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Numerical Simulation of Discharge Process in the Single Screw Compressor

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ABSTRACT

In the process of the single screw compressor, the symmetrical grooves of the compressor are periodically connected with the discharge ports and the gas enters the discharge chamber or back flows from chamber, generating gas pulsation. The gas pulsation increases the flowing loss to bring additional energy loss, meanwhile the gas pulsation is the one of main reasons for the generation of vibration and noise. The thermodynamic model of the compression chamber and the discharge chamber of a single screw compressor is established in this paper taking the discharge chamber structure and the flow resistance into main consideration. The discharge process in discharge chamber is simulated base on the thermodynamic model. In addition, the influence of speed and back pressure on the pressure in the discharge chamber is investigated in detail. The analysis results provide a basis for optimum design of structure and reduction of vibration and noise.

1. INTRODUCTION

The core component of a single screw compressor is a mesh pair composed of a screw and a symmetrical star wheel. The gas gets compressed through the closed volume composed of the casing, the screw with groove and the star wheel with teeth. Single screw compressor has been widely used in the fields of air compression, petrochemical industry and refrigeration because of the significant performance such as simple structure, high balance, small vibration and low noise.

Over the years, many scholars have investigated the working process of single screw compressor. Bein T.W. and Hamilton J.F. (1982) studied the working process of the injection single screw air compressor. Boblitt W.W. and Moore J. (1984) put forward a mathematical model of the injection single screw refrigeration compressor and analyzed the influence of some factors on the performance. Wang Z. et al. (2016) conducted the theoretical and experimental study on thermodynamic performance of single screw refrigeration compressor with multicolumn envelope meshing pair and the good agreement was obtained. Wu J. and Jin G. (1988) developed a computer model for prediction and analysis of an oil-flooded single screw compressor performance considering the influence of some important parameters. There are also some scholars who investigated the discharge process of single screw compressor. Li H. and Jin G. (1991) analyzed the discharge process and the influence of various factors to the discharge process with theory and experiment and the optimum discharge ports on different working conditions are also given. Zhou L. et al. (1997) specifically studied the discharge process of a single injection screw.

However, the previous investigations do not take the flow in the discharge chamber and the influence of the asymmetry structure of the discharge chamber on the working process is neglected. Actually the discharge chambers on both sides of single-screw compressor have irregular shapes which affect the gas state in the discharge chamber because of the flow resistance. In addition, the balance performance of the single screw compressor will be affected, which increases the vibration and noise especially for asymmetrical discharge chambers. Therefore, it is necessary to study the internal flow in the discharge chamber of single screw compressor.

The thermodynamic model of the groove and the discharge chamber of a single screw compressor with single line envelope is established in this paper taking the discharge chamber structure and the flow resistance into main consideration. The working process is simulated base on the thermodynamic model. In addition, the influence of speed and back pressure on the pressure in the discharge chamber is investigated in detail. The analysis results provide a basis for optimum design of structure and reduction of vibration and noise.

2. MATHEMATICAL MODELING

2.1 Geometry model

The single screw compressor with single line envelope geometry is shown in Figure 1. It consists of several control volumes including suction chamber, grooves and discharge chambers. The geometry parameters of the single screw compressor are listed in Table 1. The discharge chambers are asymmetry and it includes a short discharge chamber and a long one, just as shown in Figure 2. The discharge chamber can be simplified to angular tube, as shown in Figure 3. The parameters of simplified discharge chamber are listed in Table 2.

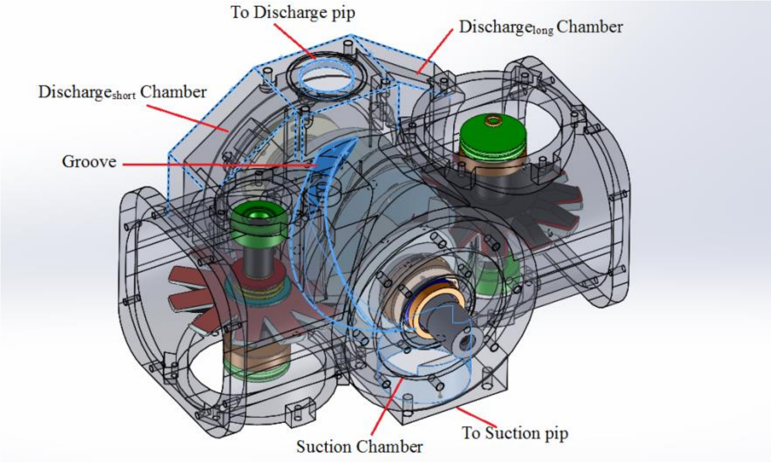


Figure 1: The 3D view of the single screw compressor

Table 1: The geometry parameters of the single screw compressor

Items	Value	Items	Value
d_1/mm	182	H/mm	6
d_2/mm	194	p_s/kPa	101
A/mm	145.5	p_d/kPa	900
l/mm	160	P	11/6
b/mm	28	fluid	air

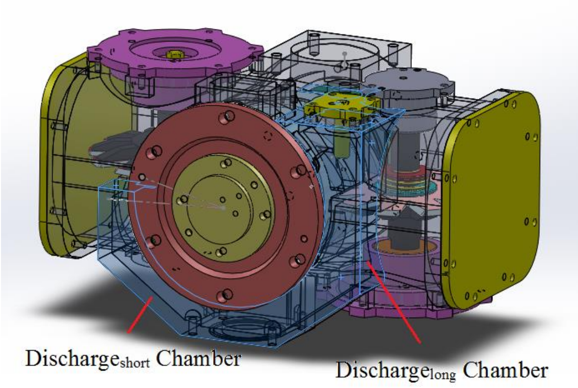


Figure 2: The 3D view of the discharge chambers

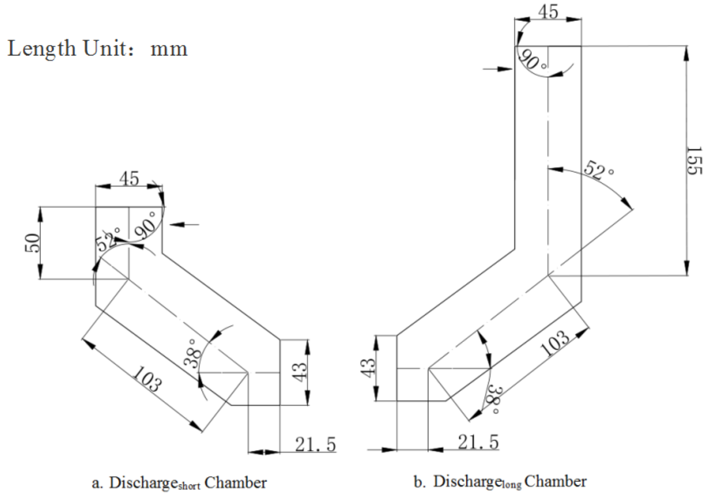


Figure 3: The simplified structures of the discharge chambers

Table 2: The parameters of simplified discharge chamber

	Volume/m^3	Equivalent diameter/m	Length/m	Area/m^2
Discharge$_{long}$ Chamber	9.7×10^{-4}	0.055	0.2795	0.0033
Discharge$_{short}$ Chamber	6.3×10^{-4}	0.055	0.1745	0.0033

The working process of the single screw compressor is shown in Figure 4. The working fluid enters the suction chamber through the suction pipe and gets preheated. Then the working fluid gets compressed in groove and flow through the discharge chamber. There is heat transfer and leakage during the working process.

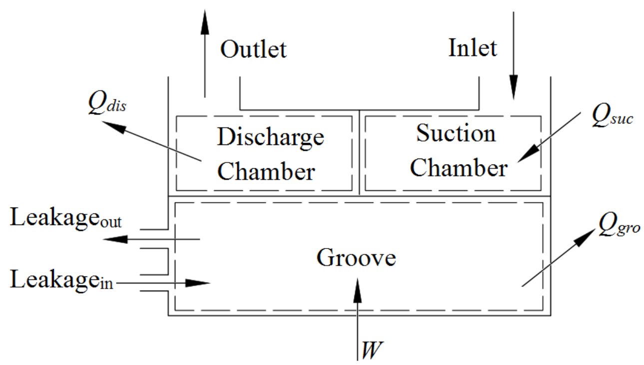


Figure 4: The working process of the single screw compressor

2.2 Model simplification

Each part in the working process is regarded as an open system. Furthermore, the discharge chamber is considered as a thermal system with constant volume and variable mass, just as shown in Figure 5.

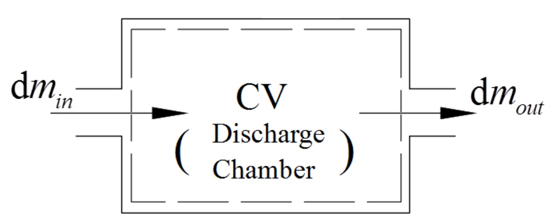


Figure 5: The control volume schematic of the discharge chamber

There are many factors that affect the thermodynamic process in the discharge chamber, including flow, heat transfer, friction and so on. In order to simplify the analysis process, the following assumptions are made:
(1) The state parameters at each point in the control volume are consistent at every moment. In other words, the working fluid keeps uniform;
(2) All external effects (mechanical energy, thermal energy and mass exchange) are transmitted to any mass point in the control volume instantaneously and uniformly;
(3) There is no change of the control volume position and the gas velocity does not change much. The kinetic energy and potential energy of the working fluid are ignored;
(4) The internal state parameters of the three screw grooves on each side of discharge chamber are consistent under the corresponding angles. Therefore, the calculation cycle is selected from the beginning of compression to the end of the discharge for one screw groove;
(5) There is no resistance loss during the gas flow in the discharge chamber, and the resistance loss is reflected in the discharge moment;
(6) In order to focus the effect of discharge chamber, the control volumes are well sealed to avoid the effect of leakage flows and adiabatic in the process of compression and exhaust after suction;
(7) The working fluid in the control volumes keep in single-phase state.

2.3 Theoretical calculation model

Under the above basic assumptions, the thermodynamic equations of the discharge chamber are established based on the energy conservation and mass conservation equations.

According to the first law of thermodynamics for variable mass systems, the energy conservation equation for a variable mass thermodynamic system can be expressed as

$$\mathrm{d}U = \mathrm{d}E_{in} - \mathrm{d}E_{out} - \mathrm{d}Q + \mathrm{d}W \tag{1}$$

Therefore, the energy conservation equation in the control volume is

$$\frac{\mathrm{d}U}{\mathrm{d}\alpha} = \frac{\mathrm{d}m_{in}h_{in}}{\mathrm{d}\alpha} - \frac{\mathrm{d}m_{out}h_{out}}{\mathrm{d}\alpha} - \frac{\mathrm{d}Q}{\mathrm{d}\alpha} + \frac{\mathrm{d}W}{\mathrm{d}\alpha} \tag{2}$$

The gas mass changing in the control volume is described as follows

$$\frac{\mathrm{d}m}{\mathrm{d}\alpha} = \frac{\mathrm{d}m_{in}}{\mathrm{d}\alpha} - \frac{\mathrm{d}m_{out}}{\mathrm{d}\alpha} \tag{3}$$

The working fluid is regarded as real gas in this paper and the parameters such as the enthalpy and internal energy are obtained by the software REFPROP 9.1.

The gas has no time to be cooled because that the compression and exhaust process are extremely short, dQ=0. The work performed by external work on the working fluid in the screw groove is described as dW=-pdV. There is no work by performed by external work on the gas in the exhaust chamber.

The 4th order Runge-Kutta method is used to solve the differential equations. In the main program, the criteria for convergence is whether the gas pressure and mass of the two successive cycles are equal. The software MATLAB (R2017a) is applied to solve the above mathematical model. The solution flowchart is shown in Figure 6.

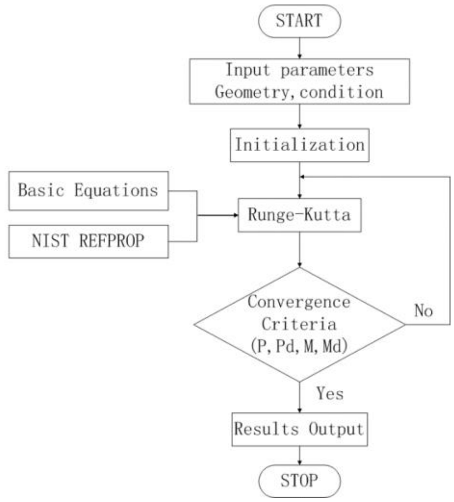


Figure 6: The solution flowchart

There is actually no resistance in the outlet of the discharge chamber, as shown in Figure 7(a). According to the above mathematical model, it is assumed that the parameters in each state of the discharge chamber are uniform. In addition, there is no flow resistance in the discharge chamber and the flow resistance exists in the instantaneous discharge process. Therefore, the discharge process of the discharge chamber is considered to be an orifice flow model with the flow resistance added in the outlet of the discharge chamber, as shown in Figure 7(b).

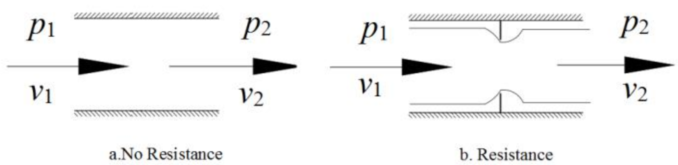


Figure 7: The simplified model of the discharge chamber outlet

According to Bernoulli equation and continuity equation, the relationship between flow resistance loss and volume flow can be expressed as follows:

$$h_s = \frac{p_1 - p_2}{\rho g} \tag{4}$$

$$h_s = \xi \frac{v^2}{2g} \tag{5}$$

$$v = \frac{q_v}{S} \tag{6}$$

The mass flow of the outlet of the discharge chamber is

$$q_m = q_v \rho = \frac{S}{\sqrt{\sum \xi_i}} \sqrt{2\rho (p_1 - p_2)} \tag{7}$$

The resistance coefficient of the discharge process is the sum of every single resistance coefficient. For both sides of the discharge chamber, there are the drag resistance and three local resistances in the outlet.

The drag resistance coefficient is related to the internal state parameters and geometry dimensions of the discharge chamber. It is found obviously that the drag resistance coefficient is different for the both sides of the discharge chamber because of the different shapes. The drag resistance coefficient can be obtained by

$$\xi_{\mathrm{ow}} = f\frac{l}{D} \tag{8}$$

$$\frac{1}{f} = 2\lg(\mathrm{Re}\sqrt{f}) - 0.8 \tag{9}$$

The local resistance coefficient can be obtained by

$$\xi_{\mathrm{lo}} = 0.946\sin^2(\frac{\theta}{2}) + 2.407\sin^4(\frac{\theta}{2}) \tag{10}$$

Where, the bending angles are 38, 52 and 90 degree respectively. The pressure of the two discharge chambers can be obtained according to the above descriptions.

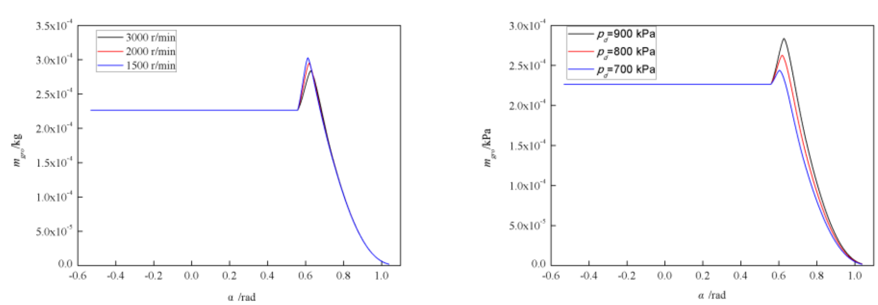
3. RESULTS AND DISCUSSIONS

Based on the above model, the working process in the discharge chambers and the corresponding screw groove is investigated. Furthermore, the influence of speed and back pressure on the pressure of the discharge chamber is analyzed in detail.

The change of pressure with angle in the given condition (n=3000 r/min, p_d =900 kPa) is shown in Figure 10. It can be seen that the pressure in the discharge chamber suddenly decreases in the initial stage of discharge because of the gas reflux caused by the under-compressed state. Then the pressure increases in the middle period of discharge process due to the increase of the inlet flow and flow resistance. Finally, the pressure gradually decreases and tends to be stable.

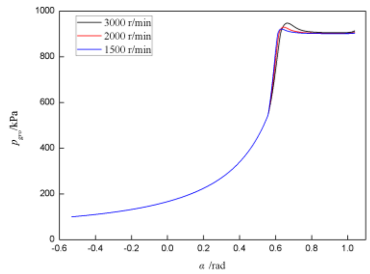
Figure 8(a) and 9(a) show the influence of speed on the change of mass and pressure of groove with angle. The influence of speed on the change of pressure of discharge chambers with angle can be obtained according to Figure 10, 11 and 12. It can be seen that as the speed decreases, the pressure change in the groove and the discharge chamber gets smaller. Meanwhile, the local oscillation is more intense and the reflux mass is more. Figure 8(b) and 9(b) show the influence of back pressure on the change of mass and pressure of groove with angle. The influence of back pressure on the change of pressure with angle can be obtained according to Figure 10, 13 and 14. It can be seen that the pressure change in the groove and the discharge chamber gets larger and the reflux mass gets more as the back pressure increases.

In addition, the pressures in long and short discharge chambers are different. When it is in the condition with n=3000 r/min and p_d =900 kPa, the difference between the pressure peaks, the variances and the angles corresponding to the pressure peak reach 0.03kPa, 0.03 and 0.17° respectively. It indicates that the pressure peak in long discharge chamber lagged behind that in short discharge chamber. As the speed and back pressure increases, the pressure difference between two chambers gets larger.

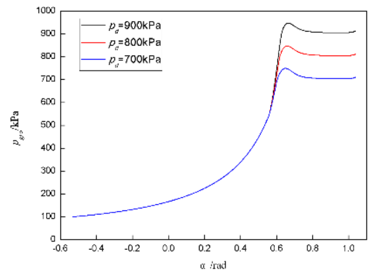


(a) (b)

Figure 8: m-α indicator diagram comparison of groove: (a) different speeds; (b) different back pressures.



(a)



(b)

Figure 9: *p*–*α* indicator diagram comparison of groove: (a) different speeds; (b) different back pressures.

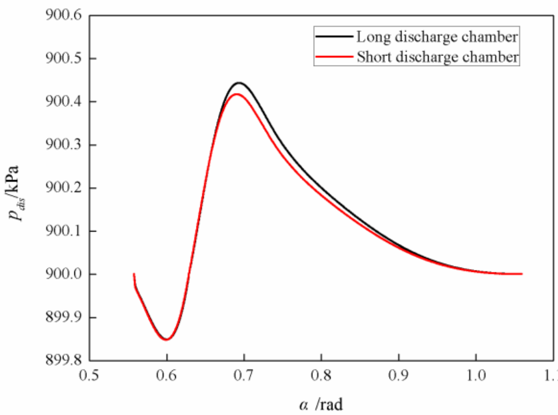


Figure 10: *p*-*α* indicator diagram comparison of discharge chamber (n=3000 r/min, p_d =900 kPa)

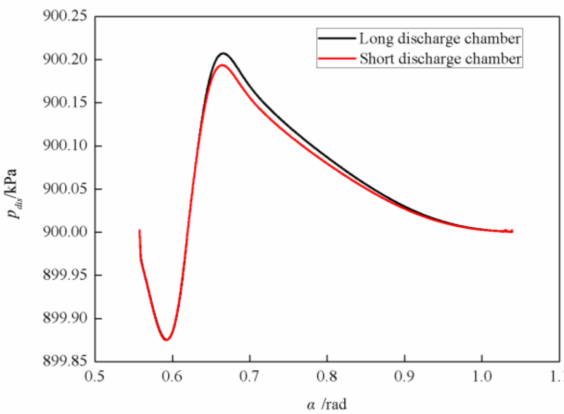


Figure 11: *p*-*α* indicator diagram comparison of discharge chamber (n=2000 r/min, p_d =900 kPa)

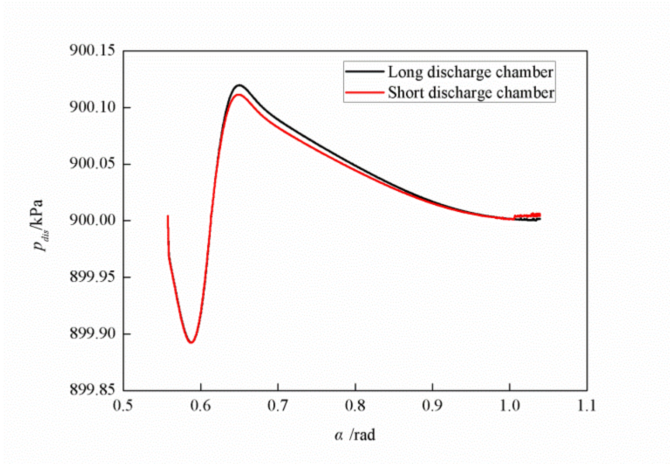


Figure 12: *p*-*α* indicator diagram comparison of discharge chamber (n=1500 r/min, p_d=900 kPa)

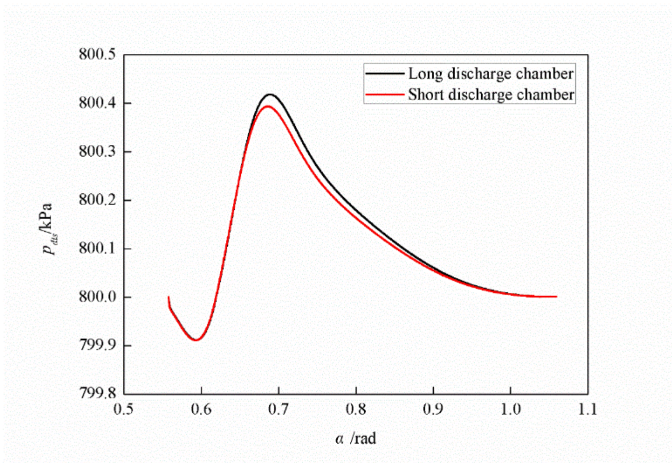


Figure 13: *p*-*α* indicator diagram comparison of discharge chamber (n=3000 r/min, p_d=800 kPa)

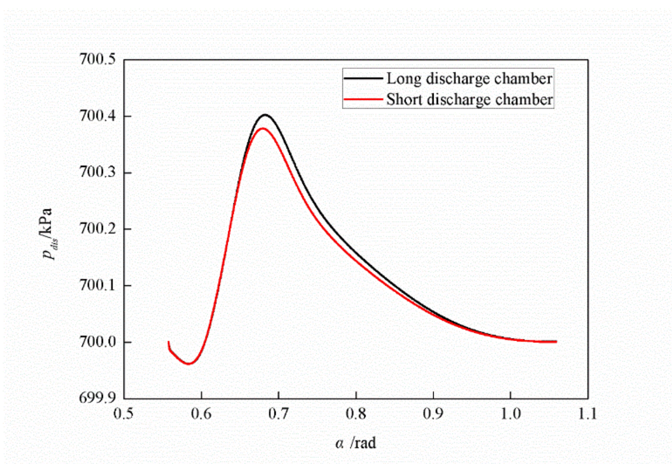


Figure 14: *p*-*α* indicator diagram comparison of discharge chamber (n=3000 r/min, p_d=700 kPa)

4. CONCLUSIONS

The thermodynamic model of the compression chamber and the discharge chamber of a single screw compressor is established in this paper taking the discharge chamber structure and the flow resistance into main consideration. The working process in discharge chamber is simulated base on the thermodynamic model. In addition, the influence of speed and back pressure on the pressure is investigated in detail. The following conclusions are obtained:

- There is a gas reflux into the groove which results in the pressure decrease in the initial stage of discharge in discharge chamber.
- The flow resistance in discharge chamber leads to the pressure increase in the middle period of discharge process.
- As the speed decreases, the pressure change in the groove and the discharge chamber gets smaller with more intense local oscillation and gas reflux. As the back pressure increases the pressure change in the groove and the discharge chamber gets larger with more reflux mass.
- The pressures in long and short discharge chambers are different. As the speed decreases, the pressure difference between two chambers gets smaller. As the back pressure increases the pressure difference between two chambers gets larger.

NOMENCLATURE

A	Central distance between the star-wheel and screw	(mm)
b	Width of star-wheel	(mm)
CV	Control volume	(-)
d	Diameter	(m)
f	Fiction coefficient	(-)
g	Gravity acceleration	(m/s^2)
h	Enthalpy	(J/kg)
h_s	Head loss	(m)
h_t	Surface coefficient of heat transfer	($W \cdot m^{-2} \cdot K^{-1}$)
H	The thickness of the star-wheel teeth	(mm)
l	Characteristic length	(m)
m	Mass	(kg)
n	Speed	(r/min)
p	Pressure	(kPa)
p_d	Backpressure	(kPa)
p_s	Inlet pressure	(kPa)
P	Teeth number ratio	(-)
q_v	Volume flow rate	(m^3/s)
q_m	Mass flow rate	(kg/s)
Q	Heat	(J)
Re	Reynolds number	(-)
S	Area	(m^2)
T	Temperature	(K)
U	Internal energy	(J)
v	Velocity	(m/s)
V	Volume	(m^3)
W	Work	(J)

Greeks

θ	Asway angle of discharge	(rad)
ρ	Density	(kg/m^3)
α	Star-wheel rotation angle	(rad)
ξ	Resistance coefficient	(-)

Subscripts

din	Inlet of discharge chamber
dout	Outlet of discharge chamber
dis	Discharge
g	Gas
gin	Groove inlet
gro	Groove
gout	Groove outlet
i	Diffrernt resistance
in	Inlet
long	Long discharge chamber
leak	Leakage
lo	Local resistance
out	Outlet
ow	On-way resistance
short	Short discharge chamber
w	Wall

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