Available online at www.dergipark.org.tr/en

INTERNATIONAL ADVANCED RESEARCHES
and
ENGINEERING JOURNAL

Journal homepage: www.dergipark.org.tr/en/pub/iarej

e-ISSN: 2618-575X

International Open Access

Volume 05
Issue 02

August, 2021

*Research Article*

# Experimental investigation of the effects of different refrigerants used in the refrigeration system on compressor vibrations and noise

***Zafer Cingiz*** [a], ***Ferzan Katırcıoğlu*** [b], ***Suat Sarıdemir*** [c], ***Gökhan Yıldız*** [d,]* and ***Yusuf Çay*** [e]

[a]*Department of Electricity and Energy, Düzce Vocational School, Düzce University, Düzce, Turkey*
[b]*Department of Electronics and Automation, Düzce Vocational School, Düzce University, Düzce, Turkey*
[c]*Department of Mechanical Engineering, Faculty of Engineering, Düzce University, Düzce, Turkey*
[d]*Department of Mechanical Engineering, Graduate School of Natural and Applied Sciences, Düzce University, Düzce, Turkey*
[e]*Department of Mechanical Engineering, Technology Faculty, Sakarya University of Applied Sciences, Sakarya, Turkey*

ARTICLE INFO

*Article history:*
Received 12 January 2021
Revised 24 February 2021
Accepted 20 March 2021

*Keywords:*
Alternative refrigerants
Noise
Vibration

ABSTRACT

Vibration and noise are undesirable effects in daily life and energy-consuming systems. In this study, the effects of different refrigerants on noise and vibration in a sealed reciprocating compressor are discussed. The study compared the noise and vibration performances of refrigerants with lower ozone depletion potential (ODP) values compared to R22, which has a high ODP value. The study was carried out experimentally in two stages. Firstly, tests were conducted for the coefficient of performance (COP) of different refrigerants. Secondly, vibration and noise data were obtained experimentally for different refrigerants. The results obtained from the experiments showed that both the COP value and the compressor vibration and noise have different values for R22 refrigerant and other alternative refrigerants, but values close to R22 are obtained. It was observed that the compressor noise values and vibration values vary depending on the type of used refrigerant. Average vibration values were determined as 0.604 $m/s^2$ in R22, 0.603 $m/s^2$ in R438A, 0.593 $m/s^2$ in R417A, 0.622 $m/s^2$ in R422D and 0.637 $m/s^2$ in R422A. When the noise values are examined, it was measured as 61.327 dB for R22, 62.913 dB for R438A, 62.187 dB for R417A, 63.715 dB for R422D and 64.913 dB for R422A. R417A, which has a 99% similar noise value to R22, was determined as an alternative refrigerant. COP values were found as 4.07 in R22, 3.98 in R438A, 3.63 in R417A, 3.37 in R422D, and 3.18 in R422A. R438A showing 95% similarity for COP can be used as an alternative to R22. Generally, it was observed that the noise and vibration values are very close to each other for all refrigerants examined.



## 1. Introduction

Noise pollution is a problem that significantly affects our health and social behavior in the developing world. The gradual increase in living standards and the shrinkage of living spaces have made the noise problem in the air conditioning and refrigeration sector important. Considering that these devices are used for almost 24 hours, acoustic and vibration properties are important in terms of quality of life. Especially at night and when people seek a quiet environment and need a state of rest, the noise generated by these devices is disturbing.

Domestic and commercial refrigeration and air conditioning (HVAC) systems constitute an indispensable part of modern life in the world [1]. Although the area of use is wide, these devices are also used to protect perishable substances, regulate temperature-sensitive processes in industries, and provide a comfortable environment [2].

* Corresponding author. Tel.: +90-850-800-8181.
E-mail addresses: zafercingiz@duzce.edu.tr *(Z. Cingiz)*, ferzankatircioglu@duzce.edu.tr *(F. Katırcıoğlu)*, suatsaridemir@duzce.edu.tr *(S. Sarıdemir)*, gokhan.yildiz1987@hotmail.com *(G. Yıldız)*, ycay@subu.edu.tr *(Y. Çay)*
ORCID: 0000-0003-3796-755X *(Z. Cingiz)*, 0000-0001-5463-3792 *(F. Katırcıoğlu)*, 0000-0002-6028-1073 *(S. Sarıdemir)*, 0000-0001-6039-9226 *(G. Yıldız)*, 0000-0003-4007-6168 *(Y. Çay)*
DOI: 10.35860/iarej.859423

Compressors used in refrigeration and air conditioning systems are one of the most important sources of vibration and noise. The most important design parameter for reducing the noise emitted to the environment by the compressors used is possible by reducing the dynamic forces transmitted from the compressor to the casing.

Vibration and noise generated by compressors are transmitted to the compressor outer casing starting from its emergence, and from here it passes to the environment in the form of noise and/or vibration through the environment in which the compressor is located or the system elements.

Li et al. emphasized in their study that the vibration value is transmitted from the compressor to the system by pipes and mounting elements that connect the compressor to the structure. It was stated that the design of these elements significantly affects the dynamic performance of the system. In the study, a sequencing explaining the compressor vibration is defined by using the solid body model. It has been shown that if this sequencing is used, the number of points to be measured in the vibration test can be reduced. The specified analytical method was applied to a compressor system, the results were compared with the measurements, and consistency was observed [3].

Hamilton evaluated the vibration and noise problems in compressors in general terms. In this study, firstly the compressors used in refrigeration systems are grouped and the vibration and noise sources that occur in the compressors and their transmission routes to the environment are emphasized with their general characteristics. Basic sound and noise properties were mentioned. The importance of measuring noise in compressors was emphasized. The measurements that can be applied in the interior of the compressors, the purposes for which they will be carried out, and how the results will be handled were evaluated. The effects of the compressor outer casing on compressor noise were discussed. Besides, measurement techniques that can be used for vibration and noise control in compressors are discussed [4].

The design and production of quieter mechanical systems have been important studies in terms of acoustics and noise control. The noise of appliances in household appliances such as refrigerators is unpleasant and undesirable. The source of noise in refrigeration systems is generally due to the operation of the component in the system and the flow of the refrigerant in the pipes. Vibrations caused by flow in the pipes pose a problem during the design of refrigeration systems. To eliminate this disturbance of refrigeration systems, noise should be reduced or eliminated [5].

Lee et al. in their study conducted different experiments to analyze refrigerator sounds. During the experiments, tests were carried out on how the refrigerator sounds were perceived by humans. For the study, sound recordings were made at home and in a completely silent room using two household refrigerators. Taking measurements from different sides of the refrigerator, it was determined that the main sound source is the compressor. Measurement records were taken from the moment the compressor started operating and the sounds generated were divided into various groups. These groups consist of the sounds during the start of the compressor, the sounds during the stable operation of the refrigerator, and the sounds that occur when the compressor stops. The voices in these groups were tested separately and analyzed [6].

Han et al. in their study dealt with noise originating from refrigerant at the evaporator inlet and the capillary pipe outlet in two different studies. The study showed that due to the transition from expansion to evaporation, bubble formation and interaction caused some degree of flow-induced noise [7] [8].

Jang et al. made a noise analysis of the rotary compressor in a domestic air conditioner. The suction pressure of the compressor is 9.12 kg/cm$^2$ and the discharge pressure is 33.45 kg /cm$^2$. The noise originating from the compressor was examined in two categories as pressure pulse of the refrigerant and structural vibration. As a result of the experiments, it was observed that the refrigerant creates noise while leaving the discharge line, and this noise increases as the speed of the compressor increases [9].

Venkatappa et al. analyzed the noise values of R134a and R1234yf refrigerants caused by the refrigerant flow in the compressor in the automotive air conditioning system. As a result of experiments, the noise and vibration energy generated due to flow is significantly higher in R1234yf compared to R134a [10].

Considering the new technological developments in refrigeration systems and the improvement of the compressor, the noise originating from the refrigerant was almost eliminated. Also, for refrigeration system manufacturers, how to improve performance and reduce noise was always a big challenge. Many consumers complained about the noise originating from the refrigerant in refrigeration systems. For a refrigeration system company with annual sales of more than 1 million in China, the return of the product due to noise resulted in financial losses of over 20 million Yuan. Therefore, it is necessary to investigate and address the noise from originating refrigerant [11].

In a 100 m$^2$ flat in Korea, the noise characteristics of different types of a refrigerator in a real-life environment were investigated. It was determined that the noise level of the refrigerator in the living room is approximately 10 dB higher than the noise level in a quiet room in the same position. A semantic differential test was also conducted using various nomenclatures to evaluate the noise quality of the refrigerator [12].

The number of complaints about noise in living spaces increases dramatically. Especially, there are concerns regarding the noise of electrical devices. The International Noise Control Engineering Institute has proposed noise labels for consumer and industrial products and researched how noise is labeled in various countries [13].

On the contrary other household appliances, users react sensitively to the noise generated by these appliances, as refrigerators operate all day. Consumer reports recommend that consumers control the noise ratio before purchasing a refrigerator. For this reason, the noise levels of the refrigerators should be reduced to create a quieter living environment [14].

Although the noise value in the refrigerator tested in a quiet room decreased, noise complaints and symptoms continue. Because of this, it is necessary to take into account the noise level, flaws, and consumer sensitivity of the refrigerators. Household device manufacturers usually measure refrigerator noise in a quiet room. However, the noise level, frequency characteristics, and differences in sound characteristics in house conditions are very different from environmental conditions. Therefore, the evaluation of refrigerator noise in people's living environments should be made. Also, the noise level is affected by background noise, noise duration, and frequency characteristics [12].

Compressors and fans are refrigerator noise sources that lead to consumer complaints. The refrigerator noise level in the living environment is approximately 40 dB. Some structural improvements to be made in the refrigerator system may reduce this noise level. For example, it was stated that the improvement in the form of rear and side panels reduces the compressor noise level by approximately 2 dB [14-16].

The method of refrigerator noise measurement in a quiet room according to ISO 3745 gives a different sound characteristic according to the noise level measured in living environments. Sound quality assessments and predictive models were proposed in various mechanical engineering applications such as automobiles, HVAC systems, and trucks used in cold transport [17-19].

Refrigeration and air conditioning systems are systems that use compressors and refrigerants. When the compressor in these devices operates, vibration occurs in the pipes due to the compressor. There are two main causes of this vibration: These are vibrations and mechanical vibrations caused by the refrigerant [20].

Pressure fluctuations consist in the compressor as a result of the closing and opening of the valve during the compression and suction of the refrigerant. The vibration occurs in the system pipes due to these fluctuations. Vibration caused by pressure fluctuations is one of the most important causes of compressor noise. Therefore, it is critical to anticipate and prevent vibration and noise [21-23].

The main cause of noise and vibration in refrigeration systems is the compressor. Therefore, it is a determining factor in the noise generated in refrigeration systems. Many scientists worked on reducing compressor vibration noise. Besides, influential methods such as improving the shape of the compressor outer layer before entering the dryer-filter, improving the suction muffler, and replacing the filter drier have been proposed. All of the driers and similar attempts have been found to significantly reduce the noise level of the refrigerator.

In this study, the effects of different refrigerants on noise and vibration in a sealed reciprocating compressor are discussed. Due to the high ozone depletion potential (ODP) of R22 used in the refrigeration system, the refrigerants with lower ODP are analyzed. The study is carried out experimentally in two stages. First, tests are conducted for the coefficient of performance (COP) of different refrigerants. Secondly, vibration and noise data are obtained experimentally for different refrigerants. As a result of the experiments, it is determined that the refrigerant that can be an alternative to R22 for both the COP and the compressor vibration and noise is detected.

## 2. Material and Method

The operation of the system is stopped by a relay because of the signal coming from the thermostat to the compressor, when the cabin temperature is reached at 0 ºC. The refrigerant flow through the evaporator also increases, when the compressor stops running [5-6]. More refrigerant flows again into the evaporator due to the pressure difference between the condenser and the evaporator, in addition to the refrigerant that cannot complete its flow in the evaporator and is therefore trapped there. The refrigerant moves in both the liquid and vapor phases, but it has been found that the noise caused by the flow mainly consists of the liquid phase [20]. When the difference between the cooled space temperature and the set temperature reaches the set level, the compressor starts to operate again. Due to the rapid drop in pressure in the suction line, the liquid refrigerant flows towards the compressor. During this flow, the refrigerant stimulates the pipe walls, causing a temporary noise [21]. Flow begins from the discharge line of the refrigeration system to the suction line when the compressor stops. Besides, some of the refrigerants that evaporate due to increased pressure at a constant temperature condense in the evaporator. After a while, the heat charge in the evaporator comes to just sufficient to evaporate some of the liquid refrigerants until the compressor starts up. A large amount of liquid refrigerant evaporates due to the rapid drop in pressure, when the cycle starts [22]. The experimental steps in the study are given in Figure 1.

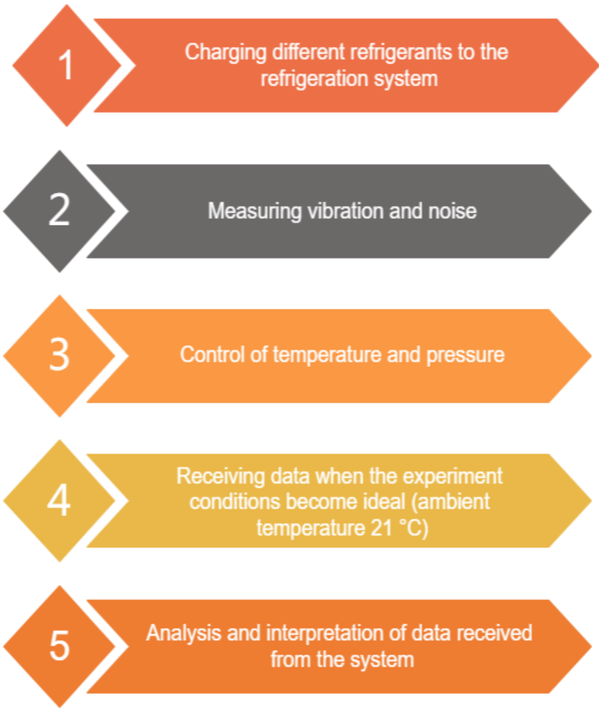



Figure 1. Flow chart for the experimental process

### *2.1 Refrigerants*

In this study, R22, R438A, R417A, R422D, and R422A refrigerants were used. R22 is a suitable refrigerant for a variety of refrigeration and air conditioning applications with a wide variety of temperature preferences. Due to this property, it has become the most common refrigerant used in many applications after the removal of chlorofluorocarbons [24]. R422D is preferred for R22 refrigerant changes in the direct evaporation component. In most systems, it shows similar efficiency and performance values to R22 [25]. R417A is used instead of R22 in direct expansion constant air conditioning and refrigeration systems in medium temperature [26]. R422A is used in some low-temperature commercial refrigeration applications as an alternative to R22. It provides lower pressure refrigerant temperatures compared to R22 and can extend the service life of the compressor. R438A, the other refrigerant used in the test system, is used as an alternative to R22 indirect evaporation systems [25]. R438A provides similar refrigeration performance and energy efficiency to R22 when operating at a lower compressor discharge temperature, similar evaporator, and condenser pressures. R22, direct expansion (DX) refrigeration and air conditioning systems were successfully realized renewal by using R438A as refrigerant [27]. The compositions and basic thermodynamic properties of the five refrigerants used in the experimental setup are given in detail in Table 1.

### *2.2 Experimental Setup*

The experimental system was designed based on the vapor compression refrigeration cycle. In the experimental system, a commercial type refrigeration system operating with R22 and alternatives R438A, R417A, R422D, and R422A refrigerants was manufactured. The experimental rig of refrigeration system is given in Figure 2. Each refrigerant to be tested was charged as 900 g, the compressor was purged with nitrogen before charging and vacuuming was performed. To be able to perform the experiments in a stable environment and to examine the performances of different refrigerants more realistically, the experiments were carried out in a closed environment with a constant outside temperature. To measure temperatures in the system, instant measurements were made with K-type thermocouples connected to the inputs and outputs of each main element of the system. Pressure measurements were carried out with pressure transmitters. All components and measurement devices used in the designed experimental system were given in Figure 3, and their technical specifications were given in Table 2.

Table 1. Environmental and physical properties of tested refrigerants [28]

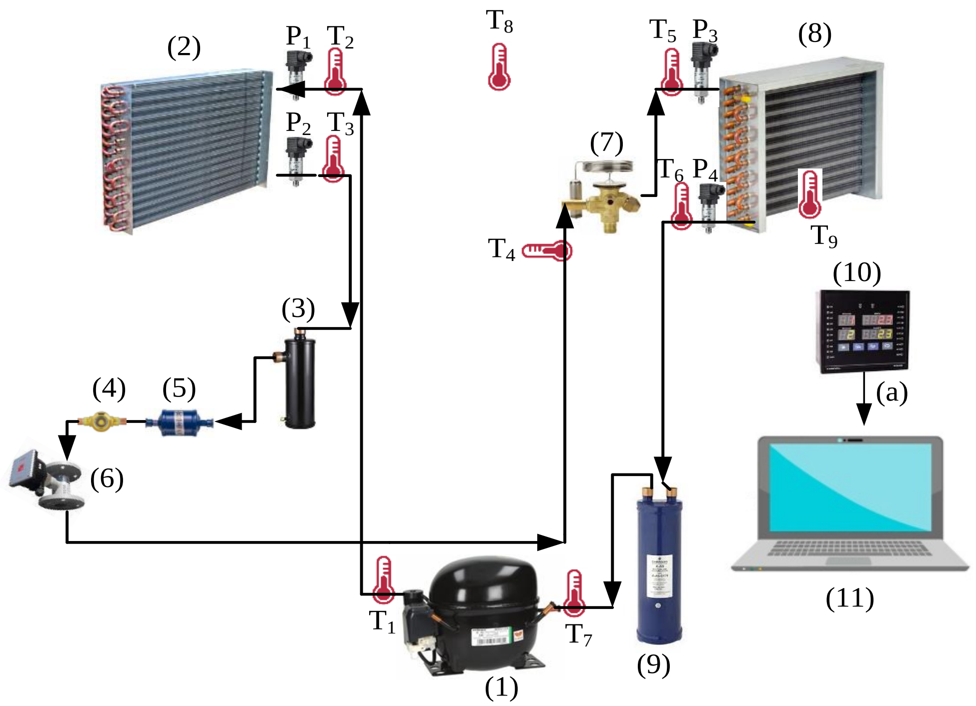
| Specification | R22 | R438A | | R417A | | R422D | | R422A | |
|---|---|---|---|---|---|---|---|---|---|
| Refrigerant Composition (Mass %) | R22 | R125 | 45% | R125 | 46.60% | R125 | 65.1% | R125 | 85.1% |
| | | R134a | 44.2% | R134a | 50% | R134a | 31.5% | R134a | 11.5% |
| | | R600 | 1.7% | R600 | 3.4% | R600a | 3.4% | R600a | 3.4% |
| | | R601a | 0.6% | | | | | | |
| | | R32 | 8.5% | | | | | | |
| Molecular Weight (kg/kmol) | 86.5 | 99.1 | | 106.8 | | 109.9 | | 116 | |
| Critical Temperature (ºC) | 96.2 | 85.3 | | 87.1 | | 79.6 | | 71.7 | |
| Critical Pressure (bar) | 49.9 | 43 | | 40.45 | | 39.18 | | 37.54 | |
| Oil | MO | MO/AB/POE | | MO/AB/POE | | MO/AB/POE | | MO/AB/POE | |
| ODP | 0.05 | 0 | | 0 | | 0 | | 0 | |
| GWP | 1810 | 2264 | | 2346 | | 2730 | | 3140 | |

MO: Mineral Oil AB: Alkaline benzene POE: Polyol ester

Figure 2. Experimental rig of refrigeration system

### *2.3 Vibration Measurement*

Accelerometer was placed at the top of the compressor and vibration data were taken from this point depending on time. First of all, the accelerometer was fixed with adhesive on the compressor and connected to the data acquisition device. Compressor vibration data were taken with a Bruel & Kjaer 4527 model piezoelectric accelerometer that can measure in three axes (x, y, z). The x-axis of the accelerometer indicates the direction perpendicular to the direction of movement of the piston in the working plane of the piston, the y-axis the direction in which the piston moves, and the z-axis in the normal directions of the working. The accelerometer was connected to four-channel VIBROTEST 80 model data acquisition devices capable of FFT analysis. The device has Bruel & Kjaer software and hardware system. Vibration signals up to 5 kHz were received for 20 seconds for each refrigerant test. Measurements were taken in the Hanning window at 6400 resolution over time. The characteristics of the triaxial accelerometer were given in Table 2.




| | | | | | |
|---|---|---|---|---|---|
| 1 | Compressor | 10 | Scanner ve Alarm Device | $T_3$ | Thermocouple (Condenser Outlet Temperature) |
| 2 | Condenser | 11 | Computer | $T_4$ | Thermocouple (TEX Inlet Temperature) |
| 3 | Liquid Receiver | $P_1$ | Condenser Inlet Pressure Sensor | $T_5$ | Thermocouple (Evaporator Inlet Temperature) |
| 4 | Sight Glass | $P_2$ | Condenser Outlet Pressure Sensor | $T_6$ | Thermocouple (Evaporator Outlet Temperature) |
| 5 | Drier-Filter | $P_3$ | Evaporator Inlet Pressure Sensor | $T_7$ | Thermocouple (Compressor Inlet Temperature) |
| 6 | Flowmeter | $P_4$ | Evaporator Outlet Pressure Sensor | $T_8$ | Thermocouple (Ambient Temperature) |
| 7 | Thermostatic Expansion Valve | a | Data Transfer | $T_9$ | Thermocouple (Cooling Cabin Temperature) |
| 8 | Evaporator | $T_1$ | Thermocouple (Compressor Outlet Temperature) | | |
| 9 | Accumulator | $T_2$ | Thermocouple (Condenser Inlet Temperature) | | |

Figure 3. All components and measuring equipment used in the experimental system

Table 2. Specifications of components and sensors in the experimental system

| Component and Sensors | Voltage | Current | Technical Specifications |
|---|---|---|---|
| Compressor | 220-240V | | Embraco NEU 6215 GK - Hermetic reciprocating, Refrigerant: R22, Power: 1/2 Hp |
| Condenser | 220-240V | | Karyer KT- with airflow, 3/4 Hp |
| Evaporator | 220-240V | | Karyer KT- with airflow, 1/2 Hp |
| Expansion Valve | | | DuNan TEX TIS externally balanced union, One way<br>PS 46 bar, Refrigerant: R22, Operating range: -40/+10 °C |
| Liquid Receiver | | | ESS-LRY, Pressure: 32 bar, Operating range: 0/+70 °C, Volume: 1.1 L |
| Accumulator | | | SAN XIN SX-204 |
| Drier-Filter | | | Sanhua DTG, Welded, Operating range: -40/+120 °C |
| Sight Glass | | | SARCOOL Welded |
| Converter | | | Ordel- SBA200- USB-RS485, USB 1.1 ve 2.0 compatible PC connection<br>300…3M Baud Communication speeds support |
| Process Monitoring, Recording and Control Program | | | Ordel OPİK16<br>Variables can be monitored, changed, and saved on the computer, 128 channels can be defined to one parameter each, Each channel can be viewed as digital, analog, and graphic, Records can be kept in Microsoft Database (mdb) format., These can be opened with Microsoft ACCESS or Microsoft EXCEL programs |
| Universal Input Scanner and Alarm Device | 100-240V AC/DC | | Ordel SCN100-03/0/2/0/11, Number of entries: 20, 2 Piece 4 Digit Numeric and 2 Piece 2 Digit Numeric Display, 22 LED Displays<br>Universal Sensor Input (TC, RT, mA, mV, V), Accuracy: ±0.2% |
| Gas Filling Device | 220V | | Gas charging station with digital scales<br>Vacuum pumped (3 $m^3$/h), Balance sensitivity: ±5 gram<br>Operating temperature: 8- 49 °C<br>Pressure Range: 15- 30 bar |
| Flowmeter | 12-24V DC | 4-20 mA | Bass- TDSS.004.015.D.A.10.S.S.N.N<br>Measuring range: 0.6... 4.5 l/dk, Accuracy: 1% T.S.<br>Temperature: -40...120°C, Pressure: 63 bar max. |
| Pressure Transmitter | 8-28V | 4-20 mA | Keller PA-21Y, for refrigeration groups<br>Pressure range: 0-30 bar<br>Accuracy: 1%, Operating range: -20....+85°C |
| Thermocouple | | | Ordel KTTE 2x0.5T 2K- K Type<br>Cross-section: 2 x 0.5 mm² Single wire, welded end<br>Insulation: Teflon + Teflon, Cable length: 2 m |
| Thermostat | 230V AC | | EVCO- EVKB 21- Digital Hold with defrost<br>NTC: -40/+105 °C, Relay Output: 1 piece16 A, Digital Input: 1 |
| Accelerometer | | | Three axis, Brüel&Kjaer 4527, Frequency range: 0.3-10000 Hz, Accuracy: 3 mV/g, Temperature range: -30/180°C, Resonance frequency: 30 kHz |
| Noise Measuring Device | | | Svantek SV 104, Filters: A, C, Z Measuring range: 55 dbA RMS/ 140.1 dbA peak, Frequency range: 30 kHz-8 kHz, Dynamic range: 95 dB |
| Data acquisition device | | | Vibrotest 80, FFT analysis can be done, four-channel, Brüel&Kjaer 4527 program |

### *2.4 Noise Measurement*

Noise is one of the most important problems in home and work life, which occurs in all air conditioning and refrigeration systems and is ignored. The dosimeter is used to determine the level of exposure to noise of people in an environment with noisy devices. The dosimeter also detects the level where the noise is at its highest level. In this study, compressor noise was measured with a Svantek 104 model dosimeter shown in Figure 4. For this, the measuring device is fixed very close to the compressor. The process of obtaining vibration and noise data from the test system is shown in Figure 5.

### *2.5 Uncertainty Analysis*

The total uncertainty value of the experimental study was calculated by the following equations. In the equation, $W_R$ is the total uncertainty of the study (%), $R$ and $w$ are the uncertainty function and the dimensional factor, respectively. In the same equation, $W_n$ expresses the uncertainties in the independent variables [29, 30].

$$W_R = \left[\left(\frac{\partial R}{\partial x_1} w_1\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} w_n\right)^2\right]^{1/2} \quad (1)$$

Figure 4. Noise measuring device

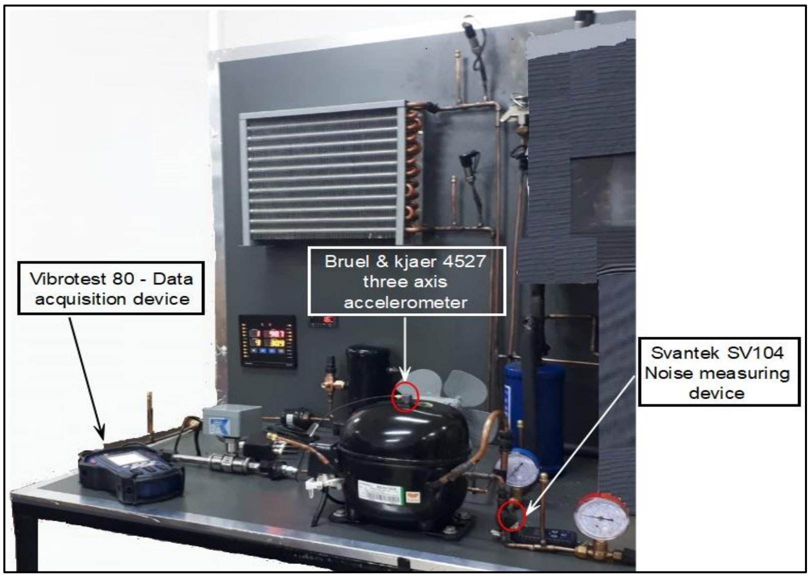



Figure 5. Vibration and noise measurement process from the experimental system

## 3. Results and Discussion

The uncertainty was calculated as 1.5% for the COP and the noise value as 2%. In this study, the variation of noise and vibration characteristics depending on the compressor load was experimentally investigated. Experiments were done with R22 and alternative refrigerants. 16384 data were taken for each measurement. When the compressor is loaded with different refrigerants, to compare the vibration amplitude values of the compressor, the root mean square (RMS) of the data taken from each axis has been calculated according to Equation (2) [31].

$$a_{RMS} = \sqrt{1/N \sum_{k=1}^{N} a_k^2} \tag{2}$$

Here; $a_{RMS}$: Root mean square of acceleration signals (g), $a_k$: k$^{th}$ acceleration value for time-domain signals, N: Total acceleration value (N:16384).

The results of the data obtained from x, y, and z axes for the compressor used different refrigerants in the loaded position, arranged according to Equation (2), are given in Table 3. As can be seen from the measurements, especially when R22 and alternative refrigerants are used in the regime, the compressor RMS values in all axes were obtained very close to each other.

As seen in Table 3, for all refrigerants when the compressor is operating, the smallest RMS value on the x-axis is obtained with R422D, while the highest RMS value is obtained in R22. While the smallest RMS value on the y-axis was obtained with R417A, the largest was obtained with R422A. While the smallest RMS value on the z-axis is obtained with R22, the largest was obtained with R422D.

Using the data in Table 3, average vibration values ($a_t$) in all axes (x, y, z) were calculated according to Equation 3.

$$a_t = \sqrt{a_{(x)}^2 + a_{(y)}^2 + a_{(z)}^2} \tag{3}$$

Here; $a_t$= Average vibration value, $a_{(x)}$= RMS value in x direction, $a_{(y)}$= RMS value in y-direction, $a_{(z)}$= RMS value in z-direction.

Average vibration values ($\boldsymbol{a_t}$) for all refrigerants were given in Figure 6. Average noise values were given for all refrigerants in Figure 7. While the highest average vibration values were obtained with R422A, the smallest average vibration value was obtained with R417A. This situation depends on the lowest COP value of R422A. The low COP value of R422A caused the compressor to operate at higher speeds, increasing the average vibration and noise values.

In Figure 7, the noise values of alternative refrigerants increase, respectively, R417A, R438A, R422D, and R422A when compared with R22 refrigerant. The noise values obtained as a result of the measurement are 99% similar for R417A, 97% for R438A, 96% for R422D, and 94% for R422A according to the refrigerant R22.

Table 3. RMS values for different refrigerants

| Axis | | Refrigerants | | | | |
|---|---|---|---|---|---|---|
| | | R22 | R438A | R417A | R422D | R422A |
| Compressor | (x), (CH1) | 0.146219 | 0.137383 | 0.117344 | 0.097732 | 0.110746 |
| | (y), (CH2) | 0.110678 | 0.1146197 | 0.107061 | 0.154545 | 0.185198 |
| | (z), (CH3) | 0.108183 | 0.111886 | 0.128004 | 0.136984 | 0.111478 |

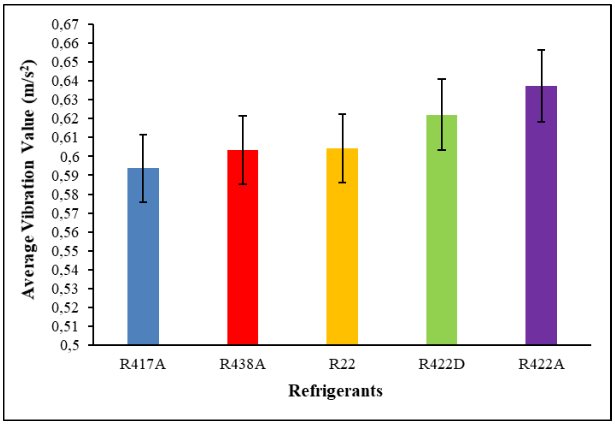



Figure 6. Average vibration values for different refrigerants

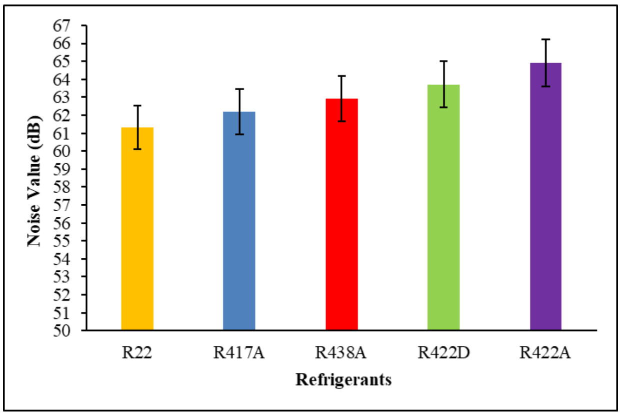



Figure 7. Change of noise according to different refrigerants

As seen in Table 1, the fact that refrigerants R438A, R417A, R422D, and R422A, which are preferred as alternatives to R22, have low hydrocarbon content by weight, keeps these mixtures at low flammability and makes them compatible with MO and AB. Besides, the used refrigerants have a similar safety level (A1) with R22. In most cases, the fact that there is no need to change the type of lubricant and the existing component during renewal and replacement shows that there are no major differences in the COP, noise, and vibration values of the used refrigerants.

One of the main reasons for the noise and vibration in the compressor is the cavitation level inside the pump. The higher the flow rate, the higher the cavitation level. Under the cavitation process, vapor bubbles increase due to the lower fluid pressure than vapor pressure. Increasing pressure fluctuations due to cavitation in the compressor cause an increase in vibration and noise. For this reason, the pressure value of the fluid in the suction and discharge line is important [32-35]. As seen in Figure 6, R417A with lower suction pressure has the lowest average vibration value.

When the experimental results of the refrigerants in Table 4 were examined, it was seen that the cooling capacity, operating pressures, discharge temperatures, and power consumption of the refrigerants used were close to each other in terms of vibration and noise values as seen in Figure 6 and Figure 7. COP values of different refrigerants are given in Figure 8.

COP is a very important indicator of refrigeration system performance. The higher the COP value, the higher the efficiency of the refrigeration system. COP values were found to be 4.07 for R22, 3.88 for R438A, 3.63 for R417A, 3.37 for R422D, and 3.18 for R422A. As seen in Figure 8, it showed similarity 95% for R438A, 89% for R417A, 83% for R422D, and 78% for R422A compared to R22. COP values showed that none of the refrigerants selected were as efficient as R22. However, their COP values revealed that each would be considered as a potential alternative for empowerment and change. While it has comparable COP with other refrigerants, the low cooling capacity of R417A makes it less attractive. R422D and R422A refrigerants with lower COP are seen as the least alternative to R22 refrigerant. R438A refrigerant was determined to be the best alternative to R22. The relationship between noise and vibration values of different refrigerants and COP is given in Figure 9.

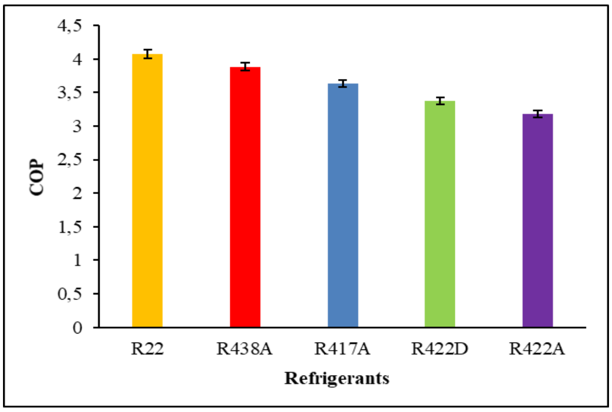



Figure 8. Change of COP for different refrigerants

Table 4. Experimental results of refrigerants

| Refrigerants | COP | Cooling Capacity (kW) | Electricity Consumption (kWh) | Evaporation Pressure $P_1$ (bar) | Condensation Pressure $P_2$ (bar) | Discharge Temperature (°C) |
|---|---|---|---|---|---|---|
| R22 | 4.07 | 1.12 | 0.32 | 1.94 | 12.49 | 91.5 |
| R438A | 3.88 | 0.94 | 0.28 | 1.88 | 12.14 | 73.4 |
| R417A | 3.63 | 0.85 | 0.27 | 1.81 | 10.93 | 71 |
| R422D | 3.37 | 0.76 | 0.26 | 1.92 | 12.57 | 70.8 |
| R422A | 3.18 | 0.69 | 0.25 | 1.97 | 14.22 | 70.2 |

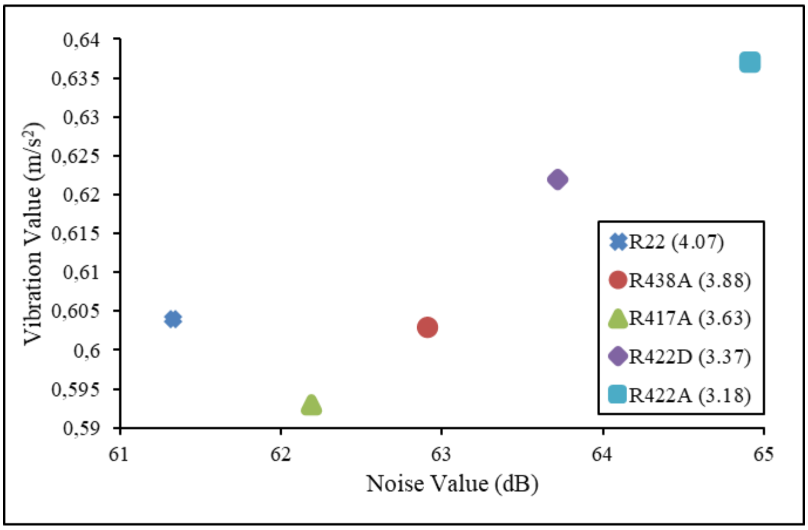



Figure 9. COP, vibration, and noise relationship graph for refrigerant

As seen in Figure 9, the relationship with the COP of the refrigerants between vibration and noise values is seen. R22 refrigerant with the highest COP value has the least vibration and noise. It is followed by R438A, R417A, R422D, and R422A refrigerants. It was observed that the vibration and noise values of the refrigerants are compatible with the COP.

## 4. Conclusions

The most important design parameter to reduce the noise and vibration emitted by the compressors used in refrigeration and air conditioning systems can be by reducing the forces transmitted from the compressor to the casing. In this study, the dynamic behavior of the reciprocating and sealed compressors, which form the backbone of the systems used for cooling purposes and which are mostly used in these systems, have been investigated. Thus, it has been possible to examine the dynamic behavior of compressors in the starting and regime states.

In this study, in which vibration and noise variation depending on the load were examined in a reciprocating sealed compressor, the average vibration and noise values of the compressor were determined by analyzing the data obtained from the x, y, and z axes. It was observed that the compressor noise values and vibration values vary depending on the type of refrigerant used. R417A, which has a 99% similar noise value to R22, was determined as an alternative refrigerant. R438A showing 95% similarity for COP can be used as an alternative to R22. It was observed that the noise and vibration values are very close to each other for all refrigerants examined. These close values obtained are also close in COP. This proximity showed that R22 and its alternative refrigerants can be used interchangeably without any changes in the system. Compressor vibrations and noise are partially affected by the refrigerant change in the system. This study showed that the refrigerants used in the experiments can be used as an alternative to R22 in terms of vibration and noise in the refrigeration system.

Hermetic reciprocating compressors are the type of compressor that is widely used almost everywhere in the air conditioning and refrigeration industry. In this study, experiments were conducted by charging five different refrigerant systems using the vibration and noise analysis method, and vibration and noise data were obtained from the experiments. It is thought that the study will contribute to the knowledge and literature of those dealing with the detection of imbalance and other problems, vibration and noise analysis caused by refrigerants in hermetic reciprocating compressors used in air conditioning and refrigeration systems. Maintaining these compressors, which serve as the most effective parts of air conditioning and refrigeration processes, ensures that they maintain their performance and efficiency. It is very important to predict the malfunctions that may occur in the compressors, to keep production down and maintenance costs to a minimum. At the same time, whether there is a difference in terms of the refrigerants used shows the importance of the study. As a result of the study, it was seen that vibration and noise analysis of the hermetic reciprocating compressor is an important method in detecting malfunctions that may occur. Considering the studies in the literature, it is foreseen that vibrations and noises can be analyzed with a smart system without the need for expert interpretation, and malfunctions or maintenance processes can be predicted with a smart decision support system. In future studies, it is aimed to develop a modern fault detection, diagnosis, and maintenance system using machine learning methods and to work on increasing the performance of the system.

## Declaration

The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article. The authors also declared that this article is original, was prepared in accordance with international publication and research ethics, and ethical committee permission or any special permission is not required.

## Author Contributions

G. Yıldız developed the methodology. S. Sarıdemir and Y. Çay performed the analysis. Z. Cingiz and F. Katırcıoğlu supervised and improved the study.

## Acknowledgment

This study was supported by Düzce University Scientific Research Projects (Project no: 2019.21.02.983), Turkey.

## Nomenclature

| | |
|---|---|
| $a_t$ | : Average vibration value |
| $a_{(x)}$ | : RMS value in x-direction |
| $a_{(y)}$ | : RMS value in y-direction |
| $a_{(z)}$ | : RMS value in z-direction |
| $a_{RMS}$ | : Root mean square of acceleration signals |
| $a_k$ | : kth acceleration value for time-domain signals |
| *AB* | : Alkaline benzene |
| *AC* | : Alternating current |
| *CH* | : Channel |
| *COP* | : Coefficient of performance |
| *DC* | : Direct current |
| *DX* | : Direct expansion |
| *FFT* | : Fast Fourier transform |
| *GWP* | : Global warming potential |
| *HVAC* | : Heating, ventilation, and air conditioning system |
| *MO* | : Mineral oil |
| *N* | : Total acceleration value |
| *ODP* | : Mineral oil |
| *POE* | : Polyol ester |
| *R* | : Uncertainty function |
| *RMS* | : Root mean square |
| *TEX* | : Thermostatic expansion valve |
| *V* | : Volt |
| *w* | : Dimensional factor |
| $W_R$ | : Total uncertainty |
| $W_n$ | : Uncertainties in the independent variables |

## References

1. Ceylan, İ., G. Yıldız, A. E. Gürel, A. Ergün, and A. Tosun, *The effect of malfunctions in air handling units on energy and exergy efficiency*. Heat Transfer Research, 2020. **51**(11): 1007-1028..
2. United Nations Development Programme UNDP. "20 years of success: Montreal protocol on substances that deplete the ozone layer.," [cited 2020 29 October]; Available from: http://www un.org. kh/undp/knowledge/publications/20-years-of-success-montreal-protocol-on-substances-that deplete-the-ozone-layer-2.
3. Li, W. L., and V. Eyo, *Dynamic Analysis of a Compressor Mounting System*. International Compressor Engineering Conference, 1996.
4. Hamilton, J. F., Measurement and control of compressor noise, West Lafayette, Purdue University, USA, 1988.
5. Celik, S., and E. C. Nsofor, *Studies on the flow-induced noise at the evaporator of a refrigerating system*. Applied Thermal Engineering, 2011. **31**(14-15): p. 2485-2493.
6. Lee, C., Y. Cho, B. Baek, S. Lee, D. Hwang, and K. Jo, *Analyses of refrigerator noises*. In Proceedings of the IEEE International Symposium on Industrial Electronics, 2005: p. 1179-1184.
7. Han, H. S., W. B. Jeong, M. S. Kim, and T. H. Kim, *Analysis of the root causes of refrigerant-induced noise in refrigerators*. Journal of Mechanical Science and Technology, 2009. **23**(12): p. 3245-3256.
8. Han, H. S., W. B. Jeong, M. S. Kim, S. Y. Lee, and M. Y. Seo, *Reduction of the refrigerant-induced noise from the evaporator-inlet pipe in a refrigerator*. International Journal of Refrigeration, 2010. **33**(7): p. 1478-1488.
9. Jang, S., Choung, H., Park, S., and Lee, S. *Investigation on noise of rotary compressors using fluid-structure interaction*. Journal of Mechanical Science and Technology, 2019. **33**(11): p. 5129-5135.
10. Venkatappa, S., M. Koberstein, and Z. Liu, *NVH Challenges with Introduction of New Refrigerant HFO-1234yf*. SAE Technical Paper, 2017. (2017-01-0172)
11. Xia, Y., Y. Liu, Y. Liu, Y. Ma, C. Xiao, and T. Wu, *Experimental study on reducing the noise of horizontal household freezers*. Applied Thermal Engineering, 2014. **68**(1-2): p. 107-114.
12. Jeon, J. Y., J. You, and H. Y. Chang, *Sound radiation and sound quality characteristics of refrigerator noise in real living environments*. Applied Acoustics, 2007. **68**(10): p. 1118-1134.
13. Berry, B. F., *The work of I-INCE Technical Study Group 2 on noise labels for consumer and industrial products*. Noise and Health, 2003. **5**(18): p. 21.
14. Kim, K. M., *Study of efficiency and noise improvement in a reciprocating compressor for refrigerator*. The Magazine of the Society of Air-Conditioning and Refrigerating Engineers of Korea, 1995. **24**(1): p. 71-81.
15. Yoo, W. H., C. G. Back, and J. H. Song, *The search for relations between compressor noise and refrigerator noise*, in Korean Society of Noise and Vibration Engineering, Autumn Conference. 1996: p. 32-36.
16. Seo, S. H., T. H. Kwak, C. J. Kim, J. K. Park, and K. S. Cho, *Noise and vibration reduction of a household refrigerator*. Proceedings of the HVAC Refrigeration Engineering, 2000. p. 1133-1140.
17. Lee, J. K., Y. W. Park, and J. B. Chai, *Development of a sound quality index for the evaluation of an intake noise of a passenger car*. Transactions of the Korean Society for Noise and Vibration Engineering, 2005. **15**(8): p. 939-944.
18. Khan, M. S., and C. Hogstrom, *Determination of sound quality of HVAC systems on trains using multivariate analysis*. Noise Control Engineering Journal, 2001. **49**(6): p. 276-283.
19. Khan, M. S., and C. Dickson, *Evaluation of sound quality of wheel loaders using a human subject for binaural recording*. Noise Control Engineering Journal, 2002. **50**(4): p. 117-126.
20. Oh, H. E., D. J. Park, and W. B. Jeong, *Numerical and experimental study on the reduction of refrigerant pressure pulsation within compressor pipes*. Journal of Sound and Vibration, 2019. **438**: p. 506-519.
21. Park, J., and S. Wang, *Noise reduction for compressors by modes control using topology optimization of eigenvalue*. Journal of Sound and Vibration, 2008. **315**(4-5), p. 836-848.

22. Birajdar, R., R. Patil, and K. Khanzode, *Vibration and noise in centrifugal pumps-Sources and diagnosis methods*, in 3rd International Conference on Integrity, Reliability, and Failure, 2009: p. 20-24.
23. Kim, B. L., J. Y. Jung, and I. K. Oh, *Modified transfer path analysis considering transmissibility functions for accurate estimation of vibration source*. Journal of Sound and Vibration, 2017. **398**: p. 70-83.
24. Linde. *Linde gas*. [Accessed 26 May, 2020]; Available from: https://www.linde-gas.com/en/products_and supply/refrigerants/h cfc_refrigerants/r22/index.html.
25. Bock Compressor. *Bock Compressor, Alternative Refrigerants Information on use of R2*. [Accessed 16 April, 2020]; Available from: http://www.bock.de/media/files/PDF/Produktinformationen/96151_ Alternative-refrigerants_R22_Gb.pdf..
26. Climalife IDS Refrigeration Limited. *Climalife IDS Refrigeration Limited*. [Accessed 15 April, 2020]; Available from: https://www.climalife.co.uk/docs/ISCEON-MO29Retrofit-Guidelines-V2.pdf.
27. Allgood, C. C., and C. C. Lawson, *Performance of R-438A in R-22 refrigeration and air conditioning systems*, in International Refrigeration and Air Conditioning Conference: 2010, USA.
28. Cingiz, Z., F. Katırcıoğlu, Y. Çay, and A. Kolip, *Buhar Sıkıştırmalı Soğutma Sisteminde R22 Alternatifi Soğutucu Akışkanların Termodinamik Analizi*. Politeknik Dergisi, 2020. **23**(4): p. 1205-1212.
29. Ağbulut, Ü., A. E. Gürel, and S. Sarıdemir, *Experimental investigation and prediction of performance and emission responses of a CI engine fuelled with different metal-oxide based nanoparticles–diesel blends using different machine learning algorithms*. Energy, 2021. **215**: 119076.
30. Sarıdemir, S., G. Yıldız, and H. Hanedar, *Effect of diesel-biodiesel-methanol blends on performance and combustion characteristics of iesel engine*. Düzce Üniversitesi Bilim ve Teknoloji Dergisi, 2021. **9**(1): p. 189-201.
31. Ağbulut, Ü., M. Karagöz, S. Sarıdemir, and A. Öztürk, *Impact of various metal-oxide based nanoparticles and biodiesel blends on the combustion, performance, emission, vibration and noise characteristics of a CI engine*. Fuel, 2020. **270**: p. 117521.
32. Al-Obaidi, A. R., *Experimental comparative investigations to evaluate cavitation conditions within a centrifugal pump based on vibration and acoustic analyses techniques*. Archives of Acoustics, 2020. **45**(3): p. 541-556.
33. Al-Obaidi, A. R., and R. Mishra, *Experimental investigation of the effect of air injection on performance and detection of cavitation in the centrifugal pump based on vibration technique*. Arabian Journal for Science and Engineering, 2020. **45**(7): p. 5657-5671.
34. Al-Obaidi, A. R., *Detection of cavitation phenomenon within a centrifugal pump based on vibration analysis technique in both time and frequency domains.* Experimental Techniques, 2020. **44**(3): p. 329-347.
35. Al-Obaidi, A. R., and H. Towsyfyan, *An experimental study on vibration signatures for detecting incipient cavitation in centrifugal pumps based on envelope spectrum analysis*. Journal of Applied Fluid Mechanics, 2019. **12**(6): p. 2057-2067.