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# Mechanistic Chamber Models: A Review of Geometry, Mass Flow and Heat Transfer Sub-Models

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## ABSTRACT

The evolution of computer resources paved the way for numerical models to substitute extensive experimental campaigns to develop and optimize compressors for the HVAC&R industry. A particular class of compressor models referred to as mechanistic (or deterministic) chamber model is well-known to be computationally efficient and has reasonable model fidelity for performance prediction and extrapolation. The overall model architecture requires inputs from various sub-models, including the mass flow model, heat transfer model, valve model and mechanical loss model. The accuracy of the comprehensive model depends on the fidelity of the individual sub-models. This article provides a comprehensive review of the different approaches available in literature which were used to model the geometry, mass flow, valves, and heat transfer processes in the compressor. The article aims to help researchers identify suitable approaches for their analysis. Additionally, it was found that there are opportunities for additional research to develop analytical correlations for discharge coefficients for mass flow model, effective flow/force area for valve model, instantaneous heat transfer coefficients and thermal conductance for multi-lumped network models.

## 1. INTRODUCTION

Performance evaluation of positive displacement compressors has always been of great interest to both academics and industrialists. This need is often driven by the regulatory changes (Heath, 2017) or industrial competition to improve the reliability and performance of the product. Introduction of novel compressors (*e.g.*, Spool compressor and Z-compressor) also require exhaustive performance evaluation for design improvements. Moreover, American Innovative and Manufacturing (AIM) Act sets the target to reduce the HFCs production and consumption by 85% in the next fifteen years, which further necessitates an exhaustive performance evaluation of existing technologies with future low-GWP refrigerants. A recent study by Tanveer & Bradshaw (2021) used the mechanistic chamber model to find the optimum scroll compressor sizes for various refrigerants. The results suggest that optimum scroll compressor size exists for each refrigerant.

The performance evaluation methods range from heuristic approaches to numerical methods. Computational-based evaluation methods are more time-efficient and less exhaustive as compared to the heuristic approaches. A wide variety of models lie between two extremes *i.e.* statistical and distributed models, which vary in fidelity and required user knowledge. Among these models, the mechanistic chamber model is a good combination of model fidelity and simulation speed. An embodiment of this type of model is available in an open-source platform named PDSim (B. Bell Ian et al., 2021). The mechanistic chamber modeling approach has been used by numerous authors for performance prediction of reciprocating, linear, scroll, spool and Z-compressor and the list goes on.

This article focuses on presenting a comprehensive review of various techniques to model the compressor geometry, mass flow, leakage, heat transfer, and valve performance across several decades of research. These models are summarized for widely used compressor types (Reciprocating, scroll, screw and rotary) and other, more novel, compressor types (linear, spool, z-compressor, among others) as well. The objective is to provide a starting point for researchers to select a suitable sub-model for their analysis and identify the possible research gaps for future work.

## 2. MECHANISTIC CHAMBER MODEL APPROACH

The mechanistic chamber model is a quasi-steady model which simulates the compression process by applying mass and energy conservation equations over different control volumes (or chambers) inside the compressor and calculates the instantaneous thermo-physical properties.

The compression process equations are derived from the mass and energy conservation equations applied to an open control volume which can be used to calculate the change of two independent thermodynamic properties such as density and temperature or density and specific internal energy with respect to time. If a secondary fluid is present in the control volume, such as lubricant oil or wet-compression, the mass fraction of the fluid can be included in the formulation (Ziviani et al. (2020)).

The general form of the governing equations can be written as,

$$\frac{d\rho}{dt} = \frac{1}{V}\left[-\rho\frac{dV}{dt} + \left(\sum_i \dot{m}_{i,in} - \sum_j \dot{m}_{j,out}\right)\right], \tag{1}$$

and

$$\frac{dT}{dt} = \frac{-1}{\rho V \frac{\partial u}{\partial T}}\left[\left(\rho V\frac{\partial u}{\partial \rho} + Vu\right)\frac{d\rho}{dt} - (p+\rho u)\frac{dV}{dt} + \sum \dot{m}_{in}h_{in} - \sum \dot{m}_{out}h_{out} + \dot{Q}_{in}\right]. \tag{2}$$

.

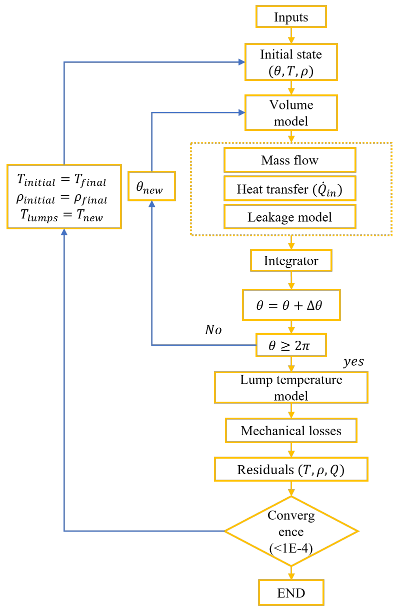
Equations 1 and 2 are the generic time dependent form of the compression process equations and can be modified to calculate the instantaneous temperature and density as function of crank angle etc, specific to compressor type. A simplified approach for the mechanistic model is presented in Figure 1a. The simulation process starts with defining the geometric dimensions, boundary conditions and guess values for initial states. The mass and energy balance equations (compression process) require inputs from the sub-models like the volume model, mass flow model, heat transfer model and mechanical loss model. The volume model provides the instantaneous volume and rate of change of volume with respect to some independent variable, either crank angle or time. The mass flow model calculates the mass flow rate along different flow paths, including suction and discharge ports and leakage paths. The heat transfer sub-model provides the instantaneous heat transfer from the refrigerant to the compressor walls, whose temperature is assumed to be constant for one cycle. The inputs from these sub-models are fed into the compression process equations and integrated over one cycle. Once a cycle is completed, residuals are calculated for the initial guess conditions and compressor wall temperatures. If the model involves a calculation of the temperature of various components in the compressor shell, *i.e.*, lump temperatures, then those heat transfer equations are solved at this step. If the residuals are greater than the tolerance, new guess conditions and lump temperatures will be used to model the compression process. The process will continue until convergence is achieved.

## 3. GEOMETRY MODEL

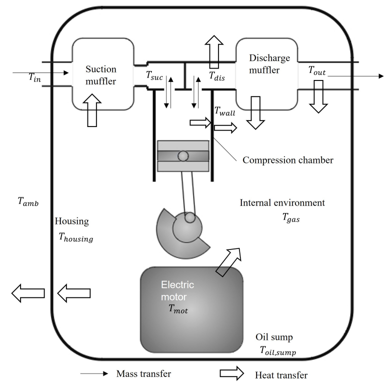
The geometry design of a positive displacement compressor is particular to the compressor type. Different methodologies are employed by various researchers to accurately calculate the instantaneous volume of the compressor chambers. The critical nature of the geometric model can be further asserted by the fact that compressor performance depends on the compressor geometry, which can be optimized for enhanced performance. In the last few decades, significant work has been carried out to improve the accuracy of the geometric model for conventional compressors. Most of the geometric models can predict the instantaneous compressor volume with acceptable accuracy for a large plurality of compressor types used in HVAC&R applications, which are summarized in this section.

### 3.1 Independent variable

The geometry model represents a core component to a mechanistic chamber model and it is often the first derived sub-model. Its functional form is based on the independent variable selected by the compression process equation formulation. This variable is often crank angle of rotation ($\theta$) for devices with a fixed volumetric behavior with each




**(a)** Simplified model schematic for mechanistic chamber modeling approach for a positive displacement compressor.




**(b)** Block diagram to represent various heat transfer phenomenon inside the compressor.

**Figure 1:** Model validation using various refrigerants and displacement volumes.

cycle (e.g., reciprocating, scroll, screw, rotary). Still, it can also be time ($t$) for non-fixed volume machines such as a linear compressor.

Yang et al. (2013) modeled the volume of a reciprocating compressor as a function of crank angle ($\theta$) using kinematic expressions. The kinematic-based models are similar to those developed for internal combustion engines and often require several geometric parameters such as piston diameter and crank length, among others. Similar to the reciprocating compressor, the scroll compressor is also a fixed volume machine and crank angle is often employed as an independent variable for instantaneous volume calculation. The functional form of the screw compressor is also based on the screw rotational angle. Chan et al. (1981) presented one of the earliest geometric models for the screw compressor, which was based on the screw rotation angle. Time is often used to integrate the volume for non-fixed volume machines where the volume depends on the compressor dynamics. The geometry of a linear compressor is similar to a reciprocating compressor but its stroke is not fixed by the crank mechanism. Instead, it depends on the compressor geometry, the linear motor and mechanical springs. H. Kim et al. (2009) presented a dynamic modeling approach for the linear compressor and used time as an independent variable for geometry modeling.

Independent variable selection can also be influenced by the selected simulation platform as well. For example, Modelica™is often used for transient simulations and requires equations to be a function of time (Tanveer & Bradshaw, 2020). So, it is more convenient to derive the equations as a function of time.

In some cases, the independent variable is selected based on the motion dynamics of the compressor. For example, Islam & Bradshaw (2021) presented a mechanistic chamber model for a novel peristaltic compressor, where pistons attached to the linear actuators were used to press the diaphragm and compress the gas inside the diaphragm. The volume formulation is derived as a function of piston displacement.

In summary, the selection of independent variables depends on the individual compressor dynamics and platform limitations. Regardless of the selected independent variable, the compression process equations can be re-cast to fit the model needs.

### 3.2 Model approach

The compressor geometry models can be classified into three main categories based on the modeling approach. Geometry models for most compressors are developed using geometric, kinematics and mathematical rules, independent of compressor dynamics. They can calculate the instantaneous volume with reasonable accuracy without using any experimental parameters. The kinematic-based reciprocating compressor model is one of the simplest models which is derived analytically. Most geometry models for the scroll compressor are also analytically derived and improved over time. Morishita et al. (1984) developed a model for a scroll compressor by defining it as an involute of a circle and established the equations of motion for orbiting scroll and Oldham ring. Similarly, analytical geometric models have been developed for rotary compressors (i.e. Cross vane, rolling piston, etc.) and novel compressors (spool compressor, z compressor and peristaltic compressor).

In some compressors, the swept volume depends on the compressor dynamics and varies with the operating conditions. The geometry model for these types of compressors is coupled with the overall compressor model. Bradshaw et al. (2013) reported that the stroke of a linear compressor is not fixed and the desired piston stroke is provided as an input to the model. A numerical algorithm was used to adjust the piston driving force to match the piston stroke to the desired stroke.

The third category of geometry models relies on precision geometry mapping techniques to develop empirical correlations of the geometry. Wu & Tran (2016) developed a semi-empirical geometry model for twin-screw compressor by developing a curve-fit equation using experimentally measured discrete points on the screw rotors. The developed equation was a function of the rotor tooth profile's direction parameter. These types of models are suitable for simulating existing machines and may not be suitable for design optimization or performance evaluation.

In summary, significant research is conducted to accurately model the compressor geometry and several different types of geometry models are available for each compressor type which varies in terms of model simplicity and accuracy. It is recommended to use analytical models because of their fidelity to easily simulate various designs. The 3D coupled or semi-empirical models may not be suitable because they involve large human effort and depends on experimental data.

## 4. MASS FLOW MODEL

The computation of the mass flow rate across the control volume is significant for compressor modeling. This mass exchange is possible through multiple flow paths, including suction ports, discharge ports and leakage paths. The models for calculation of mass flow rate through these paths can be broadly classified into two main categories 1) Incompressible flow models and 2) compressible flow models. Ishii (1996) evaluated the leakage flows through radial and flank clearances using incompressible, viscous flow theory and the results were found to be in good agreement with experiments. The authors also modeled the leakage flow using the compressible and viscous flow theory, but the computed results do not match the experimental results. It was concluded that leakage flows in the compressors can be well simulated using simple in-compressible flow theory.

In reality, the velocity of flow is very high through different paths in the compressor, which may result in compressible flow. The mass flow calculations can be incorrect with the incompressible flow assumption. The density of fluid changes significantly due to compressibility effects for mac numbers greater than 0.3. Therefore, most of the compressor modeling work in the literature utilizes compressible flow models.

In these models, the flow through suction, discharge and leakage paths are treated similar to compressible flow through a converging/diverging nozzle. Different types of models are utilized in literature for modeling compressible flows with various levels of complexity. These models include 1) Isentropic nozzle flow model, 2) Isothermal flow 3) Non-isentropic compressible flow 4) the Rayleigh flow model.

The simplest of these models is the isentropic flow model, which is an idealized flow model and utilizes the adiabatic and friction-less flow assumption. Ports and orifices in compressors are unable to move systemically enough to generate any regions of supersonic flow, so it is assumed that all gas flow in a compressor is subsonic (*i.e.* Ma$\leq$ 1). Using isentropic flow theory, the mass flow in the subsonic region can be computed as,

$$\dot{m} = \rho\sqrt{\gamma R T} A_{valve} \sqrt{\left(\frac{P_{high}}{P_{low}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} \frac{2}{\gamma-1}} \tag{3}$$

The subsonic nature of the model assumes that when the flow achieves a critical pressure ratio, choking occurs at the nozzle location such that,

$$\left(\frac{P_{high}}{P_{low}}\right) = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma+1}} \tag{4}$$

As the isentropic flow model is an idealized model, it often over-predicts the mass flow rate (I. H. Bell et al., 2020). The computed mass flow rate is often corrected by the empirical correction factor or discharge coefficient. The primary advantage of using the isentropic nozzle flow model is its computational efficiency and simplicity. But the use of the model is often bounded by the limited availability of experimental data and the empirical correction factors. Cho et al. (2000) studied the leakage phenomenon at actual operating conditions of the compressor and found a discharge coefficient to be 0.1 for the leakage model using the choking experiment. Such a low value of discharge coefficient suggests that the isentropic nozzle flow model is not a reliable option for predicting leakage flow.

Another drawback of the isentropic model is its inability to account for frictional losses. In a positive displacement compressor, the leakage length to gap ratio can be up to 500, suggesting that frictional losses in leakage flow can not be neglected. The adiabatic nozzle flow which considers the effect of frictional losses is often referred as "$Fanno\ flow$" model. Yuan et al. (1992) presented a model which considers both viscous and inertial forces to calculate gas leakage in small clearances. The proposed model shows better agreement with the experiments.

Kang (2002) compared the leakage flow rate results computed using two mathematical models (isentropic nozzle flow model, Fanno flow model) and one commercial CFD software (Fluent). Isentropic flow model over predicts the mass flow rate and compression chamber temperature compared to the Fanno flow model. The difference between the predicted mass flow rate of both models was up to 5%. The results from the Fanno flow model and CFD were found to be in good agreement.

The detailed models often required many input parameters and associated programming to make these models challenging to implement. Furthermore, these models add significant overhead to overall compressor models. I. H. Bell et al. (2013) presented a hybrid leakage model for positive displacement compressor. The model is developed by comparing the results from the isentropic nozzle flow model to the detailed model results for the number of operating conditions, working fluids, and leakage gaps. A curve fit based correlation was developed which depends on the Reynolds number, dimensionless characteristic length and dimensionless gap. The model is able to predict the mass flow rate within $\pm 20\%$ for 93% of the data points. The highest error in the model is found to be around the junction of the low and high Reynolds number.

G. W. Kim et al. (2016) developed a functional formula for calculation of discharge coefficient for radial clearance in rolling piston rotary compressor by using a combination of experimental and CFD results. The calculated flow coefficient ranges from 0.085 to 0.85 for different operating conditions. The coefficient is then applied to the isentropic nozzle flow model to calculate the leakage mass flow rate. The results are validated with experiments. Although the error in calculated results varies with operating conditions but errors were within tolerable limits.

Pereira & Deschamps (2020) presented the numerical analysis of the radial and flanked leakage in the scroll compressor. The investigation is performed for various gases, operating conditions and geometries, and is based on dimensionless parameters. The results are compared with experimental data of G. Kim et al. (2017), the theoretical model of Zuk & Smith (1969) and correlations of I. H. Bell et al. (2013). The presented correlation shows good agreement with experimental data, while the Reynolds equation presented in (Zuk & Smith, 1969) tends to significantly over-predict the radial and tangential leakage. The correlation presented by I. H. Bell et al. (2013) predicts leakage with reasonable accuracy but tends to over-predict the radial leakage as the mass flow rate and Reynolds number increase.

In summary, the isentropic nozzle flow model combined with an experimentally calibrated flow coefficient is widely used in the literature for compressor modeling. This kind of analysis increase dependency on the experiments, resulting in limiting the fidelity of the model. There is a need to develop a physics-based method, similar to G. W. Kim et al. (2016), to predict flow coefficient for different operating conditions and refrigerants. These methods should account for length to gap ratio and viscous, inertial and compressibility effects. Once a flow coefficient is calculated for a specific compressor and condition, it can be used in the mechanistic chamber model for further analysis. In this way, model fidelity can be enhanced without compromising the computational speed. The model from I. H. Bell et al. (2013) is computationally efficient and acceptable when tested with some already published experimental data. A detailed study similar to Pereira & Deschamps (2020) should be conducted to increase confidence over the model and identify the model's limitations and suggest improvements.

## 5. HEAT TRANSFER MODEL

The heating of a refrigerant is an inevitable product of the compression process. It results in an unsteady heat transfer to and from the refrigerant. Early research presents contradictory opinions about the effect of heat transfer on compressor performance. Prasad (1998) reviewed heat transfer studies and concluded that the heat transfer in the compressor is perceived as a material technology problem instead of an efficiency problem. It was in the 80's, when researchers acknowledged the effects of heat transfer on the compressor performance. Ribas Junior (2007) emphasizes the importance of heat transfer modeling by comparing the thermal efficiency with the component efficiency. The thermal efficiency of a positive displacement compressor is between 80 and 83%. As the efficiency of other compressor components is approaching their limits (electric efficiency 87-88% and mechanical efficiency 92%), the researchers focus on reducing the thermal losses to improve the overall efficiency (Ribas Junior, 2007). To accurately account for thermal losses, modeling of different heat transfer phenomenon inside the compressor is important. This section summarizes various works related to the modeling of heat transfer in positive displacement compressors.

For the mechanistic chamber modeling approach, a hermetic or semi-hermetic compressor can be divided into two time domains, 1) the instantaneous heat transfer within control volume(s) where the compression takes place at each instant throughout a rotation/cycle and 2) the overall heat transfer which generally contains the suction and discharge flow paths, motor and bearings, *etc.* Various heat flows and lumped temperatures are illustrated in the form of a block diagram in Figure 1b. Instantaneous heat transfer during the compression process is modeled by assuming a quasi-steady process, where the compression process is divided into small steps and instantaneous heat transfer between refrigerant and compressor walls is calculated using different correlations specific to a compressor type. Once the heat transfer inside the compressor is calculated, the heat transfer between various components is calculated. It is often assumed that there is no temperature gradient in individual components. The process is repeated until convergence is achieved.

The calculation methodologies and correlations adopted by different researchers for instantaneous heat transfer inside the compressor and the overall heat transfer are discussed below.

### 5.1 Instantaneous heat transfer

The instantaneous heat transfer inside the compression chambers is often calculated using various correlations which are highly specific for a specific compressor type. Many of the correlations for instantaneous heat transfer are based on the general form of the formula for turbulent flow in pipes or flat plates, which can be represented as,

$$Nu = C\mathrm{Re}^b\mathrm{Pr}^d, \tag{5}$$

where, Nu is Nusselt number, and C, b and d are constants whose values are calculated using experimental or analytical techniques. The heat transfer coefficient can be calculated from Nusselt number as,

$$\mathrm{Nu} = \frac{hD}{k}, \tag{6}$$

where h is the heat transfer coefficient, D is the cylinder diameter and k is the thermal conductivity of the fluid.

Historically, the heat transfer in reciprocating compressor was modeled using the correlations developed for the engines (Woschni, 1965). Adair et al. (1972) presented the first heat transfer correlation for reciprocating compressors in the form as mentioned above. Reynold's number is calculated as a function of refrigerant velocity, which is proportional to the crankshaft speed. The values of the constants C, b and d, in Equation 5, were reported to be 0.053, 0.8 and 0.6 respectively. Hsieh & Wu (1996) presented an updated form of Nusselt number correlation,

$$\mathrm{Nu} = (C_1 \cdot \mathrm{Re}_1 + C_2 \cdot \mathrm{Re}^{C_3})\left(\frac{\mu}{\mu_0}\right)^{C_4}, \tag{7}$$

where viscosity terms are added to account for pressure effects on refrigerant properties. An additional Reynolds number term is added to account for increased velocity during the discharge process. Different values of constants $C_1, C_2, C_3$ and $C_4$ were proposed for suction, discharge, compression and multistage compression. Over the years, different researchers proposed different values for the constants of the Equation 5.

Tuhovcak et al. (2016) presented a comparison of different heat transfer correlations with the CFD results and reported no single correlation was able to accurately predict the heat transfer with reasonable accuracy for all compression processes. They proposed a combination of different correlations for expansion, suction, compression and discharge processes to achieve better accuracy.

Similar attempts were made to formulate correlations for predicting heat transfer coefficients in other types of positive displacement compressors, which often have more complex geometries than a reciprocating compressors such as scroll, rotary and screw compressors.

The scroll compressor has a very intricate geometry which causes challenges for installing instrumentation. So, initial attempts to characterize heat transfer were mainly focused on the analytical methods to predict the heat transfer. Chen et al. (2002) used the modified Dittus-Boelter correlation for a spiral heat exchanger to simulate heat transfer between the refrigerant and the scrolls. I. Bell (2011) modeled heat transfer using an approach similar to the Chen et al. (2002) but account for the oscillation of the flow field. The correlation for a spiral heat exchanger is also found to be adequate for rotary compressor (Mathison et al., 2008) and spool compressor (Bradshaw & Groll, 2013).

Similar to scroll compressors, heat transfer in the screw compressor is also modeled using heat exchanger correlations. Ziviani et al. (2014) presented a comprehensive model for the screw compressor and used the heat transfer correlations for internally grooved tubed.

In summary, the work related to instantaneous heat transfer modeling in the positive displacement compressors is lacking in the literature. Most often researchers used the correlations developed for heat exchangers, which may have higher uncertainties for compressor modeling. Literature suggests that the compressor macro performance parameters (*e.g.* mass flow, power, efficiency) can be predicted with reasonable accuracy using these correlations. Still, these correlations may not be suitable for optimizing the thermal performance of the compressor. Significant work is required to optimize and validate the heat exchanger correlations for the compression process or develop new correlations specific to the compressors. Additionally, heat transfer correlations for some of the novel compressor technologies (*e.g.* Z- compressor or peristaltic compressor, *etc.*) should be developed and validated.

### 5.2 Overall heat transfer

Similar to the instantaneous heat transfer, in the 80's, researchers had divided opinions about the effect of heat transfer on the compressor performance and compressor models during that time often ignore heat transfer. This disagreement applied to both the instantaneous and the heat transfer between large lumps and the ambient. Brok & Touber (1980) investigated the heat transfer in a compressor and concluded that the magnitude of the effect of heat transfer is less than often portrayed in the literature. It was reported that the conclusion is not valid if the actual heat transfer coefficients in the compressor are significantly higher than the one predicted in the article.

Several researchers include the overall heat transfer mechanism into the mechanistic chamber model to understand the thermal gradients within a steady-state compressor. Additionally, the energy balance can not be satisfied without applying the overall heat transfer. Overall heat transfer modeling involves the calculation of heat transfer coefficients which are dependent on the compressor geometry. Various positive displacement compressors include complex geometries, which cause complex flows and hinder the accurate modeling of the heat transfer phenomenon. Different numerical, experimental, or hybrid approaches are often employed to model thermal behavior. The numerical methods for overall heat transfer can be further divided into three categories 1) Integral or lumped conductance method, 2) Differential or distributed method and 3) Hybrid.

The integral approach is often referred to as lumped conductance or thermal network model and involves applying the steady or unsteady energy balance for different control surfaces. Heat transfer correlations can be used to calculate the heat transfer coefficients. Zhang et al. (2020) used various correlations available in the literature to calculate the thermal resistances between different components in the shell of a linear compressor. The shell geometry can be significantly different than the one for which correlations are developed. So, experimental data is often used to calibrate the thermal resistances or overall heat transfer coefficients.

In the thermal conductance method, the heat transfer interaction between different components is modeled by using the thermal resistance, which can be calibrated using experimental data. The model is solved iteratively to calculate the temperatures of difference lumps by minimizing the residuals of overall energy balance, which can be represented as,

$$\dot{Q}_{amb} + \sum_i \dot{W}_{i,fr} + \sum_i \dot{Q}_i = \epsilon \tag{8}$$

where, $\epsilon$ is residual enforce for convergence, $\sum_i \dot{W}_{i,fr}$ is the sum of the mechanical and frictional losses and $\sum_i \dot{Q}_i$ represents the heat interactions between different components.

The simplest way to account for heat transfer is assuming a single lump, *i.e.*, the compressor wall that interacts with the refrigerant and the ambient. Various authors used a single lump heat transfer model where heat transfer between refrigerant and shell and shell to ambient is considered. This type of single lump approach is suitable for predicting the macro performance parameters but requires calibration of heat transfer coefficient using experimental data. Moreover, this type of modeling approach is not suitable for estimating the temperature of different components, which is vital for designing a hermetic or semi-hermetic compressor.

To represent the realistic heat transfer interactions in the hermetic or semi-hermetic compressor, the shell is often divided into multiple lumps. The complexity of the model increases with the increase in the number of lumps in the thermal network model. For hermetic or semi-hermetic designs, it is considered that the shell is filled with the refrigerant (generally at the discharge condition) that interacts with different components, *e.g.*, motor, oil sump, suction manifold and shell. Cavallini et al. (1996) used the multi-lump model for a hermetic reciprocating compressor and subdivided the machine into six components. The model is used to simulate two commercial units with R600a and R134a. The simulation results were found to be in good agreement with the experiments.

The second approach is referred to as the differential modeling approach which requires dividing the fluid or solid domain into small control volumes and solving Navier-stokes equations for each control volume using the computational fluid dynamics (CFD) program. This simulation approach is computationally exhaustive and requires a significant simulation time. Chikurde & Longanathan (2002) used ANSYS FLUENT to model the heat transfer inside the compressor through conduction and convection. The simulation results were found to be in good agreement with the experiments.

The third approach involves modeling of the compressor using the integral/thermal network model and calculating the heat transfer coefficients or thermal conductance using the differential approach. Oliveira et al. (2017) presented a steady-state heat transfer model for solid components and gas flows inside the compressor based on the finite volume method. The heat transfer model is coupled with the transient lumped formulation to simulate the compressor performance. It was reported that the presented model does not require experimental calibration, but it is computationally expensive.

Although all three types of modeling approaches can predict the compressor performance with reasonable accuracy, the selection process depends on the level of details required in results, dependency on the experimental data and model fidelity. A lump conductance model is the simplest of all the three methods discussed above, but its accuracy highly depends on the experimental data which is required to calculate the thermal resistances. There are correlations available to calculate the thermal resistance, but due to the complex compressor geometry, the results obtained from these correlations will have very high uncertainty. The differential modeling approach can be a more reliable option to model heat transfer and predict the temperature distribution for complex geometries, but it is highly time-consuming. Apart from the simulation time, often with differential approaches, a small design change requires an update of geometry and mesh, which restricts its fidelity. A hybrid approach can be a better combination of accuracy and simplicity. Distributed modeling and experimental techniques can be used to calibrate the thermal conductance, which can be used as an input to the multi-lump model to predict the compressor performance component temperatures.

## 6. CONCLUSION

A mechanistic chamber model requires inputs from different sub-component models, including the mass flow model, heat transfer model, valve model and mechanical loss model. The accuracy and fidelity of the overall model depend on the accuracy and fidelity of the individual sub-model. Various approaches are available in the literature for each sub-model. Similar to any model, selecting a suitable method is a trade-off between model fidelity and computational effort and time. A review of the geometry, mass flow rate, and heat transfer models are presented to provide a comprehensive summary to the readers, which will help select a suitable method and identify the potential research areas for future work. Following are some of the findings from the presented literature review.

- Various independent variables are selected for the compressor geometry which depends on the compressor dynamics and modeling platform.
- Analytical geometry models are proffered over semi-empirical and 3D coupled models because they can be easily implemented for different compressor designs.
- Isentropic nozzle flow model is widely used for compressor modeling and experimentally calibrated flow coefficients are used to improve the model accuracy with good results that tend to limit the flexibility of these models to other technologies and/or devices.
- Some correlations for flow coefficient should be developed that account for length to gap ratio, viscous and inertial effects to reduce dependency on the experimental data.
- Hybrid flow models are available in the literature that is computationally efficient and provides results with reasonable accuracy. An extensive analysis should be conducted for various flow paths and compressors to increase confidence in these models.
- Separate heat transfer correlations are required to accurately model heat transfer during suction, compression and discharge processes.
- Literature is lacking for heat transfer correlations which are specifically developed for the compressors.
- Due to the complex nature of the compressor geometry, analytical approaches for the prediction of thermal resistances have high uncertainties.
- A hybrid modeling approach, where differential methods are used to calculate the thermal resistances and integral approach is used for performance analysis, can increase the model fidelity.

## NOMENCLATURE

| | |
|---|---|
| $\dot{m}$ | Mass flow rate |
| $\dot{m}_{in}$ | Total mass entering the control volume |
| $\dot{m}_{out}$ | Total mass leaving the control volume |
| $\dot{Q}_i$ | Heat transfer from ith lump |
| $\dot{Q}_{amb}$ | Ambient heat transfer |
| $\dot{W}$ | Rate of work done |
| $\dot{W}_{i,fr}$ | ith mechanical or frictional loss |
| $\gamma$ | Heat capacity ratio |
| $\mu$ | Dynamic viscosity |
| $\rho$ | Density |
| $\rho$ | Density of the refrigerant |
| Nu | Nusselt number |
| Pr | Prandtl number |
| Re | Reynolds number |
| $\theta$ | Crank angle of compressor |
| $A_{valve}$ | Area of the valve |
| $E_{CV}$ | Energy inside the control volume |
| $h$ | Heat transfer coefficient |
| $h_{in}$ | Specific enthalpy of the refrigerant at suction |
| $h_{out}$ | Specific enthalpy of refrigerant at discharge |
| $k$ | Thermal conductivity |
| $Ma$ | Mach number |
| $P$ | Pressure |
| $P_{high}$ | Pressure on high pressure side of the valve |
| $P_{low}$ | Pressure on low pressure side of the valve |
| $R$ | Specific gas constant |
| $T$ | Instantaneous temperature in control volume |

| | | | |
|---|---|---|---|
| $T$ | Temperature of the refrigerant | $u$ | Specific internal energy |
| $t$ | time | $V$ | Instantaneous volume of the control volume |

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