

Dynamic Model Development for Highly Efficient Inverter Compressor

by

Atacan Oral

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Dynamic Model Development for Highly Efficient Inverter Compressor

**Yüksek Verimli Değişken Kapasiteli Kompresörün Dinamik Modelinin
Geliştirilmesi**

Koç University

Graduate School of Sciences and Engineering

This is to certify that I have examined this copy of a doctoral dissertation by

Atacan Oral

and have found that it is complete and satisfactory in all respects,
and that any and all revisions required by the final
examining committee have been made.

Committee Members:

Prof. İsmail Lazoğlu (Advisor)

Prof. Demircan Canadınç

Assist Prof. Levent Beker

Prof. Cemal Çakır

Prof. Özgen Akalın

Date:

ABSTRACT

The heart of a household refrigerator can be assumed as the compressor since it receives the refrigerant, compresses it and allows gas to circulate inside the refrigeration cycle. Even though there are different types of compressors in the market, reciprocating compressor is the most common one at the refrigerators.

Due to energy regulations that become more strict day by day because of global warming, the challenge between compressor manufacturers is to producing the most energy efficient compressor. In order to reduce the energy dissipation and increase the efficiency, the fundamentals behind the physics used in the compressor must be understood clearly. Moreover, even though dynamics and mechanics disciplines are investigated in this thesis, the relationship with thermodynamics and flow dynamics must be considered as well.

In this thesis, a comprehensive mathematical model of a reciprocating compressor is established by including all design parameters and considering all force sources for both steady state and startup behaviour where the highest vibration and noise are observed. The model validations are conducted by specially developed experimental setups. Moreover, the components that play a significant role in dynamic response of the compressor such as springs or discharge tube are investigated individually, further optimizations are proposed to reduce the design time and both vibration noise levels. A novel design methodology is proposed for the discharge tube for the first time in the world and mathematics is included in the design process.

Furthermore, to reduce startup and shutdown vibration and noise, a novel design addition, called transient vibration reducer (TVR), is utilized. TVR is a passive damper that operates as a pseudo-active damper due to its stepwise structure, interacting with the motion of the compressor only above predetermined limits of body displacement without affecting or compromising steady-state performance.

Finally, the startup and shutdown position map data of the crankshaft is formed with an all-round experimental setup for the first time in the literature. Suggestions are made to have a smooth startup behavior, implemented on a compressor and validated with experimental studies.

ÖZETÇE

Bir ev tipi buzdolabının kalbi, soğutucu akışkanı sıkıştırdığı ve soğutma çevrimi içinde gazın dolaşmasına yardımcı olduğu için kompresör olarak kabul edilebilir. Piyasada farklı tipte kompresörler bulunsa da ev tipi buzdolaplarında en yaygın olanı pistonlu kompresördür.

Küresel ısınma nedeniyle her geçen gün daha zorlu hale gelen enerji regülasyonları nedeniyle, kompresör üreticileri arasındaki en büyük rekabet, enerji açısından en verimli kompresörü üretmektir. Enerji sarfiyatını azaltmak ve verimi artırmak için kompresörde kullanılan fiziğin arkasındaki temellerin net olarak anlaşılması gerekir. Bu tezde her ne kadar kompresörün dinamik ve mekanik disiplinleri incelenirse de bu disiplinlerin termodinamik ve akışkanlar dinamiği ile ilişkisi de göz önünde bulundurulmalıdır.

Bu tezde, hem kararlı durum hem de en yüksek titreşim ve gürültünün gözlemlendiği kalkış davranışı için tüm tasarım parametreleri dahil edilerek ve tüm kuvvet kaynakları dikkate alınarak pistonlu bir kompresörün kapsamlı bir matematiksel modeli oluşturulmuştur. Model deneysel doğrulamaları, özel olarak geliştirilmiş deney düzenekleri tarafından gerçekleştirilmiştir.

Ayrıca, kompresörün dinamik tepkisinde önemli bir rol oynayan yaylar veya titreşim borusu gibi bileşenler ayrı ayrı incelenmiş, tasarım süresini ve titreşim ile gürültü seviyesini azaltmak için optimizasyonlar önerilmiştir. Titreşim borusu için dünyada ilk kez yeni bir tasarım metodolojisi tasarım sürecine matematik dahil edilerek önerilmiştir. Ayrıca kalkış ve duruş titreşimi ile gürültüsünü azaltmak için geçici titreşim azaltıcı (TVR) adı verilen yeni bir tasarım ilavesi önerilmiştir. TVR, kararlı durum performansını etkilemeden veya bundan ödün vermeden, kademeli yapısı sayesinde yalnızca önceden belirlenmiş kompresör gövde deplasman limitlerinin üzerinde kompresörün hareketiyle etkileşime giren aktif bir sönümleyici gibi çalışan pasif bir sönümleyicidir.

Son olarak, literatürde ilk kez çok yönlü bir deney düzeneği ile krank milinin başlatma ve kapatma konum harita verileri oluşturulmuştur. Sessiz ve titreşimsiz bir kalkış davranışı sağlamak için önerilerde bulunulmuştur, öneriler kompresör üzerinde test edilmiş ve deneysel olarak doğrulanmıştır.

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Chapter 1

INTRODUCTION*1.1. Outline*

This thesis discusses the dynamic behavior of the reciprocating compressor in household refrigerators, presents a dynamic model, investigates the parameters that affect dynamic performance, comprises novel approaches to design processes, reports solutions for better dynamic performance in terms of vibration, and demonstrates experimental work to understand the nature of the compressor and for validations.

Two main subfields are discussed under the name of steady state and transient operations of the reciprocating compressor. Each subfield comprises mathematical models and experimental validations.

A model to predict the body displacement of the compressor is developed by considering the emerging structural forces and natural frequencies of the system. A smart numerical method is established to design the discharge tube of the compressor.

Moreover, the resistive torques that occur during startup are estimated numerically. Behavior mapping studies are conducted experimentally which are shared as transient operation works. The effect of the starting position of the crankshaft is investigated, and a novel design addition is developed to suppress transient vibrations of compressors and all vibratory machines. Additionally, motor control approaches are presented to have a smooth startup for the compressor.

The thesis is organized such that Chapters 1, 2, 3, and 4 are dedicated to introduction, steady-state operations, transient operations, and conclusion, respectively. A literature review, background, and conclusion are presented for each chapter.

1.2. Overview

1.2.1. Refrigerator Working Principle and Components

The refrigeration or cooling cycle relies on a simple principle: pass the cold material around the hot object so that it loses heat.

High-pressure hot liquid enters a capillary tube, and its temperature and pressure drop as it leaves the capillary, and the refrigerant changes its phase from gas to liquid. Then, this cold liquid inside the walls of the cabinet must be moved in order to absorb the heat of the foods. After the refrigerant moves inside the evaporator, it again changes its form from liquid to gas. In order to increase the pressure of the gas, a compressor is utilized. Later, the pressurized gas reaches the condenser, which is placed behind the refrigerator. While the refrigerant moves inside the condenser, the condenser distributes heat to the room. Meanwhile, the refrigerant turns to liquid. Such a cycle occurs continuously while cooling the food down with the help of a compressor. Therefore, it is safe to say that the compressor is the heart of the refrigeration cycle, and the hermetically sealed reciprocating compressor is the most common type in refrigerators while the linear compressor takes second place.

1.2.2. Linear Compressor

A linear compressor carries a linear motor and a piston that is mounted on the motor. The spring-loaded piston is placed in the center of the electromagnet and moves back and forth to compress the refrigerant. Most of the time, it is excited at the resonance frequency in order to reduce the need for electromagnetic force [Hassan et al., 2016]. Linear compressors also demonstrate lower mechanical losses than any type of compressor since only the piston is a friction source [Liang et al., 2014].

1.2.3. Reciprocating Compressor

The reciprocating compressors employ the simplest and one of the most common mechanisms in the dynamics discipline: the slider crank mechanism. Basically, the crankshaft rotates, and the rotational motion is converted to translational motion via a connecting rod, i.e., a conrod, and the piston reciprocates back and forth.

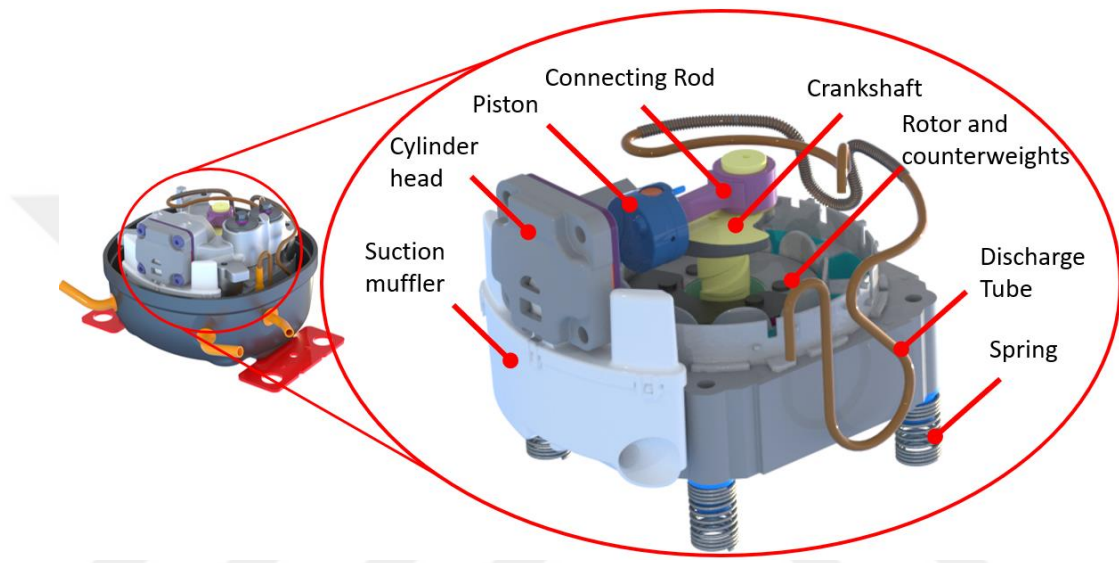


Figure 1.1. Reciprocating compressor.

The reciprocating compressor consists mainly of the following parts: body, piston, connecting rod, crankshaft, rotor, stator, counterweights, lamination steel, suction muffler, exhaust muffler, discharge tube, springs, hermetic housing, grommets, and a certain amount of oil.

The piston in a reciprocating compressor experiences the highest amount of stress during compression. Therefore, its material must be chosen wisely in order to prevent deformation so that the compressor can serve for tens of years. During reciprocation inside the cylinder, the piston is also exposed to a secondary radial motion. This radial motion may result from higher side forces and result in deformation on the skirts, whereas a stable piston trajectory is desired. Such a stable trajectory can be achieved by optimizing the clearance value between the piston and the cylinder. While smaller values of clearance help to damp out the vibration, the required power to overthrow

the viscous friction increases [Prata et al., 2001]. Higher oil viscosity also results in a more stable trajectory, yet similarly viscous friction increases, i.e., power consumption increases.

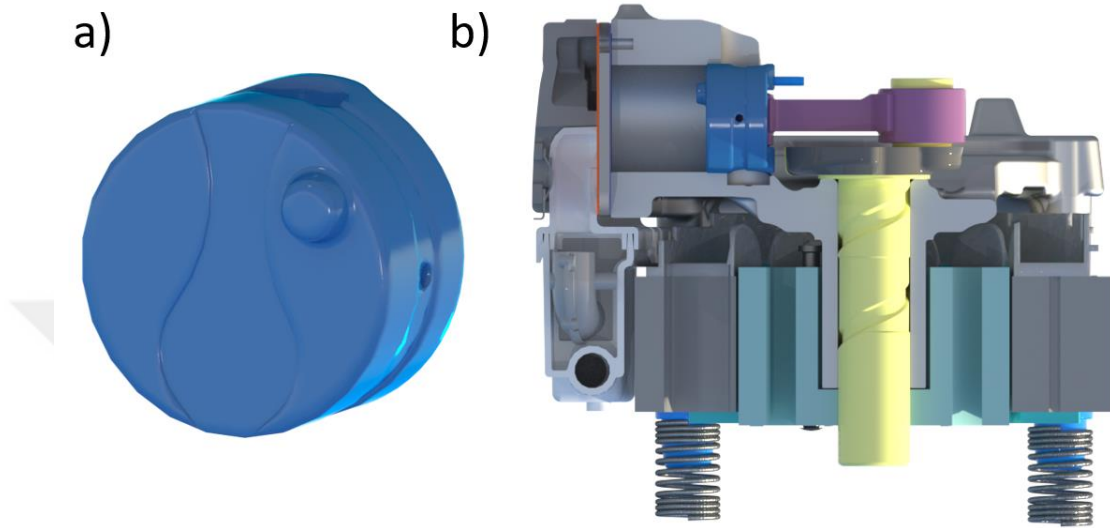


Figure 1.2. (a) Piston of a compressor, (b) section view of the compressor

The piston is excited by the crankshaft via the connecting rod. The connecting rod connects the piston to the crankshaft and transmits the forces. Thus, it should be robust enough. The mass of the conrod, as with each component, affects the vibration response of the compressor based on Newton's second law. Therefore, the conrod should be as light as possible so as not to increase vibration, but also should be robust and have a high inertia to stand against the stresses. Thus, topology optimized [Naikwadi, 2019] or I-beam shaped [Helal et al., 2021; Muhammad et al., 2020] connecting rods could be the solution. Moreover, the joint clearance also affects the performance of the compressor [Cheng and Liu, 2017], which is the indicator of importance of the design and assembly for the connecting rod. The importance of robustness of connecting rod is also reported by [Cervelin et al., 2019] by utilizing strain-gauges on the conrod to measure the forces that emerge in steady state

operation.

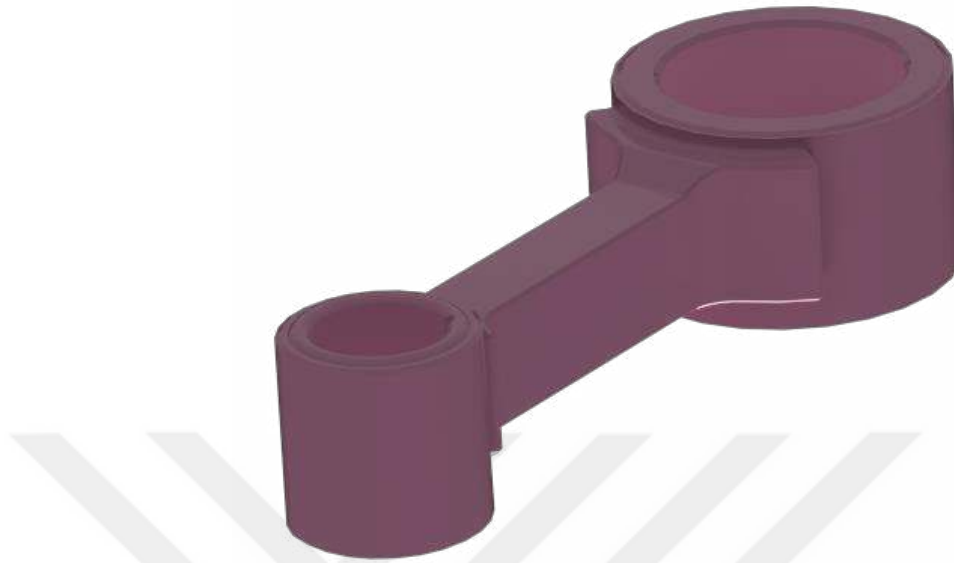


Figure 1.3. Connecting rod of a compressor.

The crankshaft is the heart of the rotating system since it transfers the rotary movement from the rotor to the conrod. In most household reciprocating compressors, it is assembled with two bearings, and has a fixed joint to the rotor. Therefore, as the rotor rotates, the crankshaft follows it. The dimensions of the bearing and crankshaft determines the capability of crankshaft to resist the stresses. FEM tools are one of the most useful tools to consider the design parameters before manufacturing [Yu et al., 2012]. It has helix grooves on the outer surface in order to carry the oil from the bottom of the housing to the top. Then oil is distributed inside the housing in splash form. In order to improve the mass flow rate of the oil, geometric parameters such as the number of helices, section area of the helices, etc. can be optimized [Ozsipahi et al., 2014]. Moreover, the oil flow and distribution can also be improved via an additional oil pump that is mounted under the crankshaft in a way that Archimedes' principle is utilized [Shahzad et al., 2022].

Furthermore, the crankshaft is also exposed to a secondary motion, which can be described as tilting behavior [Pont et al., 2017]. As the crankshaft tilts, its orbit changes, and the oil film pressure loses uniformity on the bearing. Therefore, it might be normal to expect that mechanical losses would increase as a result of higher

friction. Yet, it is reported [Estupinan and Santos, 2009] that the secondary motion of the crankshaft is not increasing the losses significantly.

The crankshaft's counterweight is yet another important design parameter in a reciprocating compressor. Such counterweights might be manufactured through casting with the crank itself, or additional plates might be utilized as well. The main purpose of the counterweight is to balance the forces and reduce the vibration [Penazzo, 2018]. Even though it is a simple passive solution, it must be placed in an optimal location so that it does not increase the tilting motion of the crankshaft. Softwares for finite element analysis [Shahane and Pawar, 2017] or even genetic algorithms [Albers et al., 2007] can be utilized in order to optimize the shape and the design, yet simple numerical approaches that consider centrifugal force, which arises from the counterweight's eccentricity, are also useful and practical. Further, another parameter called the crank throw determines the piston stroke, which is equal to the two times of the crank throw's length. It is also safe to say that as piston stroke increases, secondary motion of piston increases which results in unstable trajectory due to increasing moments around the wrist pin.



Figure 1.4. Crankshaft of a compressor.

The rotor, which carries the crankshaft, is the source of rotation. Rotor carries magnets that is affected by the magnetic field that is produced by the electrical current and rotates. Similar to the crankshaft, additional counterweights also can be mounted

on rotor [Levecque, 2011] for the purpose of reducing the vibrations.

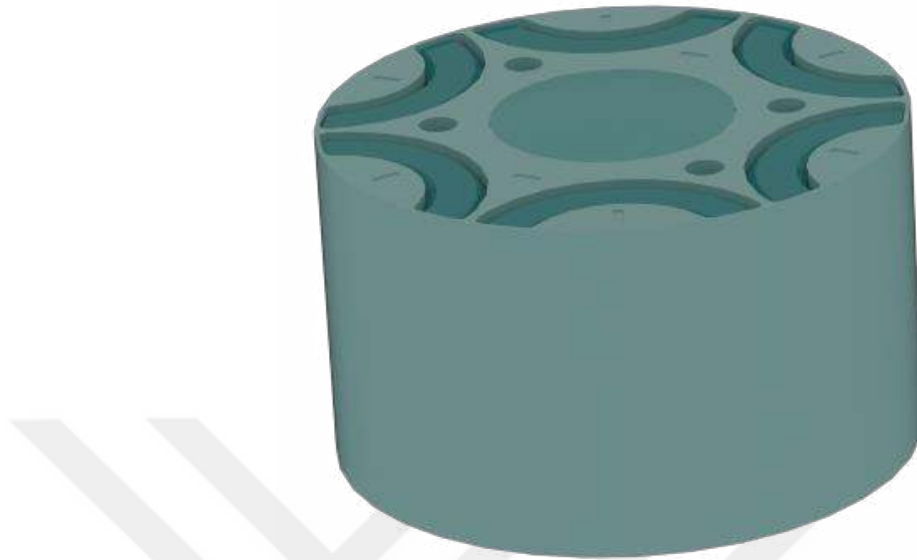


Figure 1.5. Rotor of a compressor.

The largest volume component in the reciprocating compressor is the body itself, which has significant importance in terms of temperature inside the housing during working, vibration, and resisting against the stresses. The larger the surface area that the oil covers, the better cooling performance that can be achieved. Therefore, it is important to design a large body, yet it should have a safe distance to the inner surface of the housing. Similarly, a large, and heavy body is also preferable in terms of vibration suppression. As the mass of body increases, amount of force to shake it increases. Natural frequencies of the compressor also reduce by employing a high-mass body, not only because of mass value, but also because of inertia as well. In addition, it also carries the crankshaft and piston on the journal bearing, thrust bearing, and translational bearing. Thus, it is safe to state that the surfaces on the body are the sources of friction.

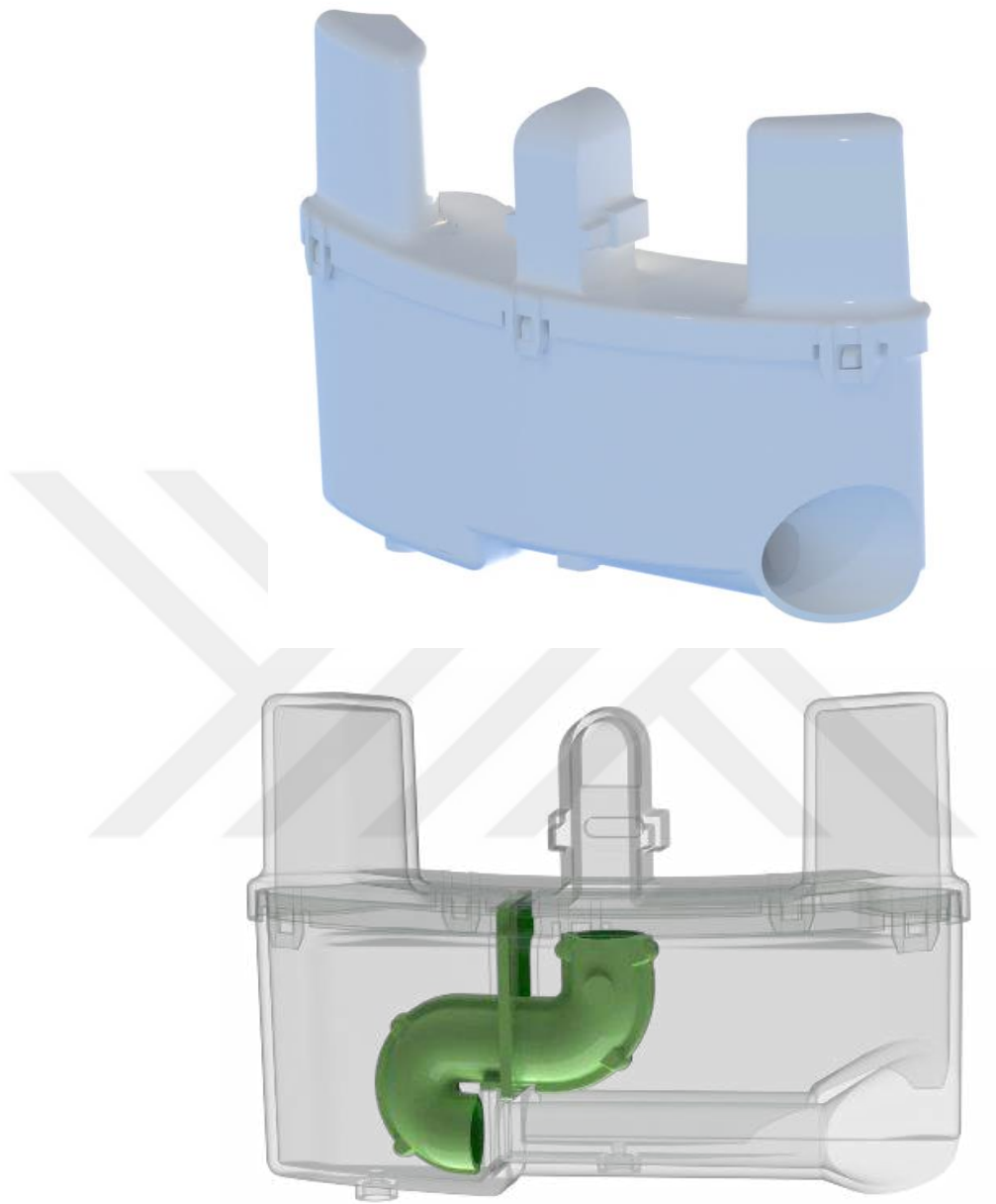


Figure 1.6. Suction muffler of a compressor.

Mufflers are utilized to accumulate the refrigerant in a compressor. While the suction muffler accumulates the refrigerant and directs it towards the piston, the exhaust muffler sends the compressed gas from the cylinder to the discharge tube so that the gas can be exported from the compressor. Especially the design of suction muffler, which affects the temperature levels of the gas, is critical for the efficiency

of the compressor [Lee et al., 2002; Kim et al., 2004]. Moreover, the inner design of suction muffler also affects the acoustic performance of the compressor [Kim et al., 2016] which in turn may disturb customers in houses. Both experimental and numerical studies are available in the literature on the design and performance of the suction muffler [Pereira et al., 2008; Nakano and Kinjo, 2008] which lead researchers to propose new designs [Rahman et al., 2005].

On the other hand, the design of exhaust muffler is relatively simpler than a suction muffler. Yet, it is quite important to reduce the pulsation effect of refrigerant. Such pulsation occurs because of piston stroke, which leads to a gas noise [George and Raj, 2018].

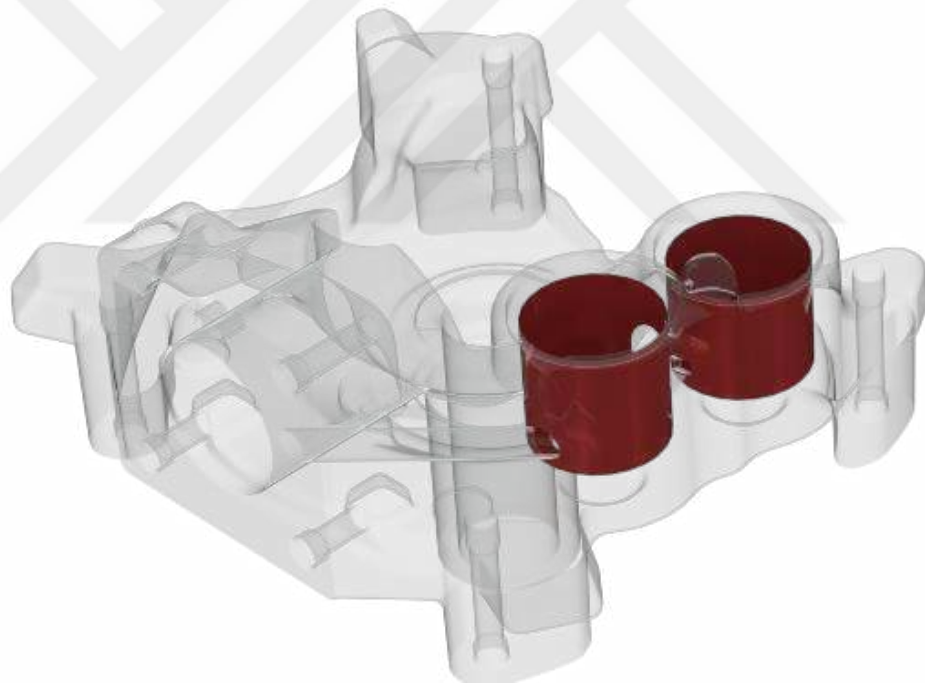


Figure 1.7. Exhaust mufflers of a compressor.

Further, the springs and the discharge tube are the components that fix compressor to the housing and the directional mechanical stiffness values of these components determine the vibration level of the body. The reciprocating compressors are mounted mostly four or three mount-springs to suppress the body vibrations. In

addition, the locations and connection points between housing and the body are extremely critical which must be utilized precisely both during developing numerical dynamic model and experimental optimizations. The locations of the springs affect the suppression capability of the compressor while also changing the natural frequencies of any system [Wu and Whittaker, 1999]. A mathematical model is developed in order to predict the natural frequencies of the 6 degree-of-freedom compressor system as reported in chapter 2.

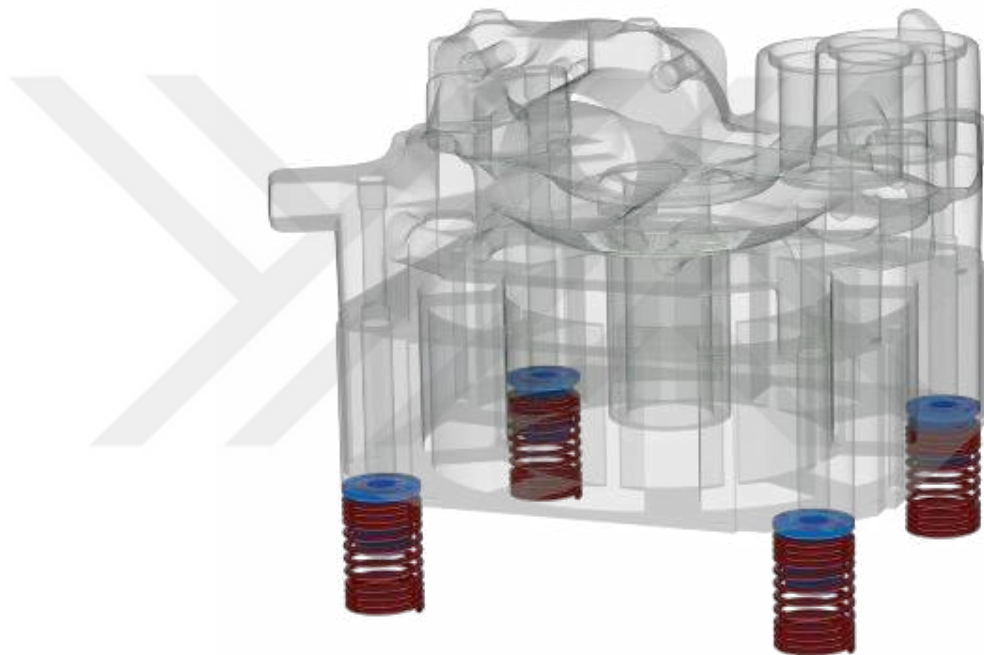


Figure 1.8. Springs of a compressor.



Figure 1.9. Discharge tube of a compressor.

The axial and radial stiffness values of the spring are dependent on number of active turns, the diameter of the spring and the wire, the material, and the total length of spring. On the other hand, the directional stiffness values of the discharge tube are dependent on the length, inner and outer diameter, shapes (radius, straight regions, etc.) and boundary conditions of the tube. The tube must be flexible enough to move with the excitation that is transmitted from the compressor body, and it must be stiff enough to deliver a smooth and successful gas flow. While the shortest tube is preferable to reduce the friction losses of the gas flow, it would result in a stiff tube, which is undesirable. Therefore, the curves are introduced to the tube during the design stage. In addition, the discharge tube experiences the highest amount of stress during startup since the body vibrates significantly [Lenz et al., 2010]. Therefore, the tube must be designed by considering the worst-case scenario. For that purpose, finite element tools would be the key [Krishna, 1994]. However, certain design methods are not presented so far, so it is safe to say that a smart method to design such critical components is needed. The discharge tube must also be exposed to a variety of quality experiments after it is designed since the compressor would end in catastrophic failure if the tube were damaged. The discharge tube design process and experimental studies

are investigated in chapter 2 in detail.

Another vibration suppression element for the compressor is grommet [Vendrami and Pellegrini, 2016] which is the connection part between housing to a plate which is mounted on refrigerator. The material properties of the grommet are crucial for suppression and life-long service. Since it has a suppressing capability, it can be modeled as a nonlinear spring element in a dynamic model [Marriott, 2000]. If the grommet is too rigid, then transmitted vibrations increase [Levecque et al., 2011]. Therefore, an optimum point must be found through either numerical or experimental method. In addition, the specifications of perfect grommet might change based on refrigerator and compressor set due to mass, and boundary conditions.

The housing also important for vibration response. It must be designed carefully by considering the mode shapes so that the resonance frequency is not excited. Moreover, it is also vital for heat transfer among the components, and must be included during a thermodynamic model [Dutra and Deschamps, 2015].



Figure 1.10. Compressor house.

Another vital component in a reciprocating compressor is the oil. The lubrication in the bearings is achieved by the mineral oil, therefore the viscosity of the oil is quite critical. While high viscous oil has better capability to lubricate the bearings, it results in an increase in mechanical losses. As the viscosity reduces, mechanical losses also reduces yet capability of adherence of the oil droplets to the surface decreases.

Therefore, chemical agents are utilized to increase such capability.



Figure 1.11. Oil splash as the crankshaft rotates in clockwise.

The oil is distributed from the oil sump via an oil pump and the helix grooves on the crankshaft based on Archimedes' principle. Then, thanks to the holes on top of crank surface the oil is splashed around the compressor as illustrated in figure 1.11. It might be possible to state that perfect lubrication might not be achieved especially during startup due to low RPMs. Oil is also a coolant element inside the housing, due to friction forces and hot compressed gas, the temperature in the compressor rises and the splashed oil helps to dissipate the heat. Additionally, as the temperature of the oil increases, the instant viscosity reduces, and mechanical losses also reduce. Therefore, it is safe to state that low viscous oil is preferred by considering the lubrication performance. Such mechanical losses can be measured by considering the torque that is emerged because of the frictional forces [Lilie and Kruger, 1990; Kerpici et al., 2019]. A detailed investigation of frictional forces that cause mechanical loss and mechanical loss measurement studies are reported in chapter 2.

Chapter 2

STEADY STATE OPERATIONS

2.1. Background and Literature Review

Although single speed compressors were used mostly in the past, the new era of refrigerator compressors relies on inverter type compressors due to the energy efficiency that is achieved by the variable speeds. Thanks to a controller, lower RPM can be used when there is less need for cooling, such as during the night when the refrigerator door opens rarely, resulting in a lower cooling need.

Further, even though some manufacturers may prefer linear compressors due to their oil-free operation, fewer moving parts, reciprocating compressors deliver a higher volumetric efficiency [Liang et al., 2014].

The household refrigerators circulate a coolant/refrigerant gas and take the heat away from the cabinet and towards the room. Even though there are different available coolants in the world, R600a is the most common gas due to its stellar thermodynamic properties [Kerpicci et al., 2019]. Such gas circulation is achieved thanks to the compressor, as discussed earlier. The refrigerant gas enters the compressor from the inlet, it is compressed, and then flows out to the condenser coils.

The gas enters the compressor from the inlet tube, and it is transferred to suction muffler. A suction muffler collects the gas and directs it to the cylinder. While the piston goes to bottom dead center, the suction valve opens, and the coolant gas accumulates in the cylinder. Afterward, as the piston goes toward top dead center, the gas is compressed, and the exhaust valve opens due to the pressure of the compressed gas.

Later, compressed gas is sent to the exhaust muffler, and then it reaches the discharge tube. The discharge tube is one of the most critical components in the compressor since it transfers the compressed coolant to the condenser tubes, and

connects the compressor body to the housing.

During this process, high amount of forces occur due to compression, friction and, inertia which are forces compressor body to displace. Not only in the direction of piston motion, but also radial effects are observed since the crankshaft is a rotary part. While the crankshaft and the rotor rotate during the compression cycle, the piston experiences a secondary motion due to unbalanced forces [Prata et al., 2001]. This secondary motion occurs such that the piston moves radially inside the cylinder. Therefore, the clearance between the cylinder and the piston is quite critical. Counterweights are mostly utilized in order to balance the unbalanced forces since they are easy to apply and a cheap solution to employ. Yet, while counterweight addition may work in a certain range of RPM, it may not be successful for each running conditions since it is a passive solution. As an additional vibration-suppression element, the compressor is mounted on four springs in order to suppress the vibration and prevent transmitting the vibration to the housing and the refrigerator. In addition, the discharge tube connects the compressor body to the housing, as mentioned earlier, therefore, the tube also must be assumed as a spring in a mathematical dynamic model. Thus, the stiffness of the tube as well as springs play an outstanding role in terms of vibration suppression. High stiffness would result in high vibration transmission, yet considerably low stiffness may result in unsuccessful vibration suppression, and the compressor body may hit the inner surface of the housing. Therefore, an optimum value must be found in order to keep the vibration at an acceptable level.

Furthermore, it is safe to say that the forces and their effects must be understood in detail before optimization studies since the compressor is just a black box without understanding the parameters that effect the performance. To understand the physics that lies behind the dynamic behavior of the reciprocating compressor, there are available studies to be investigated in the literature.

It is worth mentioning that most of the reported studies consider only single components, such as the crankshaft or piston, and only a minority have investigated

the whole compressor. While secondary motion studies dominate the literature regarding piston and effect of piston to the dynamic behavior, torsional vibration is the research topic that the scientists gave a crucial importance for the crankshaft studies. In addition to torsional vibration, orbit prediction and bearing studies are also presented. Both numerical and finite element analyses exist in addition to studies where only experimental work is conducted. Therefore, it is safe to claim that there is a lack of mathematical works where the dynamics and vibration behavior of the compressor are presented by including the whole compressor design parameters.

The piston motion is one of the key factors that affect such behavior. [Kim, 2003] conducted a numerical analysis for the secondary motion of the piston, where the Newton-Raphson iterative solution method is established. The hydrodynamic oil film characteristic is also investigated, where the solution is obtained through the Reynolds equation. Later, the kinematic equations and Reynolds equation are merged on the iterative solutions steps. It is stated that shorter cylinder length results in lower mechanical loss and lower power consumption as expected since the friction region reduces. Moreover, it is demonstrated that if the wrist pin is placed in the middle of the piston, a more stable trajectory can be achieved. Further, full piston trajectory as well as oil leakage behavior with respect to crankshaft angle are also demonstrated.

Similarly, [Silva and Dutra, 2021] investigated the secondary motion and demonstrated a piston trajectory where the hydrodynamic efficiency is increased. Moreover, the expansion and discharge duration were also reduced, which resulted in higher volumetric efficiency as validated by an experimental study. Thus, it reveals the fact that the piston behaviour plays a key role on the thermodynamic efficiency, which yields to an increase in coefficient of performance (COP).

Another study that examines the piston secondary motion where the oil film in the translational bearing is modeled as a spring and damper model is conducted by [Tan and Ripin, 2013]. A degrees-of-freedom model is developed for the piston, and oil film interaction with and without piston slap to the cylinder and lateral motion of the piston are modeled. An experimental setup is built to capture the lateral motion of

the piston via three laser sensors. The friction coefficient is also obtained from the experimental force data, which is the most critical factor in the mechanical losses.

Another contribution to piston motion modeling is made by [Patel et al., 2007] where the piston profile is also involved. Piston barrelity and ovality as well as profile defects due to the mechanical and thermal reasons are also combined. The contact forces of piston and cylinder, piston's lateral motion, the friction forces on thrust and antithrust faces, effect of clearance, and the effect of the piston pin are demonstrated.

While the secondary motion is aimed to be suppressed via design optimization studies including piston size or length, piston profile, clearance, etc., a passive control method is shared by [Dolatabadi, 2017] where a nonlinear pendulum is modeled on the piston in order to damp the secondary motion and reduce the side forces on the cylinder face. The dynamic model is built with and without the pendulum by neglecting any friction among the piston and pendulum joint, and it is shown that including such a control method is more effective than tinkering with design parameters. Yet, it must be evaluated that the manufacturing and assembly of such a pendulum may not be possible due to space restrictions inside the compressor housing and/or piston-cylinder region.

On the other hand, [Cho and Moon, 2005] combined finite element analysis and numerical modeling for piston dynamics and lubrication, respectively. A finite difference approach is utilized for the Reynolds equation based on Newtonian incompressible oil film to estimate the oil film attributes such as film thickness and pressure. It is reported that piston eccentric motion and the film pressure greatly affect the oil film, which is an indicator of secondary motion and will be reflected to the compressor by an increment in the lateral vibration. It is also stated that the wrist pin hole on the piston determines the magnitude of the piston rotation. In other words, the piston would have more stabilized trajectory if the wrist pin position were optimized, as discussed earlier.

Similarly, [He et al., 2014] combined the finite element method with Runge-Kutta iteration to solve piston lubrication by including piston dynamics. The effect of

parameters on frictional losses is investigated, and the biggest reasons for piston slap noises are explained as clearance and pin offset. It is also stated that piston skirt length and curvature of the piston is not significantly affecting the noise. As the clearance decreases, the slap noise reduces, yet higher frictional loss is observed since viscous friction force is increased.

The phenomenon that the previously discussed works were missing/neglecting is the effect of fluctuation on the angular velocity of the crankshaft. The angular velocity, or the RPM changes with respect to crankshaft position due to inertia and the gas force rather than a constant value in one cycle. [Liu et al., 2019] presented a work where it is demonstrated that such fluctuation is increasing the piston slaps which yields in a noise issue with respect to constant velocity assumption. In addition, it is also reported that torsional vibration of the crankshaft is also amplifying the piston impacts on the side walls of the cylinder.

There are also studies that investigate only the crankshaft, the crankshaft's torsional vibration, the modal response, etc. Since the crankshaft is the second main element for rotary motion after the rotor, studies on its response are crucial. [Morillo et al., 2018] calculated the torques which affect the crankshaft by studying the dynamic equations for slider crank mechanisms. In addition, thermodynamic calculations are also conducted to consider the gas force on the piston. Moreover, the oscillation of the angular velocity is also studied, and it is concluded that the flywheel can be a good option to reduce the fluctuation of the velocity. Flywheels with different inertia values are utilized for such conclusions. At this point, it is worth mentioning that eliminating the varying angular velocity would also reduce the vibration level of the compressor since the emerging forces are reduced.

Moreover, [Yu et al., 2012] combined multibody dynamics and finite element models to analyze the stress response of a crankshaft. Time dependent force data is acquired from the MBD tool, and stress is obtained via the FEA. Later, fatigue analysis is conducted for the particular crankshaft so that design could be optimized, if necessary.

[Estupinan and Santos, 2007] utilized multi-body systems for rigid components such as pistons, crankshafts, connecting rod, etc., whereas finite element modeling is used for flexible elements such as springs, discharge tube, including the oil. Since the oil also has damping capabilities, it can be assumed as a flexible element in the dynamic model. To investigate the hydrodynamic relationships between surfaces and the lubrication oil film, the Reynolds equation is solved by considering the crankshaft's secondary motion. Such secondary motion can be described as the tilting response of the crankshaft, especially due to compression force. Even though the crank can experience the tilting only in micron level, it introduces an undesired motion to the system. Therefore, [Estupinan and Santos, 2009] studied the journal bearings to demonstrate the oil film pressure distribution, crank orbit, and the effecting parameters such as the length, diameter, clearance etc. First, the equation of motion for the moving parts is built, and then forces that occur because of oil film is included via the Reynolds equation. The dynamic behavior of the rotor is included thanks to the FEM module. Then whole models are merged into the global matrix system, and a numerical solution is conducted. Thanks to the comprehensive approach, a more precise solution is converged for the crankshaft motion. It is reported that maximum oil forces and minimum fluid film thickness occurs when the piston is near to top dead center. It can be explained by the fact that as the force increases, the oil film thickness starts to reduce since the pressure on the film increases.

In addition to the static, vibratory, and modal responses of the crankshaft, the bearing design also has a vital importance since the bearings welcome the forces. Inadequate design of the journal bearing may result in excessive vibration due to high frictional forces, or even wear if the clearance is not properly adjusted. [White, 1989] reported a study where the orbit of the bearing is predicted under varying running conditions. The finite difference method is chosen to solve Reynolds equation, and the forces on the bearings are calculated based on gas and inertia forces. Then, the minimum film thickness and minimum film thickness location are observed. [Kim et

al., 2002] utilized the Newton-Raphson method to solve Reynolds' equation, and minimum film thickness and location were estimated under static loading. Static loading is achieved by a linear cylinder motor. It is stated that the motor current consumption significantly increases as the forces significantly increases which is an indicator of the bearing's carrying capacity. In addition, Sommerfeld number is utilized as the design success criteria, such that Sommerfeld number increases as the friction coefficient increases in the hydrodynamic regime.

Similarly, [Chasalevris et al., 2013] also studied the journal bearing by considering it as a finite length bearing. An analytical expression is proposed for the hydrodynamic oil film pressure. [Wisbeck et al., 2002], on the other hand, developed a model by considering the dynamic loading effect where the crankshaft's tilting motion is included. Since such secondary motion is calculated, the variation of oil film can also be investigated. Moreover, the worn volume is also integrated with the expressions by following the Archard law. Archard law utilized the hardness of the material and wear constant to estimate the material removal due to harsh running conditions. The crank's trajectory is compared with and without wear and a significant change is observed, as may expected. Such wear may also yield to vibration problems, especially at high rpms.

[Ozdemir et al., 2014] investigated the design parameters of crankshaft, which effect the coefficient of the compressor. The effect of clearance of the bearing, viscosity of the oil, and surface roughness on the mechanical losses is investigated since such parameters are highly related to frictional forces. Moreover, the efficiency that is introduced by optimizing such parameters would automatically increase the coefficient of performance (COP). Later, the optimization studies are intended to be validated via calorimeter tests. It is concluded that as the length of the crankshaft reduces, mechanical losses also reduce. On the contrary, if the clearance value decreases, power loss would increase even though the crank would experience a more stable orbit.

[Duyar and Dursunkaya, 2002] solved the dynamics of reciprocating

compressors and integrated them with elastohydrodynamic lubrication formulations. EHL is solved based on modified Reynolds equation. Minimum clearance and maximum deformation regions, hydrodynamic force is modelled and effect of viscosity, the clearance, bearing diameter, bearing length are investigated on these phenomenons and an optimum point is estimated. It is also concluded that flexible models of components such as conrod, piston, crankshaft and joints would yield more realistic simulation results, rather than rigid models.

On the other hand, the dynamic response of the crankshaft must also be investigated in detail, in addition to mechanical characteristics such as dimensions, stresses to which the crank is exposed, orbits, oil film pressure, etc. Vibration and modal properties are crucial for the moving mechanisms such as crankshaft, piston or conrod. [Kacani, 2017] investigated the vibration response of a compressor. First, a modal analysis and forced mechanical response are conducted, and torsional vibration of the crank is investigated. It is stated that the torsional vibration analysis should be conducted under steady state operations, startup and shutdown transients, and excessive running conditions so that the response of crank against the torsion can be learned.

[Liu et al., 2019] utilized the modal superposition method (MSM) to investigate the torsional vibration response of the crankshaft. The eigenbody approach is established to estimate the natural frequencies and the modal vectors. Then, MSM is employed to study transient characteristics and mode vector contributions. It is concluded that the magnitude of the torsional vibration is highly dependent on resonance frequencies in addition to running conditions.

[Yu et al., 2010] modeled the crankshaft as a 3-node timoshenko beam and solved the equation of motion to determine the natural frequencies with eigenbody approach. The joints and the components are modelled as lumped masses, inertia, stiffness and the damping properties of the components are utilized. Different running conditions, such as undamped vibration, forced vibration, are analyzed to understand the vibrational behavior.

Moreover, [Huang and Lin, 2014] used FEM software to conduct stress and modal analysis for different counterweights. The effect of counterweights on the inertial forces is simulated, the side forces are calculated, and eccentric trajectory of the conrod is plotted. It is stated that an optimized counterweight can reduce the vibration level, which leads to noise reduction.

Similarly, [Seve et al., 2003] also presented a work where the balancing the inverter-type rotor. At the first part of the study, a numerical approach is established where different scenarios are tested, and then an experimental study is conducted in the second part. Balancing planes are selected as rotor only, stator only, and both on the rotor and stator. It is revealed that the third configuration results in more successful suppression capabilities since it allows to mount masses on both the rotor and the stator.

Moreover, [Xie et al., 2019] studied the performance of an electric motor. Even though the study focuses on electrical motors, it is also important for reciprocating compressor since the compressor also carries motors that work within oil. It is reported that the winding connection of the stator is optimized, and the starting torque has been increased, the fluctuation of the electromagnetic force has been reduce and the vibrational response is also improved. Therefore, it is safe to state that an optimization study can also be made by considering the physical parameters of the motor itself.

There are also studies of the vibration response of not only the crankshaft, but also the whole system. Both experimental and numerical approaches can be seen in the literature. [Hai-yang et al., 2015] developed a multibody dynamics solution on a commercial software where the components are joints are defined. A genetic algorithm is used to optimize the stiffness and damping properties of the joint contact and achieve a predictive model of joint clearance faults. The study shows the importance of a mathematical or numerical model so that such undesired designs can be eliminated without extensive experimental work.

[Kjeldsen and Madsen, 1978] emphasized the importance of springs and

counterweights. Springs are the first components that minimize the transmission of vibration from the compressor body to the housing. Therefore, its properties, such as stiffness, diameter, location, wire diameter, outer and inner diameter, etc. must be chosen wisely. Furthermore, since the compressor is just a bulk mass on the springs, the resonance response of the springs is also vital. Since the discharge tube is yet another component that connects the body to the housing/shell, the stiffness properties of the tube are also critical. Moreover, it is stated that there is an optimum point for balancing the compressor via counterweights. In other words, there is an optimum balancing plane. In order to determine such a plane, dynamic equations are established, and it is demonstrated that vibration reduction can be achieved with such a method. In addition, it is also safe to state that if the center of gravity of the compressor is close to the piston centerline, less vibration will be observed. This can be explained by the fact that the forces experienced by the piston pushed the compressor backward and created a torsion effect. Therefore, if CoG can be adjusted in a way that it is close to piston's centerline, then the vibration would be less.

In a dynamic model of the compressor, grommets are also vibration suppression elements thanks to their non-linear elastic properties. Mostly, four grommets are used to connect the compressor to the sheet plate of the refrigerator. [Wu et al., 2016] defined the dynamic model of reciprocating compressor in a way that unbalanced forces result in vibration, springs and the discharge tube are the first components to deal with the vibration, and grommets as well as outer connection pipes play a suppressor role. First, the three-axes vibration response of the compressor is measured, and then natural frequencies are obtained through an impact hammer test so that the whole dynamic response and mapping are obtained.

[Kuyumcuoglu and Sakalli, 2014] utilized grommets with different properties and investigated the transmissibility capability of the grommets via experimental works. In addition to reducing the vibrations, the sound power level is also decreased at related frequencies. The authors reported that the vibration and noise of household refrigerators can be lessened via grommet optimization.

On the other hand, the exhaust process is another key parameter that affects the vibration response of the compressor since the compression gas force dominates the vibration. [Lee et al., 2000] employed different valves with different masses, and stiffnesses to observe the discharge pressure pulsation, acoustic impedance. It is concluded that the impact noise can be reduced with low mass or high stiffness. Yet, it is noted that high stiffness may result in an increase in the losses. Moreover, it is stated that low band frequency noise can be reduced with multiple plenums, yet such additions may also increase losses. It is also worth mentioning that the design and mass of the discharge valve must be made by considering the natural frequencies to prevent any resonance during compression.

[Silviera, 2004] listed the noise sources in a compressor as the compression system, including valves, mufflers, discharge tubes, etc., moving mechanisms, and the motor. Transmission paths are defined as discharge, suction, as well as suspension paths. Then, the noise contribution of such sources in a compressor is also investigated. Then, a new suction muffler is proposed, as well as new suspension springs and discharge tubes, to obtain lower noise values. After the optimizations, noise and the vibrations are reduced by more than 30%. This study demonstrates the importance of component designs on vibration and noise level. Therefore, smart methods rather than conventional trial-and-error must be established in order to enhance the performance of the compressor both in efficiency and the vibration. For instance, section 2.3 reports a smart method to design discharge tube by considering dynamic, modal and static response. Moreover, such approaches also reduce the time consumption during the design stage.

In addition, [Erol and Gurdogan, 2000] investigated the effect of discharge ports on vibration and noise levels. The compression force is the most dominant force for the compressor as discussed earlier, and it enhances the amount force and amount of vibration levels. Therefore, an optimized path for the compressor gas could help reduce the vibratory behavior. Four different discharge ports with varying diameters and profiles are designed and manufactured. Variable rpm and pressure values are

employed to generate a response map. It is seen that larger ports with chamfer designs result in low noise and low vibration levels.

Furthermore, vibration and noise can also be utilized for fault detection purposes. For instance, [Bardou and Sidahmed, 1994] and [Kamran et al., 2019] presented works where the accelerometers are used to detect the leakages of more significant problems for the compressor. By establishing signal processing techniques such as time-frequency analysis or non-stationary regressive modeling, vibration data is used as an indicator of the data disturbance.

In addition to vibration, the modal characteristics of a component are also critical in terms of its dynamic behavior. While components are excited with certain frequencies, one better to be careful about resonance phenomenon and excite the system away from resonance frequencies. Therefore, it is vital to know the natural frequency region for every moving system. In reciprocating compressors, modal response is formed not only by springs, discharge tubes, or mass properties, but also by the shell geometry, material, and boundary conditions. [Krishna and Durgaprasad, 2019] presented a finite element analysis to conduct a modal analysis on the housing/shell of the compressor. First, boundary conditions are defined, and analysis is run, and then experimental validation is conducted via the impact hammer tests. Afterward, various shell designs are proposed, yet these designs are manually created to increase the first natural frequency since the first mode is the closest to running conditions. However, it is better not to forget that the vibration and resonance issues are multidimensional issues, and it is not possible to attack the solution by considering only single components.

It is worth emphasizing one more time that a comprehensive mathematical dynamic model that covers each component of a reciprocating compressor is vital for design, improvement, and optimization studies, especially in trade-off decisions.

The dynamic analysis of a reciprocating compressor by [Kurka et al., 2012] and [Kim and Han, 2014] and [Pont et al., 2017] shed light on the effect of system parameters to structural forces during the steady state region. [Kurka et al., 2012]

presented a study where the focus is on flexible bearings for the compressor. Such study allows to predict the orbit of crankshaft which may affect the compressor performance and power loss, as well as stiffness and damping properties of the bearings. [Kim and Han, 2014] investigated the dynamic behavior by considering the finite and short bearing models for the journal bearing. It is reported that the finite bearing model results in higher power consumption, yet lower oil leakage than the short bearing model since circumferential pressure is neglected. On the other hand, [Pont et al., 2017] reported a numerical dynamic analysis by including secondary motion of piston, compression chamber, motor torque, and oil film effects and investigated the effects of parameters on the compressor performance metrics such as losses or power consumption. Therefore, it is safe to say that the model that is presented by [Pont et al., 2017] is a complementary study to predict the forces in a reciprocating compressor.

Overall, the design parameters, such as dimensions, clearances, etc. are the factors that affect compressor behavior. Thus, even a parametric optimization study for the sake of reducing the vibration output can be conducted after a successful mathematical model is built.

2.2. Dynamic Model Development for the Reciprocating Compressor

The reciprocating compressors contain various force sources, from gas force to hydrodynamic force due to oil film in the bearings, friction forces, and inertial forces. Gas force, indeed, is the most dominant force in the compressor since the refrigerant gas can be compressed up to 9 bars if certain temperature and velocity requirements are satisfied. In this point, it is worth mentioning that sealing of the compressor must be made very carefully, in fact sealing is made via welding operation. Since the compressor housing is welded, leakage from the compressor is impossible, yet this situation makes the compressor vulnerable to maintenance. If a component is damaged inside the housing, then the whole compressor must be replaced. Due to this

fact, the design of the components must be made by considering the long service life of the compressor, and the safety factor must be determined wisely.

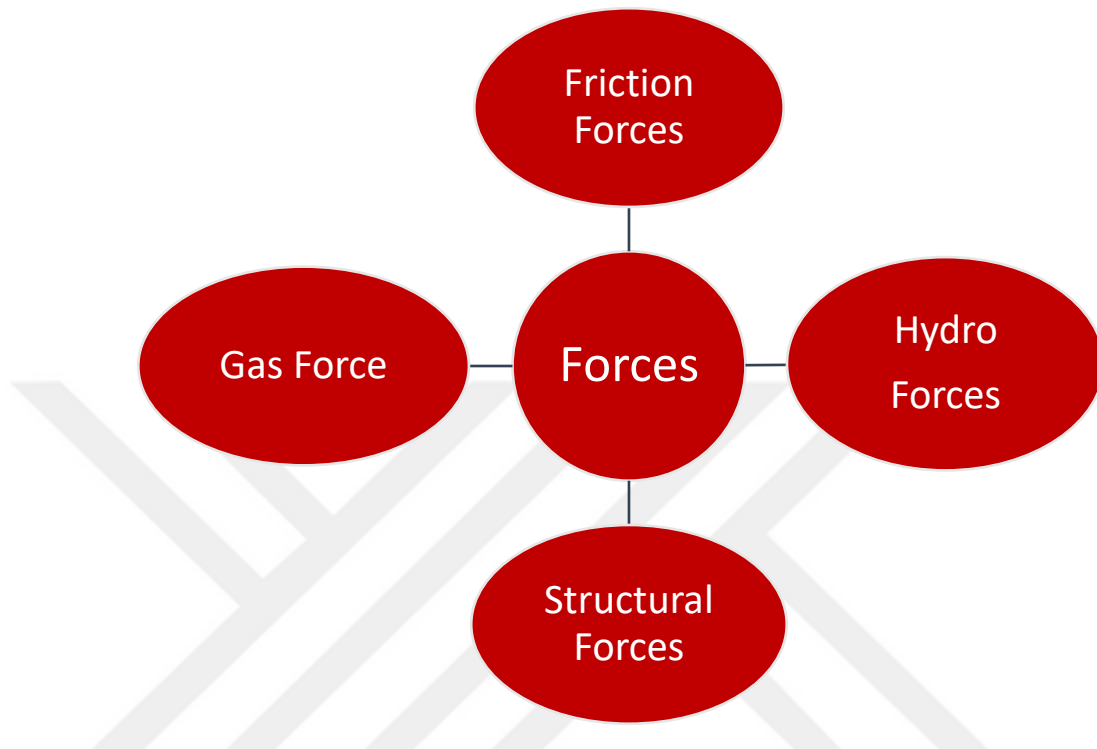


Figure 2.1. Forces in a reciprocating compressor.

Inertial, structural, and frictional forces, which are listed in figure 2.1 play a vital role, especially during startup and shutdown. High amount of electrical current is supplied to the compressor during startup in order to overcome the dry friction and inertia of the system since both phenomena introduce the resistive torques to the moving mechanism. Apart from the startup, frictional forces add mechanical losses to the compressor during steady state, which are not desired and reduce the compressor's efficiency. The change in mechanical losses can be detected either in specially designed experimental setups or in calorimeter tests. On the other hand, inertia forces introduce a vibration to the system that results in noise as well.

To optimize the design parameters against structural forces, mathematical models are the key tools so that the forces and vibrations can be predicted based on input

parameters such as lengths, diameters, clearances, etc. The important portion of the design parameters for a reciprocating compressor is shared in Figure 2.2.

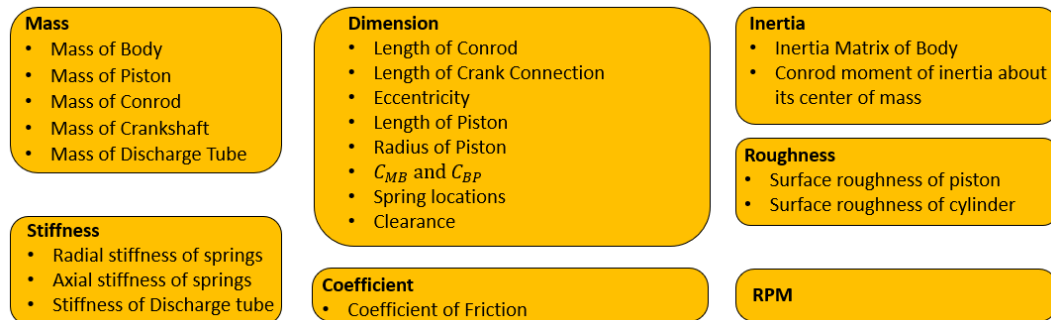


Figure 2.2. Some of design parameters of reciprocating compressor.

It is known that a reciprocating compressor consists of a slider crank mechanism that demonstrates a harmonic periodic motion. In the case of harmonic motions, the natural frequencies of the system play a fundamental role not only in transients, but also in steady state operations since they are used to determine safe working conditions so that the excitation frequency is away from the natural frequencies. If the excitation frequency of the harmonic mechanism is very close to the natural frequency, then excessive and detrimental vibrations occur. Especially for a reciprocating compressor that lies on springs, the natural frequency of the system is directly related to the springs', and small excitation frequencies are sufficient to force the system into resonance. In contrast, there is a trend, due to energy regulations, to operate the compressor at low revolutions per minute, as low angular velocity results in less mechanical loss and less energy consumption. Therefore, an optimum point must be found either with the conventional trial and error method, or mathematical models. It is obvious that such mathematical models or finite element models reduces the design and optimization time consumption.

Thus, the natural frequencies of the reciprocating compressor are predicted in this work first of all.

2.2.1. Natural Frequencies of Reciprocating Compressor

The investigated reciprocating compressor rests on four springs and directly connected to the housing over the discharge tube. Therefore, it is safe to claim that five different objects are utilized to connect the body to the housing. Such connections are intended to keep the body safe within the housing by keeping it from striking the inner shell and preventing vibrations from being transmitted to the housing. Therefore, damping properties and the stiffnesses of these components are critical and must be studied in detail.

Due to the six degrees of freedom of the compressor body, a 6x6 mass matrix (M) is created for modal analysis. The mass of the benchmark compressor body is denoted by m , and the numerical values of the inertia matrix are derived from the CAD program (Siemens NX):

$$M = \begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & I(1,1) & I(1,2) & I(1,3) \\ 0 & 0 & 0 & I(2,1) & I(2,2) & I(2,3) \\ 0 & 0 & 0 & I(3,1) & I(3,2) & I(3,3) \end{bmatrix} \quad (2.1)$$

where m is the mass, and I is the inertia matrix. The stiffness matrix is constructed by integrating the equations of motion for each degree of freedom into the Lagrangian equation of motion. Since the discharge tube joins the body and compressor housing in a flexible manner, it is viewed as a fifth spring. The distance between each spring and the center of gravity of the compressor body is measured, and the transformed stiffness matrix is represented as:

$$K = \begin{bmatrix} k_x & 0 & 0 & 0 & k_x z & -k_x y \\ 0 & k_y & 0 & -k_y z & 0 & k_y x \\ 0 & 0 & k_z & k_z y & -k_z x & 0 \\ 0 & -k_y z & k_z y & k_z y^2 + k_y z^2 & -k_z x y & -k_y x z \\ k_x z & 0 & -k_z x & k_z x y & k_z x^2 + k_x z^2 & -k_x y z \\ -k_x y & k_z x & 0 & k_y x z & -k_x y z & k_y x^2 + k_y y^2 \end{bmatrix} \quad (2.2)$$

where K is the stiffness matrix, x , y , and z are the relative distances to the center of gravity, while k_x , k_y , and k_z are the directional stiffness values. The total stiffness is computed by:

$$K_{total} = K_1 + K_2 + K_3 + K_4 + K_5 \quad (2.3)$$

where $K_1 \dots K_4$ correspond to the identical spring's stiffnesses and K_5 is the discharge tube's stiffness. Assuming no damping or external forces exist, the motion equation can be expressed as follows: [Wang et al., 2013]:

$$M\ddot{s} + Ks = 0 \quad (2.4)$$

where s represents the n -dimensional displacement vector. The non-trivial solutions of the eigenequation of Eq. 2.4 will reveal the eigenvalues (λ_i), which represent the natural frequencies of the system [Meirovitch, 2001; Jazar, 2013]:

$$\omega_i = \sqrt{\lambda_i} \quad (2.5)$$

Using the eigenvalue method, the six natural frequencies (three translational and three rotational) of the compressor assembly are computed analytically. This approach also produces modal vectors. To validate this approach, experimental impact hammer tests are performed on the investigated compressor, and the experimentally measured natural frequencies are compared to the analytically computed natural frequencies. A DAQ is utilized with an impact hammer and a triaxial accelerometer. The data acquisition program is utilized to gather and analyze the data. The impact hammer tests

are performed sequentially for each mode by aligning the accelerometer and hammer tip excitation along each degree of freedom on the compressor body (Fig 2.3a, 2.3b).

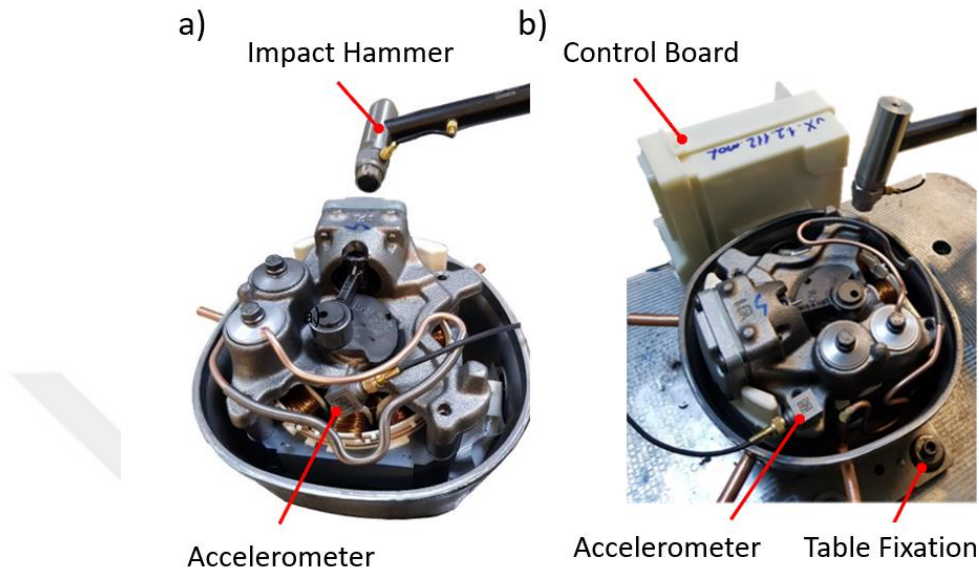


Figure 2.3 Impact hammer test for finding the natural frequencies of the compressor. (a) x-axis translational mode. (b) x-y axis rotational mode.

Moreover, spring and discharge tube stiffness parameters are determined by experiment. As illustrated in figure 2.4, the outlet end of the discharge tube is fixed, while the inlet is left free to be stimulated by a linear actuator. The fixed end is attached to a table-type dynamometer, and the linear actuator's displacement is monitored using a laser sensor.

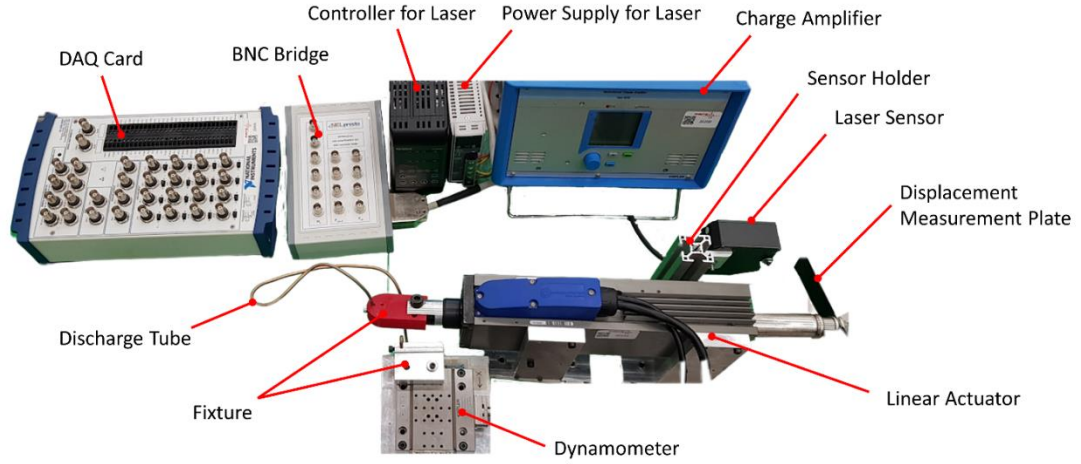


Figure 2.4. Stiffness measurement setup.

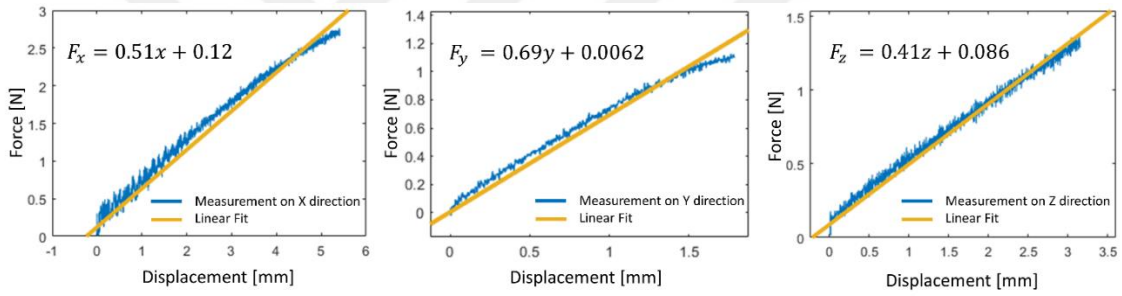


Figure 2.5. Force-displacement data for X, Y and Z axis for the discharge tube.

Measurements are taken while the linear actuator is moved. The best line fits of the recorded force-displacement data are derived to get the stiffness values, as shown in figure 2.5. The value of directional stiffness is derived from the slope of the fitted curves.

Moreover, another experimental setup is established to measure the stiffness of the springs. 3D printed parts are used to mount the spring on a fixture and to push the spring from top. The setup is assembled on table-type dynamometer and laser sensor is placed on top of the setup to measure the axial displacement of the spring. In addition, the applied force is also read thanks to the dynamometer. Setup is shared in figure 2.6.

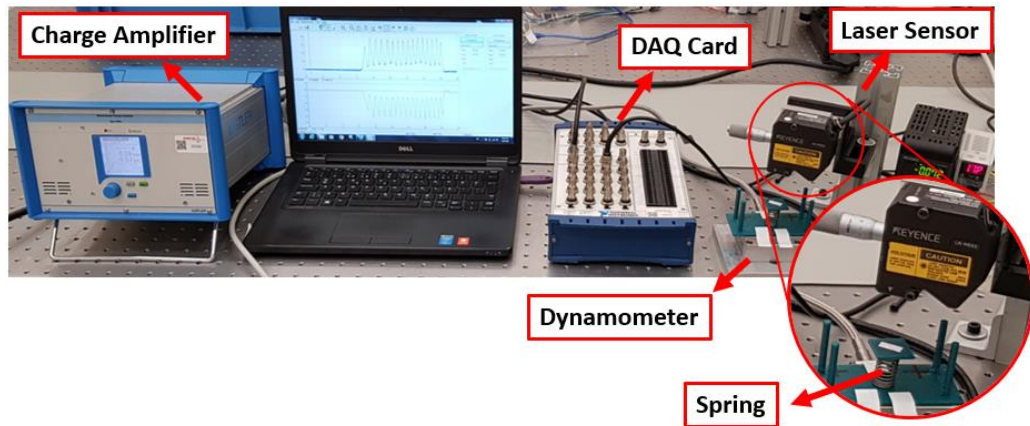


Figure 2.6. Spring stiffness measurement setup.

The same experiment is repeated 110 times for 11 springs that are manufactured at the same batch and no significant difference is observed. It is seen that the springs demonstrate a hysteresis behaviour as depicted in figure 2.7. such that loading and unloading responses are different. Nevertheless, it is assumed that no-hysteresis occurs due to the fact that calculated stiffnesses are almost the same for loading and unloading.

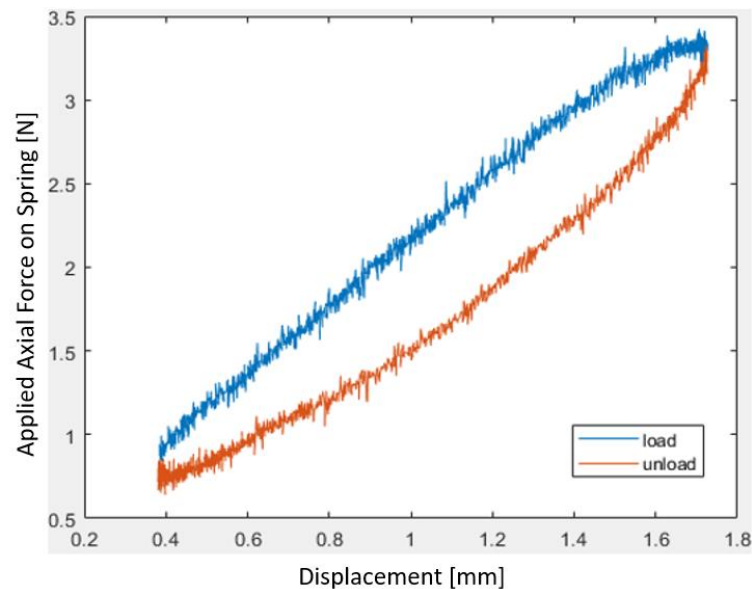


Figure 2.7. Stiffness of the spring in case of loading and unloading.

Then, the stiffness values acquired in the experimental studies are implemented

into the stiffness matrix and the eigenvalue problem is solved on Matlab. The 6th natural frequency mode of the compressor body is close to the lower bound of the excitation frequency range and avoiding detrimental resonance is critical.

Table 2.1 compares the empirically determined natural frequency modes (w_n) of the compressor with the analytically computed modes. Due to the agreement of the results, it can be inferred that the eigenvalue methodology for determining the natural frequency modes is a reliable way for making natural frequency predictions.

Table 2.1: Estimated and experimentally recorded natural frequencies.

Mode #	$\omega_{experimental}$ (Hz)	$\omega_{estimated}$ (Hz)
1	5.75	7.28
2	7.75	7.70
3	9	8.23
4	13	11.69
5	13.75	12.43
6	14.75	14.11

Moreover, by adjusting design characteristics such as stiffnesses or the position of connecting points between components and the housing, it is possible to minimize the natural frequencies and offer low steady-state operating rpms.

Further, the reciprocating compressor works based on the simplest dynamic mechanism in the world: slider crank mechanism which is shown in figure 2.8 and the forces that emerge during cycles of such system are investigated at the next chapters. The majority of the Chapter 2.2.2 is published in 2022 at International Compressor Engineering Conference at Purdue University [Oral et al., 2022a].

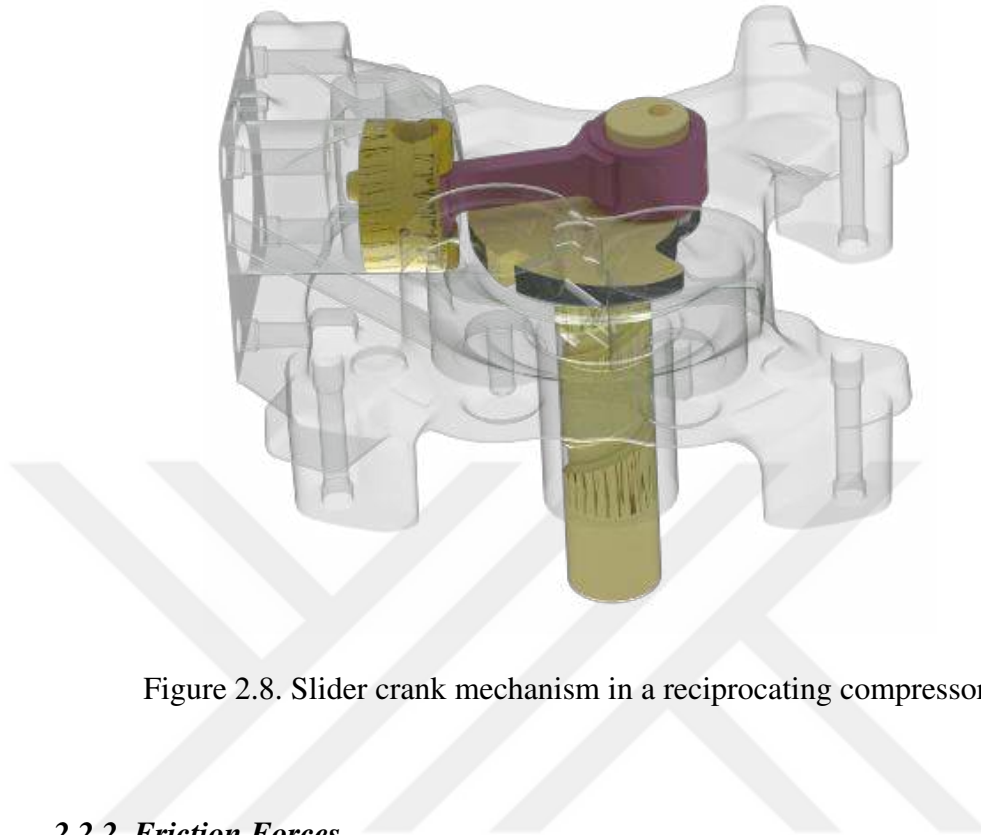


Figure 2.8. Slider crank mechanism in a reciprocating compressor.

2.2.2. Friction Forces

Due to their continuous operation to preserve food, household refrigerators are regarded as one of the most significant energy users globally among other household equipment. Consequently, increasing the energy efficiency of household refrigerators is crucial nowadays. The compressor, which is comprised of a crankshaft and an electric motor, is the principal source of energy consumption in a household refrigerator. The existence of frictional forces between the various moving components of a compressor greatly reduces its performance and raises its total energy consumption. A reciprocating compressor has three principal sources of frictional force: forces on journal bearing owing to crankshaft rotation, forces on piston-cylinder contact due to piston translation and secondary motion, and frictional forces on thrust bearing due to crankshaft mass. There are both experimental and numerical methods for determining frictional losses. [Mantri et al., 2014] employed Reynold's equation using finite-difference modeling to determine pressure gradient,

forces, and power loss on piston-cylinder contact, according to a numerical study in which the Reynolds equation is used to estimate losses on journal bearings [Posch et al., 2016].

This research assesses mechanical losses resulting from frictional forces on piston-cylinder contact, journal, and thrust bearings. After that, the estimations were confirmed using a recently developed experimental setup.

To evaluate the mechanical losses, an experimental setup similar to that of [Rao et al., 1996] was developed, in which the compressor is enclosed and operated externally. A torque meter was installed between the crankshaft and the external motor to measure the torsion caused by the motor's excitation and the forces of friction on the torque meter's rods. The torque readings were then multiplied by the crankshaft's angular velocity to estimate mechanical losses.

Theoretical Calculations

To get the mechanical loss values of the compressor owing to frictional forces, it is necessary to study each component separately. Therefore, friction forces on the piston-cylinder contact are examined first, followed by forces on the journal bearing and thrust bearing. By multiplying the forces by the path of the item, the friction force's work may be determined. In addition, losses may be calculated by multiplying the resulting torques by the constant angular velocity.

The piston's reciprocation generates a friction force that is a combination of lubricating oil shear and surface asperities. It is determined by the crank angle, piston length, piston diameter, lubricant shear pressure, friction coefficient, contact pressure, and slider/piston velocity. In this analysis, secondary motion is ignored for the piston and crankshaft. The thickness of the oil layer is considered to be constant, and the hydrodynamic pressure change over the piston surface is disregarded. Also, it is believed that the piston and cylinder surfaces are perfectly flat. Also, the influence of gas force is eliminated.

The friction force that occurs due to the piston movement is defined in Eq. 1 [Patir and Cheng, 1978].

$$F_f = -\text{sign}(U) \int_0^L \int_0^{2\pi} (p_{ls} + \mu_f p_c) d\alpha dz \quad (2.6)$$

where U is the velocity of piston, yet the signum function only considers the direction of velocity. L is introduced as the length of piston, p_{ls} is lubricant shear pressure on piston, μ_f is friction coefficient and p_c is the contact pressure between solid-solid surfaces i.e. asperities. The parameters are illustrated in figure 2.9.

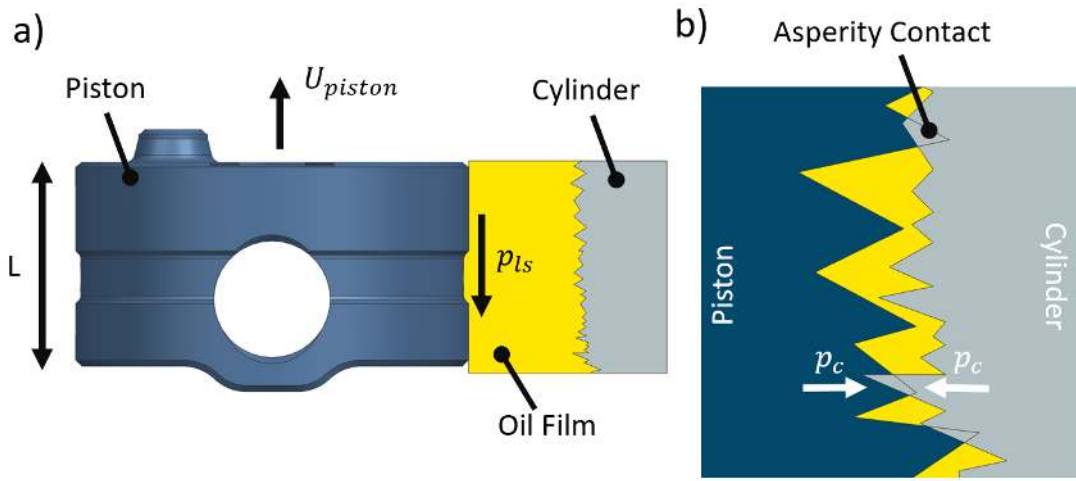


Figure 2.9. (a) Lubrication oil film and piston, (b) asperity contact in case of unsatisfactory oil supply.

[Greenwood and Williamson, 1966] created a statistical form of contact model [Jedynak and Gilewicz, 2013] which asserts that the interaction between two rough surfaces may be modeled by contact between a single rough surface and a flat surface. It is considered that rough surfaces exhibit asperities with spherical peaks. In addition, it is asserted that all asperity peaks have the same radius, but their heights vary randomly.

In addition, in the event of contact between two surfaces, although certain asperities deform plastically and elastically, asperities with comparatively shorter heights retain their original form. [Greenwood and Tripp, 1971] claimed that while

the height of an individual asperity is random, the distribution of asperity heights is quite similar to a Gaussian distribution. In addition, average contact pressure is expressed as;

$$p_c = KE'F_{2.5}(H_\sigma) \quad (2.7)$$

$$K = \frac{8\sqrt{2}}{15}\pi(X\beta'\sigma)\sqrt{\frac{\sigma}{\beta'}} \quad (2.8)$$

where X is number of asperities per unit contact area, E' is composite modulus of elasticity and the function $F_{2.5}(H_\sigma)$ relates the probability distribution of asperity height i.e. rough surface with Gaussian distributed asperities [Hu et al., 1994].

$$E' = \frac{2}{\left(\frac{1-v_1^2}{E_1}\right) + \left(\frac{1-v_2^2}{E_2}\right)} \quad (2.9)$$

Probability distribution of asperity height is also given as;

$$F_{2.5}(H_\sigma) = \begin{cases} A(\Omega - H_\sigma)^Z & H_\sigma \leq \Omega \\ 0 & H_\sigma > \Omega \end{cases} \quad (2.10)$$

The Ω is an assumed constant which is equal to 4. In addition, parameter H_σ is given as the ratio of oil film thickness over surface roughness.

$$H_\sigma = h/\sigma \quad (2.11)$$

where h is the oil film thickness.

The clearance between the cylinder and piston is approximately 7 microns in diameter, and the surface roughness is (Ra). Using a White Light Interferometer, an arbitrary surface of 1.2 mm by 0.9 mm was measured to be around 300 nanometers for the examined compressor; experimental data is presented in figure 2.10.

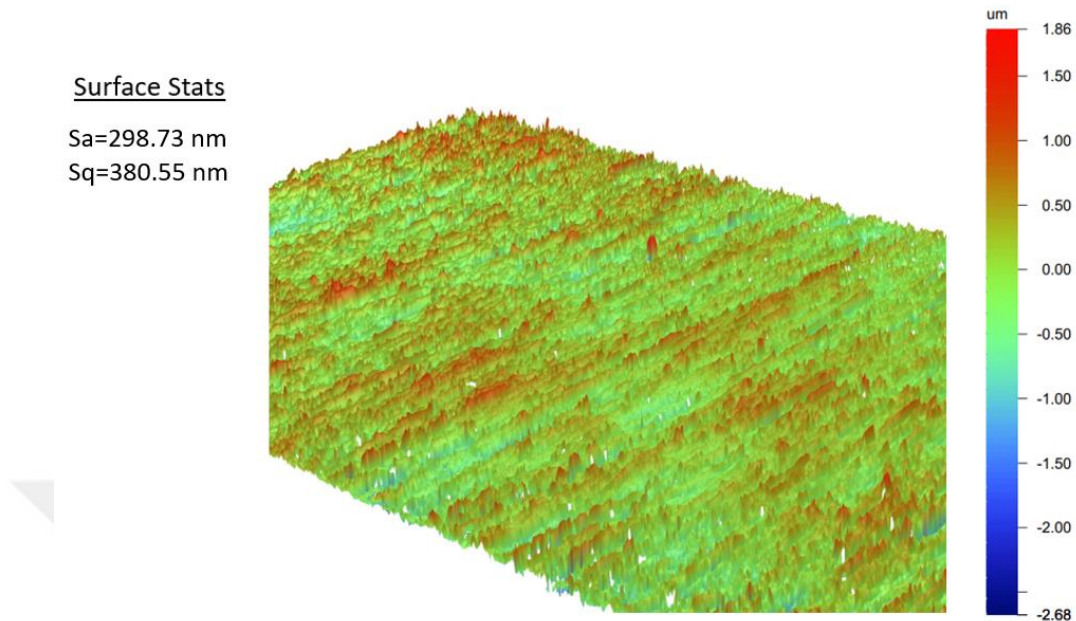


Figure 2.10. Measured surface of piston.

While asperity values range from 1.86 to -2.68 micrometers in height, where a negative value implies a valley, the average roughness was measured to be 288 nanometers. In addition, as seen in figure 2.11, asperity heights do indeed reflect a Gaussian distribution.

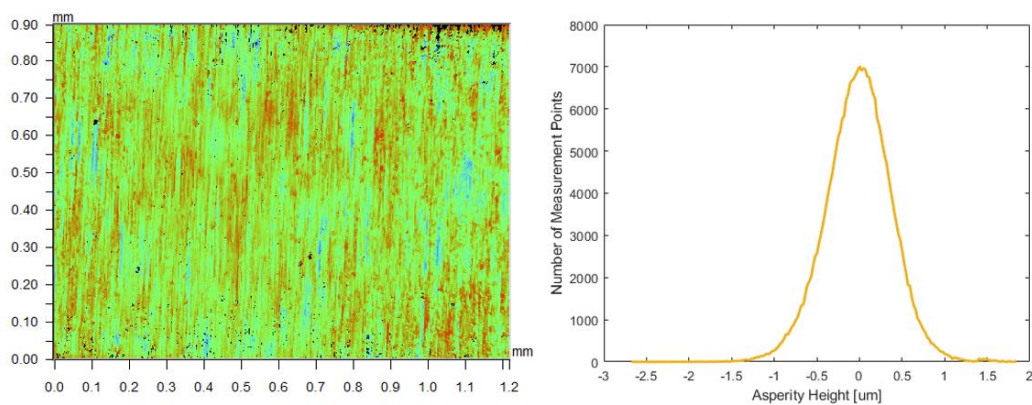


Figure 2.11. Gaussian distribution of asperities on the surface.

Since the ratio of oil layer thickness to surface roughness is greater than 4, it is reasonable to infer that the value of the probability distribution of asperities may be

assumed to be zero for the analyzed components based on equations 5 and 6. Consequently, contact pressure is disregarded in this investigation. In other words, during steady state, the forces generated during piston reciprocation are carried only by the oil layer and not by asperities.

As specified by [Zhu et al., 1992] the lubricant shear pressure acting on the surface may be estimated in this scenario.

$$\tau = -\frac{\gamma U}{h} (\Phi_f + \Phi_{fs}) + \Phi_{fp} \frac{\bar{h}}{2} \frac{\partial p_h}{\partial y} \quad (2.12)$$

Average shear stress factors, $\Phi_f, \Phi_{fs}, \Phi_{fp}$ utilized to examine the surface waviness and roughness effects and are shared in Equations 8, 9, and 10 for the case of a piston surface with zero waviness.

$$\Phi_f = \frac{h}{2\sigma} \ln \left(\frac{h+\sigma}{h-\sigma} \right) \quad (2.13)$$

$$\Phi_{fs} = \frac{3}{2} \left[\frac{h}{\sigma} \ln \left(\frac{h+\sigma}{h-\sigma} \right) - 2 \right] \quad (2.14)$$

$$\Phi_{fp} = 1 - \left(\frac{\sigma}{h} \right)^2 \quad (2.15)$$

Newton's viscosity law is used to calculate the friction forces on journal bearings caused by crankshaft rotation [Jia et al., 2018].

$$\tau = \gamma \frac{U}{h} = \gamma \frac{2\pi r N}{c} \quad (2.16)$$

and force can be written as the multiplication of shear stress with surface area;

$$F = \tau A = \gamma \frac{2\pi r N}{c} 2\pi r Z \quad (2.17)$$

where μ is viscosity, c is clearance. It is assumed that clearance is fully filled with oil as a pure uniform film and oil thickness is kept constant. Thus,

$$F \approx \pi D_b Z \gamma \left(\frac{\pi D_b N}{\bar{h}} \right) \quad (2.18)$$

where N is speed of crankshaft, $\bar{h}$ is clearance, Z and D_b are length and diameter of the bearing respectively.

While rpm has a considerable influence on the force value, it is also important to note that the effect of gas force on losses is removed and crankshaft tilting is not accounted for in the calculations. The viscosity of oil was determined based on the oil temperature recorded using thermocouples during a comprehensive compressor temperature measurement experiment.

Since the investigated reciprocating compressor comprises two journal bearings, the complete length of the bearings must be utilized.

To estimate frictional forces on the thrust bearing, a straightforward method using the mass of the crank as a surface load was adopted [Nagata et al., 2012].

$$T = \int_A^0 dT = \int_A^0 \mu_k F r dA \quad (2.19)$$

and

$$T = \mu_k \left(\frac{F}{\pi(R_0^2 - R_1^2)} \right) \int_{R_1}^{R_0} r 2\pi r dr \quad (2.20)$$

Thus;

$$T = \frac{2}{3} \mu_k F \left(\frac{R_0^3 - R_1^3}{R_0^2 - R_1^2} \right) \quad (2.21)$$

where μ_k is the coefficient of friction between a cast iron-cast iron material pair in an oil-containing environment [Barrett, 1990], F is the force created by the mass of crankshaft, R_0 and R_1 are the outer and inner contact diameters of the thrust bearing as shown in figure 2.12.

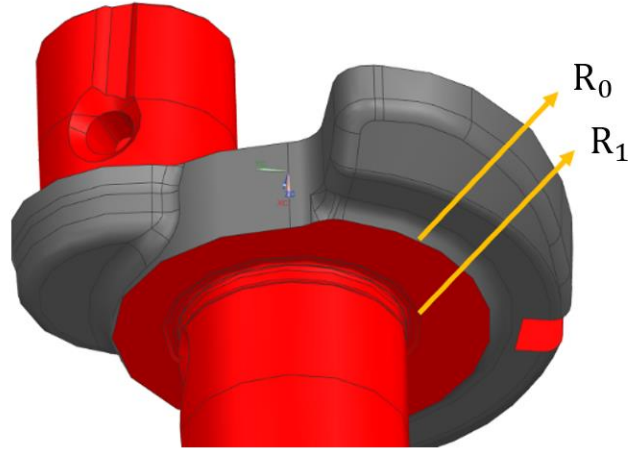


Figure 2.12. Inner and outer diameter of thrust bearing.

Experimental Setup

In a prior experimental setup [Kerpicci et al., 2019], the crankshaft's deceleration and inertia were employed in such a way that a gear and clutch mechanism attached to the crankshaft was removed once the compressor reached steady state rpm. After considering the shaft's angular velocity and angular deceleration, friction work was determined.

This method was modified since extra tools, such as a gear and clutch, were necessary. By separating the compressor from its springs and discharge tube and attaching it to a specialized fixture, a more accurate experimental setup simulating real-world conditions was created. The compressor was driven by an external motor, with the motion being conveyed to the compressor through couplings. An eccentric rod was used to eliminate the eccentricity of the crankshaft rotation axis and crank-conrod bearing, and it was attached on the coupling's upper surface. A torquemeter was positioned between the compressor and the motor to measure the torsion difference, $\theta_1 - \theta_2$, caused by the frictional forces of the compressor and the motor's excitation. The setup is shown in figure 2.13.

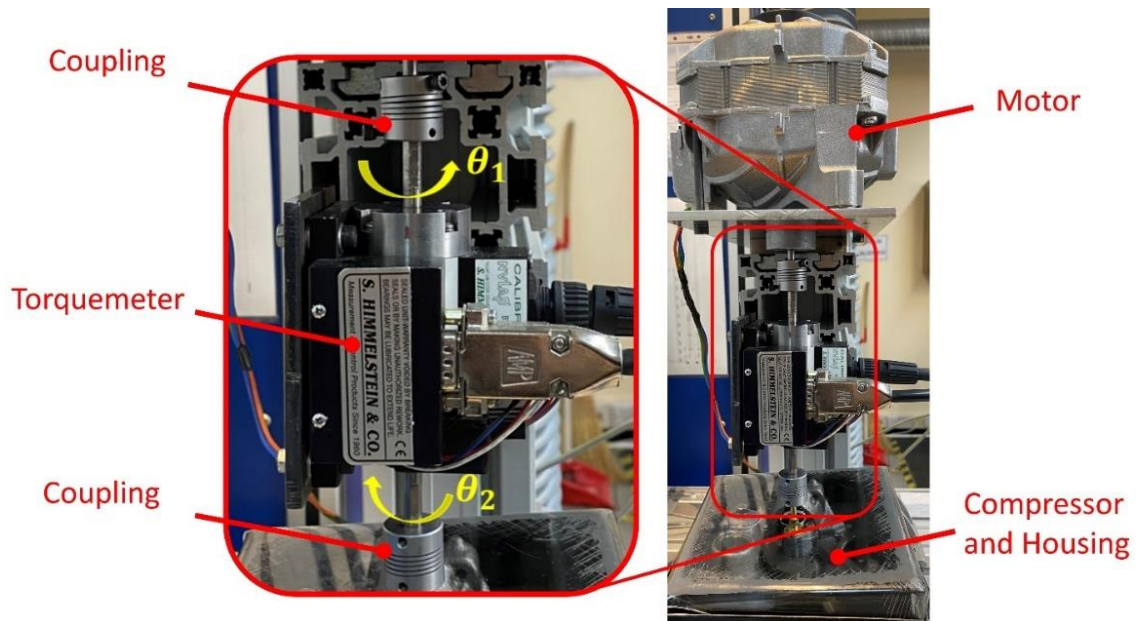


Figure 2.13. Torque measurement setup.

Thermocouples are utilized in calorimeter testing to monitor the oil temperatures at various rpms during steady-state operation. Under the housing of the compressor, a heater with a feedback loop was installed in order to manage and raise the temperature of the oil to a steady-state level in order to imitate the operating conditions temperatures in domestic applications. Throughout the studies, the oil temperature was maintained at the required level via the feedback control method. To protect the oil within the housing, a plastic plate was fitted to the housing's top. In the experimental setting, five thermocouples were used to measure the temperature of the oil and compressor body. While four thermocouples were put at the corners of the housing within the oil, the fifth was linked to the body near the cylinder through a hole. After achieving temperature equilibrium, measurements were taken for one minute and repeated four times. The torque value was then determined as the average of four separate trials.

Results

Multiplying the total of torques by a fixed rotational velocity can be used to get the mechanical losses of the compressor, or the losses of specific components can be determined separately. By using Eq 1-9 from 0 to 360 degrees crankshaft angle, friction forces on piston-cylinder contact for one crank rotation may be calculated relative to a reference variable f .

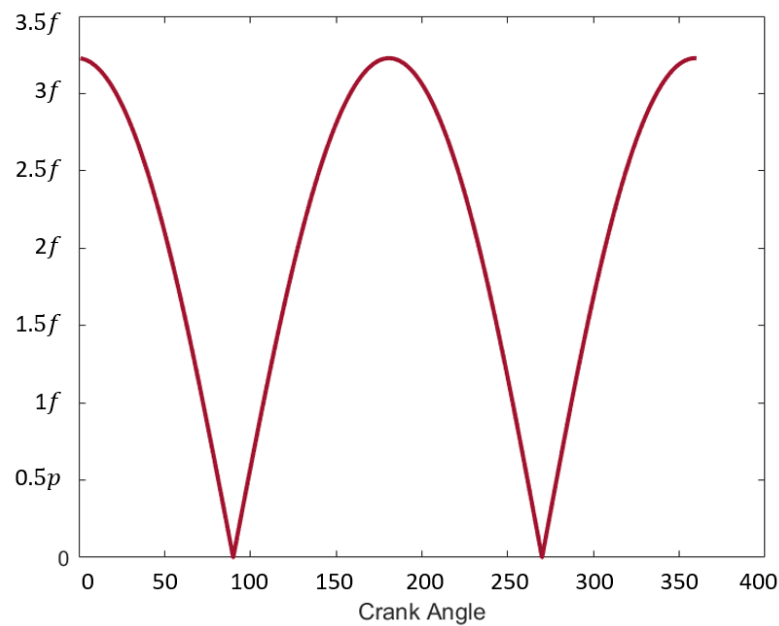


Figure 2.14. Friction force with respect to crank angle.

As expected, the lowest friction force values were achieved at 0° , 180° , and 360° , i.e., at dead positions, as shown in figure 2.14. Moreover, at 90 and 270 degrees of crank angle, friction forces reach their highest value due to the physical fact that the crank pushes the piston towards thrust and antithrust zones, so exposing the piston to greater friction. The friction work, or losses due to piston motion, may be estimated by multiplying the friction force by the stroke of the piston over the duration of a cycle. It is important to note that the secondary motion of the piston is ignored in the calculations. Also, rms values were applied to account for the impact of both the lowest and maximum piston loss values in a single crank cycle.

In addition, as the goal of journal bearing loss computations was to determine losses in the steady-state zone, the rpm was held constant throughout a cycle. The torque generated by the friction force of the crank at the axis of rotation and the torque that occurs on the thrust bearing as a result of the mass and rotation of the crankshaft were multiplied by the angular velocity and used to anticipate loss.

The amount of mechanical losses was calculated by adding the individual losses on piston-cylinder, journal, and thrust bearings under the assumption that the magnitude of the losses remains constant throughout steady-state operation.

Once the thermocouple readings reached the required temperature, the experiments were initiated. To evaluate repeatability, the tests were performed four times for 60 seconds, and the average torque values were recorded. Then, the obtained torque values were multiplied by the angular velocity to determine losses caused by friction forces alone. In addition, 10 cSt of oil was used in the studies, and its temperature was regulated based on real-world operating conditions at each rpm.

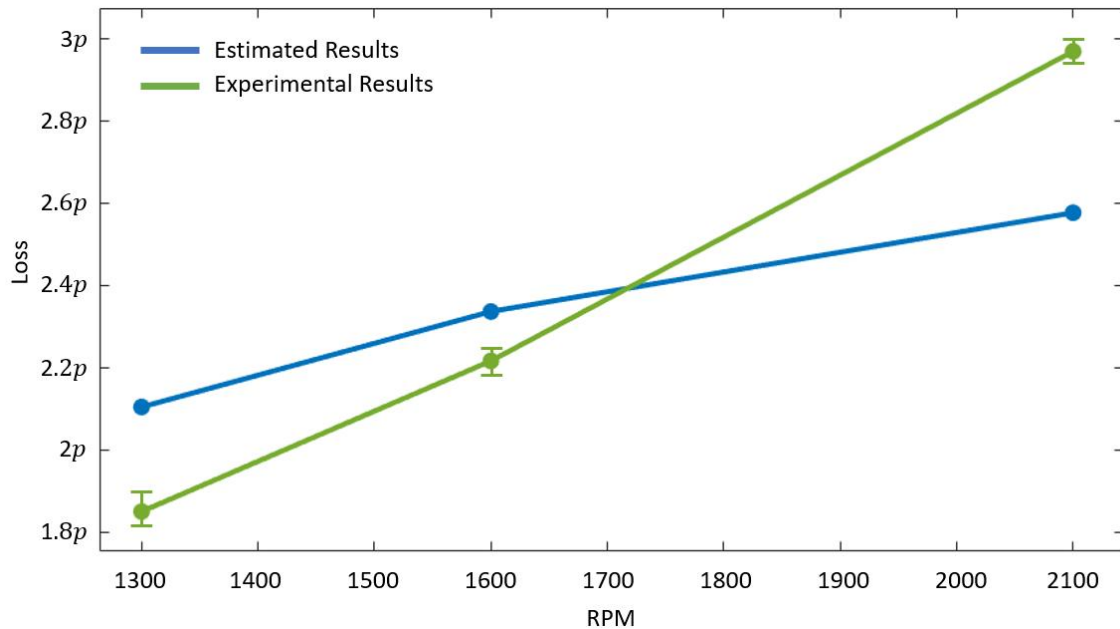


Figure 2.15. Experimental loss values and estimated loss values.

The theoretical calculations and experimental mechanical loss data are depicted in figure 2.15; values are presented relative to the reference variable p . It is safe to assert that the experimental study meets the repeatability requirement. Experiments and estimations differ by 12 percent, 5 percent, and 13 percent for 1300, 1600, and 2100 rpm, respectively.

At 1300 rpm, the moving mechanism is expected to experience decreased side forces and secondary motion. Nonetheless, oil film accumulation may not be uniform. Since it is assumed in calculations that the oil film around the crankshaft is a pure, homogeneous film, the error difference between 1300 and 1600 rpm is substantially greater. In the instance of 1600 rpm, the side forces are less than 2100, but oil distribution around the crank is achieved more effectively than at 1300 rpm, since it is known that as rpm increases, oil feed from the crank grooves is performed more effectively. Therefore, omitting secondary movements does not greatly impact the outcome, and the difference in inaccuracy between theoretical estimates is fairly minor. Even though oil suction from the oil pool is no longer an issue when the steady state rpm reaches 2100, side forces begin increasing dramatically. Since secondary movements are ignored from calculations, the influence of secondary movement of both the piston and crank plays a more significant role at high speeds in tests, and the error value rises.

Overall, mechanical losses are undesired and must be reduced to develop an energy efficient compressor. The viscosity of the oil is the key factor on mechanical loss. As the viscosity reduces, the mechanical loss reduces but the capability of oil droplets to hold the surface also reduces. In other words, the success of lubrication reduces. In order to use low viscosity oil to have less mechanical loss, chemical agents might be used to enhance the lubrication effect.

Moreover, the bearing design also plays a vital role on mechanical loss values. As the clearance of the bearings increase, thicker oil film can be formed but trajectory or the orbit of the piston and crankshaft destabilizes which yields to higher side forces. As a results of higher side forces, even wear might be observed on the surfaces.

Therefore, an optimum clearance value must be determined by considering both the loss and the side forces.

2.2.3. Structural Forces

In addition, the gas force has the highest value among other forces in compressor. Even though piston experiences the compression, the force is transferred to crankshaft over the connecting rod. The FBD of the components as well as affecting forces are shared in figure 2.16. Throughout the calculations, the gas force is evaluated as the main input trigger of the system,

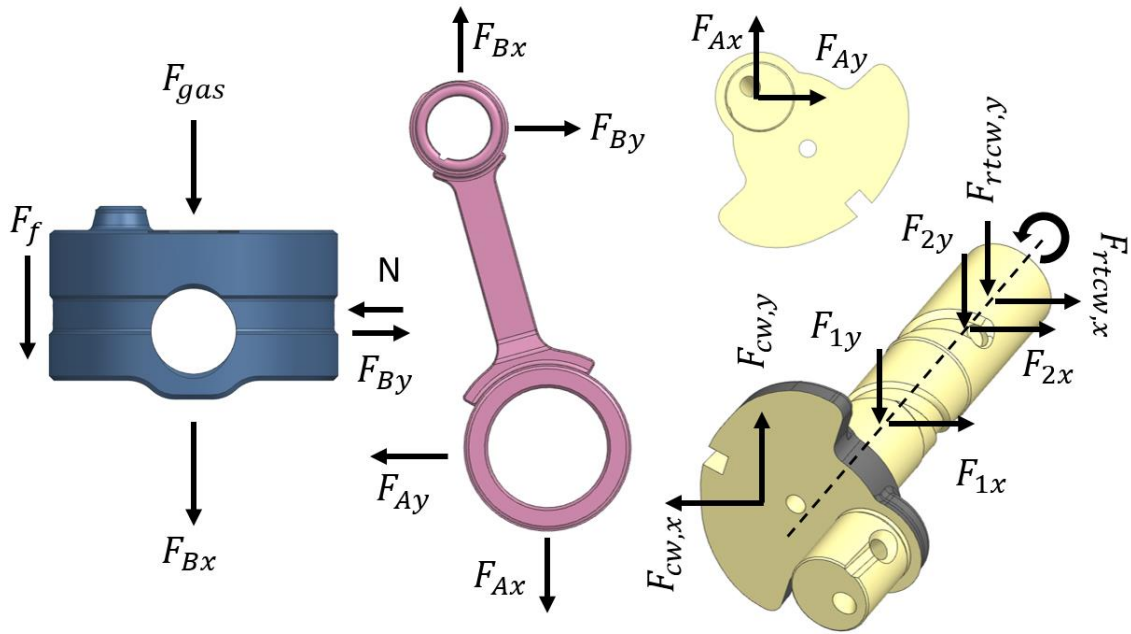


Figure 2.16. Free body diagrams.

While the piston demonstrates reciprocation, it also experiences a radial motion called secondary motion due to the clearance. Axial and radial forces that emerge in the compressor can be found as [Prata et al., 2000; Link and Deschamps, 2011]:

$$\sum F_x = 0 = F_{1x} + F_{2x} - F_{cw,x} - F_{rtcw,x} - F_{A,x} + F_{crank_{inertia},x} \quad (2.22)$$

$$\sum F_y = 0 = F_{1y} + F_{2y} - F_{cw,y} - F_{rtcw,y} - F_{A,y} + F_{crank_{inertia},y} \quad (2.23)$$

where

$$F_{A,x} = m_b a_{bx} - F_{B,x} \quad (2.24)$$

and

$$F_{B,x} = m_p a_p - F_{gas} - F_{friction} \quad (2.25)$$

Centrifugal force formula can be utilized to determine the effect of counterweights of crankshaft and of rotor on the crank's rotation center as equation 2.26:

$$F = m\omega^2 r \quad (2.26)$$

Angular velocity of the crank was obtained from the encoder data. Therefore, the rpm is known as a function of time. It is assumed that the forces acting on crank bearings such as gas force, inertia forces, friction forces, enforce compressor body to displace and vibrate. Therefore, it is safe to state that the magnitude of body displacements due to harmonic excitation depends on the bearing forces.

The following force matrix can be used to calculate the system's structural forces and emerging torque:

$$\begin{bmatrix} F_{Ax} \\ F_{Ay} \\ F_{1x} \\ F_{1y} \\ F_{2x} \\ F_{2y} \\ T_m \end{bmatrix} \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ -1 & 1 & 1 & 0 & 0 & 0 & 0 \\ 0 & -1 & 0 & 0 & 1 & 1 & 0 \\ h_m \sin \alpha & h_m \cos \alpha & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & -c_1 & -(c_1 + c_2) & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & c_1 & (c_1 + c_2) & 0 \end{bmatrix} = \begin{bmatrix} m_c a_{bx} - F_{Bx} \\ m_c a_{by} - F_{by} \\ m_{crank} a_{kx} \\ m_{crank} a_{ky} \\ h_1 m_{crank} (a_{kx} \sin \alpha + a_{ky} \cos \alpha) + I_k \ddot{\alpha} + G_{12x} r_g \sin \alpha + G_{12y} r_g \cos \alpha \\ m_{crank} h_1 \sin \alpha \\ -m_{crank} g h_1 \cos \alpha \end{bmatrix} \quad (2.27)$$

After performing the matrix solution, force and torque values relative to crank angle may be obtained. For these calculations, PV diagram that is shared in figure 2.17 is utilized for various rpms for ASHRAE condition.

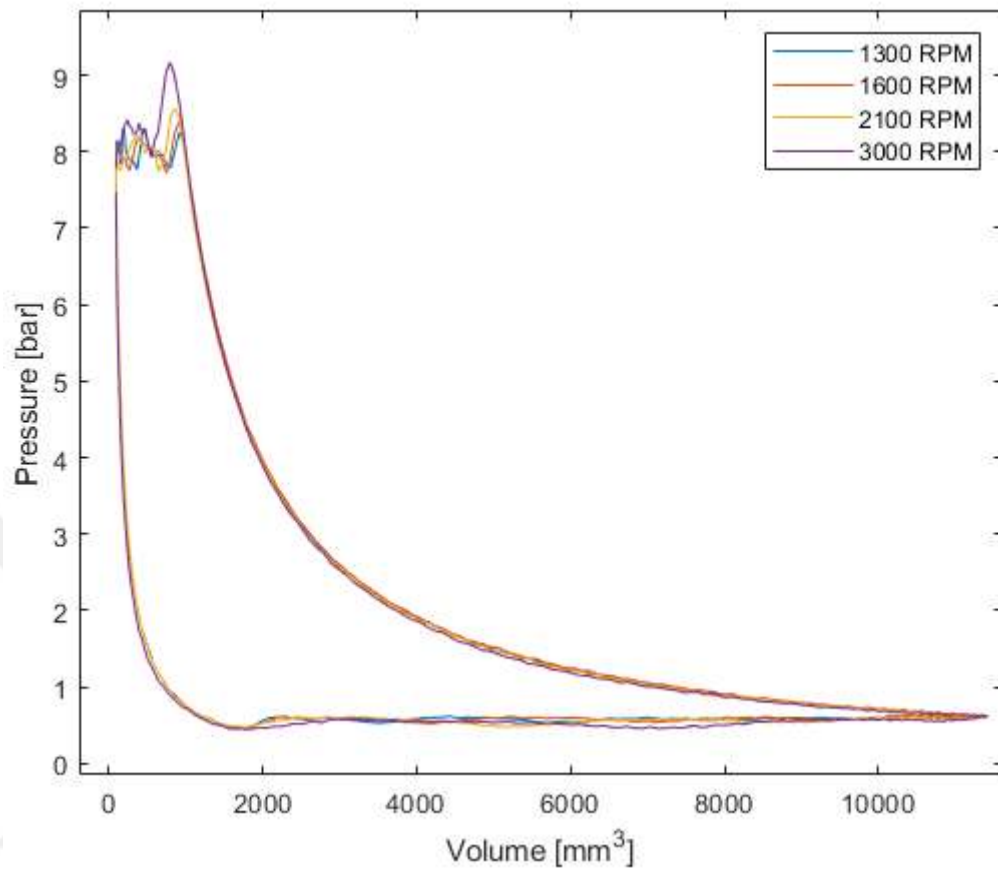


Figure 2.17. Pressure-volume diagrams for various rpms at ashrae condition.

The crank bearing forces as well as net forces along with piston and along with perpendicular axis to piston are plotted in figure 2.18 and 2.19, respectively.

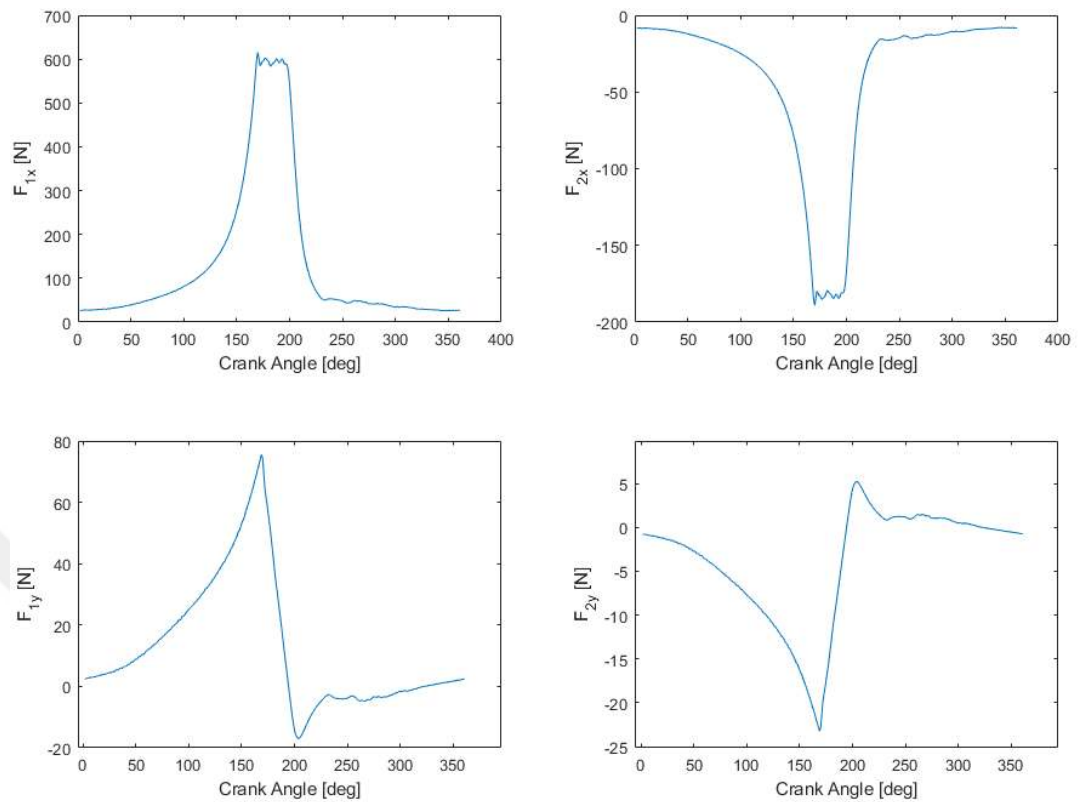


Figure 2.18. Forces on the crank bearings.

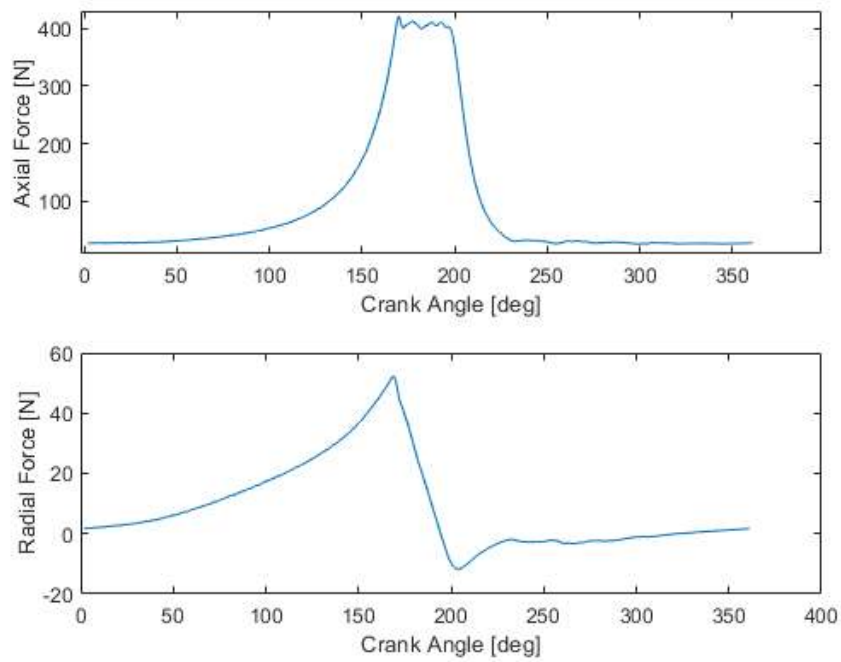


Figure 2.19. Net axial and net radial force.

Briefly, dynamic relationships are established in order to predict the structural forces. In order to validate the approach, experimental studies are conducted. However, direct force measurement on the compressor is not possible. Therefore, additional solutions should be introduced. The crankshaft is mounted on a rotary dynamometer, and the dynamometer is assembled to 5 axis machining center as shown in figure 2.20. The spindle is rotated with various rpms and axial forces are measured via the rotary dynamometer. The utilized setup only measures the forces that occurs on crankshaft because of inertia and the counterweight. In order to consider only these forces in calculations, pressure i.e., gas force, mass of piston and connecting rod are assumed zero.

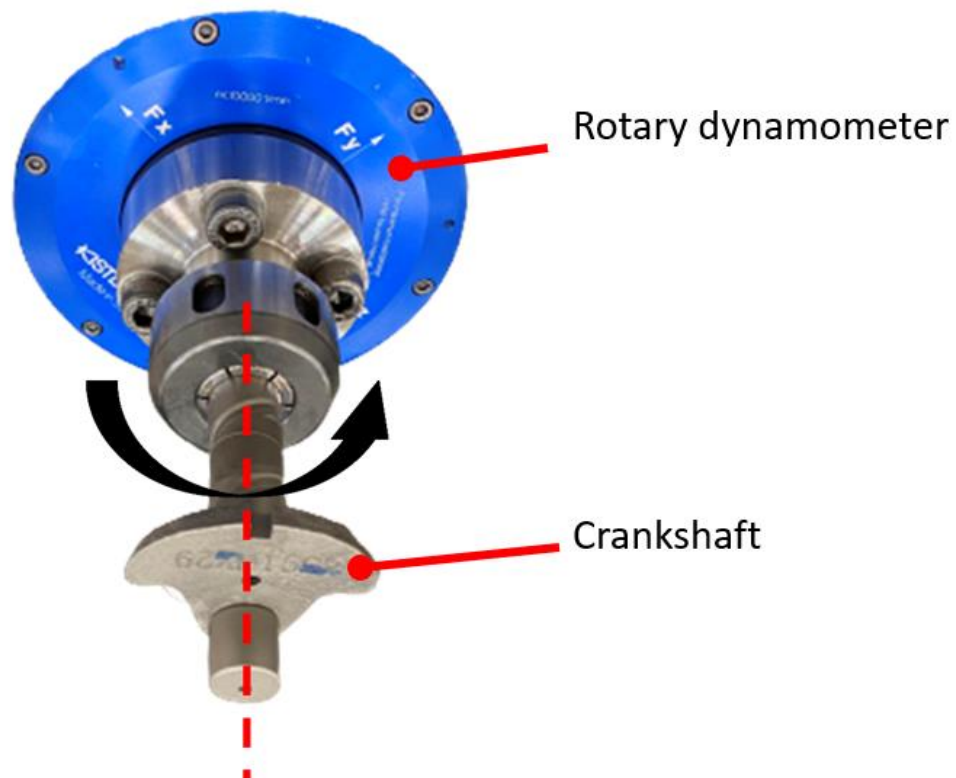


Figure 2.20. Force measurement setup.

RMS of the calculation and the experimentally measured forces are compared in table 2.2 and it is safe to state that the established prediction method is valid.

Table 2.2: Predicted and measured crankshaft forces.

RPM	Measured Force [N]	Predicted Force [N]	Error %
1600	2.98	2.97	0.33
2100	5.54	5.26	5.05
2500	7.59	7.28	4.08
2700	8.94	8.49	5.03
3000	11.21	10.49	6.42

Furthermore, it is considered that the forces operating on crank bearings are the body excitations that result in body displacements or vibrations, given that these forces comprise the effects of gas force, inertial forces, and frictional forces. It is therefore acceptable to assert that the amplitude of harmonic behavior is dependent on crankshaft bearing forces.

By examining modal vectors, the Eigenbody/Eigenvalue technique may be used to forecast the displacements or vibrations of a body under harmonic stress. The studied reciprocating compressor has three translational and three rotational natural mode forms, and coupled displacement solutions are unavailable for mechanisms with a large number of degrees of freedom. Decoupling for each axis and linearly superposing the result is thus the simplest way [Meirovitch, 2001].

In addition, the harmonic response of systems with n-single DOF is expressed as: [Meirovitch, 2001]:

$$q = \sum_{r=1}^n \frac{u_r^T Q_0}{\omega_r^2 - \alpha^2} u_r \cos \alpha \quad (2.28)$$

where ω_r is resonance frequency, Q_0 is the amplitude of force along with investigated principal axis, n is the number of degrees-of-freedom, α is the frequency of the excitation and u_r is the mode vector. Notably, the excitation frequency is derived from the change in angular velocity as the compressor accelerates. The angular velocity information was collected from the magnetic absolute encoder. Integrating

natural frequencies into the process of establishing a linear superposition yield in displacement prediction. As a result, the body displacement can be evaluated in relation to time depending on the startup position. Using the pressure-volume data for the gas force input, body displacements are displayed as illustrated in figure 2.21 for various RPMs between 1300 and 3000.

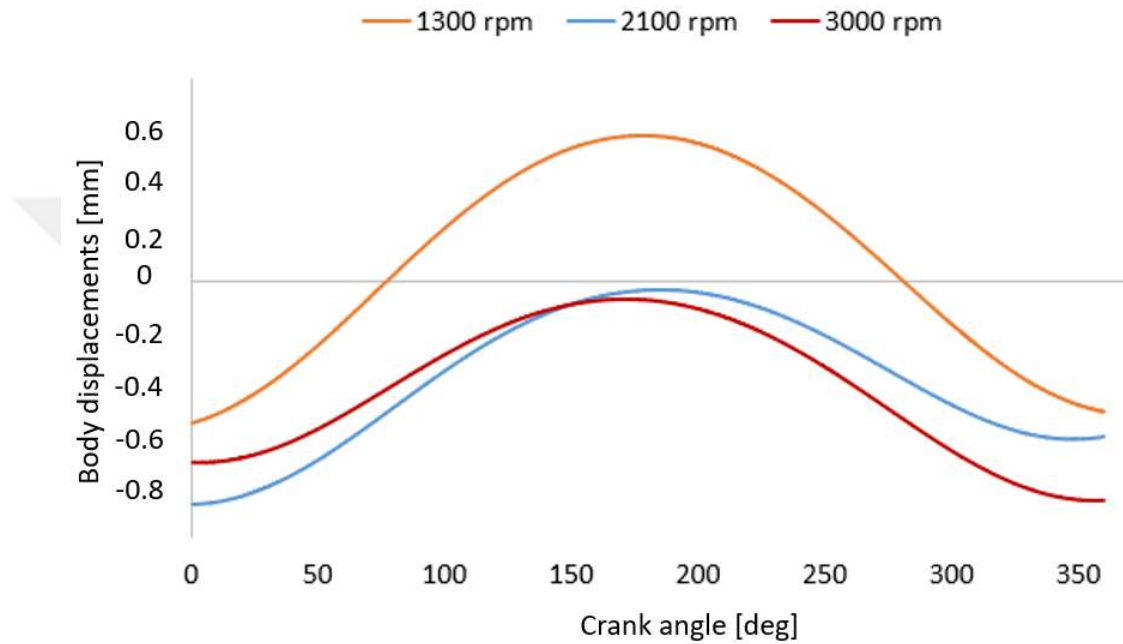


Figure 2.21. Body displacements along with piston axis for various rpm and force scenerios.

Moreover, the effect of various design parameters can also be studied and their effects on displacements can be investigated. Such parametric studies demonstrate the power of mathematical models during design stage and pave the way for the possible improvement. The effect of crank throw distance, i.e, the distance between crank-conrod bearing center to crank's rotation center, various conrod lengths and conrod mass on structural forces are investigated and demonstrated as figure 2.22, 2.23, 2.24, 2.25, 2.26 and 2.27, respectively.

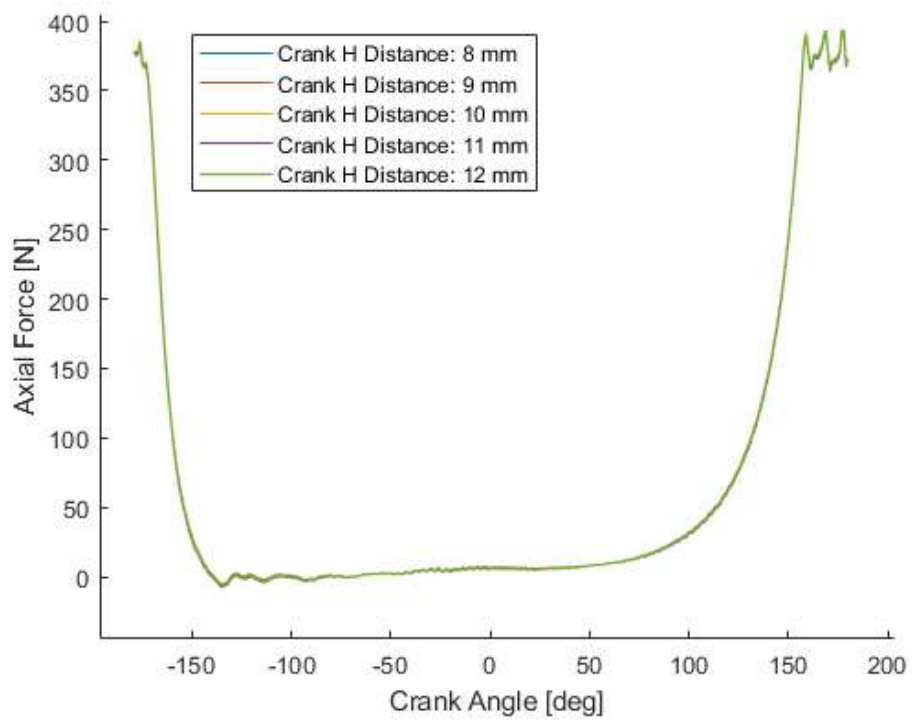


Figure 2.22. The effect of crank throw distance on net axial force.

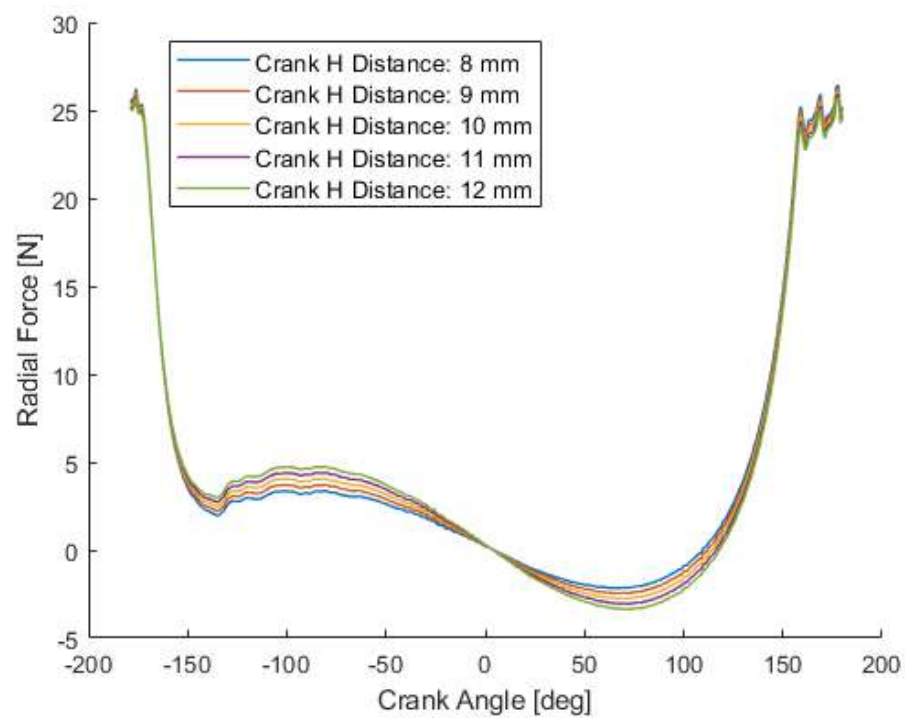


Figure 2.23. The effect of crank distance on net radial force.

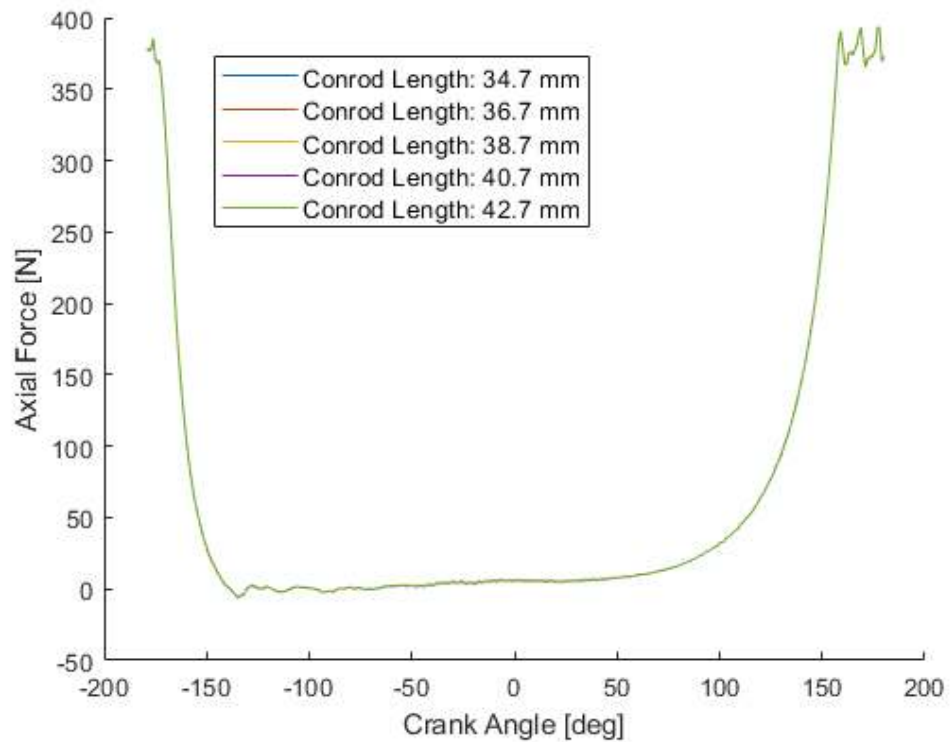


Figure 2.24. The effect of connecting rod length on net axial force.

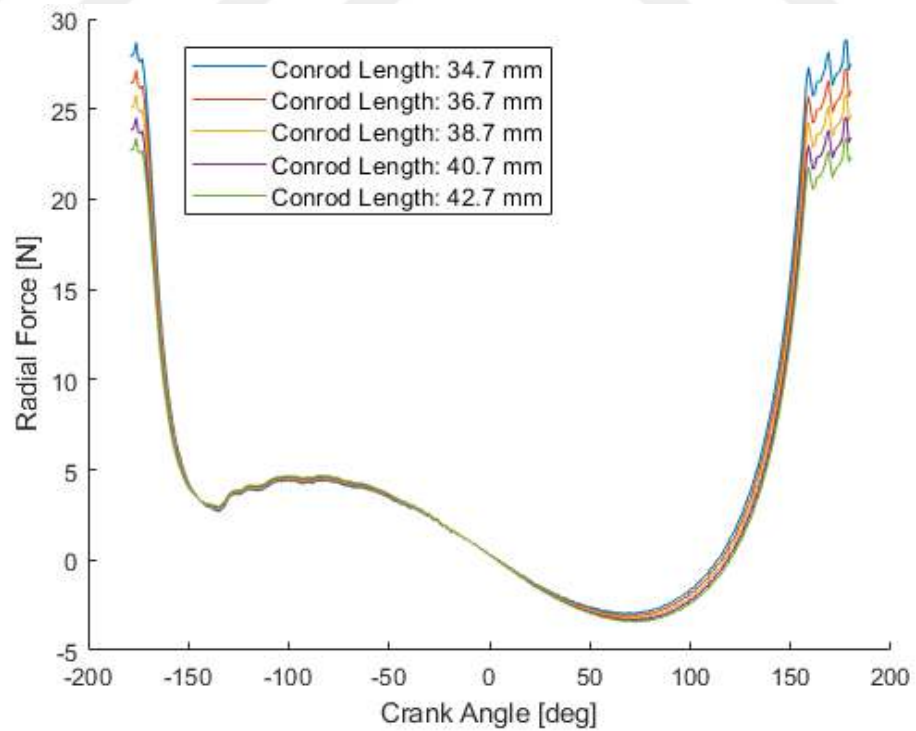


Figure 2.25. The effect of connecting rod length on net radial force.

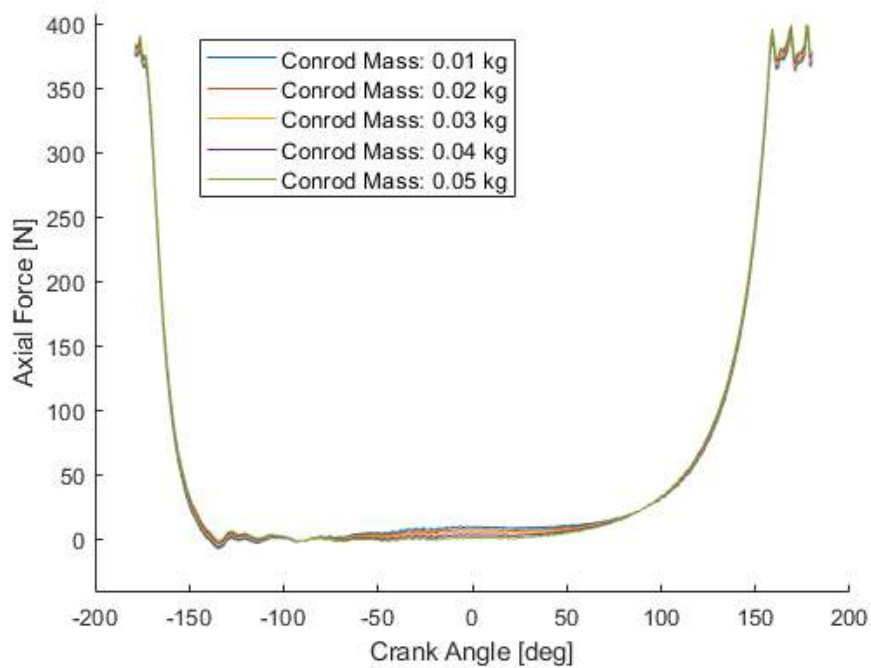


Figure 2.26 The effect of connecting rod mass on net axial force.

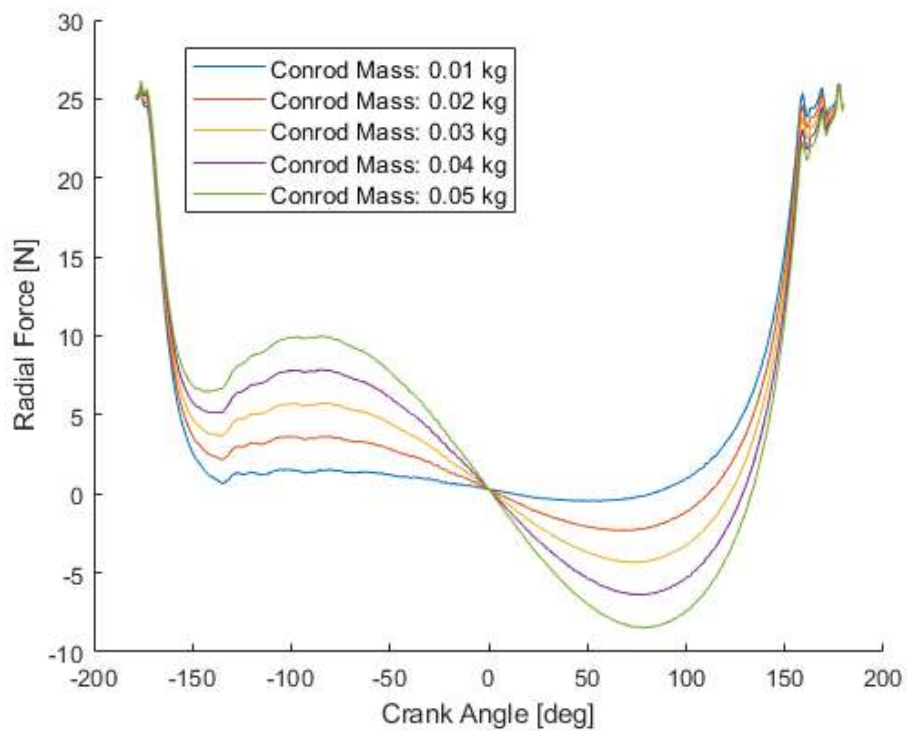


Figure 2.27. The effect of connecting rod mass on net radial force.

Moreover, the effect of design parameters on piston behaviour is also investigated. Crank's throw dimension is chosen to demonstrate the change in piston behaviour and results are shared in figure 2.28.

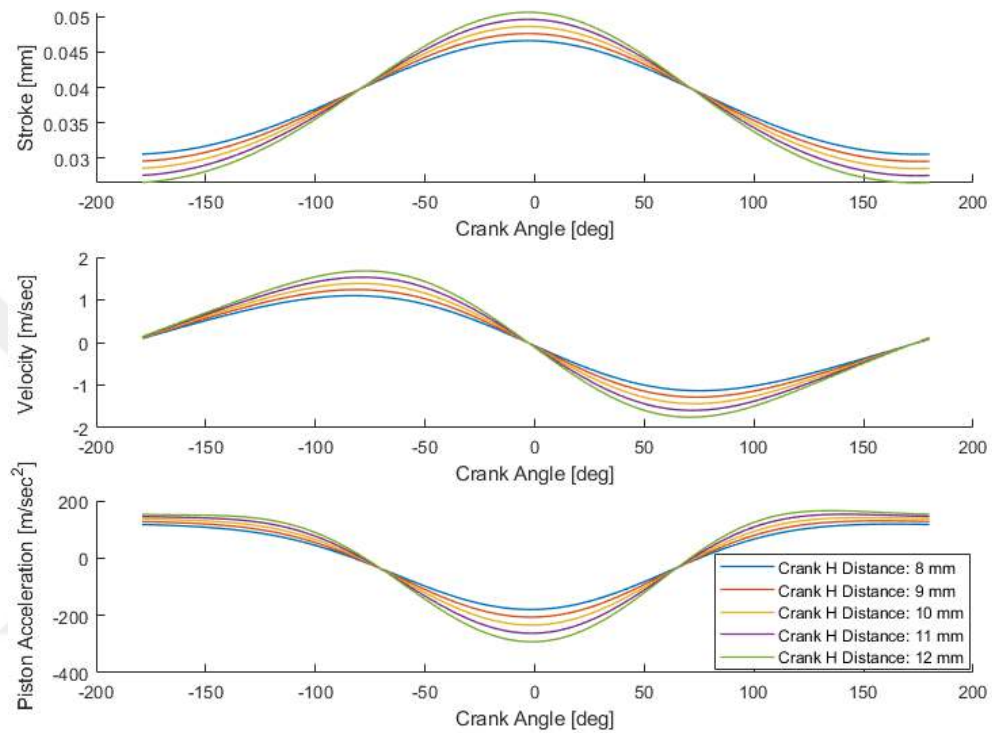


Figure 2.28. Piston behaviour in case of various crank throw distance.

After all, mathematical models that relies on physics can help to improve the dynamic resonse of the compressor by reducing the time consumption during the design process. Rather than conventional trial and error method, taking advantage of smart approaches or mathematical methods can enhance the compressor performance.

Not only rigid components, but also flexible parts such as springs or discharge tube must also be designed based on smarth algorithms. All around the world, especially the discharge tube is designed with trial-error approaches which costs to time. Section 2.3 introduces a smart and automatized method to design discharge tube by utilizing von Mises probability distribution.

2.3. Discharge tube design algorithm

The majority of section 2.3 has been published with the title “A novel approach to tube design via von Mises probability distribution” in the Engineering Optimization journal to be a part of this thesis in 2022 [Oral et al., 2022c].

2.3.1. Motivation

In reciprocating compressors, the discharge tube is the component that transports compressed gas from the reservoir to the cooling system. The tube also links the compressor body to the housing shell, and its secondary role is to absorb vibrations while preventing collisions with the housing shell [Seidel and Bernhard 1986]. It is a crucial structural component that must withstand the average decade-plus lifespan of a commercial refrigerator [Aprea et al. 2012]. The performance of a discharge tube must be evaluated from the perspectives of mechanical, fluidic, thermal, and vibratory qualities, making the optimization of the integrated design a difficult task.

All reciprocating compressors exhibit secondary motion as a result of unbalanced forces on the slider-crank mechanism [Link and Deschamps 2011; Tan and Ripin 2012] and carry/contain eccentric body masses that cause vibration [Prata et al. 2000], which can also be observed in linear compressors. To separate the vibration of the compressor from the rest of the refrigeration system, it is placed on springs from the bottom and connected to the outer shell via the discharge tube. In consideration of the desired service life, it is required to evaluate the vibratory displacements that might result in fatigue-based malfunctions or failure at stress concentration spots on the tube.

The purpose of mount-springs is to dampen the vibrations of the compressor body and minimize the transmission of movements to the exterior of the compressor case. As the sole other connection between the body and the housing, however, the discharge tube is likewise directly influenced by body vibrations. Since it functions similarly to a spring that joins the body and the case, the tube can contribute to the vibratory movements as either a dampening or amplifying source. During the

operating cycle, the largest displacements and strains on the tube occur during the start-up phase due to the motor's jerk and the overactive start trigger used to overcome dry friction conditions [Lenz, 2010].

As the discharge tube is a factor that determines body vibrations, several design solutions are applied in modern reciprocating compressors to optimize their vibratory qualities. A coil spring that partially wraps around the tube along its journey is a popular feature. According to the literature, the coil spring can play both an active and a passive function [Tasdelen and Kaya, 2016; Wang et al., 2004]. At steady-state excitation frequencies, the coil spring's mass contributes to a reduction in the natural frequencies; about 500 Hz, the spring's damping capacity prevails and minimizes the noise due to noise. Additionally, the choice of material is a vital factor. [Puff et al. 2018] proposed PFA polymer as an alternative to metallic tubes since the polymer tube exhibited promising fatigue, mechanical, and thermochemical characteristics. Analytical [Krishna et al., 1994] and computational [Lenz et al. 2010; Puff et al. 2006] research have already been conducted on the mechanical performance of the tube. The discharge tube must be both flexible enough to adapt to the body's vibrations and robust enough to bear both the internal gas and the exterior body forces. Additionally, the resonance frequency of the compressor is affected by the tube geometry's rigidity. To save production and material costs, the tube's journey should be as short as feasible; nevertheless, as the tube's length reduces, it gets stiffer, loses its elasticity, becomes susceptible to stress accumulation, and finally becomes susceptible to breaking.

For the optimization of the tube's design, fluid flow characteristics must be carefully examined alongside the opposing mechanical requirements. Due to its excellent thermodynamic qualities, R600a is utilized as a refrigerant in the discharge tube of most compressors [Palm, 2008]. The refrigerant reservoirs on the compressor body reduce the strong pulsation of the gas flowing from the piston to ensure smooth gas flow down the tube, but the tube's length, curvature radii, route, and wall thickness are also critical. The exterior diameter, placement of the spring coil, insertion length,

and tube length are the most influential characteristics on the modal response and static behavior, respectively. It is mentioned that the pressure drop between the tube's intake and outlet may grow as the tube's diameter and wall thickness decrease; the pressure drop also increases as the tube's length increases.

The compressor discharge tube path within the compressor housing and its dimensional specifications vary widely across commercial refrigerator manufacturers, indicating that there is no consensus regarding the ideal implementation technique. To the best of the authors' knowledge, the literature on tube path design is relatively unexplored, and no design algorithms such as the genetic algorithm [Renner and Ekart, 2003] or the SQP algorithm [Zhang et al., 2012] have been proposed previously, in contrast to other mechanical components such as cam shape mechanisms [Lampinen, 2003]. Similar to the discharge tube design process, the state of the art for extrusion die design is experience-based, time-consuming designs. However, [Elgeti et al., 2012] published a numerical strategy to build a smart way to design/optimize die shapes. Automated FEM studies exist in the literature to minimize design and optimization time [Coemert et al. 2021], which is another indication of the need to improve design methodologies and reduce design and optimization time.

In addition, no study has attempted to analyze the mechanical and fluidic aspects of the tube concurrently. We present a method that simultaneously analyzes tube stiffness and resonance frequency so that a designer may assess a variety of possible geometries and choose suitable parameters. In this work, the discharge tube geometry of a commercial refrigerator serves as the standard against which the algorithm-generated designs are evaluated. The mechanical performance of the tubes is confirmed using finite element calculations for static structural, fatigue, and fluid flow. The encouraging results show that the proposed tube design technique might be of substantial optimization aid to compressor manufacturers.

2.3.2. The Benchmark Investigated Compressor

Figure 2.29 depicts the CAD model of the benchmark compressor from a

commercial refrigerator manufacturer utilized in this investigation. The coordinate convention utilized for the remainder of this study aligns the piston's translational motion trajectory with the x-axis. From the tube's inlet starting point (p_{start}) to its outlet ending point (p_{end}), the u-form and -form bends are numbered (figure 2.29). These bends serve to attenuate vibrations and minimize stiffness and stress building; the remainder of the curve permits the length of the tube between the compressor body and the casing to be adjusted. Table 2.3 presents the key dimensional characteristics of the metallic discharge tube. The coil spring associated to the benchmark tube was eliminated, and the coil spring is not included in the development of new tube designs.

Table 2.3: Benchmark discharge tube parameters.

Parameter	Value
Inner Diameter (dinner)	2.2 mm
Outer Diameter (douter)	3.2 mm
Length	512 mm
Start Point (p_{start})	(0,0,0)
End Point (p_{end})	(139.2, 34.6, -39.5) mm
Curvature Radii	1 u-form
	2 u-form
	3 ω -form
	4 ω -form
	5 u-form

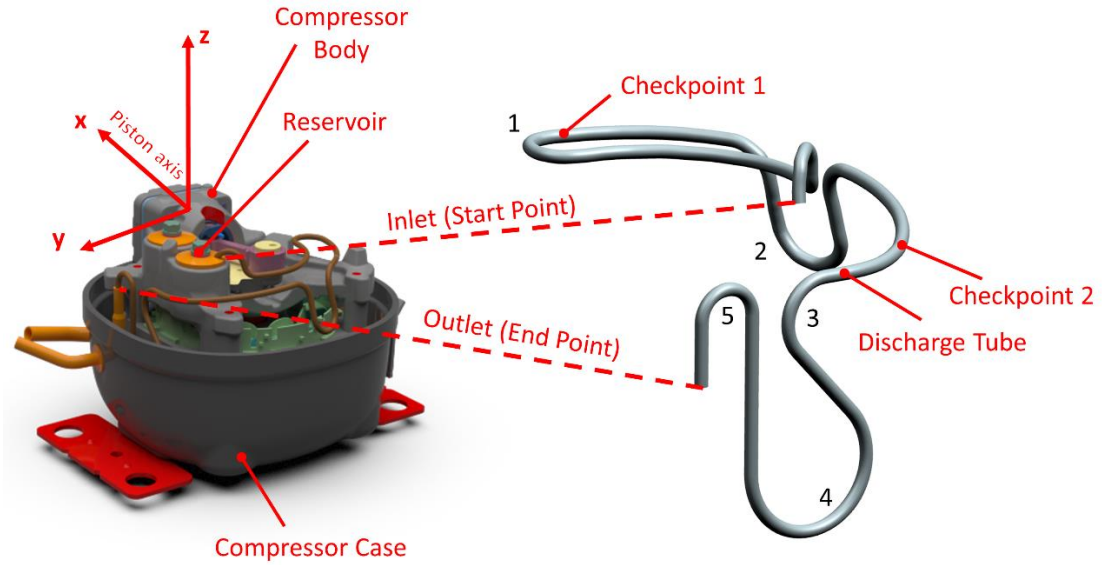


Figure 2.29. The benchmark reciprocating compressor CAD and the specific geometry of the discharge tube.

2.3.3. Discharge Tube Design Algorithm

The proposed discharge tube design strategy in this study assumes that all the parts of the compressor assembly are finalized, the p_{start} and p_{end} locations are known and the compressor body sits stationary on the bottom-mount springs. Utilizing p_{start} , p_{end} , and designated checkpoints (C_n) in between, a candidate tube path (central guideline), comprised of discrete points $p_i = \{p_{start}, p_1, \dots, p_{end}\}$, is generated by employing von Mises probability distribution [Diard et al. 2013], detailed in Section 2.3. Using the calculated trajectory points, the tube geometry is then finalized by placing inner and outer diameters (d_{inner} , d_{outer}) at each guideline point with appropriate orientation and creating the inner and outer tube surfaces by connecting the cross-sections. Then, the candidate design is passed through several chosen design criteria. The first criterion is that all the tube points should be residing in the allowable clearance between the compressor body and the housing. The next criterion is checking for self-intersection and whether the generated tube can be meshed for further finite element (FE) analysis. If the design passes meshability, the geometry is

then checked for static structural and modal conditions that ensure the stress accumulation and the natural frequency values are kept below designated thresholds. If at any point the candidate design fails a criterion, it is eliminated, and another candidate tube geometry is created. Figure 2.30 summarizes the algorithm steps and lays out the decision tree in accepting the designs.



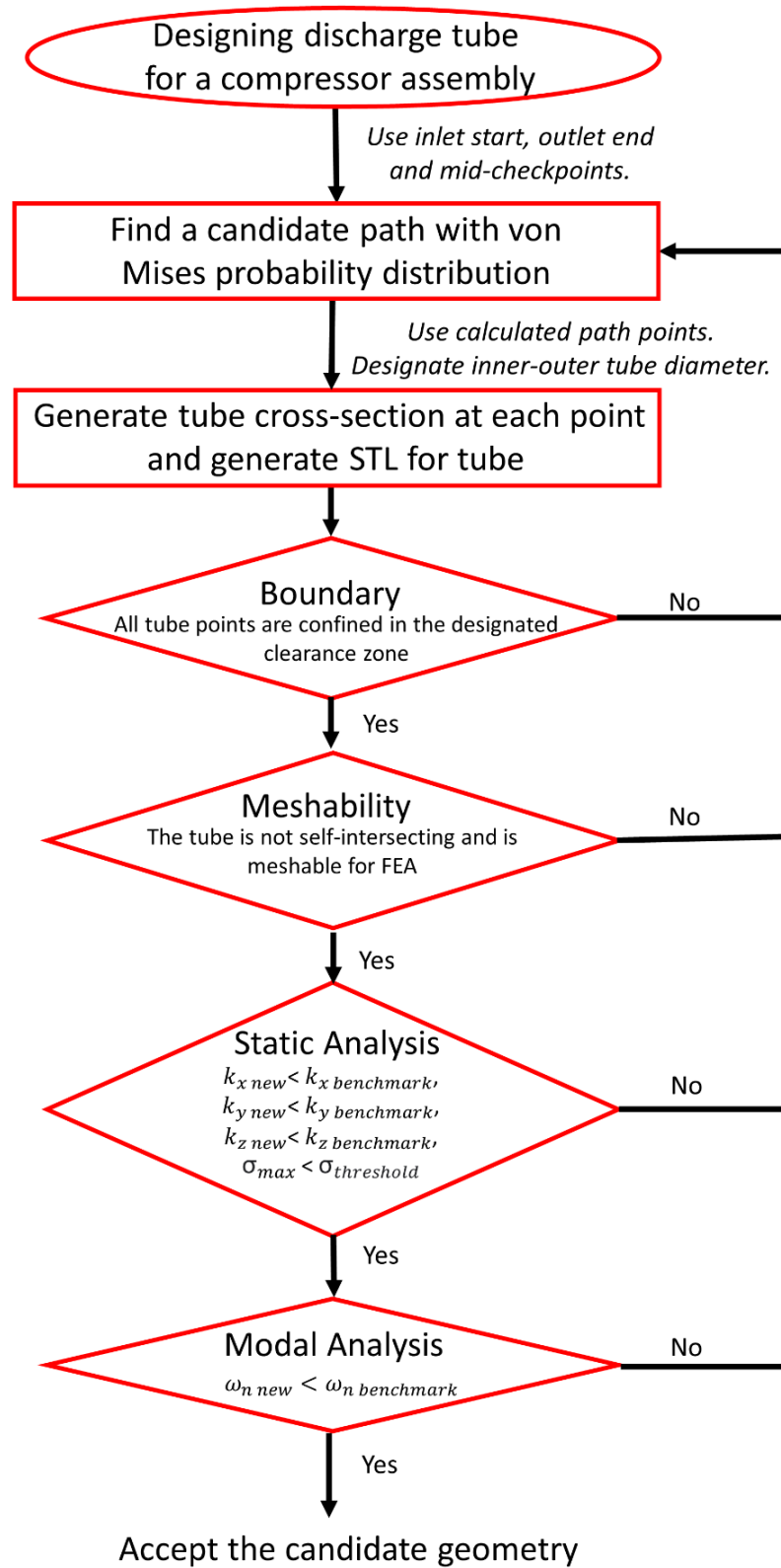


Figure 2.30. The algorithm followed for designing the discharge tube.

2.3.4. Path Generation using von Mises Probability Distribution

To generate a candidate path, the pstart and pend spatial locations should first be realized. For this study, the CAD model of the benchmark compressor is used to locate the 3D positions. In addition, certain checkpoints in between pstart and pend need to be predetermined. This is to ensure that the generated tube geometry will have sufficient length to allow flexibility; if checkpoints to partially direct the trajectory of the tube are not chosen, the algorithm will tend to create a geometry that follows the shortest path and cause it to be very rigid. Only 2 checkpoints have been used for this study as located in figure 2.29.

The artificial increase of the tube length is not only achieved via imposing checkpoints; von Mises probability that pseudo-randomly diverges the trajectory is also introduced to deviate from straight, shorter, and linear paths between checkpoints. The distribution is expressed by:

$$P = e^{\lambda \cos(\alpha - \mu)} \quad (2.29)$$

where λ is the concentration factor, μ is the preferred angular direction over a circular space and α is the swept angle range. λ is a scalar factor that dictates the tendency towards reaching target μ such that a lower λ makes it less likely to reach target μ and vice versa. Through trial-and-error, an appropriate λ value is chosen for this study, such that a lower value would cause the path to curve excessively and generate self-intersecting tube geometries, and a higher value would generate highly linear paths. Path generation is achieved in computer code by following the steps below:

- For each path point p_i with coordinates (a_i, b_i, c_i) , a step size, 1 mm, is selected akin to choosing a radius for a sphere.
- Using spherical coordinates, the points on a sphere with center (a_i, b_i, c_i) at 1 mm distance are calculated by sweeping the polar angle ϕ and the azimuthal angle θ inside chosen angular ranges with 1 degrees increments.

- The 3D distances of the sphere points to the next path target (either the upcoming C_n or p_{end}) are calculated and the nearest point ($p_{near,i}$) with angles ($\varphi_{near,i}$, $\theta_{near,i}$) is determined.
- The von Mises probability distribution equation is employed to determine the weights in pseudo-random choosing of p_{i+1} . Since two circular spaces will define p_{i+1} , two von Mises distributions are used:

$$P_{i\varphi} = e^{\lambda \cos(\varphi - \varphi_{near,i})} \quad (2.30)$$

$$P_{i\theta} = e^{\lambda \cos(\theta - \theta_{near,i})} \quad (2.31)$$

- To select the p_{i+1} from the set of candidate sphere points created in step 2, a random sample selection is conducted using $P_{i\varphi}$ and $P_{i\theta}$ as weights.
- The selected p_{i+1} is added to the series that define the complete tube guideline points from p_{start} to p_{end} .

A path is produced to connect each sequence of guideline points. Using a moving average of path vector components, the path is smoothed to prevent probable self-intersection when the outer and inner diameters are combined and to provide a tube shape that is meaningfully manufacturable and free of sharp corners. The algorithm for path generation is depicted in figure 2.31.

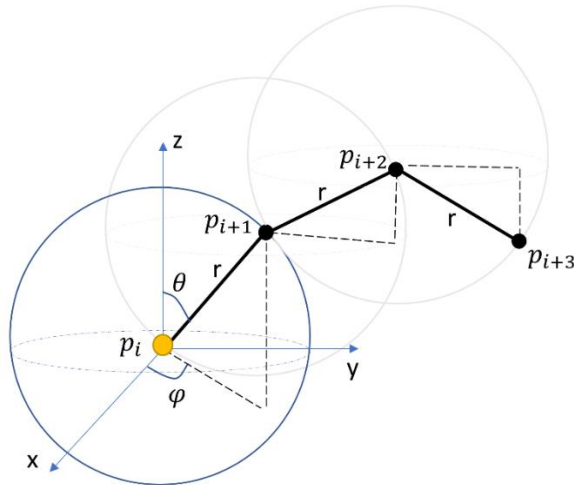


Figure 2.31. Sequential addition of path segments.

Solid Tube Generation

At each p_i generated for a candidate geometry guideline, the tube cross-section must be placed. d_{inner} and d_{outer} are chosen as identical to the benchmark tube. Each circle is defined by a certain amount of equally spaced points centered at (a_i, b_i, c_i) , d_{inner} and d_{outer} away. The spatial orientation of the tube cross-section plane is adjusted to be normal to the path vector connecting p_i and p_{i-1} . Then, the outer and inner tube surfaces are formed by sequentially connecting the calculated cross-section points in Matlab. The geometry is finalized by creating an STL file of the candidate tube.

Boundary Check

The proposed geometry must fit inside the clearance gap between the compressor body and housing. For the realization of this 3D environment, the CAD of the standard compressor is used. First, the compressor housing is solidified in the CAD program, and then the body of the compressor is removed from the solidified compressor housing to isolate the negative area the tube must occupy. The clearance CAD is then imported as a point cloud into Matlab. The `inpolygon` command is used to determine if all tube points are contained inside this negative space, and any geometries that spatially violate the 3D boundary condition are eliminated (figure 2.32a).

Meshability Check

To conduct FEA in order to analyze the static and modal behavior of the proposed design, the geometry must be meshable. At this stage, the self-intersecting geometries that cannot be meshed are deleted (figure 2.32b).

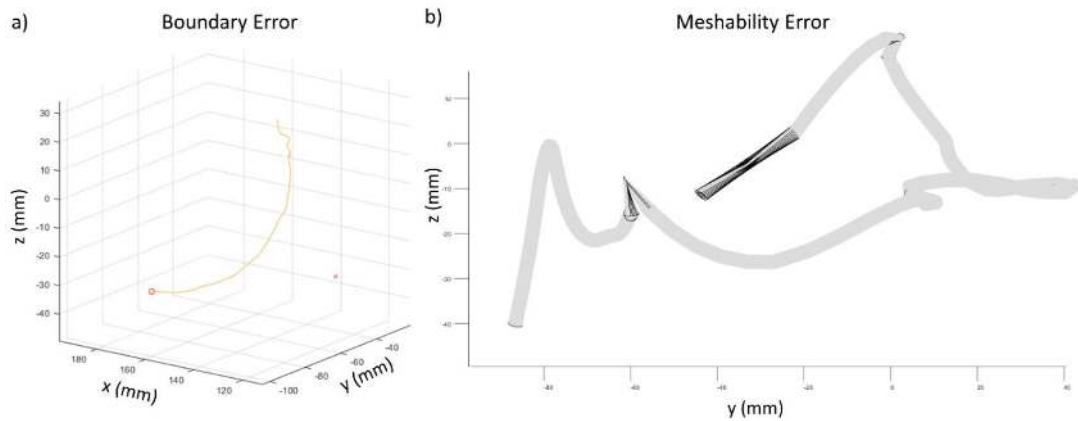


Figure 2.32. (a) Boundary error discards the design if the generated path is outside the designated clearance zone. (b) Meshability error discards the design if the tube has self-intersections and cannot be properly meshed for further FEA.

Static Analysis

Matlab's partial differential equation toolbox is used to conduct the static structural FEA in order to obtain the tube's directional stiffness values. The utilized material properties for the AISI 1015 carbon steel tube are shown in Table 2.4. The maximum mesh size allowed is 1 mm and a quadratic mesh type is used. In the study, since the tube outlet is normally welded or brazed in produced compressors, the tube is fixed at the outlet (all displacements and rotations are set to 0) and the inlet, which is the connecting point to the reservoir, is left open for displacement observation. Separately along the x, y, and z directions, a 2 Newton force is applied to the free end, and the directional stiffness values are determined based on the calculated displacements. The design requirement in this step is to aim for a tube geometry that has lower directional stiffness (k_x, k_y, k_z) in each direction in comparison to the benchmark tube, else the geometry is eliminated.

In addition, a second stress study is performed on the candidate tube. In a manner analogous to the stiffness configuration, the outlet end is fixed and the inlet has been shifted 5 mm along each primary axis. Despite the fact that the yield strength of AISI

1015 is 315 MPa, a lower von Mises stress threshold of 250 MPa is used. A lower threshold ensures a margin of safety and allows for the worst-case situations, such as the intense operating vibrations that can be induced by malfunctioning parts or the excessive displacements that the tube may encounter during shipping of the refrigerator. If the cumulative stress on the tube during any simulation exceeds the threshold stress, the design is eliminated.

Table 2.4: Mechanical material properties of the AISI 1015 tube utilized in FEA.

Parameter	Value
Young's Modulus	200 GPa
Poisson Ratio	0.29
Density	7870 kg/m ³
Ultimate Tensile Strength	420 MPa
Yield Strength	315 MPa

Computational Fluid Dynamics (CFD) Validation

The fluid flow performance of the selected candidate tube is then compared to the benchmark for validation. The program Ansys Fluent 19.0 is used to create a numerical model of the discharge tube in order to monitor pressure drop between the intake and output for each tube geometry. A time-dependent transient solver is utilized to simulate unsteady flow. R600a is modeled as an incompressible fluid with a density of 2.51 kg/m³ and a dynamic viscosity of 9.4310^{-6} kg/ms [Yesilaydin and Erbay, 2018]. Using tetrahedral components, a computational mesh is generated and then refined towards the inner tube wall. The FE statistics are presented in table 2.5. During grid creation, the Ansys Fluent User's Guide is followed to obtain acceptable orthogonal quality, aspect ratio, and skewness.

Table 2.5: FE statistics and boundary conditions for the CFD simulation.

Mesh Parameters	Values
Face Sizing	
Element Size	0.25 mm
Behaviour	Soft
Growth Rate	Default (1.2 mm)
Quality	
Smoothing	Medium
Inflation	
Inflation Option	First Layer Thickness
First Layer Height	0.02 mm
Maximum Layers	15
Growth Rate	1.45
Boundary Conditions	
Tube Walls	Wall
Inlet	Mass Flow Inlet
Outlet	Pressure Outlet

The second-order discretization method is used for the pressure and the second-order upwind discretization method for momentum conversation calculations. Turbulence is simulated with an SST k- ω model, and a mean dimensionless wall distance $y^+ < 1$ is assured near the domain walls. The SIMPLE approach is used in Fluent software to solve the governing equations for gas flow during the continuum kochphase. The experimentally measured pressure profile from 1300 rpm is set at the outlet as a transient boundary condition and a mass flow boundary condition is used at the inlet. The convergence precision for the scaled residuals is set to 10^{-4} . The time step is adjusted to 5×10^{-4} s, and the simulation time for each case is 7 seconds in real-time. A grid independence study is also conducted to eliminate mesh quality-based errors. Each simulation lasted approximately 10 hours using a 64-bit computer with 8-core Intel Xeon 2.10 GHz CPUs and 16.0 GB RAM and over 1.100.000 cells are employed. Pressure differences are checked after 10 cycles of the crankshaft revolution for each tube geometry for the same time frame [Birari, 2006].

Fatigue Validation

The fatigue life is another crucial criterion, as the tube must withstand the continual body vibrations and internal gas forces during the refrigerator's lifetime; otherwise, fatigue-based failures might limit the discharge tube's lifespan. To determine the tube's vibrational displacements, the reference compressor is rotated at 1300 rpm while the body displacements are detected with a laser sensor. Using the fatigue tool in FE software, the displacement-cycle curve for the tube is generated in accordance with the AISI 1015 life estimation. If the tube's lifespan is calculated using FEA to be 10^6 cycles (assuming infinite life) for the experimentally determined maximum displacement in each primary axis, it may be argued that the tube can safely withstand the compressor operation.

2.3.5. Results

Accepted Tube Designs

Using the design procedure and decision tree depicted in Figure 2.30, two candidate geometries are accepted. The acceptable geometries are illustrated in Figure 2.33 and will be referred to as D1 and D2 from this point forward. Only one directional stiffness value along x axis of D1 is left free for in the decision tree to highlight the algorithm's flexibility.

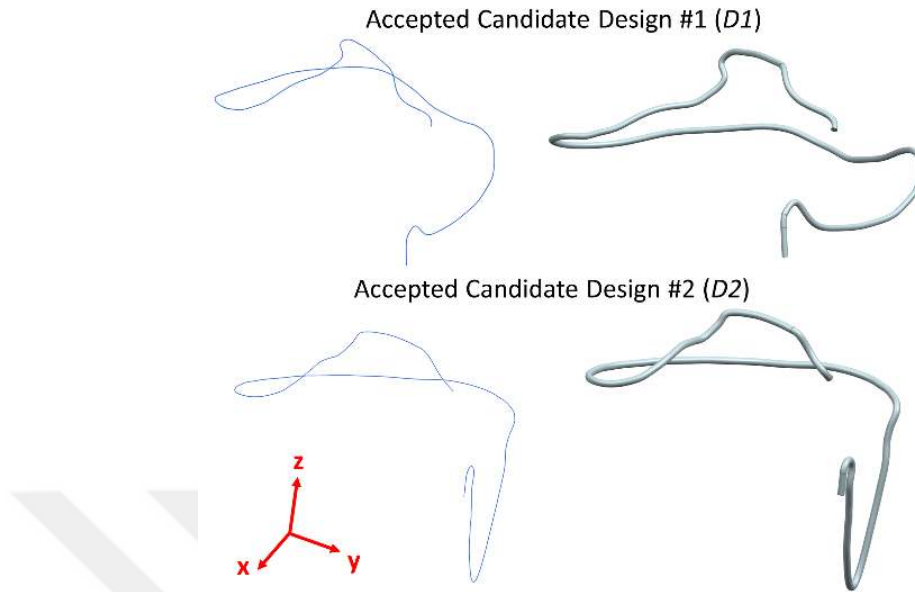


Figure 2.33. Accepted designs, D1 and D2, using the proposed algorithm.

Static Structural Results

Matlab automatically calculates the directional stiffnesses inside the procedure to accept candidate geometries. For validation, the geometries are further exported to Ansys, and the algorithm-obtained values are cross-checked with the same boundary conditions. Figure 2.34 compares Matlab FE findings with Ansys results for D1. Table 2.6 presents the computed directional stiffnesses (for 2N directional force) as well as the mesh sensitivity analysis performed in Ansys (tetrahedral mesh, maximum size 1 mm, minimum size 0.1 mm) to assure convergence. Even though the tube geometries lack symmetry, all structural evaluations are performed for the opposite directions (e.g., +x and -x directions) and the findings are only marginally different (0.01 mm).

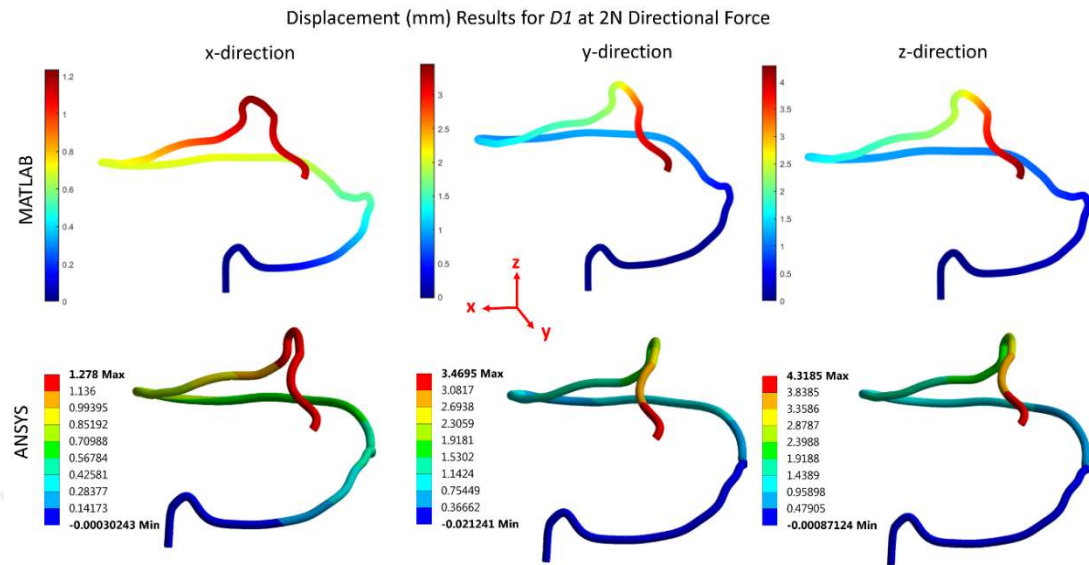


Figure 2.34. Comparison of static structural results for D1 in Matlab vs Ansys. 2N force is exerted at each principal direction for the calculation of directional stiffnesses.

Table 2.6: Directional stiffnesses of the tube geometries and mesh sensitivity analysis for 2N directional loading on tube inlet.

	k_{benchmark} (N/mm)	k_{D1} (N/mm)	k_{D2} (N/mm)		
k _x	0.545	1.662	0.414		
k _y	0.952	0.576	0.727		
k _z	0.593	0.463	0.468		
	Element #	Node #	Δx (mm)	Δy (mm)	Δz (mm)
D ₁	47840	23868	1.196	3.455	4.301
	110001	27868	1.203	3.469	4.318
	234460	42600	1.203	3.469	4.318
	491919	95207	1.203	3.470	4.318
	1373678	274641	1.203	3.470	4.319
	8225952	1787940	1.204	3.470	4.319
D ₂	53496	107236	4.820	2.744	4.267
	76046	150062	4.822	2.747	4.267
	141034	259908	4.825	2.758	4.270
	314907	530948	4.830	2.751	4.271
	658063	1078030	4.830	2.748	4.271
	2241476	3435659	4.830	2.749	4.271

Table 6 demonstrates that directional stiffnesses less than the benchmark can be obtained. In addition, mesh sensitivity analysis reveals that the findings converge to a stable value for the selected mesh parameters, supporting the algorithm's estimated results. In addition, the reported values from FE studies are confirmed experimentally for D2. The outlet end of the discharge tube is fixed, while the intake is left free to be activated by a linear actuator. The fixed end is attached to a table-type dynamometer, and the linear actuator's displacement is monitored using a laser sensor.

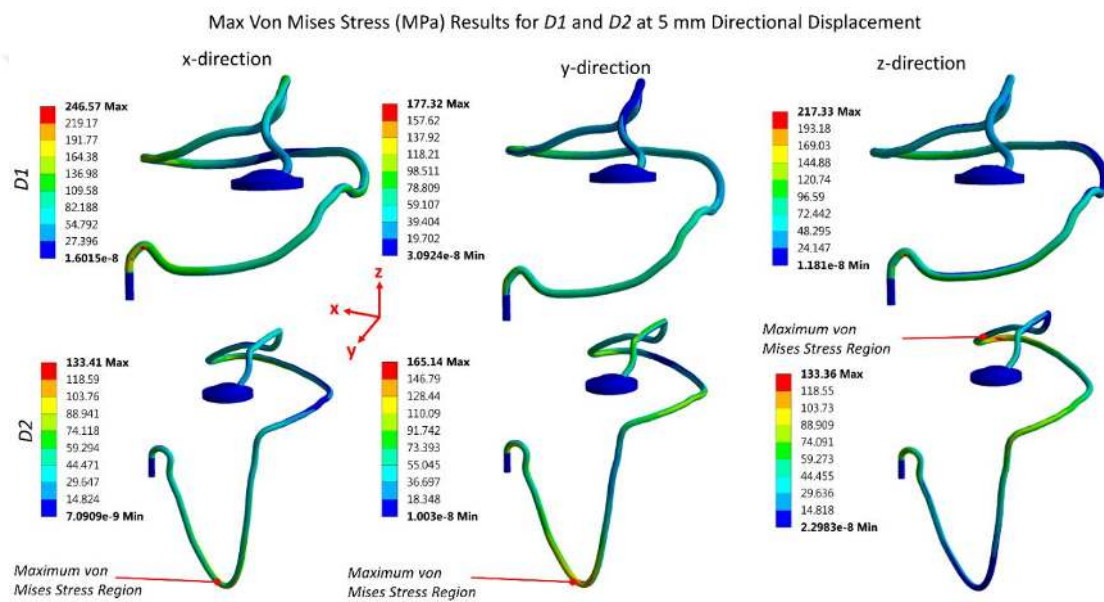


Figure 2.35. Stress results for D1 and D2 for 5mm directional displacement.

The stress analyses, which are demonstrated in figure 2.35, are confirmed by comparing the locations of maximum von Mises stress points in the FEA to experimental tube fatigue failure sites, based on the assumption that the tube will collapse at the region with the largest accumulated stress. To induce rapid rupture, the discharge tube is agitated with 4 Hz frequency and 30 mm linear displacements in the x, y, and z axes. A system identical to the stiffness measuring arrangement seen in figure 2.4 is constructed, with the exception that the outlet is fastened to the table

rather than the dynamometer. The broken tubes may be observed in figure 2.36. When stimulated in the Y and Z directions, there is an excellent correlation between the breakage experiments and the FE predictions of the sites of the maximum stress on the tubes. The only tiny discrepancy is found for the X-axis actuation of the tube, which breaks at the second most stress-accumulated position based on Ansys and Matlab findings; this may be explained by the proximity of the breaking point to the experimental excitation point.



Figure 2.36. Broken tubes.

Modal Analysis Results

Table 2.7 compares the experimentally acquired natural frequency modes (ω_n) of the compressor with the analytically calculated modes. It can be concluded that the eigenvalue approach to determining the natural frequency modes is an accurate method of prediction due to the close similarity of the results. According to the algorithm, the 6th natural frequency modes of both accepted designs fall below the 6th frequency mode of the benchmark that is closest to the possible lower limit of steady-state operation range.

Table 2.7: Compressor and tube natural frequencies for D1, D2, and the benchmark.

Mode #	$\omega_{\text{benchmark}}^*$ (Hz)	$\omega_{\text{benchmark}}$ (Hz)	ω_{D1} (Hz)	ω_{D2} (Hz)
1	5.75	7.28	6.74	6.53
2	7.75	7.70	7.51	7.96
3	9	8.23	8.11	8.08
4	13	11.69	11.60	11.44
5	13.75	12.43	12.37	12.35
6	14.75	14.11	13.95	13.93

* Experimentally obtained value. The rest of the values are analytically calculated.

CFD Results

The findings of D2 grid independence study are reported in Table 2.8. Similar to [Qyyum et al., 2019], the mesh size is reduced sequentially, and it is demonstrated that the pressure drop result is independent of the grid size. Following simulations employ over 1,1M cells. Since the discharge tube's principal role is to deliver pressured gas for refrigeration, the inlet-outlet pressure drop of the proposed geometries should be comparable to the benchmark in order to match performance. Table 2.8 demonstrates that there is a small variation in pressure drop values between D1 and D2 and the benchmark, validating the fluid flow characteristics.

Table 2.8: CFD grid independence analysis and pressure drop results.

Grid Independence			
Design	Element #	Node #	Pressure Drop (bar)
D2	648937	304442	0.161
	812420	375087	0.162
	1110411	499516	0.162
	2254220	977492	0.162
	Tube Length (mm)	d _{inner} -d _{outer} (mm)	Pressure Drop (bar)
Benchmark	512	2.2-3.2	0.156
D1	406	2.2-3.2	0.147
D2	475	2.2-3.2	0.162

Fatigue Results

Figure 2.37 depicts the results of the laser displacement experiment done on the standard compressor. Regardless of the intended rpm, the greatest displacement in each primary axis is seen during the compressor's start-up phase. For instance, while ramping up to 1300 rpm, the compressor suffers strong vibrations as it passes through the lower resonance frequencies identified by modal research. Similarly, when the stop signal is given, the compressor slows down and passes through the resonance region once again causing a peak in displacement. The highest empirical displacements for each primary axis (x, y, z) are 1.222 mm, 0.638.0 mm, and 0.611.

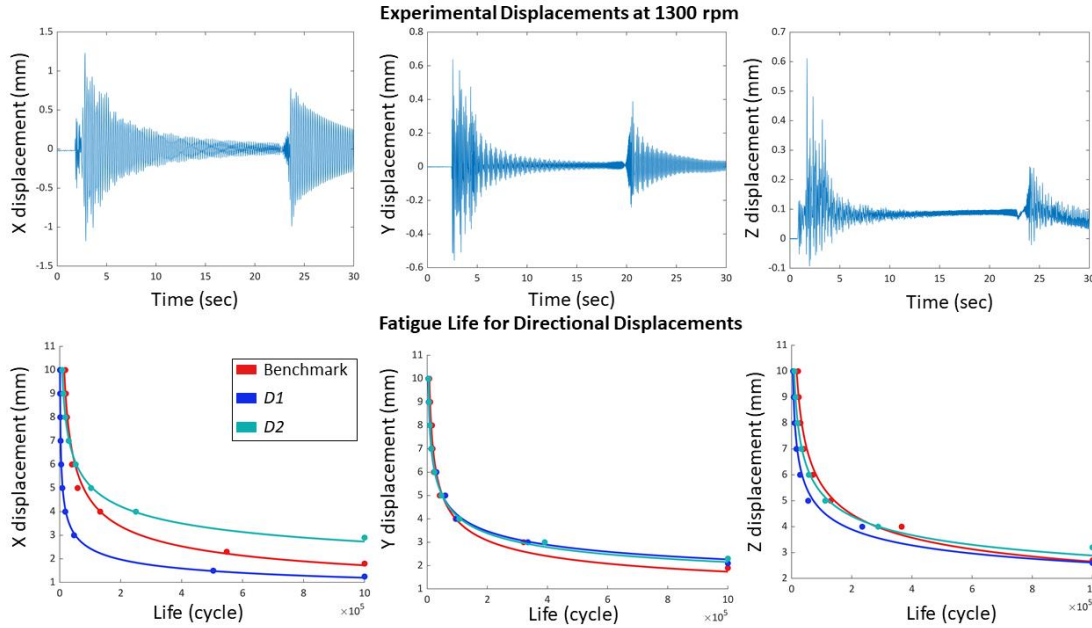


Figure 2.37. Experimental displacements of the benchmark compressor in each principal axis and fatigue life analysis of benchmark, D1, and D2.

The results of the fatigue FE analysis are also depicted in figure 2.37. The analysis is continued for each design until the largest directional displacement that causes infinite life is identified. Overall, the algorithm-generated tubes have similar fatigue properties compared to the benchmark, and the empirically found maximum displacements are expected to not cause any catastrophic fatigue-related problems.

2.3.6. Optimizations

After generating the tubes by successfully following the flowchart to consider design criteria, there is still potential to optimize the stiffness of the tube and to reduce the maximum von Mises stress against known amount of force.

In order to identify to regions to be optimized, first the maximum von Mises stress point is found by finding the minimum radius of curvature throughout the tube. Afterward, the circumcenter command is utilized in order to smoothen the path as illustrated in figure 2.38.

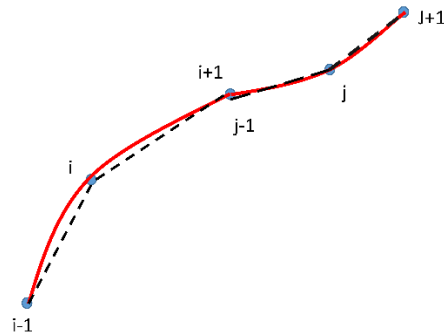


Figure 2.38. Smoothing the path.

Then, the tube is re-formed with optimized path points. The smoothed path and the initial tube form can be seen in figure 2.39.

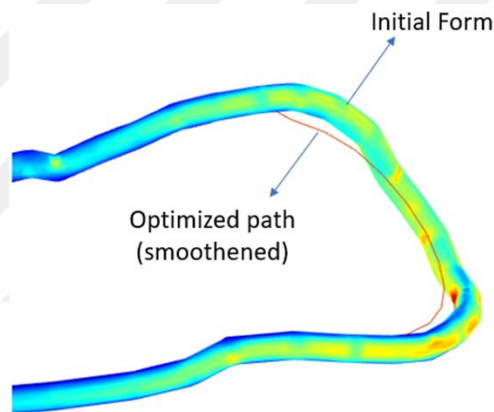


Figure 2.39. Optimized path.

After adding the optimization steps, the decision tree is also altered as follows in figure 2.40 and new steps are emphasized.

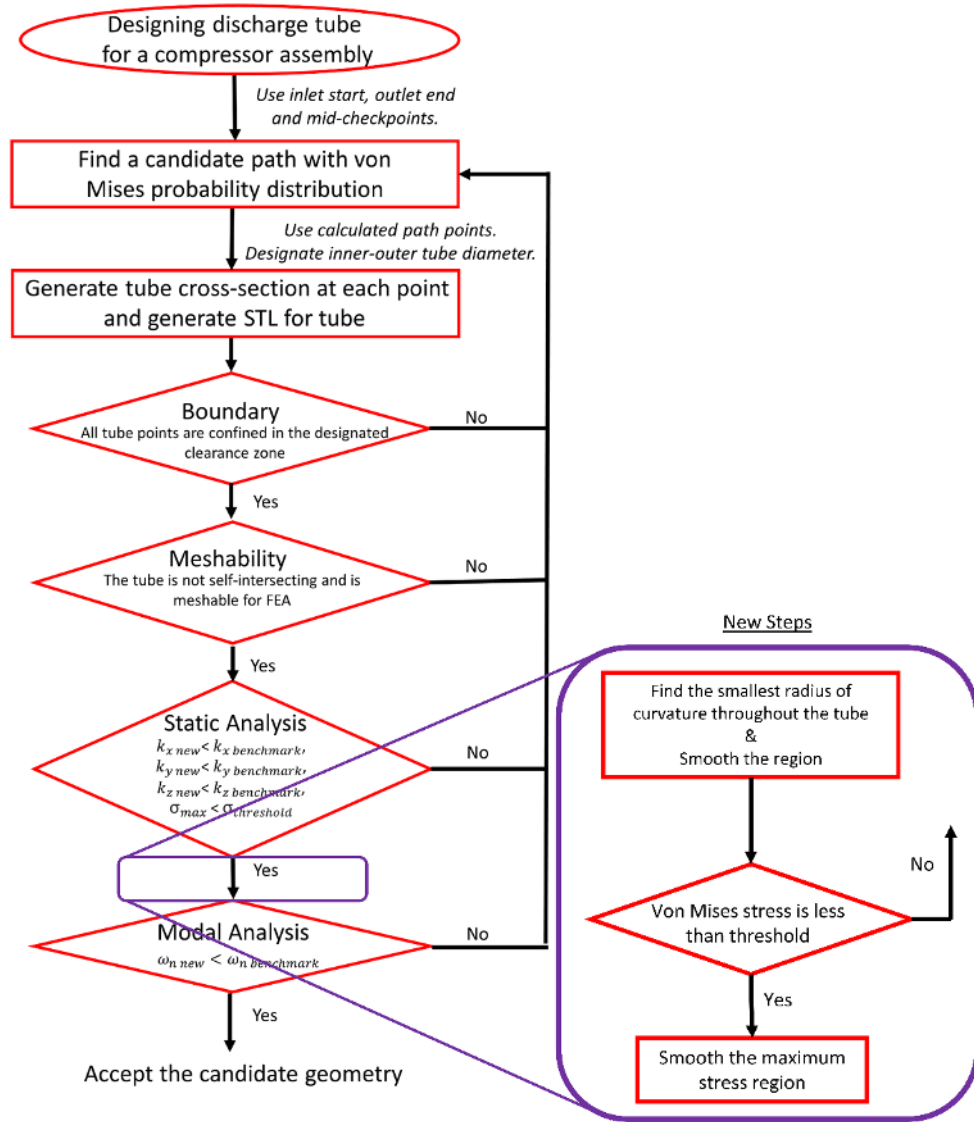


Figure 2.40. Optimized decision tree.

2.3.7. Discussion

In this paper, a novel von Mises probability-based technique for designing the discharge tube of a reciprocating compressor is proposed. The development of an algorithm that integrates important design, static, and modal factors to produce alternative tube geometries with comparable performance to an industry benchmark. Diverse makers of commercial reciprocating compressors all rely on specialized

knowledge acquired through years of technical expertise, which includes some trial-and-error. This impression is further supported by the fact that no two discharge tube geometries from various manufacturers will be similar and the overall course of the discharge tube is regarded to be somewhat arbitrary in the clearance area between the compressor body and the housing. The length is artificially expanded by means of meandering routes and u- and -shaped bends in order to achieve a flexible geometry that permits the compressor body to vibrate. However, it is impossible to speak about a full optimization procedure that takes into consideration the numerous engineering elements.

The suggested approach does not apply optimization directly, but it may yield designs that fulfill several design requirements. A candidate tube design that passes the required elimination filters is computationally assured to perform as good as, if not better than, a set benchmark. Additionally, the method is adaptable enough to accommodate differing or more stringent engineering constraints. If desired, the length of the tube can be intentionally expanded by adding additional route checkpoints or a lower modal frequency, or by aiming for static thresholds. In addition, it should be noted that the provided approach would be highly beneficial in the event of a totally new compressor design from scratch, as the design of each component will be distinct.

Among the shortcomings of the suggested technique are the absence of pressure decreases and fatigue checks in the algorithm. Despite the fact that the fatigue and pressure drop qualities of the candidate designs are checked thereafter and the findings are close to the standard, a design that satisfies certain design requirements while violating others cannot be approved. First infinity is reached at 1.25 mm, but the maximum displacement during start-up for the standard is 1.223 mm. Due to the near proximity of these values, it is feasible that fatigue-related failures might reduce D1's life expectancy over an extended period of operation. Therefore, the algorithm may be further developed to integrate the remaining technical needs and do the required threshold tests.

A remaining overlooked parameter is manufacturability, which is a complex variable to assess. While smoothing is used to build geometries without sharp corners and twists, the unpredictability in trajectory can cause a large number of minor bends to occur randomly along the path, making precise manufacture more difficult than on the entirely smoothed path of the benchmark. In addition, the acoustic property, which is considered outside the scope of this study, can be investigated in the future. High-frequency (500 Hz) noise can be caused by potential gas resonance and transmitted vibrations [Yoshimura, 1994]. Since compressed gas volume is pumped into an environment with lower pressure, gas resonance may occur during the initial handful of cycles of reciprocation.

Conclusion

The proposed novel method of designing a discharge tube using an FE-based filtering logic flow can be of great assistance to compressor part manufacturing and optimization fields. The algorithm can be readily adapted for any compressor assembly since it utilizes the specific 3D geometry such that the start, end, and mid-path checkpoints can be flexibly modified in the creation of the tube. By further tinkering with the input parameters, the performance of the discharge tube can be improved based on the designated requirements. The time to develop an operational tube geometry can be drastically shortened from weeks to minutes.

Chapter 3

TRANSIENT OPERATIONS

3.1. Background and Literature Review

In applications where the consumer interacts directly with the product, noise and vibration are undesirable because they provide an uncomfortable user experience. Therefore, suppression of these undesirable phenomena is crucial, particularly for domestic applications such as refrigerator compressors, washing machines, dryers, etc.

The reciprocating compressor is an important part of the typical domestic refrigeration system. Characterization and improvement of the compressor's operating dynamics resulting from the motion of the piston play a crucial role in improving the system's overall efficiency and performance.

Counterweights [Huang and Lee, 2012] and four mount-springs (coil type) mounted at the bottom and the refrigerant discharge tube for conveying compressed gas, both mechanically connecting the body to the outside compressor housing, are commonly used to dampen and isolate compressor vibrations during operation. The major purpose of the mount-springs is to reduce the amplitude of the compressor body's vibrations and restrict movement transmission inside the housing. Moreover, as noted by [Futakawa et al., 1980], undesirable stress builds up on suspension parts during transient operations, which can lead to the breakdown of components if the designs are flawed.

Alternately, the use of flywheels [Li et al., 2012] or even ball impact dampers [Djemal et al., 2019] might be an alternative for lowering the steady state vibrations of the majority of vibratory machines. In reality, standard vibration techniques may boost the productivity of industrial processes by allowing machines to work at faster rates [Gafsi et al., 2017]. In addition to electromagnetic suppression [Wang et al.,

2020], magnetic levitation [Taguchi et al., 2011], damping with MR (magnetorheological) fluid [Lai and Liao, 2002], and active control with a PID unit [Lai and Liao, 2002], other vibration reduction methods have been proposed in the literature for various engineering systems [Khot et al., 2011]. However, as the manufacturing cost to performance ratio is a crucial factor for commercial devices, such sophisticated and high-end solutions are neither economically practical nor appropriate for mass production.

In the spirit of maintaining a reasonable price-to-performance ratio, [Inagaki et al., 2018] proposed a spherical self-standing support design for the compressor that utilizes the gravity-caused restoration of momentum principle to reduce support stiffness for potential operation at lower frequencies. [Dianov, 2020] demonstrates experimentally that dynamic braking at a predetermined mechanical position may significantly decrease the peak stoppage vibrations.

However, such optimization efforts may be useless during transient startup and shutdown phases, where noise levels are at their height and may even impair the transient response. Therefore, transient operations must be investigated independently from steady-state operations, and solutions must be offered that do not violate steady-state constraints.

During the operating cycle between the 'on' and 'off' states, the compressor must traverse the jerky start-up phase during which the piston must overcome dry friction conditions prior to entering the steady-state phase. Before achieving steady-state at comparatively higher frequency levels, the compressor must traverse the low-frequency area that corresponds with the system's resonance frequencies, resulting in inevitable harmful vibrations during startup. These low frequencies must be passed again when transitioning from steady-state to shut down as the compressor slows down; thus, despite operating at lower revolutions, high displacements are observed in the transient start-up and shut down regions, whereas these vibrations diminish at steady-state away from system resonance.

There are two considerable benefits to reducing transient vibrations: achieving

steady-state more quickly, hence reducing the system's energy consumption, and reducing the transient over-excitation to reduce the vibration-induced noise emission. The compressor is one of the most dominating noise producers in the refrigeration system, and the transfer of vibration energy contributes significantly to the total sound power level output.

To provide a general solution, it is a certainty that the parameters that influence transitory behavior must be well understood. Otherwise, an external solution that works for one compressor might not be applicable to the others. In order to comprehend the physics underlying the vibration of a reciprocating compressor, the dynamic analyses of [Pont et al., 2017], [Kurka et al., 2012], [Kim and Han, 2014], and [Gerhold and Hamilton, 1974] illuminate the influence of system parameters on structural forces during steady-state region. Although these techniques are primarily concerned with steady-state activities, the ideas described may also be utilized to comprehend transients.

There are numerous studies of transitory regions in the literature. In the literature, finite element (FE) studies [Levecque et al., 2011] and analytical approaches [Dufour et al., 1995] are documented for dynamic study of the startup and shutdown zones where natural frequencies are taken into account. In addition, [Arqam et al., 2021] studied the effects of several parameters on the starting of a swashplate compressor, but they did not account for initial or boundary conditions such as location or velocity.

On the other hand, [Chen, 2006] incorporated the timing signals of the reciprocating compressor and a numerical solution to anticipate the motor torque-speed curve from zero to the targeted steady state rpm, allowing the speed data to be used in dynamic studies. During starting, the torque resists the acceleration; hence, the motor must overcome these resistive forces. Based on strain gauges, [Matsushima et al., 2000] design a torque measurement system for rotary compressors. The resistive torque prediction is also performed for rotary compressors, but the same technique may also be used to reciprocating compressors. Although vibrations are inevitable, their minimization is crucial to feasibly isolate the vibrating compressor body from

the rest of the refrigeration system and reduce the undesired vibration-sourced noise for home appliance user convenience. Therefore, the design and suppression capabilities of the suppression elements must be investigated both with numerical methods and experimental work in detail. Furthermore, external solutions such as extra suppression elements as well as motor control methods might also be utilized to reduce detrimental vibrations.

3.2. Transient Vibration Reducer

The majority of section 3.2. has been published in 2022 with the title “A novel transient vibration and noise reducer (TVR) for vibratory machines” in the Applied Acoustics to be a part of this thesis [Oral et al., 2022b].

3.2.1. Motivation

Soft thermoplastic polyurethane (TPU) mount-spring sleeves (henceforth referred to as transient vibration reducers - TVRs) are proposed as novel design additions to the conventional single-cylinder hermetic refrigerator compressor to dampen the severe transient body displacements and the accompanying noise in this study. The suggested accessory design addition can function as a universal solution since it can be quickly integrated to the standard coil spring support design of the reciprocating compressor and can result in a significantly smooth and silent start and stop without sacrificing steady-state performance. The straightforward yet effective technique seeks to solve the vibration concerns at their source mechanically within the housing, as opposed to utilizing external voltage control solutions. In addition, despite the fact that TVR is a passive solution, it functions similarly to an active damper due to its innovative adaptive design, only interfering with the body-spring connection in the event of excessive body displacements. Thus, it may be simply customized to lessen displacements in any vibratory machine.

3.2.2. Materials and Methods

Design and Manufacturing

Figure 3.1a depicts the proposed TVR design as a mount-spring sleeve along with its critical dimensional parameters. The principal axis convention adopted for the remainder of this study is also depicted, with the x-axis aligning with the piston-cylinder axis and the y-axis perpendicular to the piston and parallel to the ground. It is recommended that four identical sleeves be installed at the bottom of each compressor body-housing link to attenuate transient state vibrations during operation. D1, the bigger inner diameter closer to the compressor body with axial length L1 and wall thickness t, and D2, the narrower inner diameter closer to the housing floor with axial length L2-L1, are among the essential dimensions. The TVR fits the bottom section of the mount spring (outside diameter = 14.6 mm) with a degree of snugness. The dimensions parameters of D2 (14.65 mm) and L2 (22 mm) are maintained throughout the design of experiments (DOE) section of the investigation.

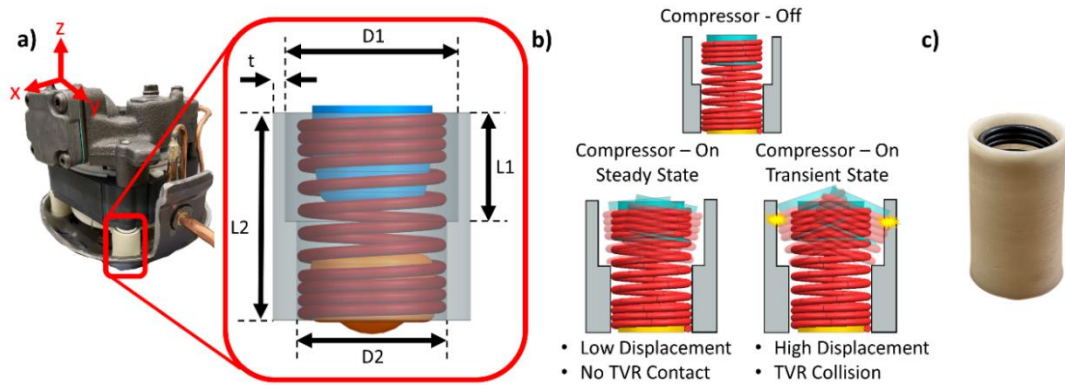


Figure 3.1. (a) Dimensional parameters of the TVR. (b) Working principle of the TVR. (c) Spring and additively manufactured TVR sample prepared for testing.

The radial clearance between D1 and D2 enables the TVR to perform similarly to an adaptive damper that is only triggered over a specific displacement threshold; the TVR's operating principle is depicted in Figure 3.1b. During the low RPM,

transient starting and shutdown phases where high body vibrations ($x_{Body} > D1-D2$) occur, the coil spring contacts and collides with the inner wall, therefore reducing displacement. During steady-state, however, the vibration levels are lower ($x_{Body} < D1-D2$), enabling the coil to vibrate freely without interacting with the inner diameter of the sleeve's top portion. In light of the constant contribution of the mount springs and the conditional contribution of the TVR to the damping, the dynamic force behavior of the TVR-equipped compressor can be expressed as a set of x- and y-axis equations (Eq. 1), taking into account the conditional contribution of the TVR to the damping (Meirovitch, 2001).

$$F = \begin{cases} m\ddot{x} + kx, & x_{Body} < D1 - D2 \\ m\ddot{x} + c\dot{x} + kx, & x_{Body} \geq D1 - D2 \end{cases} \quad (3.1)$$

where m is the oscillating compressor body mass, c is the damping coefficient introduced by the TVR, and k is the spring stiffness as depicted in figure 3.2.

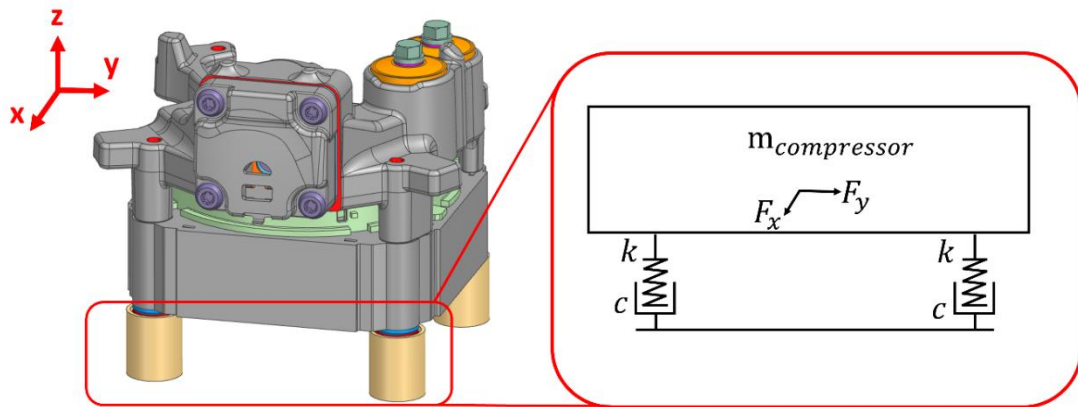


Figure 3.2. Equivalent 2D Dynamic Body Diagram

The material for TVR must be carefully selected so that no excessive noise will be produced by spring-sleeve collision, the object will maintain its structural integrity and function for long-term use, and the manufacturability costs will be low for feasible integration into commercial compressors already on the market. An

excessively hard sleeve can interfere with the natural frequencies of the system and cause damage to the elements upon collision, whereas an excessively flexible sleeve may be ineffective in suppressing the vibrations effectively. In light of these design considerations, the thermoplastic polyurethane (TPU) polymer family is regarded acceptable for use in this study as the TVR material due to its desired resilience and mechanical damping capabilities. A TPU grade [Ultrafuse® 80A LF Technical Data Sheet] accessible in filament form for fused-deposition modeling is selected for the rapid and inexpensive prototyping of TVRs. Figure 3.1c depicts a TVR sample made by additive manufacturing.

Design of Experiments with Finite Element Analysis

To examine the impact of TVR dimensions on its dynamic characteristics, a DOE research employing FE analysis is performed. . The values of D1, L1, and t are adjusted for the DOE, but D2 and L2 are held constant. The thickness of the TVR wall with which the springs engage during the transient phase is determined by the parameter t; a larger t value will result in greater damping. D1 specifies the body displacement threshold at which the spring begins to collide with the TVR inner wall; a higher D1 value will result in fewer collisions. Lastly, L1 determines how much of the spring may vibrate freely in the axial direction; a greater L1 value would limit less of the mount-spring at the bottom. Before calculating the ranges for the dimensional parameters, experimental data on the body vibrations of the compressor in each axis are collected. The laser sensor is used to monitor the displacement of a commercial compressor working at 1300 rpm in a steady-state manner (standard operation conditions). The parameter range of D1 is then established in line with the experimental data, with the upper limit chosen to be equal to the maximum displacement recorded and the lower bound set to be somewhat more than the maximum displacement at steady state. The determined range assures that the researched potential TVR designs do not interfere with steady-state vibrations, yet collisions and contact are permitted during the transient phase.

In the construction of the DOE, a face-centered cubic central composite design (FCC-CCD) is implemented. In computer-aided design software (Siemens NX, v19), 15-point candidate designs are developed and then exported to FEA software (ANSYS, v19). Since TPU was chosen for the TVR due to its good damping characteristic, its nonlinear response must be characterized accurately in the FE simulations. Mooney-Rivlin [Photiou et al., 2021] is picked among the models previously applied in the literature to simulate TPU response [Ali et al., 2010; Wang et al., 2020] since it has been deemed to adequately suit the experimental data. To model the incompressible, isotropic, and hyperelastic behavior of the TVR, ISO 527 elongation-stress data is used (Ultrafuse® 80A LF Technical Data Sheet); a three-parameter Mooney-Rivlin (M-R) fit is determined to capture the mechanical behavior adequately (Table 3.1).

Table 3.1 Hyperelastic material property used in FE simulations for the TVR

Stress (MPa)	50% Elongation	5.8
	100% Elongation	6.9
	200% Elongation	11.5
	610% Elongation (Break)	35.0
M-R Parameters	C10	0.147
	C01	3.539
	C11	0.103
	D1	0

The response to an acceleration stimulus is selected as the important output parameter to assess the vibration suppression capabilities of the TVR; the lower the peak acceleration (a_{peak}) response of the TVR, the more efficient it will be in damping transient vibrations. With the boundary conditions depicted in Figure 3.3a, 1 Hz half-sine shock base acceleration with a 1.75 m/s² peak is delivered to the TVR.

After doing a mesh sensitivity analysis, each proposed TVR is then meshed with tetrahedral elements of the proper size (Section 3.2). The peak acceleration response of each TVR is recorded, and a two-way statistical analysis is conducted to assess the

impact of design parameters on the TVR's vibration suppression capabilities. The most effective TVR design is then produced and incorporated into a standard refrigerator compressor for experimental testing.

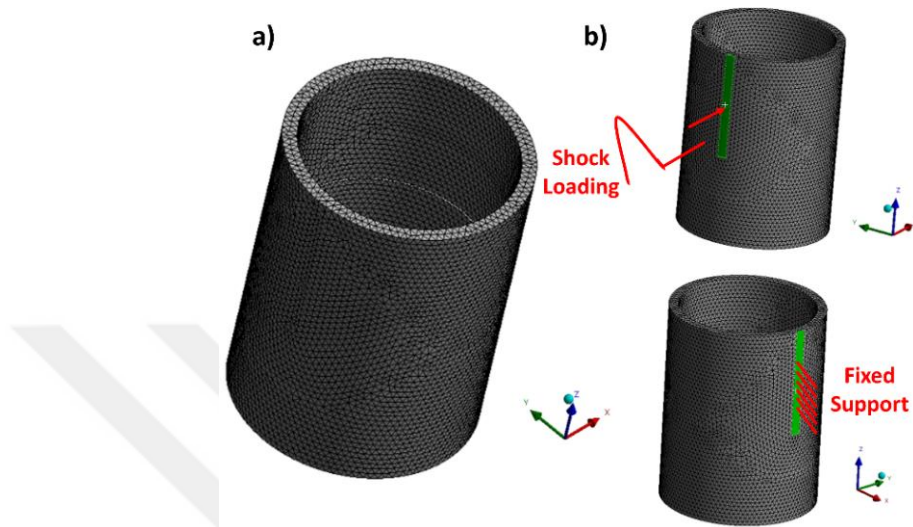


Figure 3.3. (a) Meshed TVR model and (b) boundary conditions used in the FE simulations.

Experimental Setups

Three distinct mechanical experiments are conducted to assess the influence of TVR on the dynamics of the compressor and to compare it to the baseline, unmodified design. The initial experiment is the body displacement test (Fig. 3.4b). A reflecting item is attached to the top of the cylinder, and x- and y-axis displacement measurements are collected using a laser sensor from starting to shutdown; z-axis displacement is ignored because no significant vibrations occur in this direction. The second experiment is acceleration measurement (Fig. 3.4a), in which an accelerometer is positioned above the cylinder on the crankcase and root mean square values of steady-state vibrations are recorded and compared. These experiments are done without compression force (no refrigerant) and with the cylinder head remaining connected.

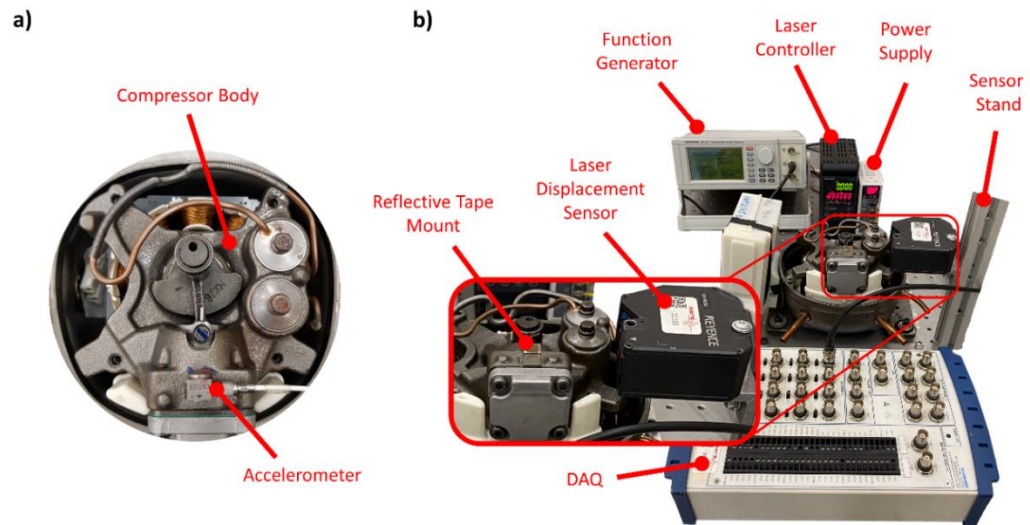


Figure 3.4. (a) Acceleration and (b) displacement measurement setup.

In a semi-anechoic environment, ISO 3745-compliant sound pressure level measurements are performed to evaluate the acoustic performance with and without TVR attachment. 10 B&K 4955 type microphones are used to record the noise data at designated spatial locations in a half-spherical surface configuration (IEC 60704-1 standard) centered around the compressor; the precise location of each microphone is tabulated in Table 2 with the assumption that the compressor is located at the origin of the setup depicted in Figure 3.5. For acoustic investigations, isobutane gas R600a is utilized. The compressor is fixed on a fixture through the bottom grommets and operated at the required speeds with the aid of a function generator in order to determine the noise level at various pressure levels.

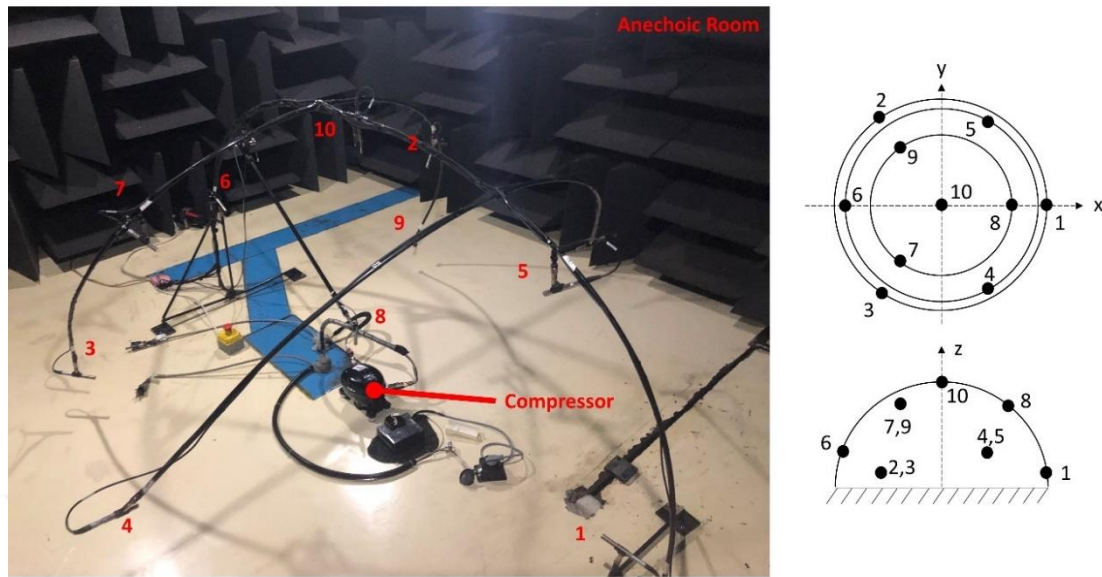


Figure 3.5. Acoustic measurement test setup and the relative positions of the microphones in half-spherical configuration.

Table 3.2 Microphone spatial locations for the acoustic test.

	X Coordinate (m)	Y Coordinate (m)	Z Coordinate (m)
Compressor	0	0	0
Mic 1	0.99	0	0.15
Mic 2	-0.5	0.86	0.15
Mic 3	-0.5	-0.86	0.15
Mic 4	0.45	-0.77	0.45
Mic 5	0.45	0.77	0.45
Mic 6	-0.89	0	0.45
Mic 7	-0.33	-0.57	0.75
Mic 8	0.66	0	0.75
Mic 9	-0.33	0.57	0.75
Mic 10	0	0	1

3.2.3. Results

Initially, peak acceleration values are acquired using finite element calculations, and the influence of design factors is explored. After that, a prototype of the design with the best vibration suppression capabilities is created. The TVR is then subjected to a variety of tests.

Design of Experiments

Different combinations of TVR dimensions will provide geometries with varied damping properties. As outlined in Section 2.2, the parameter ranges of the TVR variable dimensions and the related FCC-CCD type DOE table (Table 3) are computed based on the displacement measurement for each axis using the Fig 3.4b configuration. The displacement data is displayed in Fig. 3.6, and it is noted that the highest body displacement in the starting transient area occurs in the X-direction with 1.25 mm (amplitude of 2.5 mm). In consideration of the spring's 14.6 mm outer diameter, the D1 value is changed between 15 and 17.5 mm to provide free vibration during steady-state and TVR inner wall contact during transient areas. The values for L1 and t are approximately 10-12 and 1-2 mm, respectively.

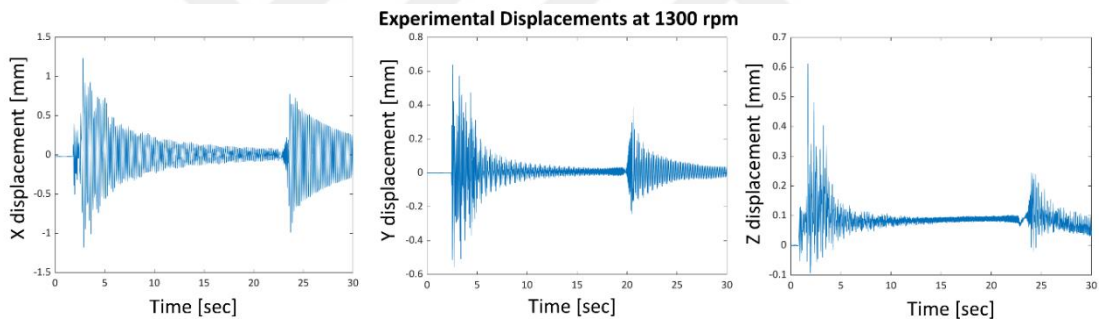


Figure 3.6. Body displacements for the compressor at 1300 rpm at each principal axis from startup to shut down.

Table 3.3 DOE points and corresponding peak accelerations are derived from FE calculations.

#	D1	t	L1	Peak Acceleration [m/s ²]
1	15	1	10	2.342
2	17.5	1	10	3.785
3	15	2	10	2.745
4	17.5	2	10	2.966
5	15	1	12	3.591
6	17.5	1	12	3.916
7	15	2	12	2.743
8	17.5	2	12	2.967
9	15	1.5	11	2.834
10	17.5	1.5	11	3.170
11	16.25	1	11	2.660
12	16.25	2	11	2.744
13	16.25	1.5	10	2.758
14	16.25	1.5	12	2.742
15	16.25	1.5	11	2.750

FEA Results and Response Surface Analysis

A mesh sensitivity analysis is performed on a proposed TVR design (D1:15, t:10, L1:1) to establish grid independence of the FE results. The size of the tetrahedral element is gradually reduced from 1 mm to 0.3 mm, and the results of the peak acceleration are recorded. Figure 3.7a reveals that the results converge adequately and that the output is independent of mesh size after 0.45 mm element size; hence, 0.45 mm size elements are used in the FE simulation of the remaining design locations. In simulations including 15 design points, the maximum number of elements is 170585 and the maximum number of nodes is 261768. In Table 3, the FE findings are collected with the DOE points.

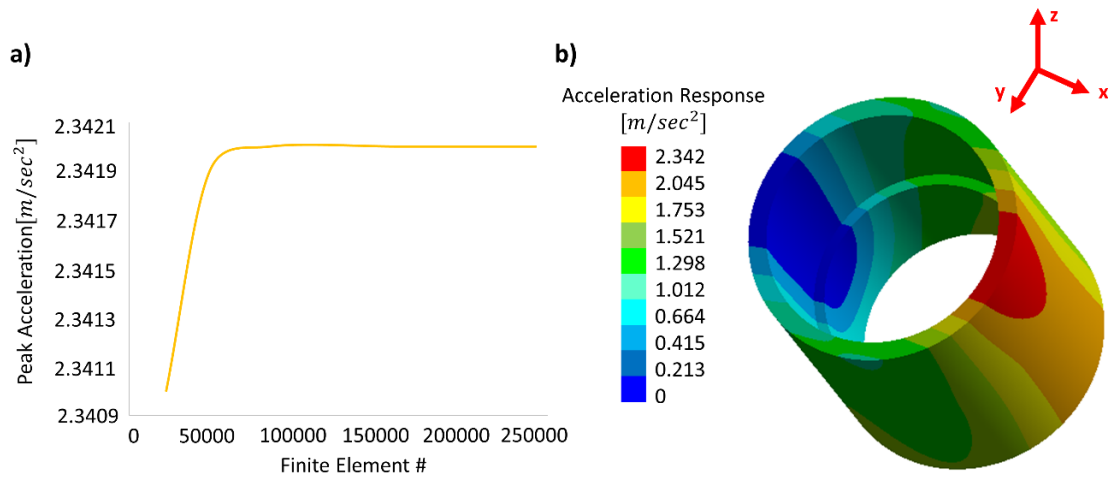


Figure 3.7. (a) Mesh independence study. (b) FE acceleration response result of the TVR with $D1 = 15$, $t = 1$, and $L1 = 10$.

Using the FCC-CCD, a response surface analysis is conducted to establish the relevance of each dimensional design parameter on the TVR's suppression feature. The observed random distribution and constant variance of the residuals verify the model's validity. The variance produced by the completed response surface regression was 85.81%. (R^2). The significance threshold for the two-way analysis of variance (ANOVA) test is set at 0.05. The estimated p-values for the linear model are 0.035, 0.062, and 0.186 for $D1$, t , and $L1$, demonstrating that the inner diameter, $D1$, is strongly linked with the change in the TVR's peak acceleration response. The 2-way interaction model revealed no significant link ($p > 0.05$) between the parameters and output (2-way p-values: $D1$ - t : 0.157, $D1$ - $L1$: 0.22, t - $L1$: 0.143).

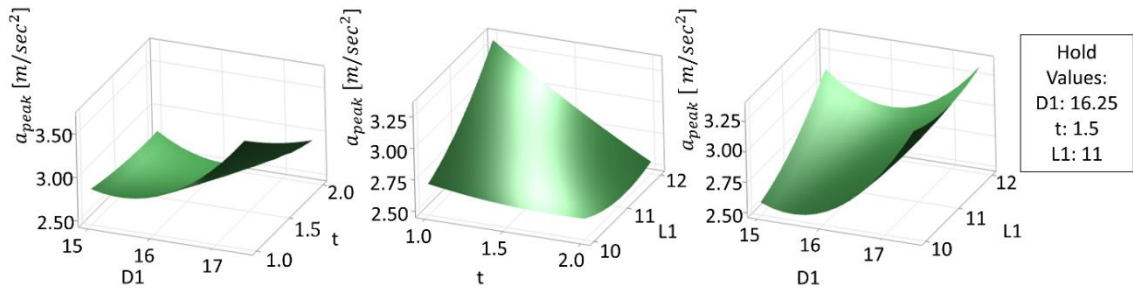


Figure 3.8. Response surface graphs of TVR design parameters D1, t, and L1 against peak acceleration response.

Figure 3.8 depicts response surface graphs with hold values $D1 = 16.25$, $t = 1.5$, and $L1 = 11$ mm to better comprehend and visualize the parameter impacts. $D1$ and a_{peak} can be interpreted to have an inverse relationship, so that a smaller inner diameter of the TVR results in more damping. Observations indicate that the effects of t and $L1$ on a_{peak} are somewhat reliant on the ranges of the other two variables. Towards the defined top limits of $D1$ and $L1$ within the selected range, t exhibits a strong inverse connection with a_{peak} , however there is no discernible influence at the lower ends of the ranges. The opposite is true for $L1$ in that there is a positive association between a_{peak} and the lower ranges of $D1$ and t , whilst the upper range shows no influence. Taking into consideration the findings of the DOE investigation, the ideal design must be determined to reduce acceleration response and successfully dampen transient vibrations. The best design in the available parameter space is determined to be design point number one, which minimizes all three variables to generate the lowest a_{peak} . Therefore, TVRs with the 15-1-10 ($D1$ - t - $L1$) combination are made for experimental testing of the compressor.

Experimental Results

The benchmark compressor is run 10 times from starting to shutdown, five times without TVRs and five times with TVRs installed at each leg. Figure 3.9 depicts the highest body displacements in the X and Y axes recorded for each test using the setup of Figure 3.4b. There are two probable causes for the same compressor to have a

different transient response between subsequent runs. The first input is the target steady-state frequency, with a higher frequency requiring more time to reach than a lower frequency; however, multiple runs of the compressor at different target frequencies revealed that this input has little effect on the change in maximum body displacement during the transient startup phase. The second explanation for the distinct transient response is the beginning location of the piston within the cylinder, which has been identified as the principal determinant of the variance in maximum body displacement in Fig. 3.9 among trials conducted under identical conditions. The source of these modest variances is because the piston starts its motion from different points within the cylinder, resulting in varying amounts of resistive torque during the engine's starter. If the piston begins its initial half-cycle just past the top dead center at lowest cylinder capacity, the piston will encounter suction force. In contrast, if the piston begins to move immediately after bottom dead center at maximum cylinder capacity, it will be subjected to compression force first. Consequently, the starter torque will vary based on the initial location, resulting in distinct maximum transient vibrations.

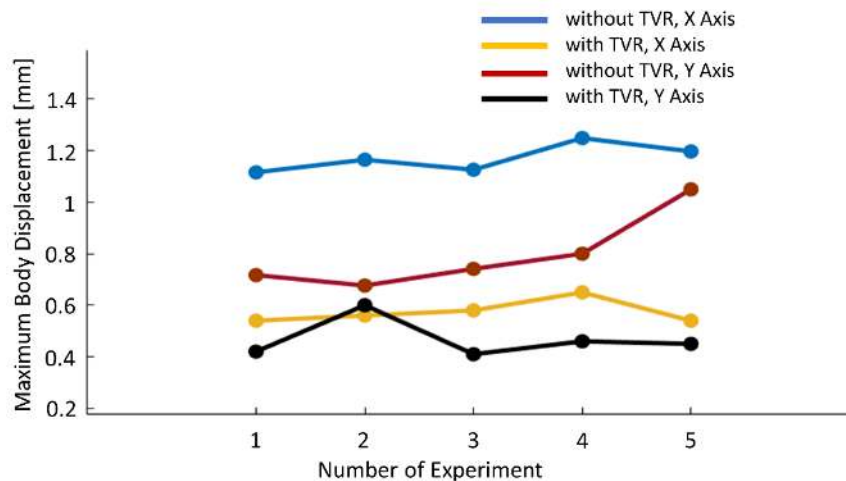


Figure 3.9. Maximum measured compressor body displacements in the X and Y axis for the compressor with and without TVRs.

The incorporation of TVRs significantly decreases the maximum displacements along the X and Y axes across all tests, as seen in Figure 3.9. On the X-axis, a 51% decrease is obtained (with TVR: mean (M) = 0.574 mm, standard deviation (SD) = 0.045; without TVR: M = 1.17 mm, SD = 0.054), while on the Y-axis, a 41% reduction is accomplished (with TVR: M = 0.468 mm, SD = 0.076; without TVR: M = 0.796 mm, SD = 0.148). In the absence of TVRs, the only suppression elements in the compressor (together with the discharge tube) are the mount springs, but the overall reduction in startup vibrations is evidence of the effectiveness of TVRs as additional damping accessories. Consequently, based on the results of the measurements, implementing the recommended design modification should result in less noise and a more comfortable refrigerator experience.

A further recognized advantage of TVRs is that it takes the compressor considerably less time to achieve steady state. This may be explained by the decreased magnitude of disruptive forces, which allows the compressor to attain the target velocity more quickly and easily. Across all studies, the time required to reach the target steady-state rpm when the body displacements stabilize, and settle is reduced by 67%. When the vibratory behaviors are superimposed on a single graph, this phenomenon and the maximum displacement decrease are clearly seen in Figure 3.10.

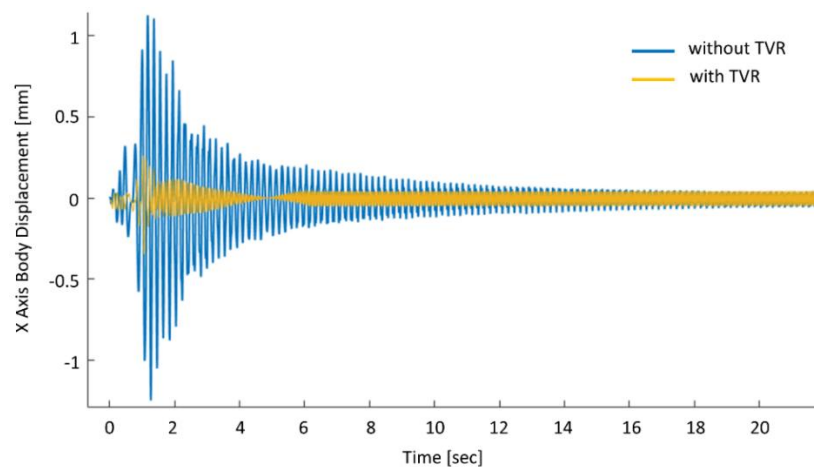


Figure 3.10. Comparison of the vibratory behavior along piston axis of the compressor from startup to steady-state with and without TVRs.

Even though the TVR is developed primarily to attenuate the huge displacements that occur exclusively in transient zones, its steady-state performance must also be evaluated. The TVR half-tightly wraps the coil spring from the bottom, and the TVRs will stay connected for the life of the compressor; thus, it must be assured that the steady-state performance is not sacrificed in an effort to improve the transient behavior. To evaluate the steady-state performance, the Fig. 3.4a configuration is employed, and the RMS acceleration response of the compressor is measured along the x- and y-axes for a variety of steady-state rpms of choice.

Figure 3.11 depicts the RMS findings of vibrational acceleration measurements obtained by sweeping a range of steady-state rpms from 1,300 to 3,000. The addition of TVR to the system has not affected the steady-state performance, as seen by the nearly similar RMS values obtained along the X and Y axes.

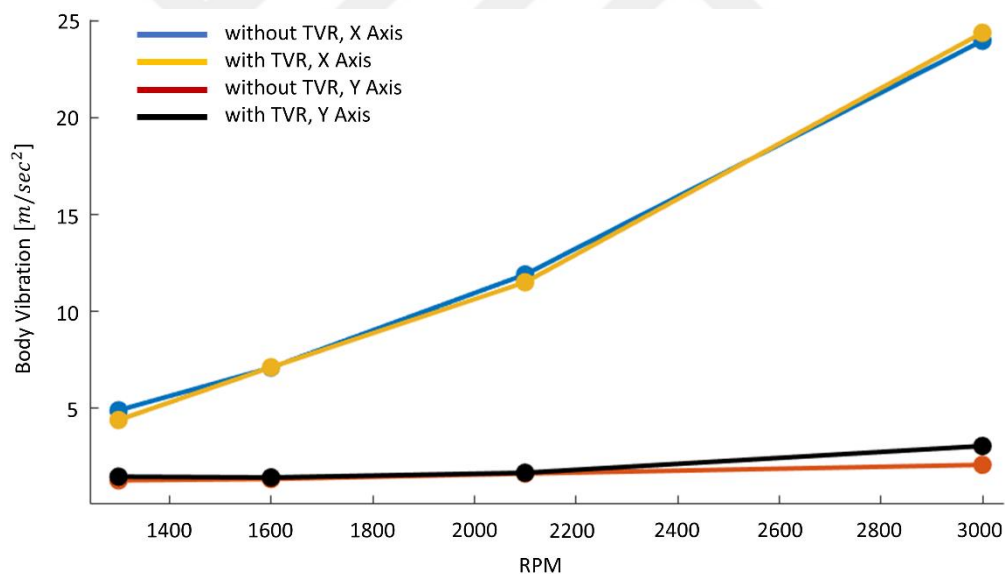


Figure 3.11. Steady-state RMS acceleration measurement comparison across different operating frequencies.

Three distinct inlet-outlet startup pressure scenarios are defined in order to assess the noise performance of the compressor (Fig. 3.5) with the addition of refrigerant to the system. The first scenario considers the operation of the family refrigerator in a

moderate climate or room temperature environment (20-30 C°) with an intake gas pressure of 1.5 bar and an exit pressure of 1.5 bar. The second instance, with an input pressure of 3.5 bar and an exit pressure of 3.5 bar, simulates an external temperature of >30 C°, which would result in a greater inlet pressure and, thus, a higher balanced outlet pressure. In the final scenario, where the input and outlet pressures may be imbalanced owing to unexpected power failures, a range of 0.5 to 3.5 bar is simulated.

Each scenario is repeated four times; thus 24 total measurements are carried out; the compiled results are presented in Table 4.

Table 3.4 Maximum noise level measurement comparison.

Pressure Values	Sound Pressure Level (dBA)					
	Without TVR			With TVR		
1.5-1.5 bar	37.2	M	SD	31.0	M	SD
	36.1			30.7		
	35.7	37.8	2.95	29.1	29.9	1.18
	42.1			28.6		
3.5-3.5 bar	26.9	M	SD	31.7	M	SD
	45.5			32.1		
	36.1	35.1	7.87	31.1	31.4	0.62
	32.0			30.7		
0.5-3.5 bar	38.4	M	SD	26.3	M	SD
	44.6			25.0		
	35.9	41.5	5.12	22.7	25.0	1.65
	46.9			26.1		

Figure 3.12 depicts the maximum noise level generated by the standard compressor. The noise pressure level of the semi-anechoic environment is roughly 16 dBA when the compressor is turned off. As a result of the compressor's beginning behavior, a rapid peak is detected as soon as power is delivered to it. Once steady-state functioning has been attained and the transient period has gone, the noise reading stabilizes.

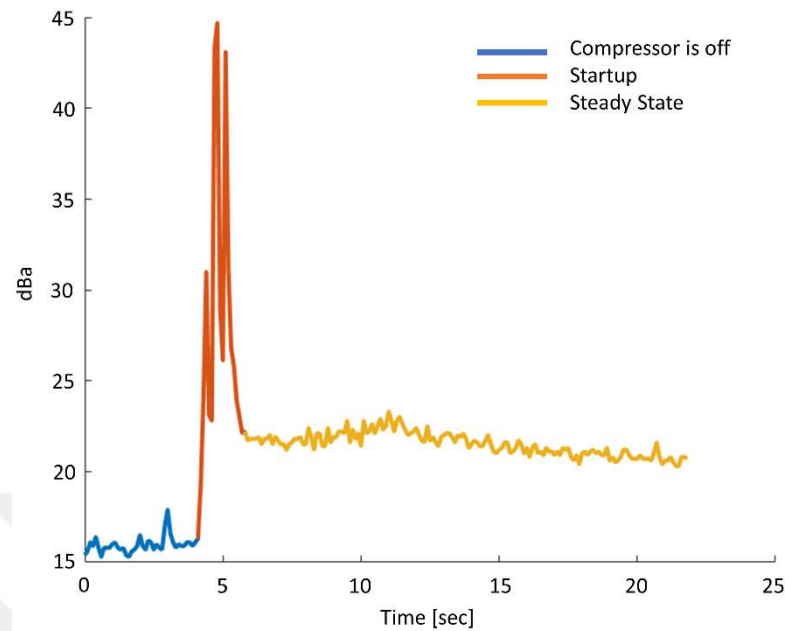


Figure 3.12. Noise measurements from 'off' state to steady-state (3.5 bar inlet, 3.5 bar outlet, 1300 rpm) for the compressor without TVR.

All comparison data reveal that the recorded noise peak during starting is lower in cases with the TVR; the suppression is most obvious for the unbalanced inlet-outlet scenario, where noise levels are reduced from 41 dBA to 25 dBA as mean values. As shown by Table 4's standard deviations, peak noise fluctuation across successive runs without the TVR was also seen to be greater. The variance in peak noise is again attributable to the beginning position of the piston and crankshaft, which is connected to the change in inertia that must be overcome and the cylinder volume that must be swept. Nonetheless, this variance is practically minimized when the TVRs are added to the system due to the increased damping. This is an additional realized advantage that has the ability to standardize the start-up response independent of piston starting position.

Discussion and Conclusion

In this work, an unique TPU sleeve is proposed for the compressor mount spring to decrease the excessive vibrations and noise levels that occur during the startup and shutdown transient stages of the operating cycle. Even though a reciprocating compressor is used in the experiment, the TVR is applicable to any vibratory machine. The described TVR is a passive damper that functions as a pseudo-active damper due to its stepwise design, interacting with the motion of the compressor only beyond certain body displacement thresholds without affecting or compromising steady-state performance. Among the benefits of the TVR is its simple integration into the current compressor construction procedure. Therefore, the TVR is a manufacturer-independent design addition, since it neither interferes with the conventional circuit nor requires a modification to the electrical operation or management of the compressor. The compact sleeve design easily accommodates the limited dimensional clearances between the compressor body and the outside housing, making it a versatile and universal solution to the specified problem.

The producers of machines using slider-crank mechanisms in the industry prioritize steady-state operations while attempting to maximize performance. Similarly, research addressing the transitory state problem is scarce in the academic literature. According to the experimental findings provided in this paper, the most important source of vibration and noise is not during steady-state operation, but rather at startup. An uncontrolled startup can cause not only unwanted vibration and noise, but also excessive power consumption and undesired stress accumulations at structural elements, leading to wear, fatigue, damage, and component malfunction, which is exacerbated by insufficient initial oil supply and lubrication. Therefore, the unavoidable transitory stages at both startup and shutdown must be traversed as rapidly and as precisely as feasible.

The inevitability of the violent starting reaction is mostly attributable to the large amount of current given to the motor in order to overcome the dry friction, inertia,

and the emerging torque on the journal bearing induced by the early compression pressures at the first few of revolutions. Once the crankshaft's angular velocity settles to a consistent value, the huge magnitude of body displacements lessens, indicating steady-state functioning. Incorporating an extra damping component in addition to the mount springs and the discharge tube, the suggested TVR not only lowers the transient body motion, but also offers a relatively quiet operation. According to the findings of the performed trials, the inclusion of TVRs reduces peak displacement by a maximum of 51% and peak noise by a maximum of 40%, while the steady-state performance is practically identical to that of the benchmark. Given the encouraging results, it is reasonable to assert that TVRs can successfully reduce the undesirable transient vibrational behavior to more tolerable levels. The selection of material and the technique of production might be optimized further to improve the design. When compared to alternative passive/active electromagnetic solutions addressed in the literature, the cost-effectiveness of the suggested approach is a noticeable benefit that makes it comparatively attractive. For the proof of concept, a TPU grade that can be FDM 3D printed is chosen to quickly and easily generate TVR samples for testing. Alternatively, other elastic materials with comparable stiffness, durability, and oil resistance may substitute TPU as the TVR material. As long as the material preserves its physical shape and dynamic qualities under extreme working circumstances such as high temperatures, oil-filled surroundings, and repeated impact, it can serve as a viable substitute. In addition, plastic injection and molding are more durable manufacturing techniques that would eliminate the inherent structural flaws associated with additively created parts. However, the exploration of alternative materials and production processes is outside the scope of this project, and the primary focus is on optimizing the dimensional design parameters through a thorough DOE analysis.

In the spirit of acknowledging the study's constraints, a preliminary FE analysis was undertaken to select the most optimal parameters within the provided design range. While the utilized acceleration response analysis is adequate for assessing the

proposed TVR's capacity to dampen vibrations, more computationally intensive approaches exist to model the interaction between the compressor and the TVR in greater detail. A transient structural study that includes the models of the mount spring and the compressor top/bottom legs can disclose the reaction forces at the TVR inner wall contact during impact, enabling the prediction of fatigue life for an additional layer of output that can be improved. In further research, it is intended to undertake a comprehensive comparative computational dynamic study for improved TVR and compressor response prediction. In addition, longer-running operation cycle testing under hermetically sealed and oil-filled circumstances are necessary to reliably predict the performance of a refrigerator during its lifespan.

In conclusion, it is fair to reiterate that although the proposed design improvement is applied to reciprocating compressors in this study, it is readily applicable to other vibratory machines.

3.3. Startup Vibration Modelling

Under regular running conditions, it is safe to evaluate that the compressor of the refrigerator starts its motion from balanced pressure scenerio such that the inlet and the outlet are equalized. However, the compressor might be forced to start at unbalanced pressures, which yield high vibration, in the case of undesirable situations such as an electricity shortage. In addition, high amount of electrical current considering a safety limit is supplied to the compressor during startup in order to overcome the dry friction, inertia, and mostly unbalanced pressures. The gas force that affects on piston must be evaluated in this regard. By assuming that zero degrees of crank angle corresponds to top dead center, the piston will experience the compression immediately if it starts its motion from bottom dead center. However, if it starts to move before the bottom dead center, the suction force will affect on piston, which yield to lower amount of net force.

In order to confirm this theory, an experimental setup is established where the

electrical current data, vibration, crank position, and pressures are measured with an oscilloscope and current probes, an accelerometer, an encoder and transducers, respectively.

The majority of section 3.3. has been published in 2022 with the title “Effect of Starting Position of Crankshaft on Transient Body Vibrations of Reciprocating Compressor” in the International Journal of Refrigeration to be a part of this thesis [Oral et al., 2022d].

3.3.1. Motivation

Despite the fact that there are works that search transient areas, to the best of the authors' knowledge, no study has been published that discusses the significance of the crankshaft or piston's beginning position. This work explores, for the first time in the literature, the influence of starting location on startup displacement using both computational and experimental methods. The displacements of the body are determined based on the crankshaft's initial position, and then verification trials are undertaken.

3.3.2. Materials and Methods

In this work, the forces that develop on the compressor components were first computed using experimentally collected pressure-volume data and encoder data as input parameters, followed by the solution of displacement equations for harmonic excitation using the eigenbody method. Later, trials of validation were done. As seen in figure 3.13, the x axis coincides throughout the research to the piston axis.

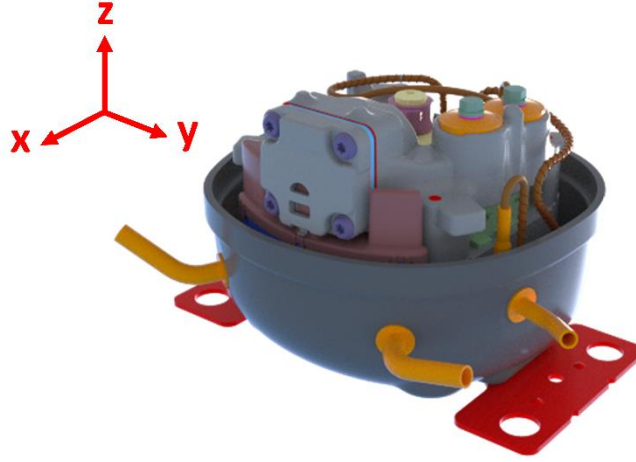


Figure 3.13. Reciprocating Compressor

Displacements

Compressors with reciprocating motion demonstrate force sources including gas force on the piston, inertial forces, and friction forces on the moving components. Moreover, (Meirovitch, 2001) derives the harmonic displacement response of mechanisms with n-single degrees of freedom by considering modal vectors as follows:

$$q = \sum_{r=1}^n \frac{u_r^T Q_0}{\omega_r^2 - a_f^2} u_r \cos \alpha \quad (3.2)$$

where ω_r is resonance frequency, a_f is excitation frequency, Q_0 is the magnitude of force along with investigated principal axis, n is the number of DOF, and u_r is the mode vector. It is worth mentioning that the excitation frequency was obtained through the angular velocity change while the compressor accelerates. The utilized speed data was obtained from the magnetic absolute encoder. The excitation force was also calculated as explained at Chapter 2. The resultant net force on the center of rotation of the crankshaft along with principal axes is assumed as the excitation force to displace the compressor body.

Further, establishing a linear superposition yields displacement prediction by

including the consideration of natural frequencies. Thus, the body displacement can be observed with respect to time based on startup position, i.e., based on startup compression force, since the amount of gas in the cylinder will vary based on piston location. The setup can be seen at figure 3.14.

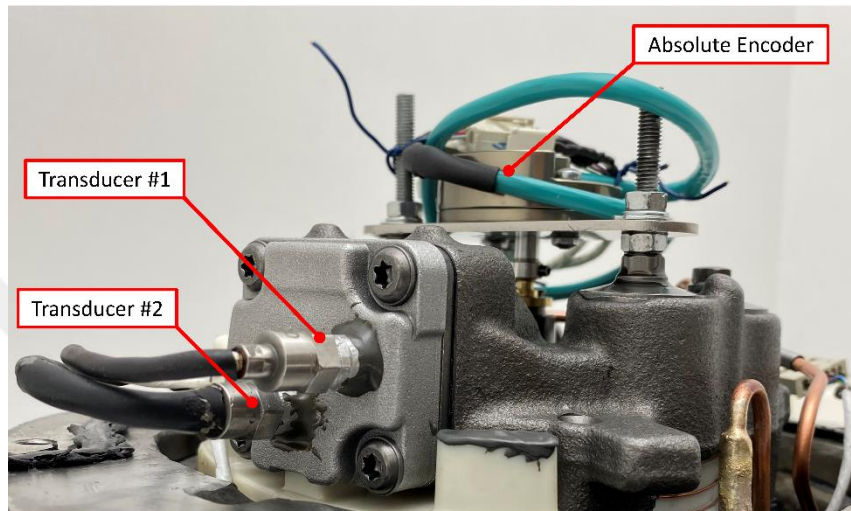


Figure 3.14. Encoder and the pressure transducer on the valve table

Experiments employed R600a refrigerant to imitate real-world circumstances. To ensure sealing, flanged housing was constructed. When the intake and exit pressures converged to a constant value, i.e. when the pressure balance is attained, the pressures were read from transducers and experiments were initiated. Since the established apparatus incorporates a bypass valve between the intake and outlet tubes, the inlet and outlet pressure may be instantaneously balanced, allowing for rapid experimentation. A specific amount of time must elapse until the pressures are equalized; otherwise, the compressor would have an imbalanced start-up. In addition, the absolute encoder offers crankshaft position information at any moment. Consequently, both the crankshaft's starting location and angular velocity may be determined by calculating the rotation's time derivative.

The relationship between crank position, vibration, and pressures were observed thanks to the all-around experimental setup, which is illustrated in figure 3.15.

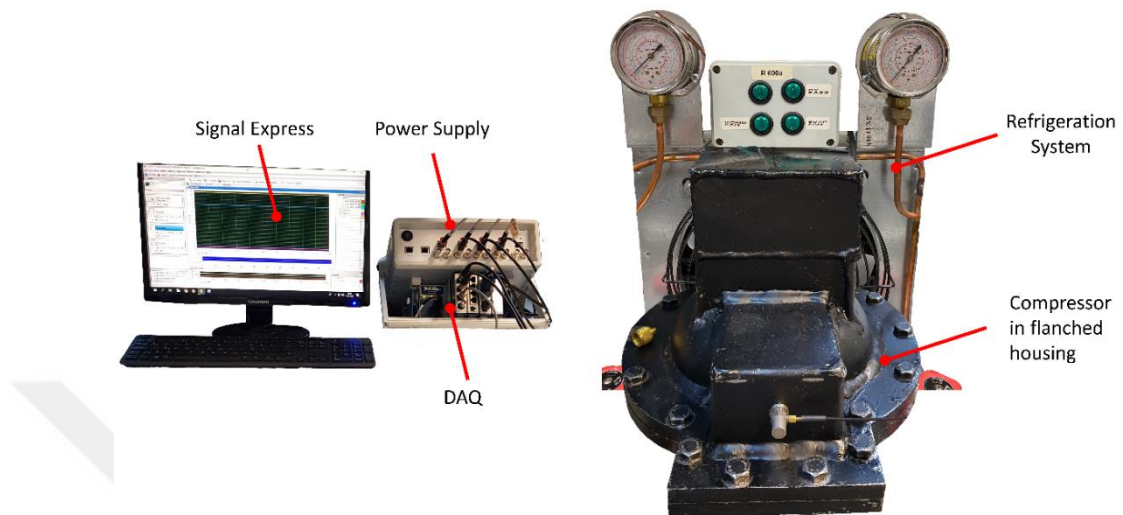


Figure 3.15. Experimental setup

3.3.3. Computational and Experimental Results

Cylinder pressure is also correlated with the influence of crank position on starter vibration. If the piston begins its motion close to the bottom dead center, it will first feel compression. However, if it begins to reciprocate before reaching the bottom dead center in a compressor with a counterclockwise rotating crank, the suction force will initially influence the piston and less force will act on the compressor. Similarly, if the piston begins to move after the bottom dead center (i.e. 200 degrees, 220 degrees, 270 degrees, etc.), the exhaust pressure will be reduced. The pressure within the cylinder is an input parameter for the analytical solution. As a result, the axis of the piston was employed throughout the investigation, as the greatest effect was observed in conjunction with the piston. In addition, when the crankshaft accelerates from zero revolutions per minute to its steady state speed, the bearing forces vary. Consequently, angular velocity data was also incorporated into the formulas.

Eventually, the experimental cylinder pressure data when the crankshaft starts at

110, 181, and 221 degrees were employed for research, as shown in figure 3.16. In these tests, 2.25 bar is the level of equilibrium pressure; hence, evaporation and condensation pressures begin at 2.25 bar. The only difference between successive experiments is the position of the crankshaft. It is evident that if the piston starts at a crank angle of 181, the cylinder pressure builds up more quickly, resulting in a greater amount of forces during the initial phase of operation, i.e. more transient vibration.

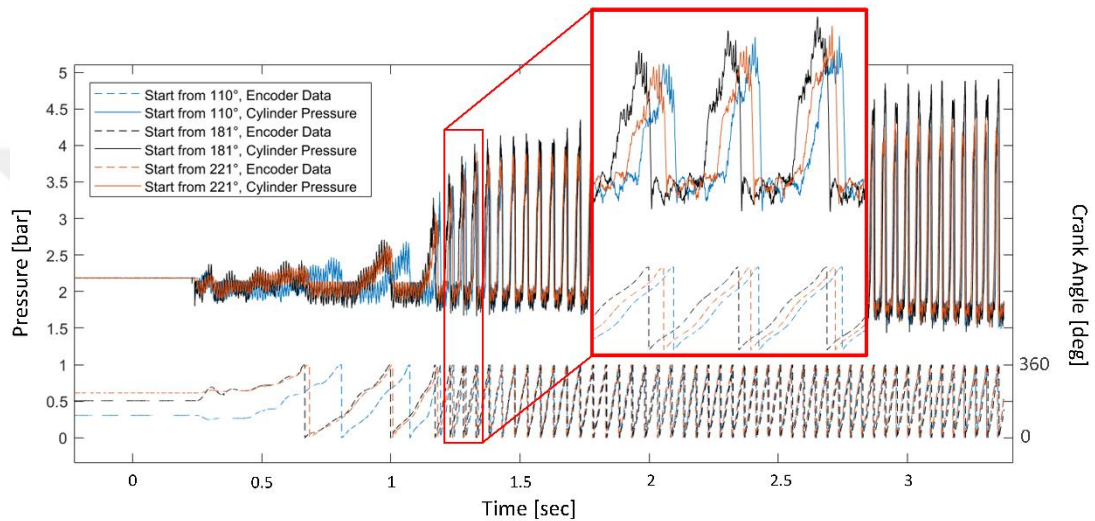


Figure 3.16. Experimental cylinder pressure and encoder data

As seen in Fig. 3.17, equation 3.2 was applied with the predicted bearing forces accounting for friction and inertia as well as experimentally determined gas forces owing to cylinder pressure, and body displacements were computed for these various situations.

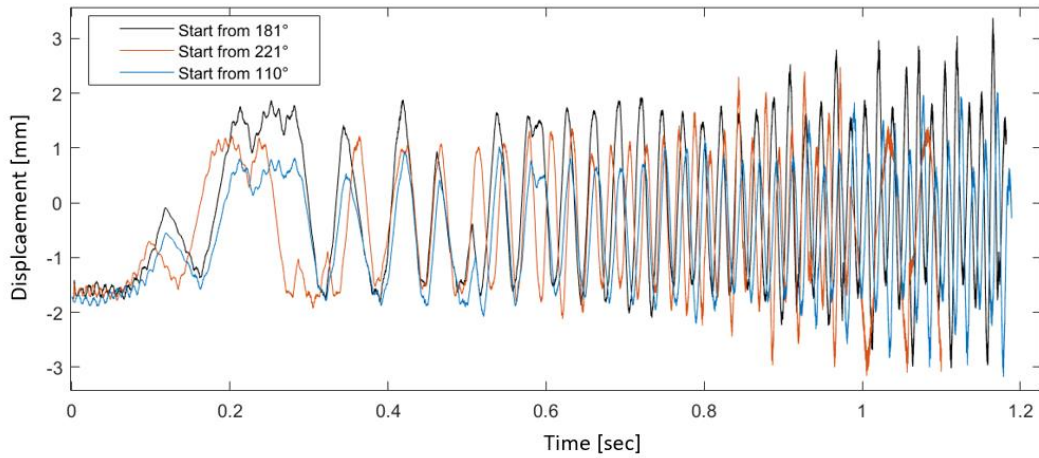


Figure 3.17. Predicted displacements

In the analytically generated displacement data, the displacements begin with a non-zero value since the compressor begins the experiment with a cylinder pressure of 2.25 bar owing to the volume in front of the piston, which results in a non-zero gas force. In figure 3.16, experimental cylinder pressure data, it is evident that there is no considerable increase in pressure during the first couple of revolutions, but that after the third revolution a pressure buildup is evident. Therefore, it is acceptable to assert that the influence of crankshaft beginning position becomes significant after the third rotation. Thus, the initial 10 crankshaft rotations were incorporated into the projections. While the desired speed is 1300 rpm, the crank speed was found to be 1200, 1016, and 1176 rpm after 10 revolutions in circumstances where the crank position was 110° , 221° , and 181° , respectively.

Even though the computed peak-to-peak displacement values are greater when the crankshaft starts at 181 degrees, the RMS and maximum displacement values are also compared. In addition, the vibration of the compressor body is computed by calculating the second derivative of the predicted displacement values with respect to time, as seen in figure 3.18.

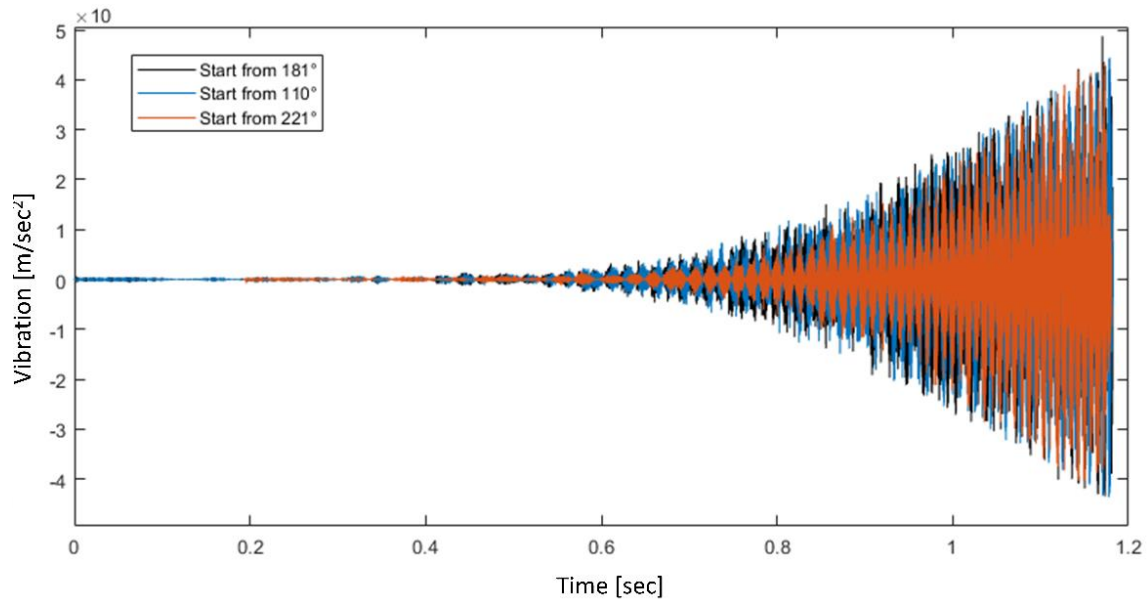


Figure 3.18. Predicted vibrations

Table 3.5 RMS and maximum values of the predicted displacements based on start angle and experimental vibration values

Start Angle [°]	Displacement [mm]		Vibration [m/sec ²]		Experimental Vibration [m/sec ²]
	RMS	Maximum	RMS	Maximum	
110	1.14	2.01	5.27	42.1	14
181	1.271	3.378	5.43	48.7	45
221	1.187	2.479	5.34	43.3	15

The data for maximum displacement, root mean square displacement, anticipated vibration, and observed vibration with respect to beginning angle are displayed in Table 2. If the crank starts at 221 or 110 degrees, the RMS and maximum anticipated displacements as well as the vibrations are reduced.

Moreover, while recording the cylinder pressure and encoder data of these comparative experiments, acceleration of the flaring housing was also recorded along with the piston axis, as shown in table 2, and the analytically calculated displacements and vibrations were examined alongside the experimental vibration measurements. It is important to note that the calculations assume the original geometry and boundary conditions of the compressor. Due to the encoder assembly, pressure transducers, and flaring of the housing, the boundary condition of the compressor had to be altered

during the trial work. Moreover, the stiffness and damping qualities of the grommets are not accounted for in the computations. Since the only source of variation in the experiments is the crankshaft's starting position, it is necessary to assume that the trend of measured startup vibration based on crankshaft starting position should match the trend of analytically calculated body displacement values in order to make a comparison.

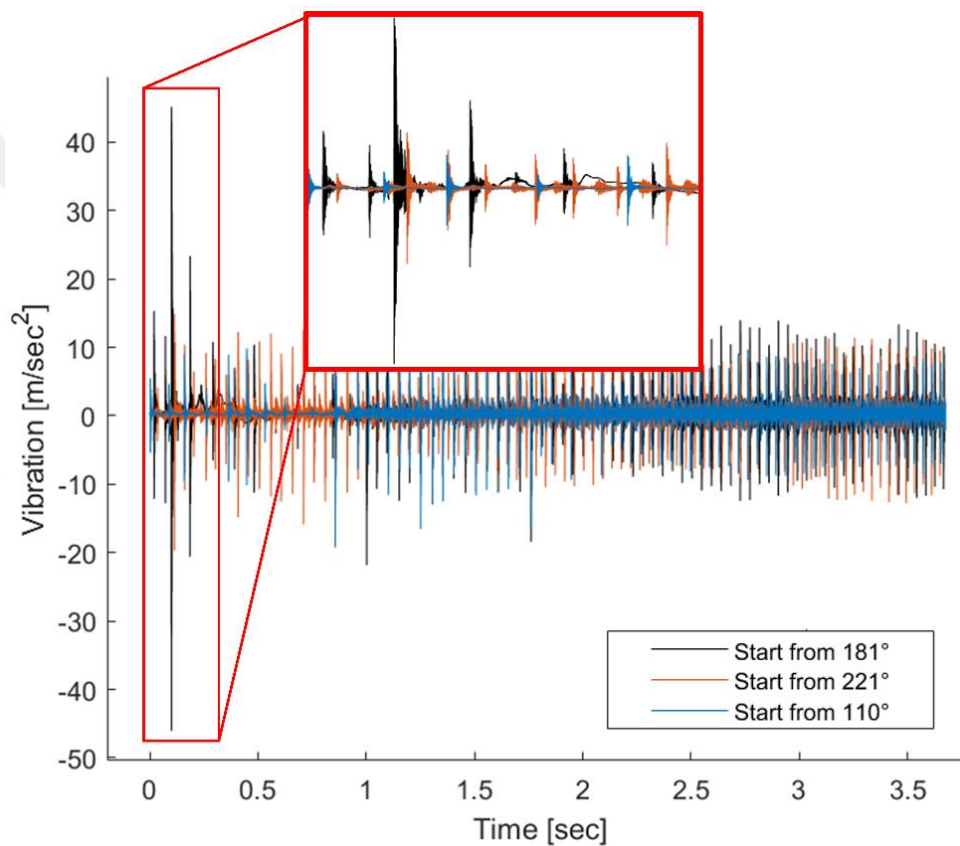


Figure 3.19. Experimental vibration data for start from 110, 181 and 221 degrees.

Figure 3.19 depicts the observed accelerations together with the piston axis. When the crankshaft starts at 181 degrees, the initial vibration peak reaches a larger value, however when it starts at 221 or 110 degrees, the compressor has a smoother starter.

In addition, the same experiment was performed 50 times, and measurements were taken of cylinder pressure, system suction pressure, encoder data, and vibration.

In the meanwhile, the behavior of the crankshaft as illustrated in figure 3.20 is monitored, and mapping study is done for the crankshaft's start and stop locations.

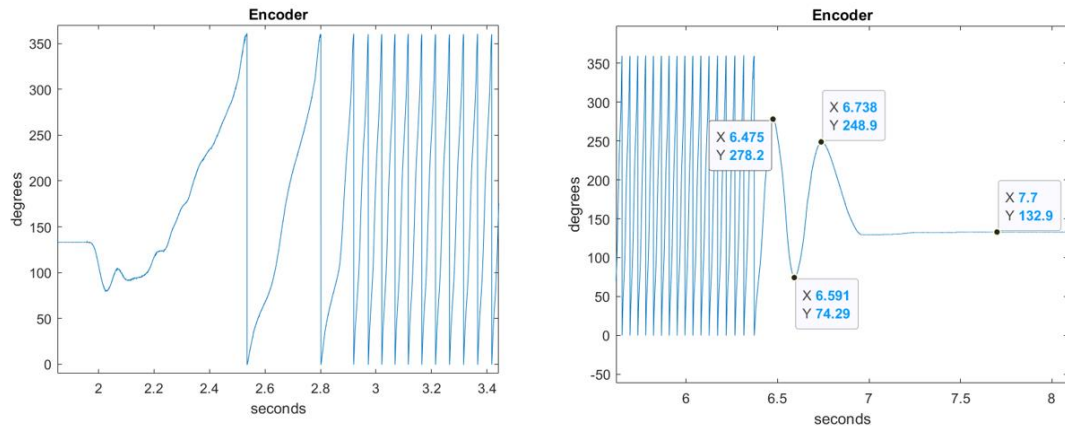


Figure 3.20. (a) Crankshaft startup behaviour, (b) crankshaft stopping behaviour

The oscillation of the crankshaft during phase alignment of the electric motor can be seen at figure 3.20(a), whereas figure 3.20(b) illustrates the shutdown oscillations of crankshaft. Moreover, it is clear that crankshaft changes its rotation direction during shutdown when the rpm reduces significantly. Such phenomenon can be explained by the fact that the gas in front of piston creates a braking effect while piston goes top dead center and pushes it towards back dead center. Afterward, since the system still has kinetic energy, the piston goes to TDC, and this sequence continues until the kinetic energy is dissipated.

Figure 3.21 illustrates that the number of oscillations that the crankshaft demonstrates changes between experiments. This fact can be explained by the fact that the kinetic energy that the compressor carries is different when the stop command is received. Thus, the time required for energy dissipation may differ among the experiments.

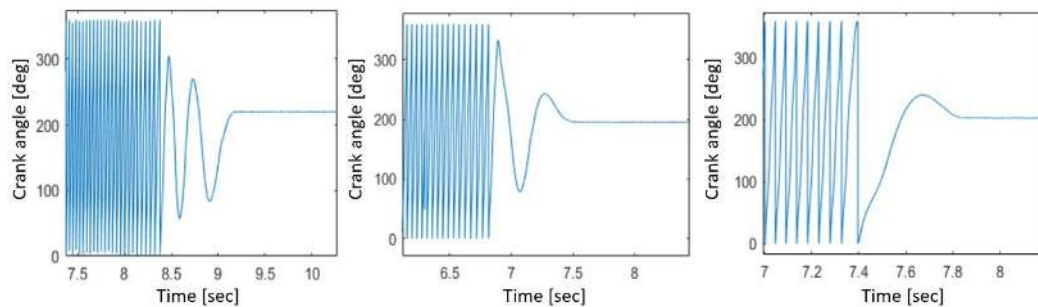


Figure 3.21. Stopping behaviour of the crankshaft from various experiments

Moreover, the start and stop positions of the crankshaft is recorded and a histogram graph is plotted as shown in figure 3.22.

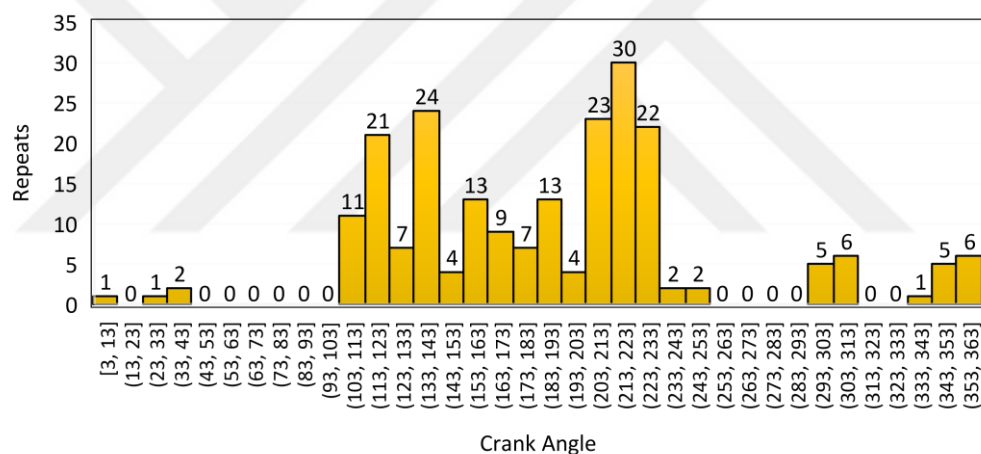


Figure 3.22. Crankshaft's start and stop position map

It is clear to see that the investigated compressor stops and therefore starts mostly either from 100-140 degrees or 200-230 degrees. Furthermore, the third most common position is observed as 150-190 degrees which corresponds to region that is close to bottom dead center.

The peak value of vibration from the transient operations was plotted with respect to starting position, as can be seen in fig. 3.23.

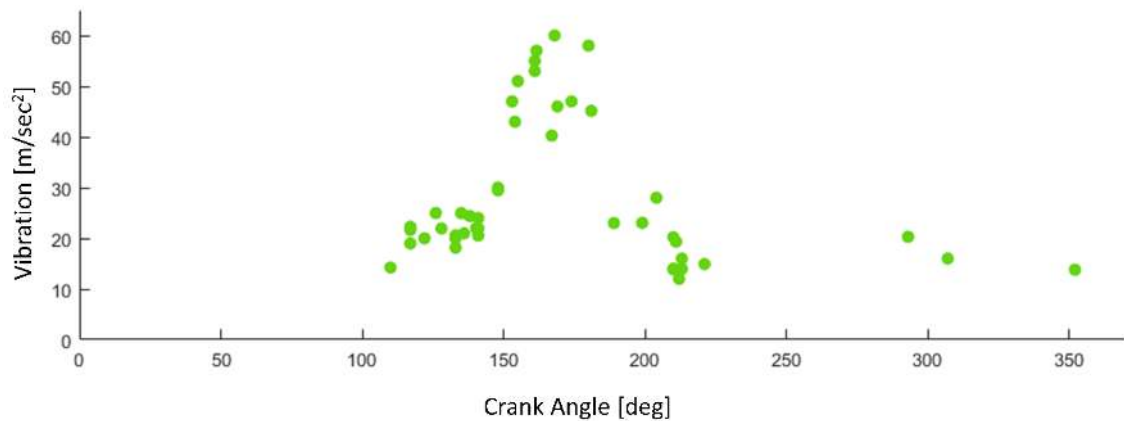


Figure 3.23. Maximum vibration from startup based on initial position of crankshaft

It is clear that as the crankshaft starts from the range between 150-190 degrees, the vibrations are high whereas a smooth start can be achieved if it starts from 100-150 or 200-300 degrees.

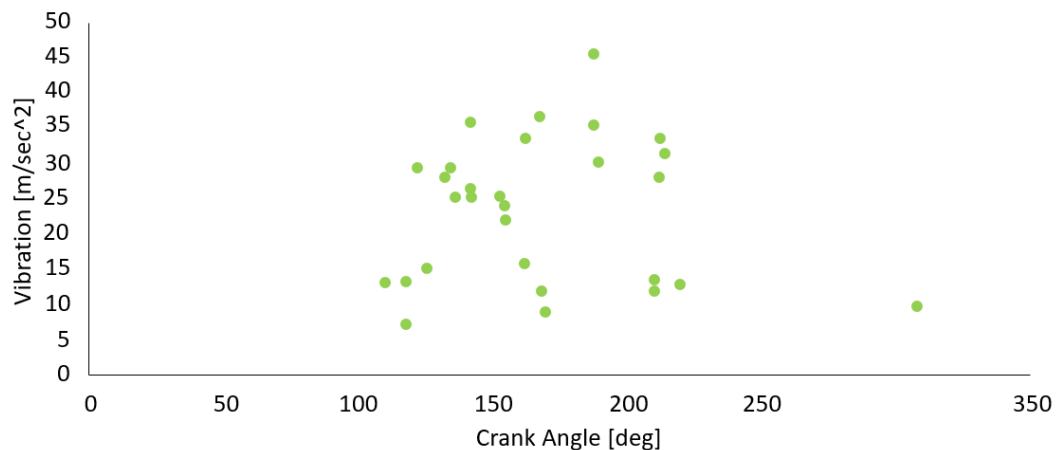


Figure 3.24. Maximum shutdown vibration and stopping location of the crankshaft

Even though a clear correlation is observed between starting position and vibration level during startup, the same behavior was not observed at shutdown, as illustrated in figure 3.24. Rather than the relationship between stopping position and maximum vibration value, the shutdown vibrations must be related to the moment that the compressor receives the stop command. In other words, the kinetic energy that the compressor carries plays a vital role in the shutdown vibration response, as discussed earlier.

Again, the experimental work validates the idea that the compressor will vibrate more if the piston starts around BDC. It will have a smoother startup if the initial position is before or after the BDC, as both the predicted body displacements and the experimentally measured housing vibration agree.

3.3.4. Startup Optimization

In order to optimize the startup response of the investigated compressor, motor control parameters are investigated. It is revealed that 1.5 amps of electrical current are supplied to the compressor for 0.2 seconds to start, which causes an impact-like vibration, before steady state regions take place. In order to smooth the startup, it is proposed that a lower amount of current can be supplied to the motor for a longer period of time. Therefore, 1.1 amps are supplied for 1 second, as illustrated in figure 3.25.

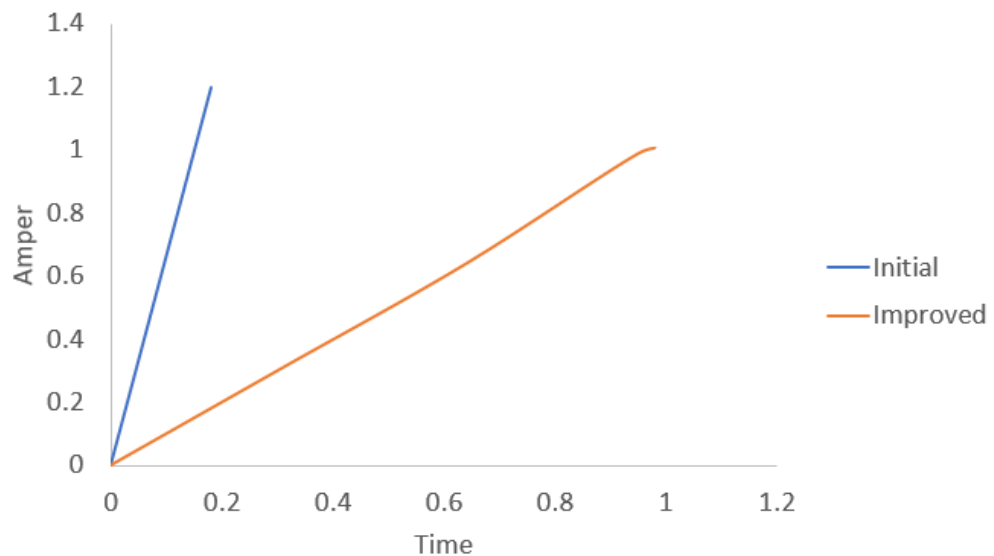


Figure 3.25. The current ramp in original and improved/proposed software

The experimental work is conducted while the investigated compressor is connected to the refrigerator. The inlet and outlet pressures are measured via pressure transducers, while the vibrations along with the piston axis are measured with an

accelerometer. The experimental setup is shown in figure 3.27.

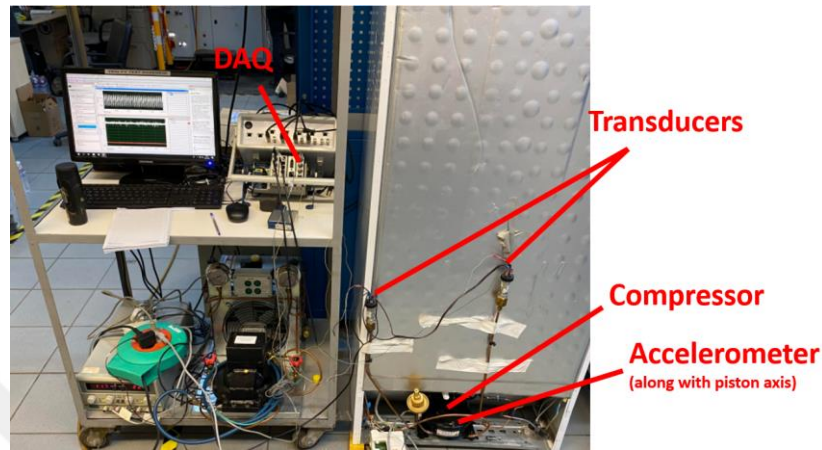


Figure 3.26. Experimental setup to measure vibration and pressures from the refrigerator

The experiments were started when the inlet and outlet pressures were balanced and were repeated four times. It is observed that the maximum peak vibration value has reduced from 6.6 m/sec^2 to 6.2 m/sec^2 thanks to the proposed startup control methods as depicted in figure 3.25.

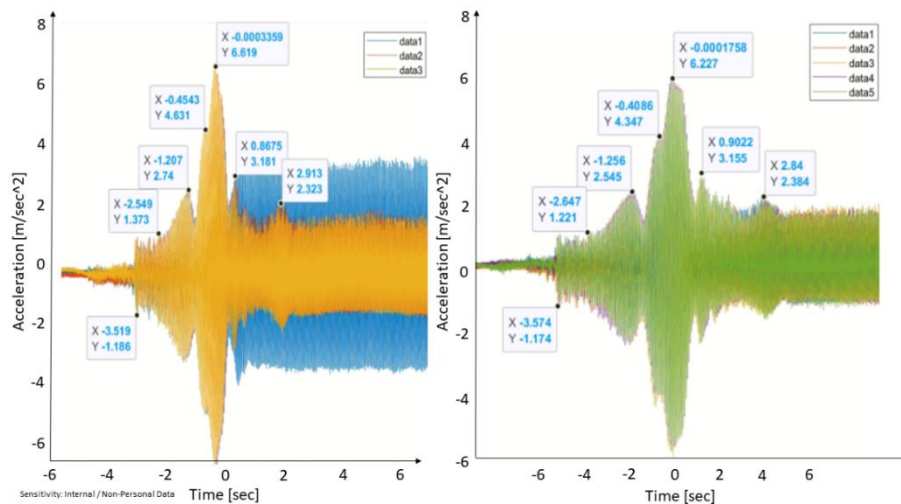


Figure 3.27. Measured vibration from the compressor

Moreover, the sound pressure levels were also measured in a full anechoic room with the refrigerator, as depicted in figure 3.28. The data that was recorded with a microphone that is close to the compressor, i.e., microphone number #4, is used in

comparisons.



Figure 3.28. Sound pressure level measurement in full anechoic room

Both RMS and the peak values are recorded in Pascal as sound pressure level from startup region.

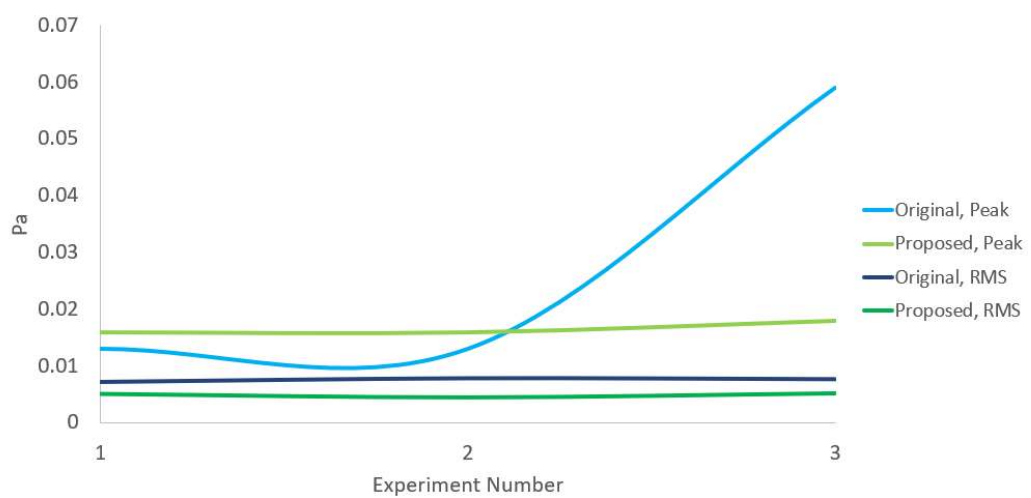


Figure 3.29. RMS and peak value comparison of original and enhanced software

As demonstrated in figure 3.29, the original control methods demonstrate the most varying results whereas the improved control methods deliver stable and lower

sound pressure values.

3.3.5. Discussion and Conclusion

Reciprocating compressors must contend with imbalanced forces that subject the system to vibrations, particularly during startup and shutdown transients. Initially, the friction forces of the compressor were modeled in this study. On the basis of the Greenwood and Tripp contact model, piston-cylinder contact mechanics were examined, and the friction force of journal bearings was computed using Newton's viscosity technique. The friction of thrust bearings was then estimated.

After that, the forces operating on the mechanism were computed using mechanical design characteristics such as diameters, lengths, and so on. Incorporating experimentally determined cylinder pressure into formulations and comparing three sets of experimental data in which the crankshaft was started from various places. In the experimentally measured cylinder pressure data, it is observed that pressure builds up more rapidly if the crank starts at an angle of 181 degrees. Therefore, it is safe to conclude that the cylinder pressure and, the crankshaft location, determine the transient vibration response.

Then, the body displacement together with the piston axis was computed by considering the 6-degree-of-freedom modal vector of the compressor. The eigenvalue method was applied to forecast natural frequencies and modal vectors, and the method and results were confirmed by means of an impact hammer test. After that, the body's displacement was computed using force input and linear superposition of modal vectors. Because the influence of location is negated when the compressor accelerates, only the first 10 crankshaft rotations were accounted for in the calculations.

Acceleration and encoder readings were used to indirectly validate the modification in the transient body displacement response of the compressor. Due to the difference in boundary conditions between computations and experiments, trends are utilized for validation. In calculations, the stiffness and damping qualities of the grommets are disregarded, whereas grommets are used in trials. In addition, the

studied compressor's center of gravity differed from the calculations due to its flanged housing and encoder assembly. In spite of this, it is noted that the crank position plays a significant influence in initiating vibrations owing to the change in gas force, and a significant correlation is detected. Analytically and experimentally, it is seen that if the crankshaft starts at 110 or 221 degrees, the examined benchmark compressor has a smoother starter than if the crank starts at 189 degrees. Among all the experiments, the crankshaft starting angle of 110 degrees produces the least amount of vibration. Taking into account the fact that the crankshaft will start from its stop position, control algorithms may be customized, or active and passive methods can be offered to regulate the crankshaft's stop position in order to achieve smooth and quiet starting operation.

Furthermore, a crankshaft mapping study was also conducted so that the positions where the crank stopped and started its motion were recorded. Since an absolute encoder is utilized during the experiments, exact locations and behavior of the crank are also recorded. It is revealed that the crankshaft oscillates during startup and shutdown due to motor phase alignment and the kinetic energy-gas force relationship.

In addition, an enhancement is also proposed to obtain a smoother startup response by reducing the amount of current supply and increasing the current supplement ramp duration. The vibration measurement experiments are conducted on the compressor that is assembled to a refrigerator, then the refrigerator is moved into an anechoic room for sound pressure level measurements. It is observed that a quieter startup is obtained with the enhanced motor control, which emphasizes the importance of motor control parameters on startup optimization.

Chapter 4

CONCLUSION

The new era of household refrigerator compressors relies on inverter type compressors due to energy efficiency that is achieved by the variable speeds. Thanks to the controller, lower RPM may be utilized when less cooling is required, such as at night when the refrigerator door is rarely opened, resulting in a lower cooling need. Since a compressor that can run at variable RPMs demonstrates different dynamic and mechanical responses against various forces, the optimizations must be made accordingly. To achieve an optimized compressor design in terms of vibration suppression, the behavior of the compressor must be understood clearly.

Therefore, in this thesis, the mathematical model of the dynamic response of a benchmark compressor is developed by considering the required input parameters, such as dimensions, the natural frequencies of the compressor, and the spring and discharge tube stiffness values. A lagrangian eigenbody approach is used to predict the natural frequencies, and impact hammer tests are conducted for validation. Spring and discharge tube stiffnesses are directly measured dynamometer and laser displacement sensor. Then, gas forces are measured with pressure transducers, and inertial forces and frictional forces are calculated analytically. The formulations that include all force sources are established by considering the harmonic motion of the compressor and the body displacements of the compressor are obtained.

Afterwards, in order to enhance the design process of the discharge tube, which is the 5th element that connects body to the housing in addition to four springs, a novel methodology considering von Mises probability distribution is developed. The tubes created by computer code is exposed to static and modal analyses through the finite element method. A decision algorithm is employed to determine a new tube design that performs better than the investigated compressor's discharge tube.

Afterwards, the transient behavior of the compressor is also investigated. Because of the fact that the transients are the conditions where the harshest forces emerge, i.e., employ sudden changes, during a running cycle of the compressor, reducing the effects of transient period is crucial. Similar to the mathematical model developed for steady state operations, the startup behavior of compressors is also modeled. The effect of starting position of crankshaft or piston is investigated, and it is revealed that the starting position affects the transient vibrations and noise significantly. Afterwards, the optimal starting position is proposed.

Moreover, since the highest vibration and the noise emitted from the compressor emerge in this region, a novel design addition called Transient Vibration Reducer (TVR) is developed and implemented to the compressor in a way that it surrounds the springs. In addition, despite the fact that TVR is a passive solution, it functions similarly to an active damper due to its innovative adaptive design, only interfering with the body-spring connection in the event of excessive body displacements. Moreover, the transient vibrations are reduced by 51% and the emerging noise is reduced from an average of 41 dBA to 25 dBA in peak transient noise level, which are testaments to the success of the prospective design addition to the vibratory machines.

4.1. Future Work

This section provides some suggestions to enhance the dynamic performance of the reciprocating compressor.

The crankshaft tilting behavior inside the bearing may be affecting performance, especially in case of high compression forces. Therefore, the secondary motion of the crankshaft must be investigated for future enhancements. Such a model will allow the compressor engineer to identify the most effective design parameters of the crankshaft so that further optimizations can be conducted. Not only the crankshaft secondary motion, but also piston secondary motion should also be studied in order to realize the noises and vibrations due to piston slap and radial motion of

the piston. While studying secondary motions, the hydrodynamic effect of oil must also be considered in order to include the damping capabilities of oil. If the clearance of the bearings is not a subject of optimization, then utilizing more viscous oil yields lower crankshaft displacements, but it would increase the mechanical losses. Thus, it is safe to state that it requires optimization.

Furthermore, not only the axial stiffness, but also the radial stiffness of the springs must be investigated, measured, and optimized in order to reduce the vibrations. Such optimization study can be made thanks to the mathematical models of finite element models and should also include the properties of the discharge tube. Moreover, having a control mechanism where the stop and start positions can be controlled allows for the elimination of vibration variation during startup and the determination of the optimal position.

Piston diameter also affects the forces that emerge on the bearings, therefore vibrations, yet since the piston diameter effects the stroke volume and thermodynamic capacity, it is not suggested to change the diameter for only vibration reduction.

Overall, the forces creating vibration must be reduced by altering the input parameters, including the control algorithms. Reducing vibration yields a compressor with lower noise emission, lower bearing forces, and therefore longer product life.

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