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Development of a Mechanistic Chamber Model of a Novel Peristaltic Compressor for Air-conditioning and Refrigeration Applications

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ABSTRACT

The air-conditioning and refrigeration industry are constantly searching for efficiency improvements to vapor compression refrigeration systems. A valve-less compressor with variable volume ratios can significantly increase efficiency and add flexibility to these systems. The novel peristaltic compressor is introduced, which has the ability to operate at variable volume ratios and without valves. This compressor operates by means of a progressively actuated diaphragm that compresses vapor and stimulates flow through a cylindrical chamber. In this study, we will present a discretized thermodynamic model of a peristaltic compressor by splitting the mechanism into a series of conjoined segments. Mass and energy balance equations for each compression segment analyzed to create a mechanistic chamber model of the compressor with the segments interacting through flow of mass between them. This model includes a geometric model of the compression chamber created by the interface between the flexible diaphragm and the cylinder. The geometric model is coupled with the mass and energy balance to predict the compressor performance metrics. The preliminary model results show that the peristaltic compressor presents unique attributes that will be explored in future work.

Keywords: Peristaltic movement, Valve-less compressor, Novel Compressor, Geometric Model, Thermodynamic model.

1. INTRODUCTION

In the United States, 24% of the nation’s total energy consumption is used by refrigeration and air-conditioning systems (U.S. EIA, 2018). These two applications use the vapor compression cycle under different operating conditions. Consequently, efficient and innovative technology in the vapor compression equipment has the potential to reduce energy consumption and reduce operating cost. The energy consumption of a vapor compression cycle is highly dependent on the compressor. Energy consumption of vapor compression cycles increases with inefficiencies (losses) in cycles and components. Significantly more losses of vapor compression cycles occur in the compressor compared against the condenser, expansion valve, and evaporator (Ahamed et al., 2011; Pearson, 2015). Improvements to leakage, porting and over/under-compression losses can reduce these losses. A scroll compressor was introduced to eliminate the losses associated with valves and leakage but the compressor has internal volume ratios, which results in over/under-compression losses (Diniz and Deschamps, 2016); similar is true for screw compressors. Reciprocating and rolling piston compressors do not have over/under compression losses but require valves, which results in leakage and/or porting losses associated with those components. A compressor that is able to operate without valves and over/under-compression losses will provide a significant advantage over legacy technologies.

A peristaltic compressor is a promising technology that provides an effective mechanism to eliminate the aforementioned drawbacks. A peristaltic compressor includes a series of segmented compression pockets, similar to a scroll compressor. However, in contrast to the radial compression of a scroll compressor, the peristaltic compressor performs its compression along a linear axis. The segments are formed via a sandwiched diaphragm material (or tube) that is actuated via electromechanical actuation to isolate adjacent portions of the diaphragm. Compression then occurs by reducing the number of actuated electromechanical devices along the linear direction. The plurality of segments allows for isolation of the suction and discharge ports from one another, which removes the need for valves and allows for adjustment of the volume ratio on the fly. These features, combined, have the potential for large improvements in compressor efficiency.
2. LITERATURE REVIEW

2.1. Peristaltic and Diaphragm-based Compression

The peristaltic compressor as a concept is very unique (Bradshaw et al., 2020) but has been explored, in limited capacity, by a few studies in the past. Lawless et al. (1987) designed a miniature compressor which introduced the peristaltic actuation mechanism in the compressor. The system used a hollow deformable elastic tube to flow the gas and electrostrictive ceramic blocks for gas compression. The ceramic blocks were motion amplified that helps them to move back and forth to the tube channel by controlling the voltage to the blocks. The synchronized compression of the blocks on the tube operates the gas to flow and the process reduced the requirement of an inlet and discharge valve. However, 71% of energy is absorbed by the circuit to run the process and the size of the capacitors is very large which reduces efficiency.

Sulfridge and Miller (2001) presented a new model for a mesoscopic peristaltic compressor. A diaphragm placed in a center of a chamber and connected with a programmable logic device. The membrane was driven by an electric field which helped to make a wave pattern and flow the fluid throughout the channel. The compression ratio can be controlled by changing the depth of the cavity which would be helpful to operate the compressor for variable volume.
ratios. The computational analysis of the paper showed that the peristaltic compressor can perform the required work for a compressor. The analysis of physical testing is missing to evaluate the result and the author did not examine the heat transfer of the system which may influence the behavior and results of the compressor. In the peristaltic process, the pressure rise and volume of the discharge gas depend on the forward stroke of the diaphragm. A maximum pressure rise can be obtained if the stroke operates both compress and discharge of the gas. On the other hand, the flow rate and performance of the system can be maximized when the forward stroke of the diaphragm cover all the compression chamber. (Mathew and Hegab, 2013)

The peristaltic mechanism relies on deformation of a flexible membrane, most similar to a diaphragm in pump and compressor applications. In recent years, diaphragms have been examined as a mechanism to be used as a compressor in vapor compression cycles. This synergy to the peristaltic compressor has shown some useful outcomes to leverage for this work. Previous work showed that the diaphragm offered a less leaky mechanism and the ability to generate higher pressures. Hartley (1999) offered a pump model where the peristaltic actuation mechanism used to move the fluid. The pump had an electrically conductive membrane that was actuated using a voltage applied between a conductive diaphragm layer and a base layer, separated by a non-conductive material. The process is reversed when the voltage is reversed. The process creates pressure and when coupled with an electrostatically actuated valve, creates a mechanism to move fluid. This work was restricted to pumping applications and only considered the electrostatic actuation mechanism which has limited displacement potential.

The existing studies of the diaphragm compressors and pumps highlight the potential for a compressor with low leakage and others suggest it is possible to operate similar mechanisms without valves. However, the existing studies have some limitations and they are all either only a theoretical model of pumps/compressors or are not specifically applied to HVAC&R applications.

2.2. Mechanistic Chamber Models
Mechanistic Chamber models have been used for modeling positive displacement compressors by various individuals (rotary -Mathison (2011), scroll -Bell (2011), linear -Bradshaw (2012), spool -Bradshaw and Groll (2013)). It was first generalized by Bradshaw (2012) and further formalized in recent work and accompanying software by Bell et al. (2019). The model is a discretization of a compressor into a series of control volumes. These control volumes represent the various compression chambers at a given instant in a compressor.

Navarro et al. (2007) proposed a phenomenological model for a reciprocating compressor to predict the overall isentropic and volumetric efficiency. The approach to the model uses an unsteady mass and energy balance on the compression chamber. The volumetric and isentropic compressor efficiencies were presented as a system of implicit equations and several correction factors are needed in this model to evaluate the results. Xie et al. (2004) applied a comprehensive model for the purposes of studying the effect of structural parameters and peristaltic pumping operation on the performance of a hermetic reciprocating compressor. Mainly, bearing frictional power loss, valves, leakage and heat transfer models are presented as sub-models. The predicted results are compared with the published compressor map values. The model was restricted to measure the effects of structural parameters on the compressor performance. Mathison et al. (2008) developed a comprehensive model that investigates the pressure and temperature at different crankshaft angle and overall efficiency of the system by considering the effects of compressor chambers leakage and heat transfer between the wall and fluid. Later, Mathison (2011) used the same method to characterize the performance of a novel spool compressor with multiple injection points. The prototype and comprehensive model measured the mass flow rate and power consumption accurately where error of the results varied within 5%. However, error of the predicted discharge temperature and intermediate pressure was very large. Bell (2011) offered a geometric model and overall compressor model to investigate the liquid flooded compression in a scroll compressor. The numerical method helped to improve the compressor performance for use in oil-flooding. Recently, Peng et al. (2017) presented a comprehensive thermodynamic model with differential geometry to study an oil-free scroll compressor. The changes of the different parameters calculated by using the improved Euler method. This work presented that under different discharge pressure, the temperature and power change range is large. Bradshaw (2012) presented a miniature-scale linear compressor, which is a free-piston mechanism where the piston is attached to a linear motor and mechanical springs and operated at a resonant frequency. The free-piston embodiment of this technology coupled the dynamics of the piston movement with the thermodynamic model in a unique way. These mechanistic chamber models require geometric and a series of sub-models to solve the compression process equations and evaluate the target parameters to design a novel compressor. The present work develops a mechanistic chamber model for a peristaltic compressor to get better understanding on the actuation technology and improve overall efficiency.
3. PERISTALTIC COMPRESSOR PROTOTYPE

The segmentation of the peristaltic compressor allows to reduce pressure differences between each compression pocket and improve leakage characteristics due to valve-less operation. The main purpose of developing a peristaltic compressor is to get better volumetric efficiency and an optimized actuation technology to improve overall efficiency. The mechanical actuation technology has never been studied to operate a peristaltic compressor. A series of linear actuators are considered in the design to put load on the diaphragm for actuating the flow in the compressor. Figure 2 presents a schematic of the proposed re-configurable peristaltic compressor prototype. The diaphragm is constrained and sealed against a female die which would create a cylindrical compression chamber. A male die is attached with the linear actuator to press the diaphragm against the female die to provide compression by sealing the chamber synchronously. In the model, 10 linear actuators are used to control the mass flow of the compression chamber. Preliminary testing will be done using air as a working fluid in the compression chamber.

![Diagram of peristaltic compressor prototype](image)

**Figure 2**: Reconfigurable prototype platform of a peristaltic compressor that has been designed for this study. (Bradshaw, 2017)

The 1<sup>st</sup> and 10<sup>th</sup> piston will operate as a suction and discharge valve of the compressor. Gas is compressed, and different volume ratios are achieved, by using a pivotal piston which indicates when the fluid from the compression chamber will begin to discharge. For example, consider a selection of the 7<sup>th</sup> piston as the pivotal piston, this is illustrated in Figure 3. Initially, the 10<sup>th</sup> piston block the compression chamber at the discharge side and the working fluid fills the diaphragm. Then, at step 2, the 1<sup>st</sup> piston isolate the fluid within the compression chamber. At the start of compression in step 3, the 2<sup>nd</sup> to 6<sup>th</sup> piston will press the diaphragm sequentially and compress the fluid to a significantly smaller volume. Then in steps 4-5, when the system hits the 7<sup>th</sup> piston, simultaneously 7<sup>th</sup> to 9<sup>th</sup> piston will press the diaphragm in a sequence and the 10<sup>th</sup> piston will rise to allow fluid to discharge into the system.

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This process can be controlled to achieve a large variety of different volume ratios of operation by selecting different pivotal pistons. By adjusting the pivotal piston from the 3rd to the 9th the prototype can achieve volume ratios from 1.143 to 8. The results from this large variety of volume ratios will allow for an efficient study of the influence on the volume ratio on relevant compressor metrics such as the volumetric efficiency.

4. MECHANISTIC CHAMBER MODEL OF A PERISTALTIC COMPRESSOR

The thermodynamic model of the peristaltic compressor utilizes the unsteady mass and energy balance of a control volume assuming all work is done by boundary work. Density (\( \rho \)) and temperature (T) are the two unknown parameters in the thermodynamic model, which are related using the mass and energy conservation relation. To account for the plurality of embodiments and actuation techniques the peristaltic compressor may encounter, the thermodynamic relationships that govern it was developed as a function of vertical displacement, \( Z \). The actuation process gradually increases the pressure (P) of the fluid by reducing the volume (V) of the compression chamber. Figure 4 shows the computational algorithm for the mechanistic chamber model of the peristaltic compressor highlighting the compression process solver and its required sub-models, which are discussed in the following sections.

Figure 3: Schematic diagram of the prototype compressor operation with pivotal piston 7 selected.

Figure 4: Computational algorithm of the mechanistic chamber model for the peristaltic compressor.
4.1. Compression Process Equations

The compression process equations are derived using an unsteady mass and energy balance equations. The equations provide the state of the working fluid in the compression chamber at any point during the compression process and are solved for each control volume at a particular displacement of the actuator (Z). The density of the system can be determined by differentiating the equation with displacement, which is computed as,

\[
\frac{d\rho}{dZ} = \frac{1}{V} \left( \frac{1}{\rho} \left( \Sigma m_{in} - \Sigma m_{out} \right) - \rho \frac{dV}{dZ} \right).
\]

(4.1)

For the model we have considered actuation speed, S is 0.254 m/s and compression chamber volume, V which will be changed for each compression step. The parameters from the mass balance equations would solve the energy balance equations in order to find the other derivatives. The conservation of energy for a general control volume can be written as follows,

\[
\frac{dE_{cv}}{dt} = \dot{Q} - \dot{W} + \Sigma m_{in}h_{in} - \Sigma m_{out} h_{out},
\]

(4.2)

The equation can be expended in terms of changes in internal energy and mass. The following relation is the energy balance equation, which can be obtained by differentiating the expanded equation with displacement,

\[
\frac{dT}{dZ} = \frac{1}{\rho c_v} \left( \dot{Q} + \Sigma m_{in}h_{in} - \Sigma m_{out} h_{out} \right) - \frac{u \frac{dV}{dZ} - u \frac{\partial P}{\partial Z}}{\rho c_v} \frac{dV}{dZ}.
\]

(4.3)

Here, the only temperature is the independent variable. Several numerical integrators are used to calculate the temperature and density for each step during the compression. Pressure and enthalpy of the system calculated with the known temperature and density using *CoolProp* (Bell *et al.*, 2014), which is an open-source database of fluid and humid air properties.

Equations 4.1 and 4.3 require a series of inputs that are also functions of the instant in time, called sub-models as highlighted on the right-hand side of Figure 4. The following sections outline the sub-models required to provide complete solutions to the compression process equations including a sub-model for geometry, mass flow, heat transfer and leakage.

4.2. Geometric Model

The mechanistic chamber model requires an instantaneous volume of the peristaltic compressor. In the designed reconfigurable prototype peristaltic compressor, linear actuators operate pistons sequentially to generate volume change, as described in Section 3. Various diaphragm shapes can have impact on the overall efficiency of the compressor. To explore this, two types of diaphragm arrangement are considered, circular and semi-circular compression chamber. During the prototype test run, it is shown that the circular compression chamber doesn’t able to rebound the diaphragm properly after the linear actuation. For that reason, we have chosen semi-circular compression chamber to analyze the model for the peristaltic compressor.

If we seal a flat diaphragm on the female die. The diaphragm creates a semi-circular compression chamber which is illustrated in Figure 5. The following equation is used to determine the deflected volume \(V_d\) of the chamber during the actuation:

\[
V_d = \left( R^2 \cos^{-1} \left( 1 - \frac{Z}{R} \right) + (R - Z) \sqrt{2RZ - Z^2} \right) Y,
\]

(4.4)

Here, Y is depth of the piston and Z is changing from 0 to 0.0127m, meaning at the initial the displacement is 0 and sequentially when the piston touches the female die, at the end the maximum deflection would be 0.0127m.
Axial view

Linear actuator

Removeable male die

Diaphragm

Compression chamber

Figure 5: Geometric schematic depicting the geometry model approach of a single-piston is deflecting the diaphragm which is creating a semi-circular compression chamber.

However, there is a curve in both sides of the compression chamber with radius 0.00254m ($R_c$) which we have to consider during the volume change. The following equation is used to derive the volume of the curve segment ($V_c$),

$$ V_c = \left( R_c^2 \cos^{-1} \left( 1 - \frac{h}{2R_c} \right) - \frac{1}{4} (2R_c - h) \sqrt{(4R_c - h)h} \right) Y, \quad (4.5) $$

We considered the displacement of the curve section, $h$ is 0 to 0.00127m. To calculate the semi-circular compression chamber’s final volume ($V$) we can subtract the $V_d$ and $V_c$ from the total volume and the following equation is

$$ V = \left( \frac{\pi}{2} R^2 + \left( 2R_c^2 - \frac{1}{2} \pi R_c^2 \right) \right) Y - V_d - V_c, \quad (4.6) $$

Figure 6 (a), shows how the volume of the compression chamber change during one compression cycle for semi-circular diaphragm arrangements. Here, 1st to 9th pistons are pressing the diaphragm in sequence to reduce the compression chamber volume and release the fluid. After that, all pistons rise together to pump the fluid into the channel and reach the initial compression chamber volume. However, the chamber volume for the discharge flow area would be different for the different pivotal piston. For example, figure 6 (b), demonstrates how the discharge volume will change for the pivotal piston 7. Here, 1st to 6th piston will press the diaphragm sequentially to compress the fluid. For the pivotal piston 7, when the system hits the 7th piston the discharge will start. Sequentially 7th to 9th piston will press the diaphragm to move out all fluid from the chamber, throughout this segment, we will get the discharge volume. During compression and suction, there would be no discharge, so the discharge volume would be zero.
4.3. Mass Flow Model

When the actuators load the diaphragm, the mass flow changes with respect to the area of the compression chamber. The basic equation of mass flow rate for a fluid with a specific heat ($\gamma$) is:

$$
\dot{m} = \rho \vartheta A, 
$$

(4.7)

Here the density of the fluid is $\rho$, working fluid velocity $\vartheta$ and area $A$. To calculate the fluid velocity the Mach number of the flow is calculated using the following relationship,

$$
Ma = \sqrt{\left[\left(\frac{P_{\text{low}}}{P_{\text{high}}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \frac{2}{\gamma-1}}. 
$$

(4.8)

We consider the compression chamber as a nozzle where pressure ratio is the difference in pressures between compression chamber and exit control volume. The next unknown function is fluid velocity, which we can get from the following equation:

$$
\vartheta = \sqrt{\gamma R_g T Ma}, 
$$

(4.9)

Here, $R_g$ is the gas constant and $T$ is the temperature of the fluid at each step. The temperature of the fluid can be evaluated from the energy balance equation.

4.4. Heat Transfer Model

Heat transfer is modeled between the working fluid and the walls of the compression chamber at each instant. The quantity is calculated using the approach of Adair et al., 1972,

$$
Q = h_t A (T_w - T), 
$$

(4.10)

where Heat transfer coefficient is defined as,

$$
h_t = 0.0278 (1 + 0.38\vartheta)\sqrt{(P^2T^2)}. 
$$

(4.11)

Here, $P$ is the fluid pressure at the compression and $T_w$ is the wall temperature. The compression process solver and overall energy balance which are shown in the Figure 4, converge and specify the guess wall temperature during the iteration by using the secant method.
4.5. Leakage Model

The leakage model only focused on the leakage between the diaphragm and the female die when the piston is fully engaged. 1% area of the total compression chamber for single piston is considered as leakage area ($A_{\text{leakage}}$). Following equation is used to determine the leakage mass low rate:

$$m_{\text{leakage}} = \rho \dot{V} A_{\text{leakage}}$$

(4.12)

Where, $A_{\text{leakage}} = \frac{\text{total compression chamber volume (V')}}{V} \times 0.01$

5. PRELIMINARY MODEL RESULTS AND ANALYSIS

The thermodynamic and the geometric model of the peristaltic compressor from Section 4 implemented in a computational model. Air is used as the working fluid to run the model. The suction pressure 104 kPa, and discharge pressure 364 kPa are considered. Figure 7 shows, for the pivotal piston 7 how the thermodynamic properties are behaving during the compression given a corresponding volume change associated with the semi-circular diaphragm as shown in Figure 6.

![Figure 7](image)

Figure 7: (a) Density of the compressor for one cycle, (b) Temperature of the compressor for one cycle, (c) Pressure-Volume diagram for the peristaltic compressor.

As Figure 7(a) shows, the density of the compression chamber increases until the pressure of the fluid reaches the discharge pressure. As discharging of gas occurs the mass velocity and heat-transfer of the chamber start decreasing. Therefore, the density begins to decrease when discharge start and returns to its initial position when the fluid starts entering the compression chamber again. Figure 7(b) highlights that the temperature of the compressor rises modestly until the final piston compresses when the temperature increases exponentially. Similar to the density, when the pistons start open the inlet segment for the fluid flow, the pressure drops significantly and volume of the compression chamber start increasing from nil. For that reason, the temperature starts dropping during the expansion and reaches to its initial condition when the suction completed. Figure 7(c) shows the fluid pressure as a function of compression chamber volume for the peristaltic compressor. This pressure-volume diagram is also referred to as an indicator diagram.

![Figure 8](image)

Figure 8: PV diagram for different actuation mechanism for semi-circular compression chamber.
Figure 8(a) shows that the chamber pressure for pivotal piston 9 greatly exceeds the system discharge pressure. This over-compression is a result of the volume change when compressing pistons 1 to 8 exceeding the volume change required to achieve the system discharge pressure. This requires more work (power) to execute and is therefore a loss. The re-configurable nature of this prototype allows study of these changes, as Figures 8(b) and 8(c) show with the results for the same system conditions with two different pivotal pistons with sequentially lower volume ratios. Pivotal piston 8 (Figure 8(b)) is also over-compressed but to a lesser degree than with pivotal piston at 9. When the pivotal piston is at 7 (Figure 8(c)) the system presents with neither over or under compression. This will result in minimal power and maximize compressor efficiency. A scroll compressor, for example, does not have the ability to modify its volume ratio, so if an operating condition results in significant over-compression the scroll compressor will be less efficient. In contrast, this result suggests that a peristaltic compressor could overcome that inefficiency by adjusting its volume ratio, on the fly.

6. CONCLUSIONS

The total system energy of a vapor compression system can be reduced by improving the efficiency of the compressor. The peristaltic compressor presents an adaptable technology potentially increase the efficiency of these systems. This study presents the mechanistic chamber model of a peristaltic compressor including the relevant sub-models including geometry to match a re-configurable prototype compressor, heat transfer, and mass flow. Preliminary results of the model are presented and highlight the influence of adjusting the volume ratio by adjusting the pivotal piston. These preliminary results will allow to predict most appropriate mechanical activation technology and various design parameters such as different aspect ratios of the diaphragm, different operating speed and conditions for the re-configurable prototype peristaltic compressor. The mechanistic chamber model of the peristaltic compressor will next be validated against data from the reconfigurable prototype compressor and then leveraged to explore the most useful geometric arrangements of this device applied to HVAC&R applications.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>Area of the compression chamber</td>
</tr>
<tr>
<td>$C_V$</td>
<td>Specific volume</td>
</tr>
<tr>
<td>$h$</td>
<td>Displacement of the piston when acting on the curve segment</td>
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<td>$h_{in}$</td>
<td>Enthalpy in</td>
</tr>
<tr>
<td>$h_{out}$</td>
<td>Enthalpy out</td>
</tr>
<tr>
<td>$h_t$</td>
<td>Heat transfer constant</td>
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<tr>
<td>$Ma$</td>
<td>Mach number</td>
</tr>
<tr>
<td>$\dot{m}_{in}$</td>
<td>Mass flow in</td>
</tr>
<tr>
<td>$\dot{m}_{out}$</td>
<td>Mass flow out</td>
</tr>
<tr>
<td>$\dot{m}_{leakage}$</td>
<td>Mass flow leakage</td>
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<tr>
<td>P</td>
<td>Pressure of the fluid</td>
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<tr>
<td>$\dot{Q}$</td>
<td>Heat transfer rate</td>
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<tr>
<td>$R$</td>
<td>Radius of the piston/compression chamber</td>
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<tr>
<td>$R_c$</td>
<td>Radius of the curve segment in the compression chamber</td>
</tr>
<tr>
<td>$R_g$</td>
<td>Gas Constant</td>
</tr>
<tr>
<td>S</td>
<td>Actuation speed</td>
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<tr>
<td>T</td>
<td>Temperature of the fluid</td>
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<td>$T_w$</td>
<td>Wall temperature</td>
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<td>Volume of the semi-circular compression chamber</td>
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<tr>
<td>$V_c$</td>
<td>Volume of the curve segment in the compression chamber</td>
</tr>
<tr>
<td>$V_d$</td>
<td>Volume of the deflected segment</td>
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<td>Depth of the piston</td>
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<td>Displacement of the piston</td>
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<td>Fluid velocity</td>
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<td>$\gamma$</td>
<td>Specific heat</td>
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REFERENCES


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