Computational Analysis of the Effect of Balancer on the Vibration Performance of the Engine: Experimental and Simulation

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Abstract: Torsional torque is considered as one of the loads that the crankshaft is constantly transferring. This force causes torsional stress (torsional stress due to torsion) in the fixed crankshaft bearings and shear stress in the movable bearings. Due to the effect of different parameters in the balancer design, e.g. inertial forces, the mass of weights, off-center, and distance from the axis of the crankshaft the need for extensive research in this field is sensed. The research investigates the effect of the balancer on reducing engine vibration. The main objective was to study the kinetics of the engine using Adams engine software by application experimental data. And finally comparing test results with software results to confirm created simulation. By analyzing the output signal, the root mean square (RMS) value of experimental data, the balancer reduced the amount of RMS value of engine vibration at 750 rpm in no-load mode by 40.46%, and 12.73% at the rotational speed of 2400 rpm. Accordingly, using the balancer with no-load was more effective at lower speeds. In the case of no-load mode, the use of a balancer reduced vibrations by 28.83%. It was found that in full load condition, the RMS of the engine vibrations increased with increasing engine speed. Experimental and simulated by software results were highly consistent with experimental results, which confirms the validity of the created simulation.

Keywords: Computational Analysis; Balancer; Vibration; Diesel Engine; Torsional Torque; Crankshaft

1 Introduction

Internal combustion (IC) engine as a power source in industry and agriculture [1, 2] can be considered one of the major factors in the vibration of automobiles. A balanced IC engine is one in which the forces acting on the supports and bearings are constant in terms of quantity and direction under normal performance conditions. In an unbalanced engine, the force on the supports and bearings changes frequently and causes the engine and the vehicle to vibrate [3]. The vibration by an IC engine can be related to friction and wear on parts, improper engine adjustment, incomplete combustion, and using poor fuel [4, 5]. The identification and elimination of these agents, will save costs, reduce fuel consumption and improve the ergonomic impact on the operator [5]. Over the past 30 years, there are few studies investigating the vibration impacts of an IC engine which requires more fundamental studies. The vehicle drive system is composed of different parts such as crankshaft, piston, connecting rods, and flywheel [6]. One of the most important, expensive, and complex mechanical parts of a car in terms of design, construction, and analysis is the crankshaft. Engine components need to be optimized by an excellent design for reducing the engine noise, producing high power levels, and increasing the engine durability and vehicle's comfort [7]. In recent years, reducing noise and vibration in IC engines and vehicles has become one of the design goals as well as the engine's durability and efficiency [5]. Customer perception of an engine and a vehicle quality is highly correlated with the generated noise and vibration which can be considered a competitive parameter for manufacturers. The crankshaft is always subjected to the application of various, large, instantaneous, and variable loads. It also affects the vehicle performance and tolerates different loads, which are very difficult to be predicted and estimated [8, 9].

Torsional torque is considered one of the loads that the crankshaft is constantly transferring. This force causes torsional stress in the fixed crankshaft bearings and shear stress in the movable bearings. The joints between the crankshaft bearings are also exposed to this force, and the most critical points in terms of stress are related to the connection of this part of the crankshaft with the movable bearings [10, 11]. In addition to the torsional torque required to the vehicle movement, another undesired effect is torsion and bending followed by the tension in the various parts of the crankshaft, which is considered as the important effect of vibrations. Since the value of stress caused by vibrations is significant, then studying and providing solutions to eliminate those vibrations have been considered by different studies. In order to deal with these problems, various methods have been proposed to estimate, model, and calculate the vibrations of the crankshaft in engines. The force on the piston is transmitted to the crankshaft through the connecting rod at the power stage. The reciprocating force of the connecting rod is transmitted to the crankshaft, and due to its large mass, it exerts a large force on the engine body. In addition, the performance of the valves knocks the engine body four times in every crankshaft rotation. In the fuel system, engine shocks cause fuel turbulence in the tank, which has high frequencies [12]. Therefore, it is necessary to study the common methods of vibration detection in more detail and use more advanced methods to investigate the problem. The rotational imbalance can be known as one of the main reasons for the crankshaft imbalance. The eccentricity of the crankshaft related to the axis of rotation as well as the weight of the end section of the connecting rod attached to the crankshaft, are two factors of crankshaft imbalance [13]. Typically, two methods are used to balance the crankshaft. In the first method, balancing weights are used against each crank (located directly on each movable bearing). The second method uses balance weights in the middle of the crankshaft. The second method is not very useful due to the presence of bending torque in the middle of the crankshaft [13].

One of the most powerful, widely used, and well-known software for simulating mechanical systems and motion analysis in industrial applications and even research centers is Automatic Dynamic Analysis of Mechanical System (Adams) Software [14]. Using this software can easily create a complex mechanism consisting of a number of rigid and flexible components with their connections, relative motion, with or without friction, can easily load them, and then display the full motion of the system in three dimensions [15]. It is also possible to calculate the force at the joints, position, speed, and acceleration of each component. The main feature of this software is to design and create a virtual test sample [16]. The software was presented about thirty years ago by a group of University of Michigan elites and to date has gained a significant position in the industry. Also, checking the control and vibration of the systems and the possibility of performing tests for flexible parts are among the unique features included in this software [17]. The software uses the method of Lagrange to obtain the equation of motion for a multiple body system [18].

$$\frac{d}{dt}\left(\frac{\partial T_p}{\partial q_i}\right)-\frac{\partial T_p}{\partial q_i}+\frac{\partial U}{\partial q_i}+\frac{\partial R_e}{\partial q_i}=Q_i \tag{1}$$

Where n refers to the number of degrees of freedoms of a multiple body system, qi refers to the generalized coordinates, Tp refers to the Kinetic energy of the system, U refers to the potential energy of the system, Re refers to the dissipative energy, and Q refers to the Generalized external force and moment for the qi coordinate. By use of the Lagrange method, a number of differential equations, equal to the number of degrees of freedom are derived. For general multiple body system problems, these equations are non-linear and require solutions via numerical integration methods. The ADAMS solver can be used to solve all problems of dynamic like shooting the ball, for bar mechanism and, etc. which is able to extract exact results as we solve analytically. However, this solver was developed to solve multibody systems which solving analytically is difficult and time-consuming. Then, it is quite a scientific solver to solve dynamic problems.

The main parts of this software include the following sections [19]:

Adams/view: It includes three-dimensional modeling, the definition of various constraints and joints, three-dimensional animation, display of forces, displacements, model stresses, and so on.

Adams/engine: It provides a special platform for modeling internal combustion engines. It is possible to quickly create and assemble subsystems related to an internal combustion engine such as cylinders, pistons, crankshafts, valves, etc., and finally to examine the performance of the system.

Adams/solver: It provides the numerical solution of the dynamic system designed using the Euler-Lagrange method. An interesting feature of this section is solving nonlinear equations using different solvers and the possibility of programming.

Mohammad Ranjbar Kohan (2011) conducted a study on the crankshaft tension of a four-cylinder 24Z gasoline-based engine. In this study, modeling and application of boundary conditions on the crankshaft were performed using Adams engine software and Nastaran finite element software was employed to analyze the crankshaft stress. It was concluded that the maximum stress was created at high gear speeds and reverse gear and it was at the fourth movable bearing fillet. Also, the number of crankshaft fatigue cycles was determined. Finally, the natural frequencies of the crankshaft in the form of different modes were extracted to study the phenomenon of resonance in the crankshaft by analyzing the free vibrations of the crankshaft by finite element software [20]. Yilmaz and Anlas (2009) investigated the effect of balance weight and position of the weight on the main bearing load and flexural stress of the crankshaft in a linear 6-cylinder diesel engine using Adams engine simulator software and in their analysis three modes of the rigid, beam, and 3D solid were used. Twelve balance weight combinations with zero-degree balance angle and eight balance weight combinations with 30-degree balance angle were considered at 0, 50 and 100% balance. It was found that with increasing balance, the maximum load on the main bearing and the flexural stress of the crankshaft increased and the average load on the main bearing decreased [21]. Combustion pressure is considered the main source of engine driving. Pressure in the cylinder is a function of engine speed, crankshaft angle, and engine load [22]. It also depends on other parameters such as the amount of fuel spraying, which is a function of engine rotation speed. A cylinder pressure curve is shown as a function of flywheel rotation (Figure 1) [23]. The torque is produced one time over two round rotations of the flywheel. This torque affects the crankshaft during one-third of a rotation [23]. The moment inertial of flywheel drives the engine for the remaining two-thirds of a rotation [24]. This process leads to high engine vibration and abnormal behavior. Having more cylinders and distributing their torque peaks in each cycle leads to smoother engine operation.

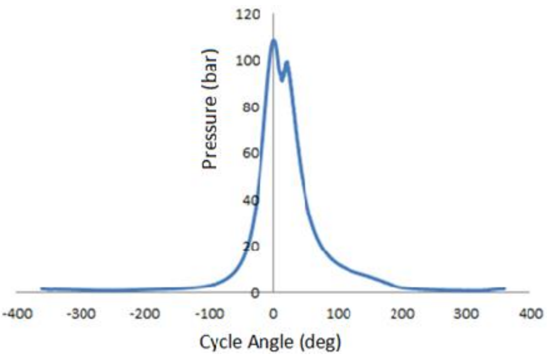

Figure 1
In-cylinder pressure in terms of crankshaft rotation

According to the statistics obtained from Iran Tractor Factory, the main products of agricultural tractors are MF-285 and MF-399 models. The highest production of the tractor factory in 1991 is related to the MF-285 (single differential) product with nearly 11,000 units per year. Low cost, small farms, high rain-fed lands in Iran, ease of use, low fuel consumption, and reasonable created power are the main reasons for the frequent usage and production of this tractor in Iran. A study on the statistics of breakdowns and repairs of various parts of the MF-285 tractor indicated that the internal parts of the engine were in second place after the injector pump [25]. The existence of this literature reveals the necessity of research on the rotating components of this tractor engine as well as the importance of vibration analysis for mechanical parts that are subjected to different loads. Dynamic simulation of the engine using ADAMS enables the researcher to extract exerted force and moments in each part in every moment. Also, by designing and optimization any of the parts we can see its effect on the system operation immediately and easily without any cost. Hence, the MF-285 tractor balancer analysis was selected as one of the internal rotating components of the tractor engine for the study in order to optimize balance and see its effect on engine vibration. The aims of this research are: (1) kinetic analysis of the crank mechanism by Adams engine software, and (2) analyzing experimental results and comparing them with software results.

2 Methodology

2.1 Experimental Procedure

The engine used in this research was a 4-cylinder, four-stroke diesel engine that was made in Iran Tractor Manufacturers Company (ITMCO). Table 1 presents the specifications of the tractor engine.

Table 1
Technical specifications of MF-285 tractor engine

Model	MTI440C-105AD, Made by Motorsazan Co. Iran
Fuel system	Direct injection
Number of cylinders	Four-cylinder
Standard of pollution	ECE-R96-Stage 2, ECE-R24
Compression ratio	17.5:1
Max. power	78 kW in 2200 rpm
Weight	380 kg
General dimensions	Length: 0.72 m, Width: 0.55 m, Height: 0.98 m
Air inlet system	Turbo charge and air to air cooler
Cooling system	Water cooling system
Max. torque	375 Nm in 1400 rpm

Data collection began after the preparation of all equipment as well as the vibration sensors. The experiments were conducted in ITMCO (Figure 2)

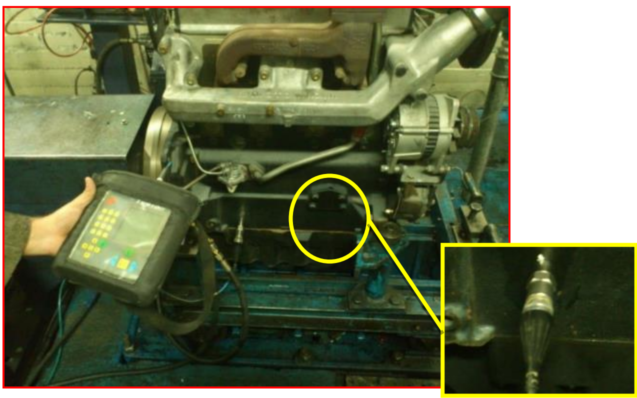

Figure 2
Connection and location of the Vibro-meter on the engine body

The engine is studied in full load and no-load conditions. In no-load mode, the vibration value was measured at 750 and 2400 rpm, then in full load mode, at 1400 and 2200 rpm. Finally, by keeping the rotational speed of 1400 rpm, the amount of load on the engine was halved to compare the vibration change of the engine in the balanced and unbalanced mode. Eddy Carnet dynamometer E400 made by Dynamic Micro Tools Industrial Development Company was used to measure torque which was attached to the flywheel. To collect the engine vibration signals, a vibrometer with Easy-Viber specifications equipped with a piezoelectric sensor model AC102-1A made by VMI (vibration measuring instruments) factory was used [26]. The vibration gauge was connected magnetically, such that there was no gap between the vibrometer and the engine body. The location of the vibrometer on the cylinder was at the location of the second fixed bearing of the crankshaft from the flywheel (ISO 8525-9). It was 17 cm from the frontside of the cylinder and 9 cm from the bottom of the cylinder (Figure 2).

2.2 Creation of balancer in Solid-Work 2018

The components of the balancer including balance weights, the balancer shaft, pin, spur gear, and the helical gear and the shell were created in SolidWorks 2018, respectively (Figure 3). By modeling the parts mounted on the balance shaft and assigning the density of the materials of each part, the weight of each of them was obtained in the following order where the weights of the balancer, shaft, gear, and oil pump are 1.66kg, 1.06kg, 0.42kg, and 0.082kg, simultaneously.

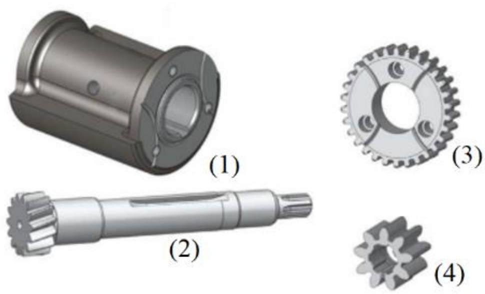

Figure 3
Weight of different parts of the balancer (1) balancer, (2) Shaft, (3) gear (4) oil pump

After creating balancing weights, balancing shafts were created. The balancer weight was attached to the shaft by a pin mounted on the shaft so that the weights would also rotate as the shafts rotate. Figure 4 shows the weights, shafts, roller bearings, helical and spur gears. All parts were assembled inside the balance case.

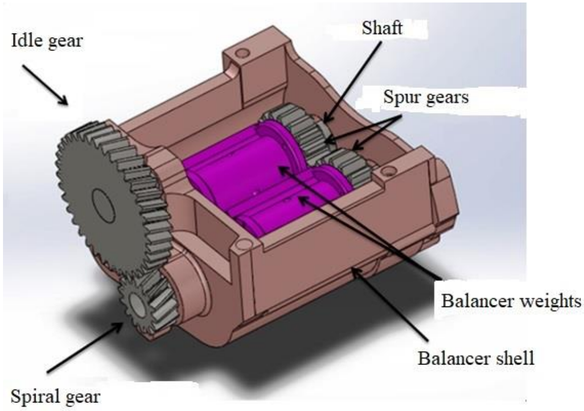

Figure 4
Final assembly of the balancer

2.3 Dynamic Design and Analysis in Adams Engine

In this research, the Adams engine software was used. The balance model created for dynamic analysis was then converted to *. MNF format, which is the Adams program input file for flexible components. At the first stage of building, the subsystem started. Figure 5 introduces the fixed and movable bearings. Information about the number of cylinders, the direction of engine rotation, the effective length of the connecting rod, the length and diameter of piston pin, piston boss spacing and the order of combustion, etc. were determined for the software.

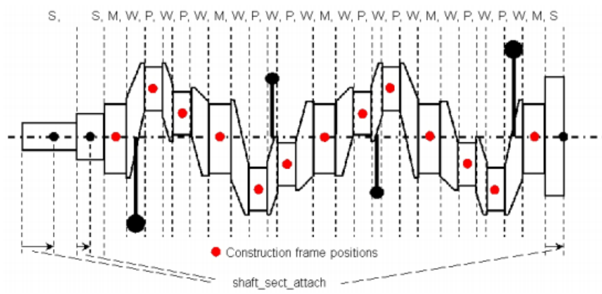

Figure 5
An example of introducing fixed and movable bearings to Adams engine software

The combustion engine analyzed in this study has characteristics such as effective length. Then all the components of the crank mechanism including the crankshaft, piston, connecting rod, pin, flywheel and fixed and movable bearings, fixed bearing diameter, fixed bearing length, movable bearing diameter and distance between movable bearing and fixed bearing center, elastic modulus, coefficient of elasticity, curvature modulus defined in the software. One of the obvious advantages of this software compared to other software is the ability to parameterize them and connect them with each other so that the user can apply more maneuvers in their design [27]. To make the crankshaft, the feature of 1/n of the crankshaft (n number of cylinders) was drawn introduced to the Adams software. To do this, the Sequence command was used. Hence, 1/4 of the crankshaft was drawn and defined as the basis and the rest of the parts and components were assembled in relation to it. The dimensions of the crankshaft are independent of its motion kinematics and the strength was the only determining factor in choosing the dimensions of that [13]. On the other hand, the balancer in the software could be defined in two ways, rigid and flexible. Despite the time-consuming nature of this analysis, the flexible model was used due to its approach to the real mode. To define the crankshaft balancer, the flexibility box option was selected from the definition of flexible elements. After defining all the parts, if there is a need to change or replace a part, the subsystem adjustment step was performed. For example, the rigidity or flexibility of a part is done at this stage. After defining all the parts around the balance shaft, its rotation speed was defined. At this stage, all the components of the crank mechanism, including the crankshaft, piston, connecting rod, pin, flywheel, and fixed and movable bearings were assembled (Figure 6).

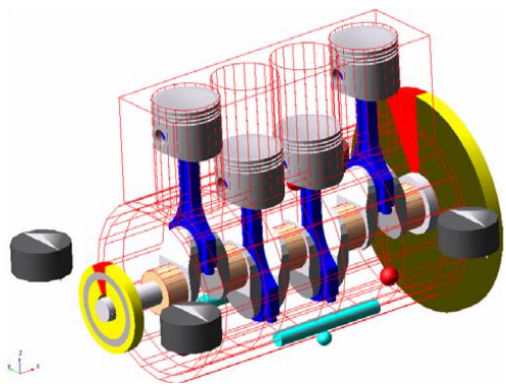

Figure 6
Simulated engine in Adams Engine Dynamic Analysis software

dynamic analysis of the engine based on the operating temperature and the effect of friction on the parts of the engine is one of the capabilities of using this software. The working temperature of the engine studied in this research, which was recorded in the test cell, and also other required data, recorded in the engine test cell of the Manufacturer Company, presented in Table 2, and these data inserted into the software. The software then automatically, by following these data, applies the engine operating temperatures based on the speed and torque applied to create a simulation close to the actual state.

Table 2
Operating temperature of the studied engine

Operational Condition	Load Zero, Rotational speed 600 rpm	Full load, Rotational speed 1400 rpm
Inlet air temperature to the air manifold (˚C)	15	63
Air temperature after compressor (˚C)	14	60
Engine inlet air temperature (ambient) (˚C)	15	64
Inlet air temperature to the intercooler (˚C)	-7	-28
The temperature of the air leaving the intercooler (˚C)	-4	-18
Engine water outlet temperature (˚C)	21.5	92.2
Inlet water temperature to the engine (˚C)	20	80
Oil temperature (˚C)	119	499.8
Fuel temperature (˚C)	14.5	63.1
Exhaust gas temperature (˚C)	-0.3	-0.7

One of the important parameters that are used to determine the forces acting on different parts of the crank mechanism is to consider the looseness at the joints, which has a significant impact on engine vibrations [26]. This looseness was defined in the software as a *.txt file. After applying all these modes, the engine is ready to define speed, load, etc. (Figure 7).

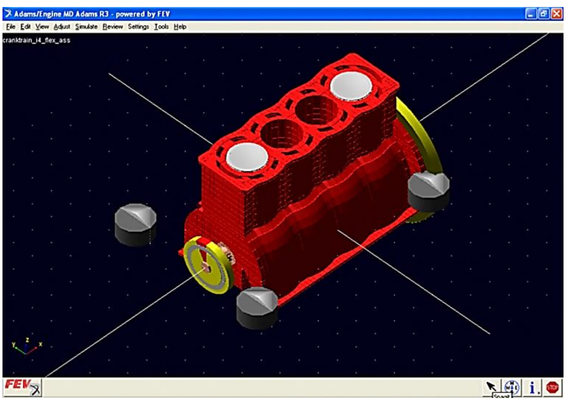

Figure 7
Final simulated engine using Adams Engine software

In this analysis, the engine is considered to be stable to approach the real state according to the practical test, different rotational speeds and different loads were applied to the engine in the simulation presented in Table 3.

Table 3
Different rotational speeds and loads entered in the software

Without balancer		With balancer	
Torque (Nm)	Rotational speed (rpm)	Torque (Nm)	Rotational speed (rpm)
0	750	0	750
0	2400	0	2400
125	1400	125	1400
250	1400	250	1400
250	2200	250	2200

Combustion pressure in the form of numerical data was separately inserted at different rotational speeds, in no-load (idle) and full-load modes, while the part-load was applied at a constant rotational speed. Then, the engine vibrations were completely simulated and extracted with and without the balancer in no-load and full-load modes. After this step, it was Adams solver's turn to solve the equations. This software solves the designed dynamic systems using the Euler-Lagrange method. After solving the equations, the defined results were extracted and displayed in the Adams/post processor environment. After dynamic analysis of the engine, diagrams related to kinematic and kinetic analysis of the crankshaft as well as engine vibrations were extracted as output results.

3 Results and Discussion

3.1 The Results of the Kinetic Analysis Section of the Crank Mechanism

Combustion pressure is considered the main driver of the system. The pressure inside the cylinder is a function of engine speed, crankshaft angle, and engine load (power or torque taken from the engine) [26]. It also depends on other parameters such as the ignition angle and the amount of fuel spray. Figure 8 shows the pressure change curves inside the 4, 3, 2, 1 cylinders in terms of the crankshaft angle. The maximum amount of pressure inside the cylinder was 140 bar.

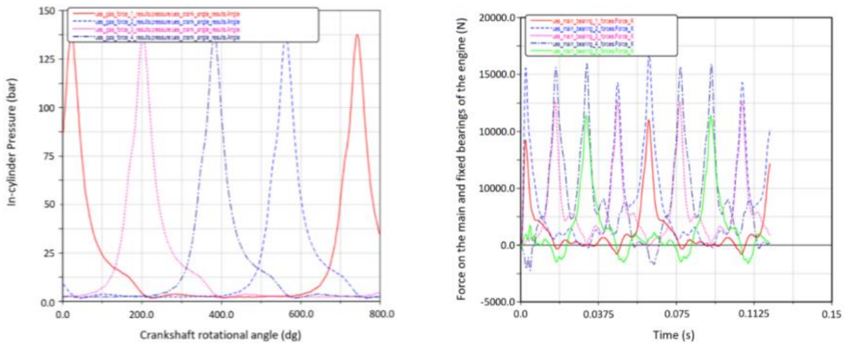

Figure 8
Pressure variations inside the 4, 3, 2, 1 cylinders according to the rotation angle of the crankshaft (left) force applied to the main and fixed bearings of the engine at a rotational speed of 2000 rpm over time (right)

Figure 8 shows the force applied to the main and fixed bearings of the motor at a rotational speed of 2000 rpm. The maximum force on the bearings was 16,950 N.

3.2 Root Mean Square (RMS) Value of Vibration

In terms of vibration, RMS is used to display the engine vibration by three parameters of vibrational displacement signal, vibration velocity signal, and vibration acceleration signal [28]. In this research, the RMS vibration velocity signal was used to analyze the vibration in m/s. Figure 10 shows an example of the results of a motor vibration signal at a rotational speed of 1400 rpm and a load of 125 Nm in the time domain for experimental and simulated results, respectively.

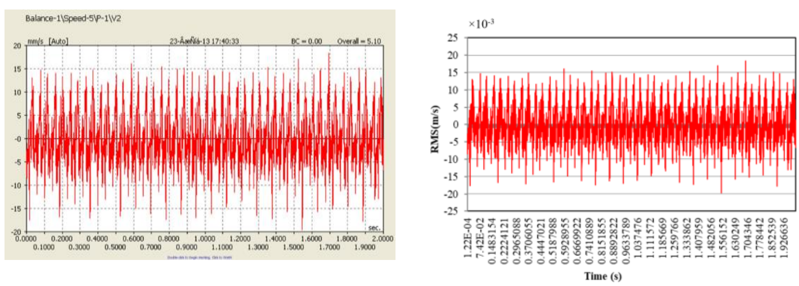

Figure 10
Vibration velocity signal obtained from the experiment (Left) and simulation (right) at a rotational speed of 1400 rpm and a load of 125 Nm

3.3 Results of No-Load Condition

Figure 11 shows that using the balancer reduced the amount of RMS of vibrations by 40.46% at 750 rpm in no-load mode, and reduced the RMS value by 12.73% at the rotational speed of 2400 rpm. Accordingly, using the balancer in no-load mode was more effective at lower speeds. In the case of no-load mode, the use of a balancer reduced vibrations by 28.83%.

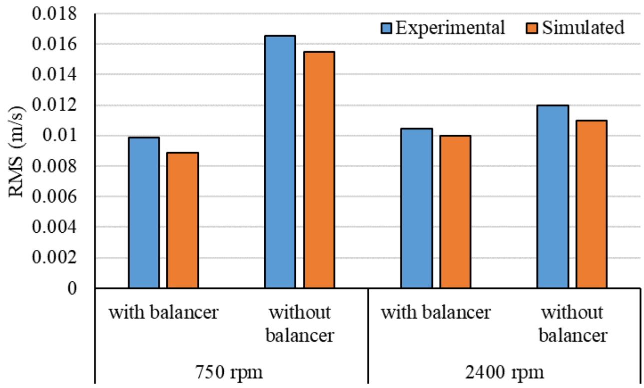

Figure 11
Experimental and software results in no-load condition at 750 and 2400 rpm

According to Figure 11 in no-load mode both at 750 and 2400 rpm in both balanced and unbalanced modes, experimental and simulated by software results were highly consistent with experimental results, which confirms the validity of the created simulation. At all tests, experimental vibration results were a little more than simulation results. It was because of not considering valves, camshaft, cylinder head, and its components that are effective in engine vibration.

3.4 Vibration Performance of the Engine at 1400 rpm and Different Loads

Figure 12 shows that the motor operates in both balanced and unbalanced modes at 250 and 125 Nm of torque and at a constant rotational speed of 1400 rpm. At torque of 250 Nm using balancer decreased the RMS vibration by about 21.33%. Therefore, using a balancer is more effective at minimum torques. Accordingly, the use of the balancer reduced the vibrations by about 29.65%. It is also observed that with increasing load, the number of engine vibrations increased. According to Figure 12 at full load and 1400 rpm in both balanced and unbalanced modes, the laboratory and software results are highly consistent so that the results of laboratory tests, the results of simulations confirm engine dynamics.

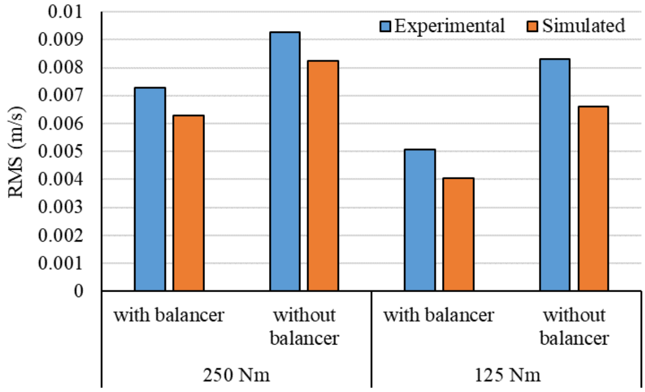

Figure 12
Results of balancer performance underload of 250 and 125 Nm and rotation speed of 1400 rpm

3.5 Balancer Operation under Full Load at Different Rotational Speeds

Figure 13 shows that the engine operates in both balanced and unbalanced modes under a full-load of 250 Nm, at rotational speeds of 1400 rpm and 2200 rpm.

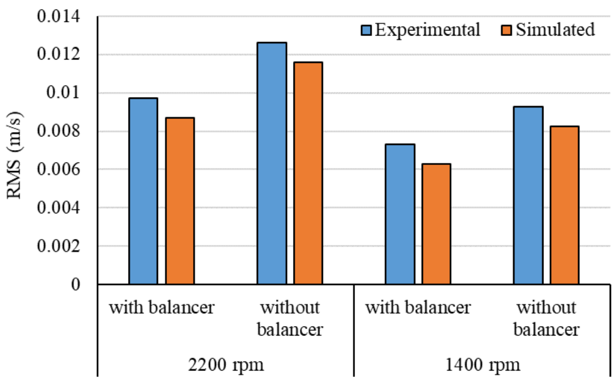

Figure 13
Results of balancer operation under full load (250 Nm) at two rotational speeds of 2200 and 1400 rpm

Under full load, the balancer reduces the RMS value of vibration by 21.31% at 1400 rpm and also reduces the RMS value by 23.05% at 2200 rpm. On average, the use of balancer reduced vibrations by 22.43%. Therefore, using a balancer at full load was more effective at higher rotational speeds. Figure 21 shows that at full-load conditions, the RMS of the engine vibrations increased with increasing engine speed. Taghizadeh-Alisaraei (2017) has achieved a similar result in this regard. The increase in vibration with rising speed can be due to the increase in combustion intensity and pressure inside the cylinder because as the speed increases and more fuel is injected into the combustion chamber, which is followed by an increase in combustion energy causing more force to the crankshaft and engine body and ultimate increment of vibration [28]. For the future research using advanced machine learning methods are suggested, e.g., [29-32]. Optimization methods and mathematical modeling are a core part of the current information systems for advancing machine learning methods which should be used in the future research [33-34].

Conclusions

Using a balancer reduced the amount of RMS vibrations at 750 rpm in no-load mode by 40.46% and 12.73% at the rotational speed of 2400 rpm. In a no-load mode, both at 750 and 2400 rpm in both balanced and unbalanced modes, experimental and simulated by software results were highly consistent with experimental results, which confirms the validity of the created simulation. At a torque of 250 Nm, using the balancer decreased the RMS vibration by about 21.33%. It was found that with increasing load, the number of engine vibrations increased. Increased vibration with load increment can be due to increased intensity and pressure of combustion inside the cylinder. With increasing rotational speed from 1400 to 2200, the value of vibration with and without balancer increased. Under the full-load conditions, the balancer reduces the RMS value of vibration by about 21.31% at 1400 rpm and also reduces the RMS value by about 23.05% at 2200 rpm. Therefore, using a balancer at full-load was more effective at higher rotational speeds. In full-load conditions, the RMS of the engine vibrations increased with increasing engine speed. It was concluded for boating exact results from simulation whole parts of the engine like the camshaft, the cylinder head, the water pump should be considered in future research to show their effect on the vibration of the engine.

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