2000

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E. Ayder
Instanbul Technical University

H. H. Arikan
ARMAS Arikan Makine San. Ve Tic. A

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Erkan AYDER
Istanbul Technical University
Faculty of Mechanical Engineering
80191 Gümüşsuyu, Taksim, Istanbul, TURKIYE
Tel :+ 90 212 249 8811 ; Fax:+90 212 245 0795
E-mail: erkan@burgaz.mkn.itu.edu.tr

H. Hüseyin ARIKAN
ARMAS Arıkan Makine San. Ve Tic. A.Ş.
Ümraniye, Istanbul, TURKIYE
Tel:+90 216 313 9504
Fax:+90 216 313 9505

ABSTRACT

The main purpose of the present work is to show the influence of a developed cylinder head on the free run performance characteristics of the piston compressors used in the brake systems of the heavy vehicles. When the discharge chamber pressure reaches to the desired value, the compressor delivers the air to the atmosphere through the suction or discharge port of conventional cylinder heads. In the developed system, the pressurized air is used to recuperate the energy taken from the vehicle engine. Therefore, less energy is consumed compared to that for conventional types of cylinder heads.

Effects of three different cylinder heads on the performance of piston compressor in the free run mode are studied experimentally. The detailed cylinder pressure and torque measurements are used to explain the low energy consumption.

INTRODUCTION

The piston compressors used to produce the pressurized air required in the brake systems of heavy vehicles are driven by the engine. When the pressure in the discharge chamber reaches the desired value, the compressor should stop filling the chamber. Since the compressor is directly connected to the engine, it can not be stopped. However, its running mode can be changed to a mode in which less energy is consumed. This mode is called free running mode in the present study.

During the free run of the piston compressor used in the brake systems, different methods are applied in order to decrease the required energy for the compressor. In the so-called conventional system-1 in the present study, the suction valve is kept in the open position and air flows into the cylinder and is sent to the atmosphere through the same suction port. In the second system named as conventional system-2, the air taken inside the cylinder through the suction port is sent to the atmosphere through the discharge valve and the unloader valve. All the air flow process is done by the piston that uses the energy of the engine. The similar conventional systems are also used in order to control the capacities of the industrial compressors and explained in [1] and [2]. The energy required for the free run mode might be reduced by use of already compressed air inside the piston cylinder system.

In the present study, working principles of the developed system are explained and its energy saving character is described through experimental measurements. Results concerning the energy consumption are compared with those of conventional systems.

DEVELOPED SYSTEM

The piston compressor head with the energy saving system is developed based on use of the energy of the compressed air inside the cylinder during the free run. One can stop the air flow in and out of the cylinder. Therefore, a nonnegligible part of the energy taken from the engine during the compression phase of a cycle might be recuperated during the expansion phase. This leads to a reduction in the consumed energy.

In the system developed in this study, a volume increase is required. This is accomplished by opening the discharge valve by means of a specially designed mechanism triggered by the pressurized air in the discharge chamber. A one way valve between the deactivated discharge valve and the discharge chamber prevents air from flowing towards cylinder. Due to the increased volume of the cylinder, the maximum pressure becomes much lower than that in normal running mode. Lowered maximum pressure in the cycle of the free running mode causes less energy consumption. The air pressure should be as low as possible but not low enough to open the suction valve when the piston is at the bottom dead centre.

In the applications, due to the leakage flow through the gaps between piston and cylinder, the pressure inside the cylinder can drop below suction pressure during the expansion phase, and small amount of air is sucked into the cylinder. Since the amount of the air sucked into the cylinder is very small, it has a very little effect on the energy saving behavior of the developed system in this study.

The volume of the cylinder-head system can also be increased by placing a second suction valve into the suction port and keeping it open during the free run. The piston compressors working with this principle exist and are already used in the brake systems of the heavy vehicles. Addition of the second suction valve into the suction port increases pressure losses of the incoming air to the cylinder causing to a reduction in the volumetric efficiency of the compressor in the normal running mode.

EXPERIMENTAL FACILITY

Some aspects of laboratory testing of the reciprocating compressors are given in [3]. The experimental facility used in the present study consists of a piston compressor with single cylinder driven by an AC motor and a data acquisition system. The speed of the AC motor is controlled by the frequency controller (Figure 1). The main geometrical dimensions of the compressor are given as follows

Diameter of the piston (Bore)	: 92.075 mm
Sweep volume	: 252.5 cm^3
Stroke	: 37.6 mm

The pressure inside the cylinder are measured by means of variable reluctance pressure transducers with a sample rate of 5 kHz. To assure the best possible dynamic frequency response, transducers are connected as closely as possible to the cylinder.

The torque transducer located between two bearings is inductive type with strain gauges and is capable of sampling with 10 kHz.

The outputs of the pressure and torque transducer are in volts and are collected by a personal computer equipped with a data acquisition card with a sampling frequency of 100 kHz.

RESULTS

The torque measurements are made without installing the compressor in the experimental set up in order to verify the alignment of the shaft, bearings and motor. The variation of the shaft torque with the crank angle at 2000 RPM for three cycles is shown in Figure 2. The friction torque in the bearings is about 1 Nm and almost independent from the crank angle.

The measurements are performed without compressor head in order to evaluate the frictional torque of the compressor. The dotted line in Figure 2 shows the variation of the torque to overcome the mechanical friction with the crank angle.

The variations of the cylinder pressure and shaft torque with the crank angle for the compressor with the developed head in normal running mode are shown in Figures 3 and 4, respectively. The variations are shown over three cycles and 60 samples have been taken for each cycle. The discharge chamber pressure is kept constant at 0.4 Mpa and simulates the continuous consumption of the pressurized air. The tank pressure has not been increased up to the characteristic value (0.9 Mpa), for the brake systems of the heavy vehicles, due to the mechanical limitations of the torque transducer. This experimental limitation does not change the results for the free run mode in the present study, where the air pressure values in cylinder are low for free run modes in general.

The comparison of three different free run modes for the piston compressor is made by using the same head geometry and valves. Some modifications are made on the head in order to simulate the conventional systems.

For the conventional system-1, the suction valve is fixed in such a way that it will stay open in the expansion and compression phases of the cycle. The geometrical characteristics of the gap between the head and valve through which the air flows are similar to those of the cylinder heads developed for this purpose in conventional system-1. In order to simulate the conventional system-2, the discharge port of the compressor head has been opened to the atmosphere. During the cylinder pressure and torque measurements for conventional system-1 and 2, the mechanism in the head developed in this study has been turned off.

The measured cylinder pressure and torque values for three different free run modes are compared in Figures 5 and 6, respectively. For the conventional system-2, the cylinder pressure increases up to a value required to open the discharge valve and therefore has the highest value among other modes (thin solid line in Figure 5). The lower value of the peak pressure in the conventional system-1 (dotted line) is a result of the special design where suction and discharge are made through the same suction port. Therefore, hydraulic losses just due to suction port causes a lower peak value for the pressure. The peak pressure value in the system specially developed in this study is in between those of conventional system-1 and conventional system-2.

The pressure curve below the atmospheric value (0 Mpa) shows the air intake part of the cycle. Air intake part of the expansion phase occupies large portion of a complete cycle in conventional systems 1 and 2. As mentioned above, any air flow during the expansion phase is not recommended. However, as seen from Figure 5, during the expansion process of the developed system air intake is observed. During the compression phase, small amount of air escapes through the gap between cylinder and piston. This causes a pressure drop below the atmospheric value during the expansion process, which results in a negligible amount of air intake.

The shapes of the shaft torque variations of conventional systems and developed system are quite similar in the entire cycle(Figure 6). The peak values differ due to the different cylinder pressures. In the expansion process, the torque value of the developed system is quite lower than those of conventional systems. This clearly shows that the compressed and trapped air in the cylinder is used to help the engine to drive the piston down during the expansion part of the cycle. This leads to the energy saving behavior of the compressor in the free run mode.

Pressure-Volume diagrams of three free run systems are compared in Figure 7. The area swept by the Pressure-Volume curve for the developed system is smaller than those of the conventional systems. The smallest area indicates the least energy required for free run mode among others.

CONCLUSIONS

Concerning the power consumption during the free run, the developed system in this study is better than conventional systems. This system does not cause a decrease in the volumetric efficiency since no modification is required in the suction port of the cylinder head.

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ACKNOWLEDGMENT

Support for this work was partially provided by the Technology Development Foundation of Turkey (TTGV) under the Contract No: 230.

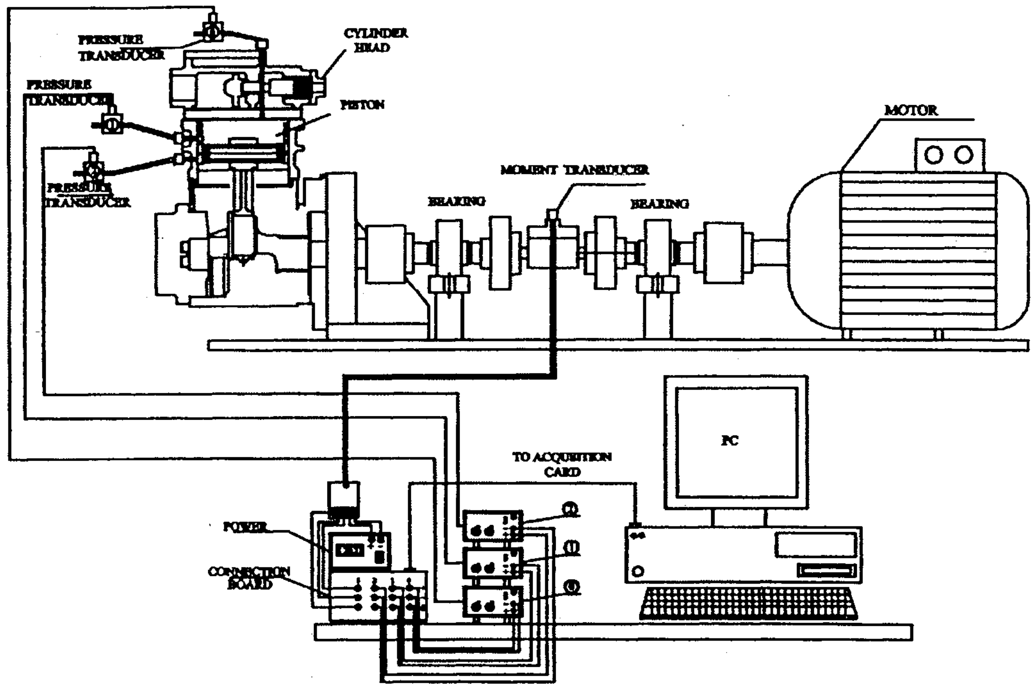

Figure 1. Experimental setup and measurement system

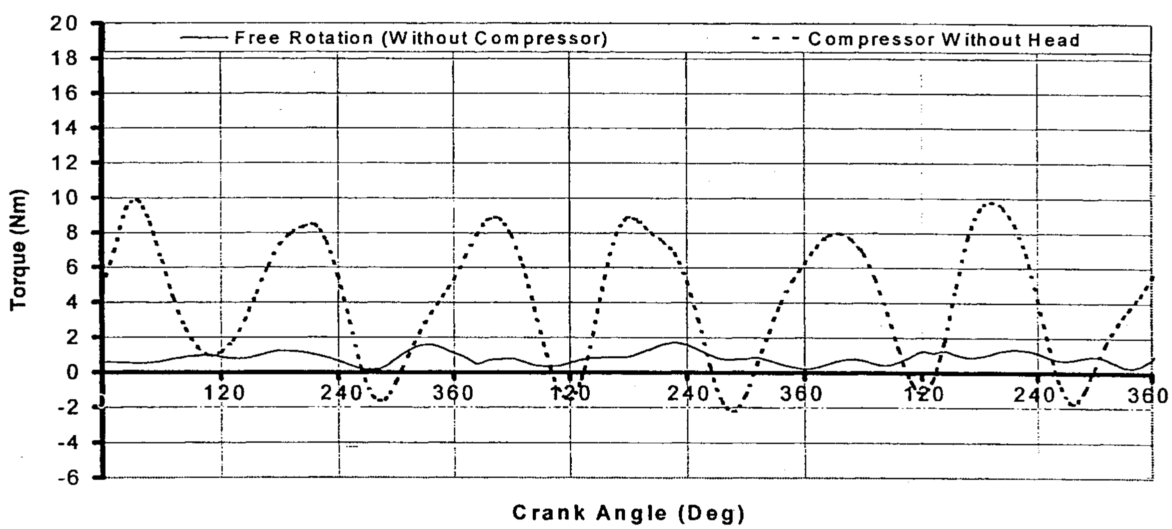

Figure 2. Variation of the shaft torque with crank angle; without compressor and with compressor without cylinder head

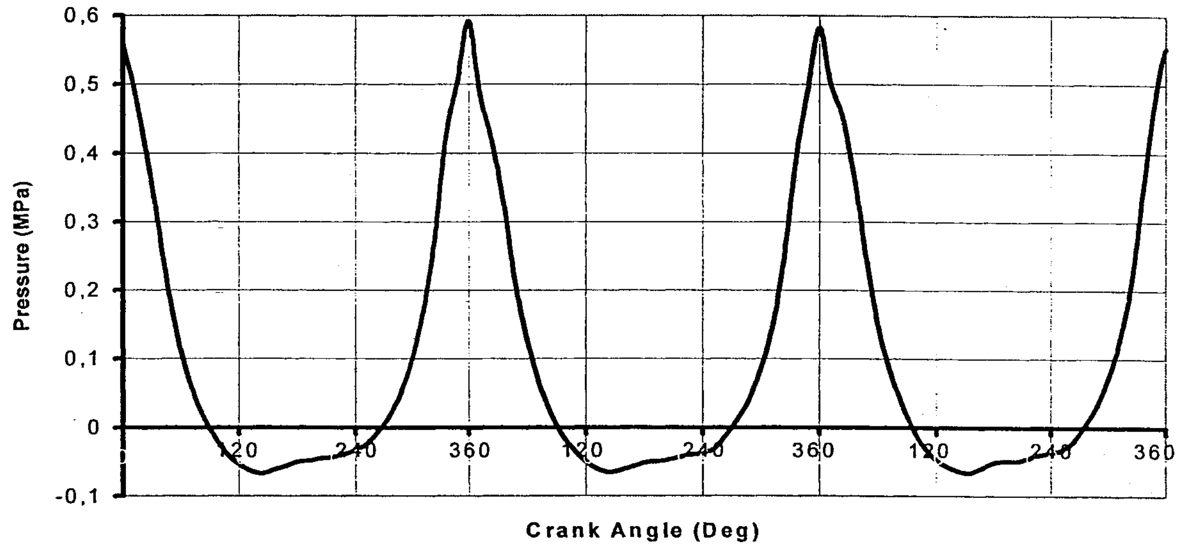

Figure 3. The variation of the cylinder pressure with crank angle of the compressor in normal running mode

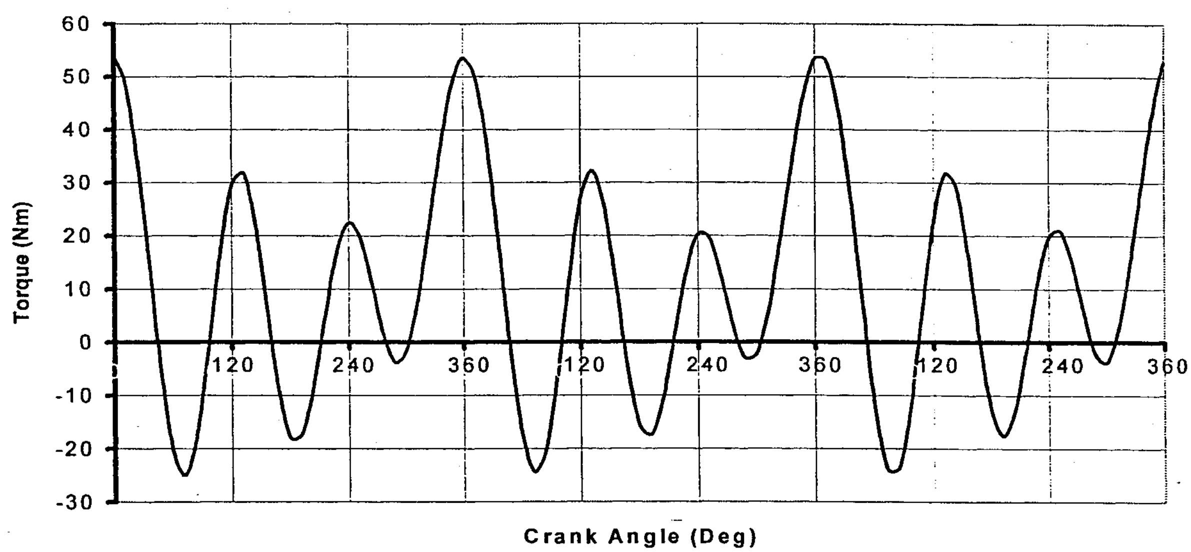

Figure 4. The variation of the shaft torque with crank angle of the compressor in normal running mode

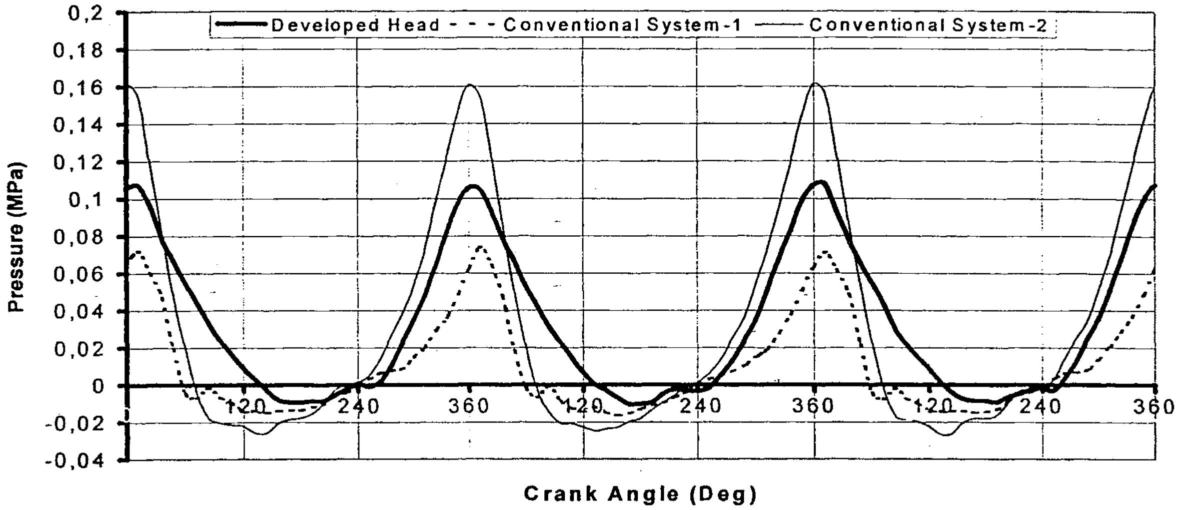

Figure 5. The comparison of pressure variationsin cylinder with crank angle for three different free run systems

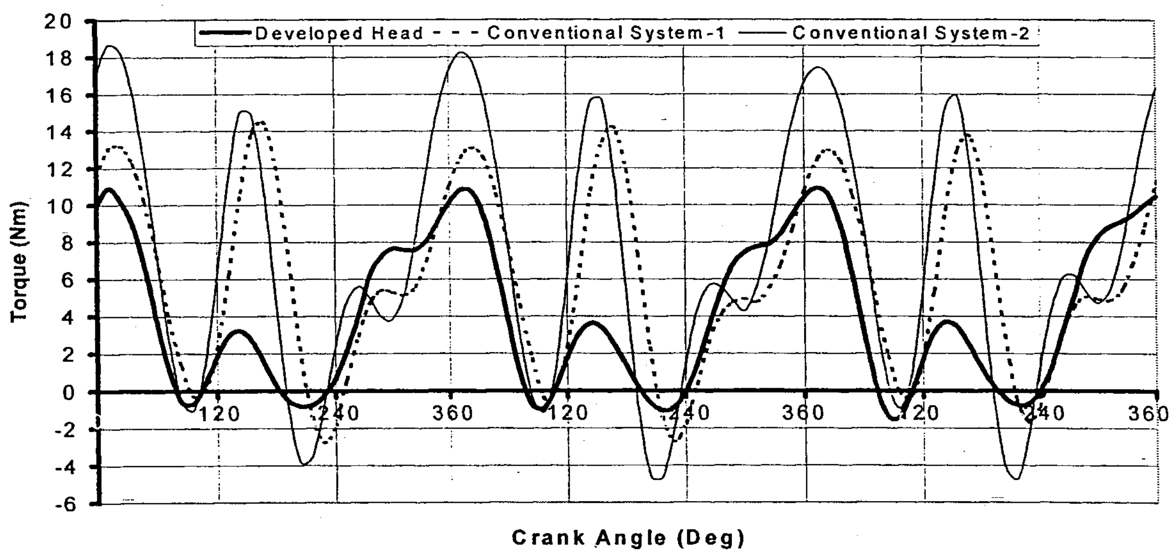

Figure 6. The comparison of shaft torque variations with crank angle for three different free run systems

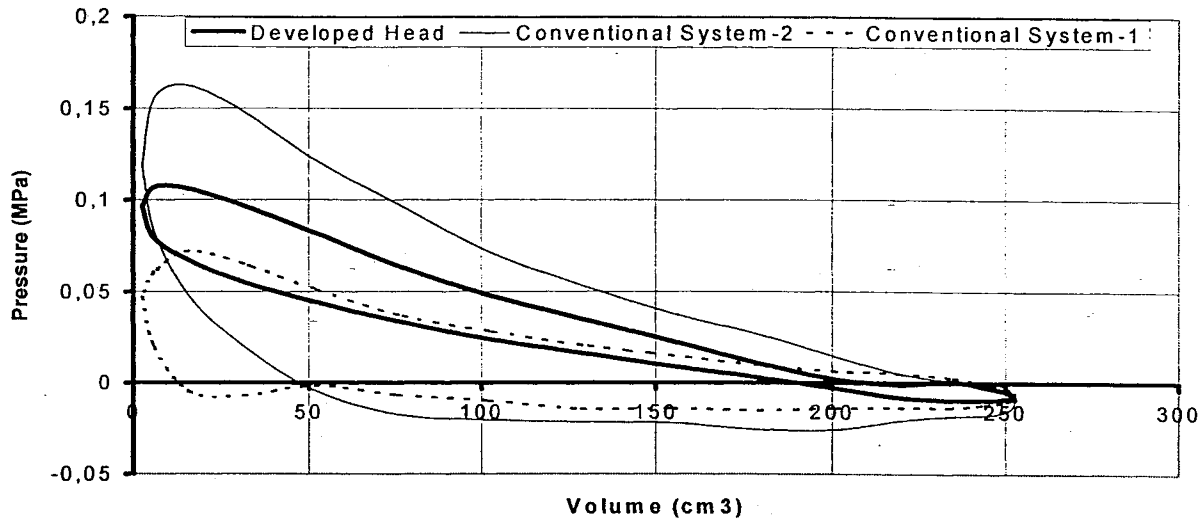

Figure 7 . The comparison of pressure-volume diagrams of three different free run systems