | Angular displacement                     |
| kₙ     | Constant variable                        |
| W      | Work                                     |

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