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Theoretical and Experimental Comparison of Revolving Vane Compressors and Rolling Piston Compressors

Duygu GURKAN KOCATAS[1,*], Melih OKUR[2], İhsan KORKUT[1]

[1] *Gazi University, Faculty of Technology, Department of Manufacturing Engineering, Ankara, Turkey*
[2] *Gazi University, Faculty of Technology, Department of Automative Engineering, Ankara, Turkey*

Highlights

- This paper focuses on different rotary compressor design.
- The two designs were compared with each other theoretically and experimentally.
- The advantages and disadvantages of two compressor designs are evaluated.

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Keywords

Rotary compressors, Revolving vane compressor, Rolling piston compressor, Double stage compressor

Abstract

Rotary compressors have some advantages such as lower noise, vibration, lower cost and few parts. For this reason, it is preferred more than its substitutes such as piston, screw and sliding compressors in cooling and air-conditioning systems. In this study, two different designs, rolling piston and revolving vane, were made to increase the performance of the double-stage, rotary compressor with a volume of 22.6 cc. When the test results are examined, it is seen that the rolling piston compressor consumes 1.3 times less power than the electric motor for 2700 rpm and 3 bar pressure, due to the lower inertia forces compared to the revolving vane compressor.

1. INTRODUCTION

Energy is of great importance for countries' social, economic and technological progress and energy need is one of the the most significant problems of this century. [1-2]. With the increase of the world population and the need for energy in technology, energy consumption has increased dramatically worldwide, especially in developing countries (such as China, Turkey and India) [3-6]. For this reason; developed countries are looking for new energy sources to solve the energy problem. [7]. In electrical energy consumption, compressed air systems rank 3rd in many industrial plants. Compressed air demand is 9.4% of industrial electrical energy consumption in China, and it constitutes 10% of the USA and Europe [8]. The world calls for continuous improvement in heating, ventilation and air conditioning (HVAC) systems mainly in terms of greater system capacity, thermal efficiency and reduced power usage [9]. Air compressor, which is the heart of the compressed air systems, work with the principle of loading potential energy and bringing it to the desired pressure by shrinking the air intake volumetrically at ambient pressure. Recent studies have focused on rotary compressors due to the difficulty in manufacturing and the maintenance of piston compressors and screw compressors. In 2014, more than 100 million rotary compressors were produced in China [10]. Combining their lightness and low cost with high performance, rotary compressors are becoming more and more popular than piston and screw compressors in air conditioning and cooling systems that require low capacity [11]. Rotary compressors consist mainly of cylinders, rotors, vane and eccentric shafts. Study volume; the outer surface of the rotor is limited by the inner surface of the cylinder and the vane. The vane is spring-loaded to the cylinder, and the other end of the vane slides as a hinge in the vane cavity located inside the rotor. By the rotational movement of the cylinder or eccentric shaft; the volumes trapped between

*Corresponding author, e-mail: duygugurkan@gazi.edu.tr

the rotor, vane and cylinder are changing and the working fluid performs the suction, compression and discharges the processes [12]. Rotary compressors have advantages such as lightness, compact design, easy balancing, less vibration, less moving parts and low production costs. However, there are also disadvantages such as sealing, leakage, friction and inertia problems deployed and from other algorithms. While some of the literature studies focus on the mechanical friction problem encountered in the vane [13-20] most of them have turned to new designs in order to decrease the pressure loss and increase the compressor efficiency [21-26].

Okur and Akmandor have replaced the vane mechanism driven by the spring in traditional compressors with the hinge vane mechanism. Thus, they obtained better results than the pallet driven with a spring, especially at high speeds and pressures. The isentropic compression efficiency of the hinged vane compressor is around 85% for pressures reaching 9 atm [27]. Sung, Lee and Oh applied TiN coating on the vane surfaces in order to prevent abrasion the vane. As a result of the experiments, they observed that the TiN coating increased the abrasion resistance compared to the uncoated vane [28,29]. Shuxue and Guoyuan experimentally compared single-stage and double-stage rotary compressors. They observed that the double-stage compression system improved the cooling capacity and the COP value in the range of 5% -15% and 10% -12%, respectively, compared to the single-stage system [30]. Ba et al., created a three-dimensional dynamic numerical model with the CFD method to analyze the axial force in the rotor system of a rotary compressor. The CFD analysis performed showed consistency with the results of the experimental study. They also emphasized that the space between cylinder and rotor is important in compressor performance [31]. Cai and his colleagues examined the leakage problem and volumetric efficiency of the rolling piston compressor. They found that with reduced rotation speed, volumetric efficiency decreases and smaller radial spaces increased the compressor efficiency [3]. Shin et al., used the inner cavity of the cylinder as an additional working volume in the rolling piston compressor to increase cooling capacity and mechanical efficiency. In this way, cooling capacity has been increased by 31.28-37.99%. In addition, with the structure where the vane is fixed between the cylinder and the rotor, the mechanical loss in the vane has decreased [32].

In order to prevent friction between the rotor and the cylinder in rotary compressors, Ooi et al. made a design that rotates the pallet by fixing it to the cylinder instead of rotating the rotor. With this design, they aimed to prevent torque fluctuations, vibration, mechanical friction, leakage loss in the radial space and increase the compressor performance [33-36]. The prototype of the revolving vane compressor, whose theoretical analysis gave positive results, has been tested. The difference between the theoretical results and the results obtained from the experimental study was found under 10% [37,38]. Hu et al. used bearings in a revolving vane compressor that can convert the friction between the end of the vane and the inner wall of the cylinder into rolling friction. They observed that the bearing causes extra power loss, but the overall power consumption can be reduced by 170.17W compared to the conventional revolving vane compressor [39]. Ooi et al. have made a new design in order to reduce the compressor dimensions and make the compressor more compact [40-43]. They carried out studies on increasing the performance of the new type of compressor [44] and compared the new design with the conventional rolling piston type rotary compressor [45]. They also designed a multi-vane, revolving vane type rotary compressor where high cooling capacity and low vibration are desired [46,47]. Based on the need to evaluate technological developments, they also carried out a compilation study on sliding vane and rolling piston type rotary compressors [48]. When the literature studies are examined, it is seen that the studies on the rotary compressor are generally on the rolling piston type compressors, limited study has been done in revolving vane type compressors, but they are not compared with the revolving vane compressor.

In this study, two different designs of a 22.6 cc double-stage rotary compressor as revolving vane and rolling piston were made and compared theoretically and experimentally. The most important feature of this study and its difference from other studies is that two different designs can be evaluated in terms of efficiency and performance on a single compressor system. In the last section, results are provided and references to further studies are given.

2. MATERIAL METHOD

2.1. Compressor Design

Compressor generally consists of cylinder, rotor, and vane. The vane is rigidly fixed to cylinder and separates the pressurized and non-pressurized volumes. In this study, the inlet and outlet channels of the designed compressor change according to the design (rolling piston and revolving vane). Figure 1 shows the compressor parts and Table 1 shows the compressor part dimension.

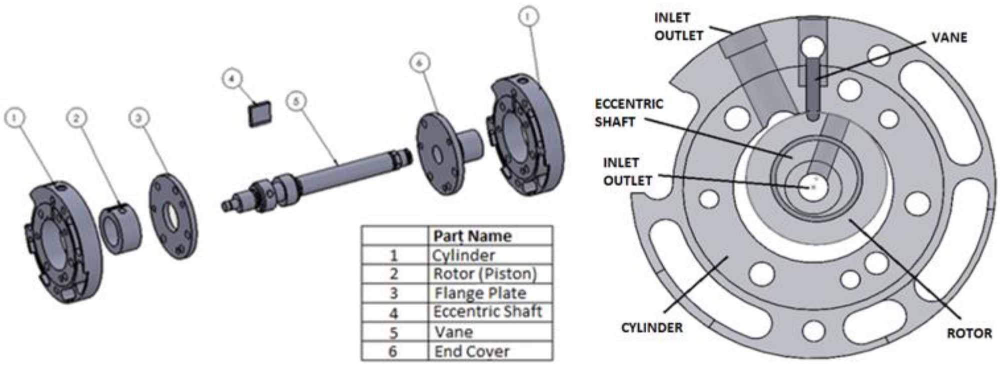


Figure 1. *Compressor parts*

Table 1. *Compressor part dimension*

Cylinder inner diameter (mm)	Rolling piston(rotor) outer diameter (mm)	Rolling piston (rotor) thickness (mm)	Cylindrical journal diameter (mm)	Vane thickness (mm)
53.5	47	22	29.5	4.7

In this study; pressure, flow and power values of rotary compressor for two different designs (rolling piston compressors and revolving vane compressors) were theoretically and experimentally compared. In the first design, with the general use in the industry, the eccentric shaft was rotated by keeping the cylinder fixed. In the second design, in order to reduce the pressure leaks and friction losses, eccentric shaft is kept fixed and the cylinder and vane are rotated. The parameters such as compressor size and pressure values were kept the same for both designs, then the rotation speed and torque values were compared. Figure 2 shows the rolling piston and revolving vane compressor structures.

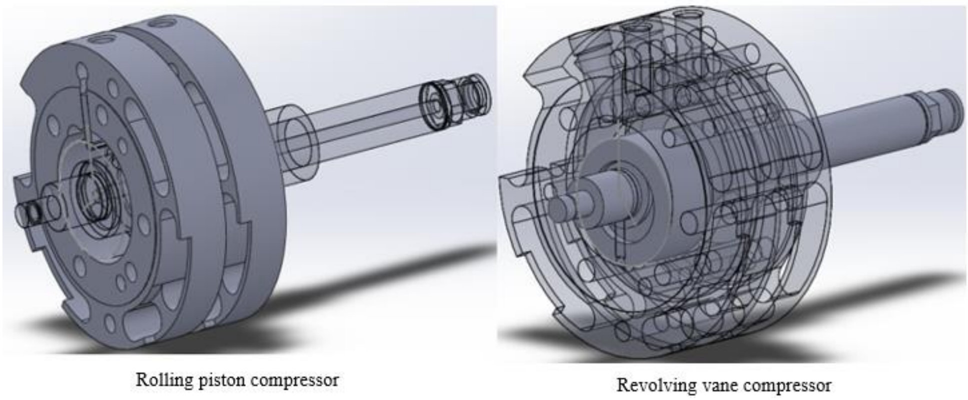


Figure 2. *General structure of rolling piston compressor and revolving vane compressor*

As it can be seen in Figure 2, in this study, a double chamber, rotary compressor with a volume of 22.6 cc, which is widely used in the industry for cooling purposes, was compared for two different designs.

Theoretical calculations and experiments were made by changing only the positions of the inlet and outlet channels on the same compressor. For both designs, the moving parts are shown in solid and the fixed parts in transparent (Figure 2). In the first design, the rolling piston compressor, the outer cylinder is kept fixed while the eccentric shaft rotates. Air is sucked from the center of the compressor and sent to the pressurized chamber through a special channel and hole on the eccentric shaft and rotor. Figure 3 shows the working principle of the rolling piston compressor.

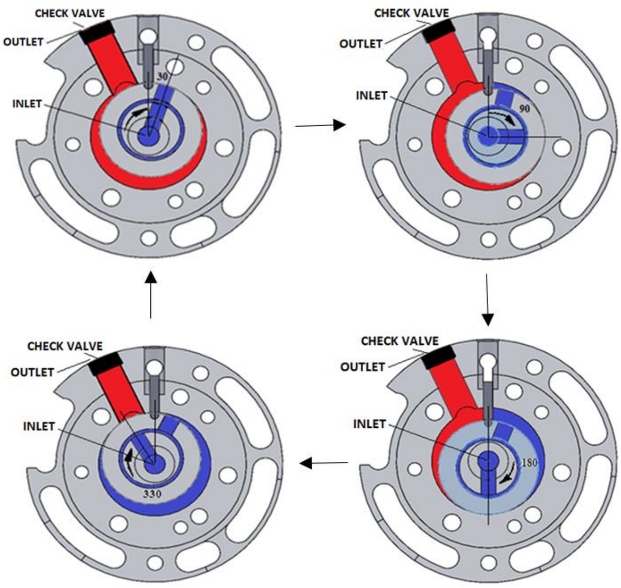


Figure 3. *Working principle of rolling piston compressors*

In the rolling piston compressor design shown in Figure 3, the areas seen in blue represent suction and the areas seen in red represent the compression areas. Due to the structure of the inlet and outlet sizes, the compressor does not perform suction and compression at the first 30 and the last 30 degrees. For this reason, the working areas are determined between 30 °-330 °.

In the second design, the structure of revolving vane compressor, is exactly the same as the rolling piston compressor. In this design, differently, the eccentric shaft is kept fixed and the vane is rotated with a cylinder. Air is sucked from the hole on the cylinder, compressed with the help of a rotor and sent to the pressure tank through the hole in the center. Figure 4 shows the working principle of the revolving vane compressor.

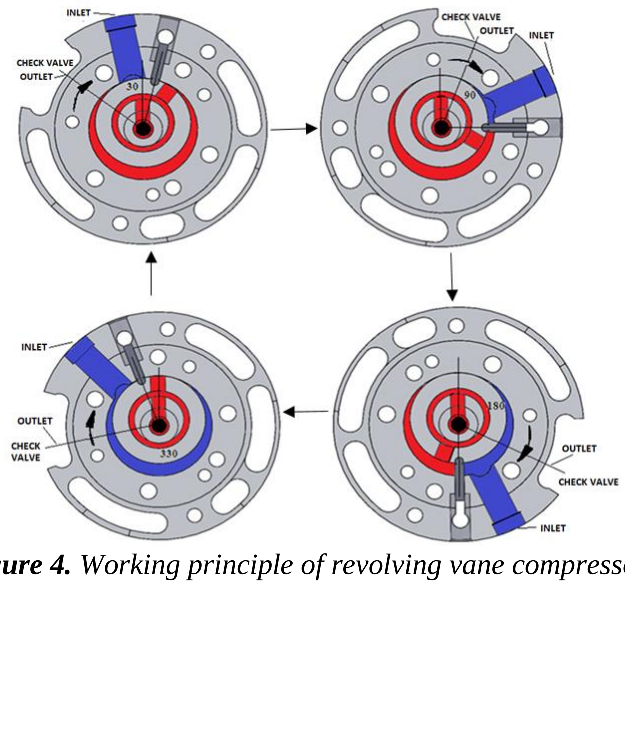
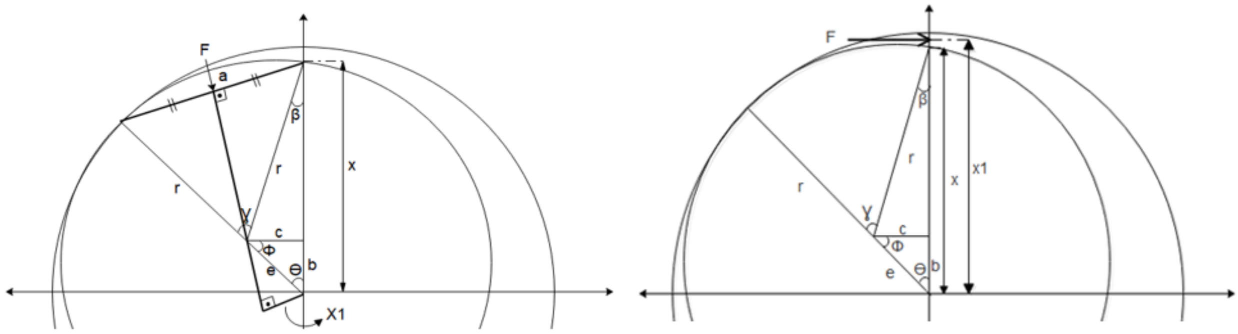


Figure 4. *Working principle of revolving vane compressors*

2.2. Design Calculations (Theoretical Work)

Design calculations were made at 30 ° intervals between 30-330º rotor angle for both designed compressors. While calculating the torque and power values of the compressors, the pressure values were kept constant as 0.5-1-1.5-2-2.5-3 bar. Figure 5 shows the calculation parameters for rolling piston and revolving vane compressors.



Rolling piston compressor   Revolving vane compressor

Figure 5. *Calculation parameters of rolling piston compressor and revolving vane compressor*

In the rolling piston compressor, the force acts on the center of the rotor. For each 30 ° angle between 30 ° - 330 ° rotor angle in Equation (1), the area where air is compressed between the cylinder and the rotor has been analytically calculated. As shown in Equations (2), (3) and (4), the force was found by multiplying the area and the accepted pressure value for each constant pressure (P) value from 0.5 bar to 3 bar between 30 ° -330 °. The torque generated by the force effect (T_1) is calculated by multiplying the perpendicular distance of the force to the rotor center (X_1) for each angle

$$A_r = \left[\pi r^2 \gamma / 360\right] t \quad . \tag{1}$$

P pressure is taken as constant (0.5 bar, 1 bar, 1.5 bar, 2 bar, and 3 bar)

$$F_r = P.A_r \tag{2}$$

Torque arm from the similarity of triangles;

$$X_1 = a/2 .e.r \tag{3}$$

$$T_1 = X_1.F \quad . \tag{4}$$

In the revolving vane compressor, the force only acts on the vane. Therefore, the area to be considered in the torque calculation is the vane area. As can be seen in Equations (5) and (6), the torque length is determined as the distance of the force applied to the vane from the center of rotation

$$x = \sqrt{r^2 + c^2} - b \tag{5}$$

$$X_1 = x + [(r + e - x)/2] \tag{6}$$

$$A_v = (r + e - x)h \tag{7}$$

$$F_v = P.A_v \tag{8}$$

$$T_1 = X_1 \cdot F \tag{9}$$

$$W_1 = {T_{1ort}.n}\big/_{9,549} \;. \tag{10}$$

In order to calculate the torque caused by inertia in revolving vane compressor, Equation (11) the angular velocity, in Equation (12) moment of inertia, in Equation (13) the calculation of torque due to inertia (T_2) is seen

$$\omega = {\pi.n}\big/_{30} \tag{11}$$

$$I = m(r_o^{\,2} - r_i^{2}) \tag{12}$$

$$T_2 = I.\omega^2.\rho \tag{13}$$

The operating rotation speed range of the compressor is determined as 600-2700 rpm at 300 rpm intervals; Angular velocity and T_2 torque values were calculated according to this rotation speed range. As seen in Equations (14) and (15) the average T_1 torque calculated for each pressure and T_2 torque values obtained by the above equations are summed up and calculated as the total torque (T_{top}) of the revolving vane compressor and the total power (W_2) is consumed by the electric motor

$$T_{top} = T_1 + T_2 \tag{14}$$

$$W_2 = {(T_1 + T_2).n}\big/_{9,549} \cdot \tag{15}$$

The compressors are driven by a 4 kW electric motor operating in the range of 300 - 3000 rpm for both designs. In the experiments, pressure, flow and power values were measured between 600-2700 rpm for each design at intervals of 300 rpm. The experiments were repeated 3 times and the results were evaluated as 5% relative error for each measurement value. Test setup diagram for rolling piston and revolving vane compressors is shown in Figure 6 and test setups are shown in Figure 7.

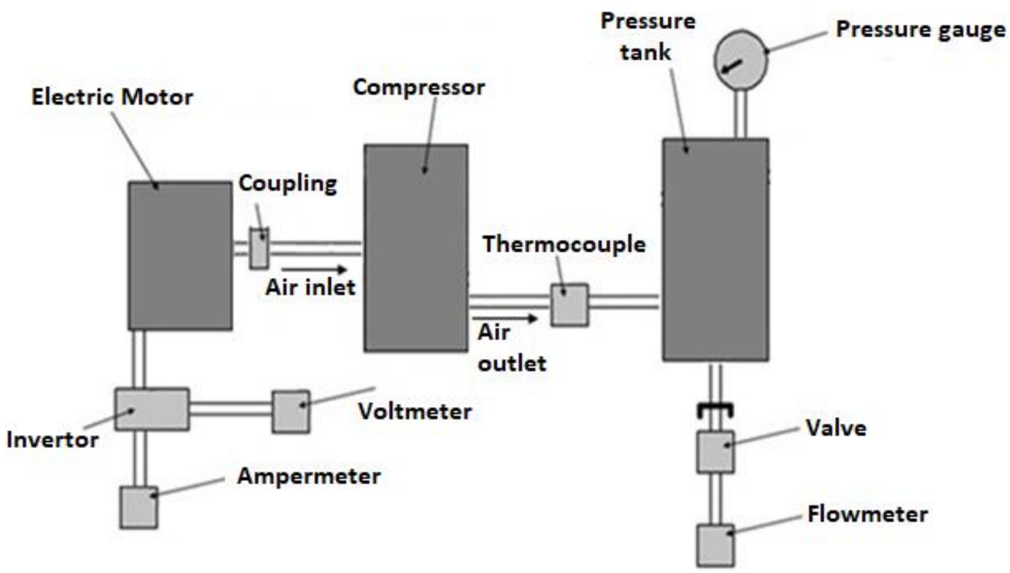


Figure 6. *Experimental setup diagram for compressors*

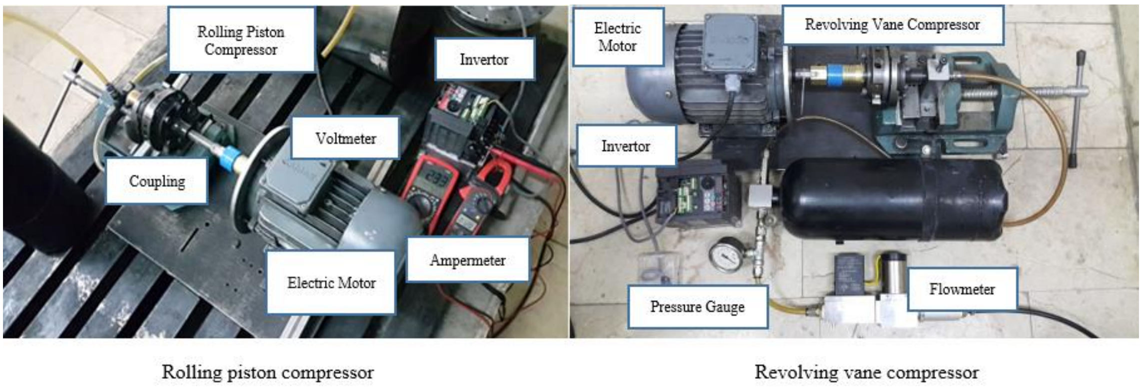


Figure 7. *Experimental setup for rolling piston compressor and revolving vane compressor*

In the experiments, the maximum pressure that the compressor can produce for each fixed rpm was determined. Then, the pressure was reduced at 0.5 bar intervals and flow, current and voltage values were measured for each pressure value. For rolling piston compressor test; the cylinder was held fixed with the help of a crescent-shaped apparatus and the eccentric shaft was rotated by an electric motor. For the revolving vane compressor test, the eccentric shaft is fixed by clamping to the vise with an auxiliary holding apparatus. The cylinder was turned by the electric motor with the help of a coupling. All the tolerances are the same for both compressors and the same parameters were measured in both experiments.

3. THE RESEARCH FINDINGS AND DISCUSSION

In this section, the results found for both designed compressors are evaluated in three main sections. While the calculated theoretical values are compared in the first part, the experimental results are examined in the second part. In the third part, theoretical and experimental results are compared at 1800 rpm, which is an average value for both compressors.

3.1. Evaluation of Theoretical Calculations

As a result of the theoretical calculations made in Equations (4) and (9) in Material Method, torque values are calculated at 30-degree angle intervals for a 360 degree of both compressors. Comparisons were made by taking the average values of the calculated torque values. Figure 8 shows the torque curves for rolling piston and revolving vane compressors.

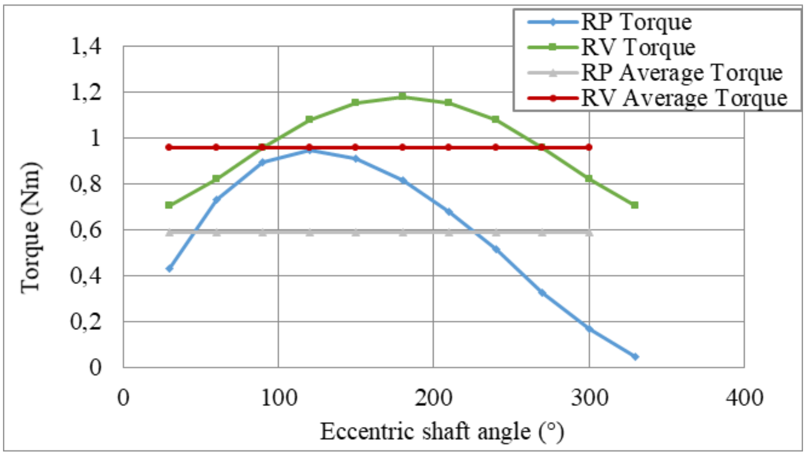


Figure 8. *Torque curves for rolling piston and revolving vane compressors*

When Figure 8 is examined, it is seen that the average torque for the rolling piston compressor is 0.588 Nm while the average torque for the revolving vane compressor is 0.96 Nm. The pressure surface area (Ar) in the rolling piston compressor is about 20 times larger than the pressure surface area (Ap) in the revolving vane compressor. However, the torque arm (X_1) in the revolving vane compressor is approximately 8 times more than the torque arm in the rolling piston compressor, and due to the high inertia force, more torque occurs in the revolving vane compressor. In the theoretical calculations, the power consumed by the compressors was calculated according to different pressure values at 1800 rpm as an average revolution (Figure 9). When the curves are analyzed, it is seen that the power consumption increases as there is an increase of pressure for both compressors.

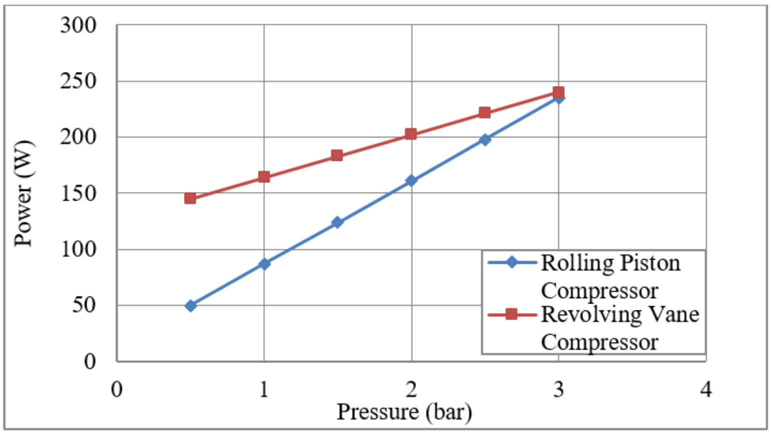
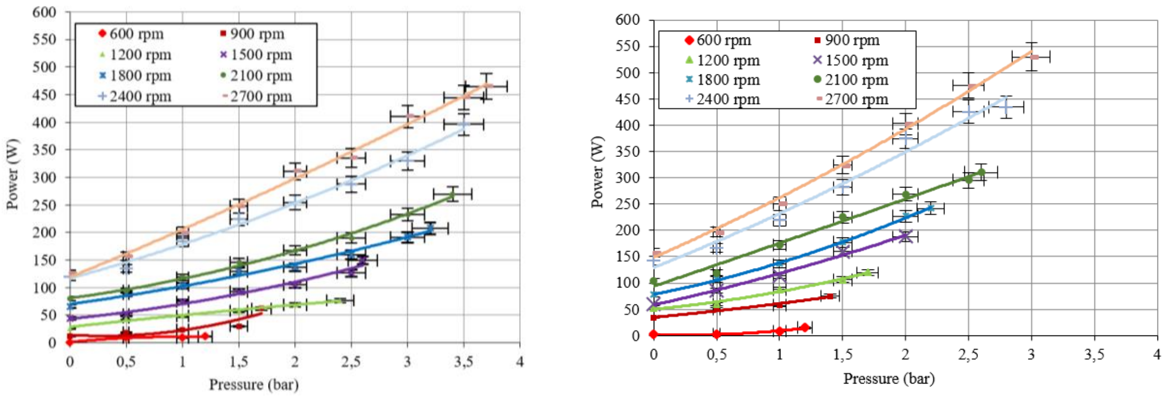


***Figure 9**. Power consumption curves of compressors with force and inertia effect*

When Figure 9 is examined, it is seen that the rolling piston compressor consumes less power than the revolving vane compressor for each pressure value. However, as the pressure produced by the compressors increases, it is seen that it consumes proportionally more power than the rolling vane compressor. The reason for this is that the pressure surface of the rolling piston compressor is approximately 20 times higher than the pressure surface of the revolving vane compressor.

3.2. Evaluation of Experimental Studies

Laboratory experiments were conducted to compare rolling piston and revolving vane compressors to see the effect of inertia. Figure 10 shows the power-pressure curves for rolling piston and revolving vane compressors.



a b

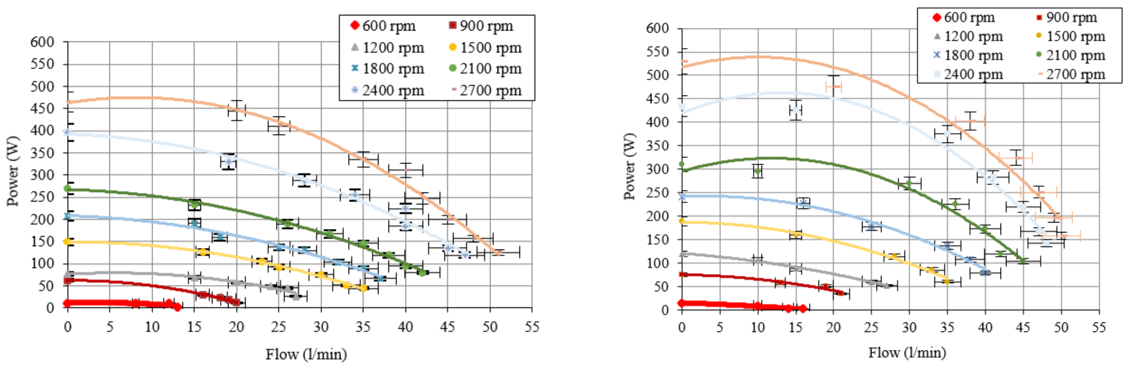
***Figure 10.** Power-pressure curves; a) Power-pressure curves for rolling piston compressor b) Power-pressure curves for revolving vane compressor*

When Figure 10 is examined, it is seen that as the generated pressure in both designs increases for all the rpms, the power consumption increases. The maximum pressure obtained for the rolling piston compressor is 3.7 bar at 2700 rpm; the maximum power consumed from the electric motor is 464.79 watt. In the revolving vane compressor, it is seen that the maximum pressure is achieved as 3 bar at 2700 rpm and the highest power consumed by the compressor reaches 530 Watts. In Table 2, the ratios of the power consumed by the rolling piston and revolving vane compressors for each rpm and the pressure are compared.

Table 2. *The ratio of the power of the rotary vane compressor to the power of the rotary eccentric shaft compressor according to the pressure and circuit*

Pressure(bar) / RPM	0,5	1	1,5	2	2,5	3
600	1,46	1,18	1,36			
900	2,78	2,48	2,58			
1200	1,33	1,81	1,86	1,98		
1500	1,67	1,53	1,73	1,81		
1800	1,2	1,32	1,37	1,67	1,7	
2100	1,27	1,46	1,55	1,61	1,63	
2400	1,23	1,2	1,26	1,47	1,48	1,47
2700	1,26	1,26	1,31	1,3	1,42	1,29

In Table 2, since the revolving vane compressor produces less pressure than the rolling piston compressor, the ratios of the power consumed by the compressors at low rpm could not be obtained for each pressure. When the rates are examined, it is seen that the revolving vane compressor draws approximately 1.5-2 times more power than the rolling piston compressor for all revolutions except 900 rpm. At 900 rpm, this rate goes up to 2.8. This can be explained by the fact that the revolving vane compressor operates more stable after 900 rpm. The second important parameter after pressure, and generation in rotary compressors is the flow rate of the air obtained. Figure 11 shows the power-flow curves for the rolling piston compressor and revolving vane compressor.



a b

Figure 11. *Flow rate curves; a) Change of power according to flow rate for rolling piston compressor b) Change of power according to flow rate for revolving vane compressor*

When Figure 11 is examined, it is seen that in the rolling piston compressor, as the flow rate increases for all the rpms, a decrease in power occurs with the effect of decreasing pressure. In the rolling piston compressor, the maximum flow rate of 51 l / min is produced, while the power consumed from the electric motor is 125.46 Watt. In the revolving vane compressor, the maximum flow rate achieved at 2700 rpm is 49 l / min, while the power consumed by the compressor is 149 Watt.

3.3. Comparison of Theoretical and Experimental Results

In this section, theoretical calculations for both designs (rolling piston and revolving vane compressors) and experimental studies are compared for an average rotation speed of 1800 rpm. Figure 12 shows the comparison of the theoretical and experimental studies of rolling piston and revolving vane compressors.

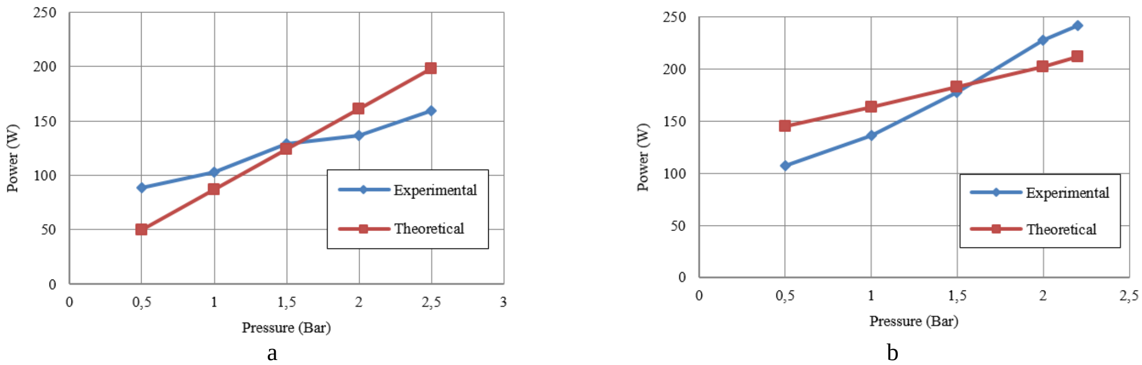


Figure 12. *Curves of theoretical and experimental comparison; a) Rolling piston compressor at 1800 rpm b) Revolving vane compressor at 1800 rpm*

When Figure 12.a is examined, it is seen that similar curves are obtained from the theoretical and experimental study for the rolling piston compressor. In theoretical calculations, as the pressure increases for both compressors, there is a linear increase in the power consumed. However, in the experimental results, the increase in power is not linear since different pressure leaks and friction losses occur at each pressure value. As seen in the test results of the rolling piston compressor in Figure 12-a, the power consumed at low pressures is higher than the theoretical calculations. However, as the pressure increases, there is a decrease in power due to the increase in pressure leaks. When the experimental results of the revoiving vane compressor in Figure 12-b are examined, less pressure leaks occur at low pressures than theoretical calculations, while at high pressures pressure leaks increase and results close to the theoretical values are obtained.

4. RESULTS

In this study, two different designs have been made for the rotary compressor, which is generally used in the cooling industry. In the first design, with the common use in the industry, the eccentric shaft was rotated by keeping the cylinder fixed. In the second design, the cylinder and vane were rotated by keeping the eccentric shaft fixed. Theoretical and experimental comparisons were made for both designs, and the following results were obtained.

• Rotary compressors; although they have a simple structure, small size, lightness, and stable working, they also have some disadvantages [11-12]. When operating at low speed, the compressors show -insufficient cooling capacity. When working at high speeds [28,30], large friction losses occur in compressors [13-19].

• In previous studies, a revolving vane compressor was operated at 2,4 bar pressure in the range of 2850-3800 rpm [35]. In this study, maximum 3.7 bar with rolling piston type compressor; Maximum 3 bar pressure was obtained with revolving vane type compressor.

• The maximum pressure obtained for the rolling piston compressor in the experiments is 3.7 bar at 2700 rpm, whereas the maximum consumed power from the electric motor is 464.79 watt. In the revolving vane compressor, the maximum pressure obtained is 3 bars at 2700 rpm, while the maximum consumed power from the electric motor is 530 Watt. The rolling piston compressor for the same 3 bar pressure consumed approximately 1.3 times less power with 410 watt. Based on the same revolution, it has been observed that the power consumption is consistent with the literature [36].

• In the experiments, the flow rates obtained from the revolving vane compressor at all pressure values for 600 rpm are higher than the rolling piston compressor. For other fixed rotation speeds (900-2700 rpm), the revolving vane compressor produces higher flow rate at low pressures, while the rolling piston compressor produces more flow rate as the pressure increases. This is because the friction losses reduce the amount of flow rate when the revolving vane compressor is used at high rpms and pressures.

• Due to the effect of inertia forces for all speeds, the revolving vane compressor consumes 20%-25% more power than the Rolling piston compressor.

• Considering the power consumed and the pressure produced as a result of the theoretical and experimental studies, it is seen that it would be more appropriate to use rolling piston systems in compressors and revolving vane systems in turbines.

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CONFLICTS OF INTEREST

No conflict of interest was declared by the authors.

NOMENCLATURE

a: for angle θ on the rolling piston compressor beam length perpendicular to the force (mm)
A_r: area of rotor (piston) (mm^2)
A_v: area of vane (mm^2)
b: vertical length of the triangle used in calculations (mm)
c: horizontal length of the triangle used in calculations (mm)
e: eccentricity (mm)
Fr: force acting on the rotor (N)
Fv: force acting on the vane (N)
I: moment of inertia (mm^4)
m: mass (kg)
n: cycle (rpm)
P: pressure (bar)
r: rotor radius (mm)
r_i: inner radius of the cylinder (mm)
r_o: outer radius of the cylinder (mm)
t: thickness of the rotor (mm)
T_1: torque generated by force (Nm)
T_{1ort}: average of torque generated by force (Nm)
T_2: torque generated by inertia (Nm)
Ttop: total torque (Nm)
W_1 : power generated by force (Watt)
W_2 : power generated by force and inertia (total power) (Watt)
X: sum of rotor radius and eccentricity (mm)
X_1: torque length (mm)

Greek Symbols
γ : rotor angle (°)
θ: Eccentric shaft angle (°)
ρ : irregularity coefficient
ω : angular velocity (rad/sec)

Abbreviations

CFD: computational fluid dynamics
COP: coefficient of performance
RP: rolling piston
RV: revolving vane
TiN: titanium nitride coating

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