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Vibration Analysis of an Air Compressor Based on a Hypocycloidal Mechanism: an Experimental Study

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A B S T R A C T

In this paper, the experimental vibration analysis of a single cylinder air compressor based on a hypocycloid straight line mechanism (HSM) is investigated. The HSM mechanism uses planetary gears to convert rotational motion to purely linear motion. In the conventional air compressor, the slider-crank mechanism is replaced by the HSM mechanism with appropriate counterweights. The constructed setup is fixed over a base plate supported by an elastic foundation and is driven by a speed-controlled electric motor. During the experiments, the driving frequency is varied and the vibrations of the base plate due to unbalanced forces are recorded in three directions. The results are compared with the experimental results of a compressor based on the conventional slider-crank mechanism. Time and frequency domain analyses indicated that the mechanism can be balanced effectively, and the shaking force and induced vibrations can be significantly suppressed.

1. INTRODUCTION

The slider-crank mechanism developed one hundred years ago is still used today in many high-speed machines. The simplicity of the mechanism promotes its employment in many applications, such as internal combustion engines, some air compressor types, and pumps. However, it has been proven that the slider-crank cannot produce a purely or symmetrical reciprocating motion. In addition, because of the gyrating motions of the connecting rod, dynamic balancing of the slider-crank is a serious problem, possible only by using complex linkage mechanisms. The general planer-type motion of the connecting rod produces piston side loads which increase the amount of friction force between the piston and the walls of the cylinder holding it. Different methods have been developed for dynamic balancing of the mechanism [1, 2], but the complexity and restrictions of some proposed methods have caused new approaches to be reported. Balancing based on imposing counterweights to the

links and redistributing the mechanism's mass is a conventional method for balancing slider-crank mechanisms [1-4], but it is difficult to apply for balancing the shaking force of the mechanism. Employing two equal counter rotating masses, the Lanchester balancer makes it possible to balance some frequency components of the shaking forces and moments in slider-crank mechanisms where balancing is achieved by attaching the balancing elements to the crank shaft [5]. This method has found extensive application in engine balancing; however, when the slider-crank mechanism has an eccentricity or the sliding axis does not pass through the fixed pivot of the input crank, this method should be improved. The generalized Lanchester balancer has also been investigated by Arakelian and Makhsudyan [6]. In another study, Tsai and Walter [7] suggested balancing an in-line four-cylinder engine by using an Oldham-coupling balancer. They balanced the second harmonic of the shaking force and moment where the balancer mechanism turned with the same velocity as the initial unbalanced system. Moreover, the shaking force and moment of the slider-crank mechanism was balanced

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based on the copying properties of the pantograph and parallelogram mechanism [8]. This method involves connecting a two-link group, forming a pantograph with the crank and coupler, to the mechanism to be balanced. In addition, balancing the shaking force and moment of the slider-crank mechanism has been achieved using two identical slider–crank mechanisms which are executed in a similar manner but in opposite directions [9]. The slider-crank mechanism has also been balanced employing planetary systems [8, 10] and a cable and pulley arrangement [11]. Cam mechanism is the other technique used to balance the slider-crank mechanism [12, 13]. In other words, the shaking force and moment are cancelled via a cam mechanism carrying a counterweight, and the cam-driven masses are employed to keep the center of the mechanism's mass stationary. Based on the cam-spring mechanism for minimizing input torque fluctuations, Arakelian and Briot proposed a method for balancing the shaking force and shaking moment in the slider-crank mechanism [14]. Imposing a second cam to the input crank, they developed an instrument to balance both shaking force and moment. Employing multiple cylinders with static counterweights in the crank shaft is another method for balancing the slider-crank mechanism; however, the vibrations of the compound mechanism could cause a considerable amount of power to be lost.

To surmount the aforementioned restrictions, some researchers proposed a different mechanism which converts the purely rotational motion of the input shaft to a linear reciprocating motion in the connecting rod [15-17] and vice versa. This is known as the hypocycloid mechanism (HSM). This mechanism includes a fixed gear ring (or gear) and a planetary gear (or pinion). When the pinion rolls inside the gear, a point on the pitch circle of the pinion generates a curve called a hypocycloid. When the diameter of the pinion is one half that of the gear, the curve traced by a point on the pitch circle of the pinion is an exact straight line. This principle is used to design a planetary gear pair to produce a reciprocating motion with no gyrating motion in the connecting rod.

To the authors' best knowledge, there is no experimental result concerning the dynamic test and force balancing of the HSM mechanism in the literature. The present study, for the first time, examines the dynamic balancing of the HSM mechanism experimentally. Employing an HSM mechanism in an air compressor, the dynamic balancing of the mechanism was evaluated. In a conventional air compressor the slider-crank was replaced by the HSM mechanism with appropriate counterweights. The setup was fixed over a base plate supported by an elastic foundation. Then, employing a variable speed motor, the compressor was driven with different angular velocities, and the vibrations of the base plate recorded in three directions. Acceleration records of the HSM

mechanism were compared with test results of an equivalent compressor based on the conventional slider crank mechanism. In the experimental setups, the effects of the compressed air flow as an unbalancing force were ignored. Results showed that utilizing the balanced HSM mechanism notably suppressed the shaking forces and consequently the vibrations of the system. The test results were compared in time and frequency domains.

2. EXPERIMRNTAL SETUP AND THE RESULTS

Prior to explaining the experimental setup, the kinematic diagrams of the conventional slider-crank and the HSM mechanism were compared. Schematic views of the slider-crank and HSM mechanisms are shown in Figs. 1 and 2, respectively. In the conventional slider-crank mechanism, the piston side loads are reduced by selecting the length of connecting rod L as more than two times that of crank arm R. Consequently, the displacement of the slider will not be a purely harmonic motion and is described by the following equation:

$$x = L(1 - \sqrt{1 - \frac{R^2}{L^2} sin^2\alpha}) + R(1 - cos\alpha) \qquad (1)$$

where x is the displacement of the piston and α the crank angle. Due to the complicated structure of the reciprocating equation and the gyrating motions of the connecting rod, the dynamic balancing of the mechanism is difficult. In this mechanism if the connecting rod length is equal to that of the crank arm ($L=R$), the motion will be a purely harmonic one :

$$x = 2R(1 - cos\alpha) \qquad (2)$$

It should be noted that the side load is the main reason that L is always greater than R in real applications. In addition, when R is equal to L, the mechanism can be locked as α approaches 90 degrees, where the side load approaches its maximum value. However, in the proposed HSM mechanism (Figure 2), the gear diameter is two times the diameter of the pinion, and the connecting rod is attached to the pinion at point *P*. Therefore, point *P* and the rod are reciprocated in a straight line (over the vertical diameter of the gear). In other words, any point on the circumference of the pinion traces a straight line. For instance, point E on the opposite side of the point P reciprocates along the horizontal diameter of the gear. Results of the current study revealed that the planetary motion of the pinion and the reciprocating motion of point P can be produced by an artificial slider-crank shown by red dashed lines in Figure 2. In the fictional mechanism, the crank arm and the connecting rod have the same length. However, due to the presence of the pinion, the dead point and the

lateral load between the pinion and the gear are eliminated. In the HSM mechanism, the displacements of points P and E are represented as:

$$x_p = 2R cos\alpha$$
$$x_E = 2R sin\alpha \qquad (3)$$

A schematic view of the experimental setup is shown in Fig. 3. It includes the HSM mechanism with the appropriate balancing weights. It converts the purely rotational motion of the input crank to a purely harmonic linear motion of the connecting rod and consequently of the piston. In the experimental setup for symmetry, greater stability of the system, and the elimination of unwanted shaking moments, double HSM mechanisms with balancing weights are constructed and assembled to carry the connecting rod and the piston (see the side view in Figure 3).

Details of the dimensions and specifications of the conventional slider-crank and the HSM mechanism used in this study are listed in Table 1. It should be noted that replacing the HSM mechanism within the crank cases of a conventional compressor caused the stroke of the HSM mechanism to increase to 46 mm and, consequently, (because of the fixed height of the holding cylinder) the height of the piston and its weight were decreased.

Balancing the HSM mechanism took place in two steps. At first, to eliminate the shaking force induced by the connecting rod and the piston, a counterweight m_E is attached to the pinion on the side opposite point P (Figure. 3). Therefore, two components of shaking forces could be addressed. The shaking force induced by the connecting rod and piston acts in a vertical direction, and that related to the imposed counterweight m_E in a horizontal direction. In the second step, another counterweight m_C is attached to the crank shaft to eliminate the shaking forces resulting from the first step. Taking into account the locations of the counterweights, their masses have the following relations:

$$2m_E = m_L + m_p$$
$$2m_C = m_L + m_p + 2m_{pi} \qquad (4)$$

where m_L, m_p, and m_{pi} denote the weights of connecting rod, piston, and pinion, respectively. The performance of the imposed counterweights or dynamic balancing of the constructed HSM mechanism was evaluated by measuring the induced vibrations due to unbalanced forces. To this end, a compressor based on the balanced HSM mechanism and another compressor based on the conventional slider-crank mechanism were tested sequentially. They were fixed over a base plate with an elastic foundation and driven with a variable speed motor. The elastic foundation includes six helical spring with total stiffness of 240 KN/m. Overall weight of the moving parts including the motor, compressor and base

plate masses is 24 kg and the bouncing natural frequency of the system is about 100 rad/sec. Pictures of each setup are shown in Figures 4 and 5, respectively.
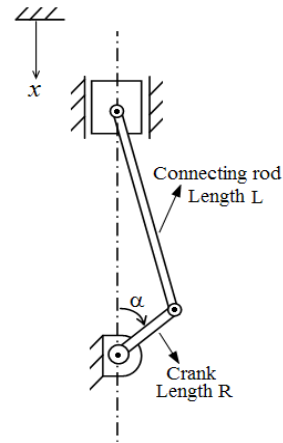

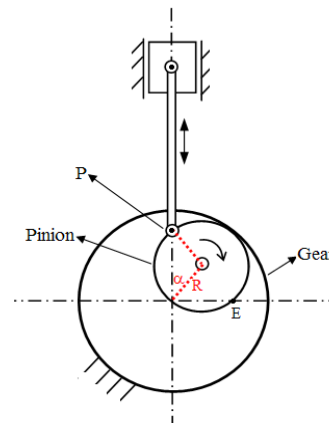**Figure 1**. Schematic view of the conventional slider-crank mechanism.

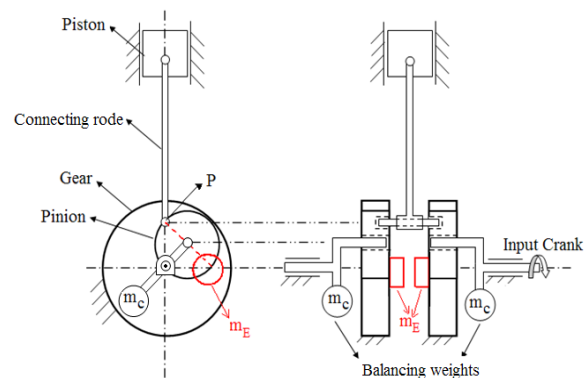
Figure 2. Schematic view of the HSM mechanism

Figure 3. Schematic view of the constructed HSM mechanism and its balancing weights.

TABLE 1. Dimensions and specifications of the experimental setup

Electric motor Power (Kw)	0.75
Driving speed (rpm)	0-1250
Piston diameter (mm)	65
Slider crank	
Stroke (mm)	40
Volume (cc)	133
Piston weight(gr)	295
Connecting rode weight(gr)	90
HSM	
Stroke (mm)	46
Volume (cc)	153
Pinion pitch diameter (mm)	23
Gear diameter (mm)	46
Piston weight m_p(gr)	266
Connecting rode weight m_L (gr)	64
Pinion weight m_{pi}(gr)	58
Counter weight 'm_E' (gr)	2×165
Counter weight 'm_C' (gr)	2×223
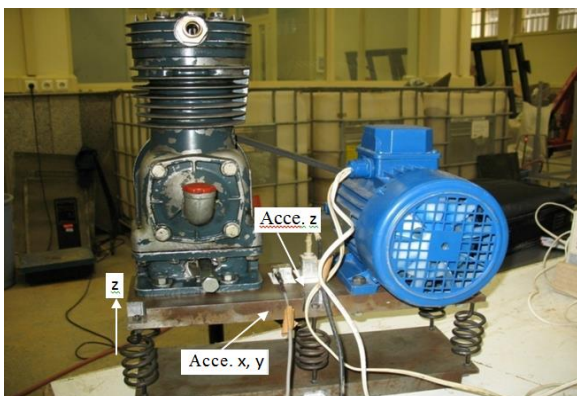
Figure 4. Experimental setup based on conventional slider crank mechanism.
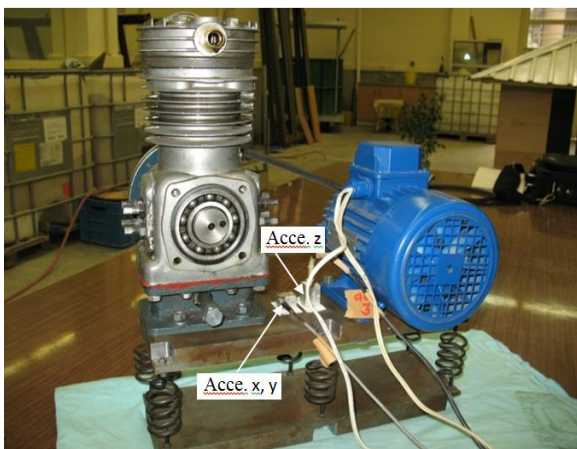
Figure 5. Experimental setup based on the balanced HSM mechanism.

Driving both mechanisms at up to 1250 rpm induced vibrations of the base plate due to shaking force, and moments are recorded in three orthogonal x, y, and z directions. The normal direction to the base plate is assigned as the z axis, and x and y axes are in the plane of the base plate and orthogonal to each other.

It should be noted that, during the tests, the output pressure pipe was disconnected from the storage tank, and the effect of the output air pressure on the dynamic forces was neglected. Throughout each test, the angular velocity of the input crank was varied steeply, and the accelerations of the base plate were recorded in three directions. Employing the conventional slider-crank mechanism, the time history of the base plate acceleration in z direction is shown in Fig. 6a. It is depicted that the shaking force increased with increases in velocity, especially when the driving frequency was close to the bouncing natural frequency of the system. Peak acceleration amplitude of 2g was recorded when the forcing frequency was about 1000 rpm. Fig. 6b shows the acceleration of the base plate in z direction where the slider-crank was replaced by the new balanced HSM mechanism. Results revealed that the shaking force and the induced vibrations were effectively suppressed. The emergent low level of acceleration, "up to 0.2 g", could be related to the imprecision of the constructed elements, which is inevitable. During each test, in addition to the acceleration in z direction, the acceleration components in x and y directions were also recorded. Due to effective balancing of the HSM mechanism, it was seen that the vibrations were also significantly suppressed in x and y directions (Figures 7 and 8).
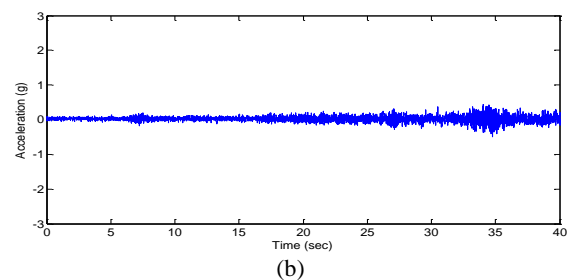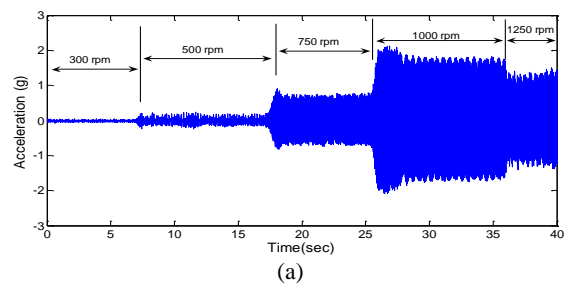

Figure 6. Comparison of the base plate accelerations in 'z' direction for different angular velocities, a) Conventional slider-crank mechanism, b) Proposed HSM mechanism.

Table 2 gives the root mean square 'RMS' of the recorded accelerations that is related to the consuming energy in the system. The results show that the RMS values of the accelerations are suppressed effectively in the proposed mechanism. A comparison of the test results is also reported in frequency domain. Taking into account the acceleration components in z direction, the frequency spectrum of the conventional and HSM mechanisms are shown in Fig. 9. It was seen that the frequency content of the conventional mechanism included distinct forcing frequencies. Some low energy levels of vibrations were also included in the frequency content. These frequencies could be related to transient vibrations and/or higher harmonics of the shaking forces in the slider-crank mechanism; however, all of the components were suppressed in the balanced HSM system.

In spite of the exact balancing of the HSM mechanism, some low-levels vibrations also emerged in the results. These vibrations could be related to exhaust air and some imprecision of the constructed gears and pinions, which were unavoidable in the present study (Figs. 6b, 7b, and 8b). This mechanism could be used in internal combustion engines to reduce fuel consumption and conserve considerable energy compared to the conventional slider-crank mechanism. However, in comparing the proposed HSM mechanism with the conventional slider-crank mechanism, complexity was seen as the main drawback of the proposed HSM mechanism.
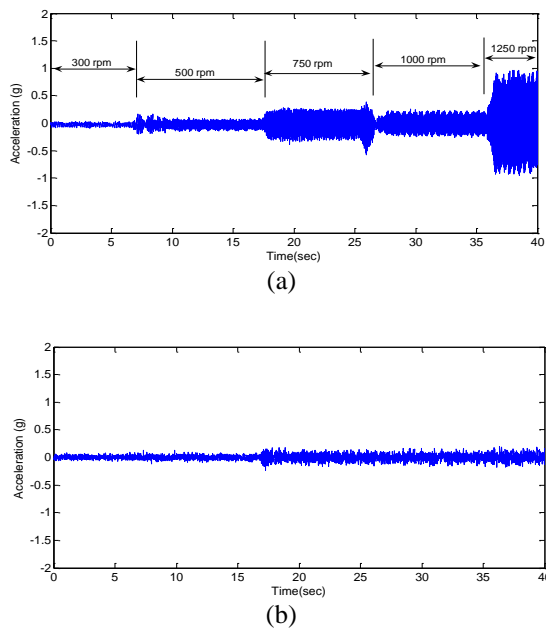

Figure8. Comparison of the base plate accelerations in 'x' direction for different angular velocities, a) Conventional slider-crank mechanism, b) Proposed HSM mechanism.
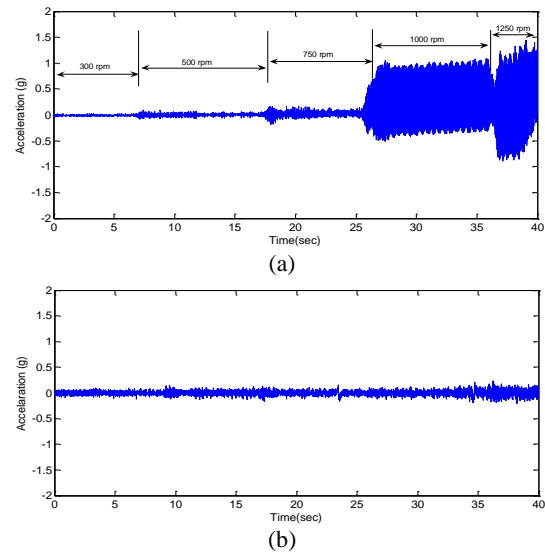
TABLE 2. Root mean square 'RMS' values of the acceleration records.

Direction	Conventional mechanism	HSM mechanism
Z	0.7562	0.0868
x	0.3753	0.0524
y	0.2245	0.0506

Figure 7. Comparison of the base plate accelerations in 'y' direction for different angular velocities, a) Conventional slider-crank mechanism, b) Proposed HSM mechanism.
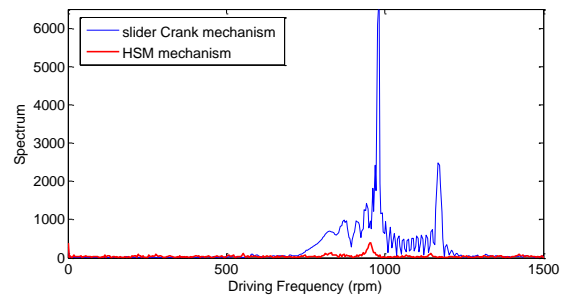

Figure 9. Comparison of the frequency spectrum of the slider-crank and proposed HSM mechanism in z direction.

3. CONCLUSION

Despite many efforts to balance the slider-crank mechanism, it was not possible to be balanced exactly except by using complex mechanisms and linkages. As an alternative, a new HSM mechanism is proposed which converts a purely rotational motion to a purely linear motion without any gyrating in the connecting rod and which can be balanced effectively. Employing the HSM mechanism, a single cylinder air compressor was made and balanced using appropriate counterweights. The apparatus was installed over an elastic foundation

in which its performance was studied by recording the shaking force induced vibrations in three directions. Deapite some imperfections in the constructed gear assembly, results showed that the mechanism could be balanced effectively, and the side loads on the piston and the shaking loads could be significantly eliminated. The balanced HSM mechanism can be used in internal combustion engines to reduce fuel consumption and conserve considerable energy, compared with the conventional slider-crank mechanism.

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چکیده

در این مقاله آنالیز ارتعاشات یک کمپرسور تک سیلندر بر مبنای استفاده از مکانیزم حرکت مستقیم هیپوسیکلوئید بررسی شده است. مکانیزم هیپوسیکلوئید شامل چرخ دهنده‌های سیاره‌ای است که حرکت دورانی را به حرکت رفت و برگشتی تبدیل می‌کند. در کمپرسور مورد نظر مکانیزم لنگ و لغزنده مرسوم با یک مکانیزم هیپوسیکلوئید جایگزین و پس از بالانس کامل آزمایش شده است. برای بررسی عملکرد مکانیزم پیشنهادی، سیستم مورد نظر بر روی یک بستر الاستیک قرار گرفته و توسط یک موتور الکتریکی و درایور مربوطه در دورهای مختلف آزمایش شده است. در طی آزمایش و در بسامدهای مختلف ارتعاشات پایه که ناشی از نیروها و گشتاورهای نامیزانی می‌باشند در سه جهت توسط شتاب سنج‌هائی اندازه‌گیری شده‌اند. نتایج به دست آمده با نتایج حاصل از تست ارتعاشات یک کمپرسور معمولی (بر مبنای مکانیزم لنگ– لغزنده) مقایسه شده‌اند. مقایسه نتایج در حوزه زمان و حوزه بسامد نشان داده است که مکانیزم پیشنهادی به صورت کامل متوازن شده و نیروهای نامتوازن ایجاد شده میتواند کاملاً توسط مکانیزم پیشنهادی حذف گردند

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