A simplified method of dynamic balancing for automobile wheel-tire assembly

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Abstract. The method of balancing the wheel tire assembly, so that it rotates smoothly at high speed is known as wheel balancing. The most common type of equipment is called a dynamic balancer, commonly referred to as a "spin balance" It detects dynamic balance by mounting a tire on a test wheel, rotating the assembly up to 300 rpm or more, and then evaluating the forces of unbalance as the tire rotates. Dynamic balancers identify the exact amount of counterweight which needs to be applied to correct an unbalance as well as to determine its position. The technique of unbalance identification for the vehicle wheel tire assembly is addressed in this study. A specific technique is discussed for calibration of wheel balancer machine and a means of identification is proposed in order to identify the location and magnitude of the unbalance. The robustness of the technique is also validated with a specific case of wheel tire assembly balancing. The vehicle wheel balancer machine producer can easily implement the method discussed in the present investigation to build up the balancer machine.

1 Introduction

A vehicle's wheels must be perfectly balanced and as closely aligned with one another as feasible. These dynamic forces are activated if they are not appropriately balanced. These forces result in increased bearing load, stress on different vehicle components, and uncomfortable and hazardous vibrations in those components. All of a vehicle's wheels must be precisely balanced; otherwise, there would be wheel bubbling, which will affect driving control. One of the most frequent causes of unanticipated vibration, non-smooth movements and even instability in mechanical systems is unbalance [1]. Vehicle safety is significantly impacted by wheel tire assembly balancing [2-3]. Tires which are unbalanced can wear unevenly and need more frequent replacement. As modern cars and light-duty trucks, are extremely tuned vehicles, a slight unbalance can have a detrimental impact on anticipated performance, driver comfort, fuel economy, and tire life [4]. Since unbalance is a vector that contains information about its orientation and amplitude, the goal of wheel balancing system is to determine the precise position of the unbalance and determine its magnitude, respectively [5]. Luciano Paiva Ponci [6] presented streamlined process for dynamic balancing and vibration analysis in systems exposed to beat frequency. The

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outcomes demonstrate the good efficiency of the process and there are noticeable advantages in terms of cost, ease of use, and speed, particularly when contrasted with other kinds of more intricate methods that are currently being employed to address related issues. Tire condition monitoring systems (TCMS) are safety devices that measure the temperature, pressure, balancing, and other aspects of a tire in a car. These technologies are extremely important for safety in this day [7]. Zhan Wang et al. [8] demonstrated how an optimization approach can significantly lower vibration amplitude and increase dynamic balance efficiency, which offers the theoretical foundation required to increase the spindle system's running precision. Joshua E. Siegel et al. [9] showed that wheel tire assembly imbalance in a car can be effectively detected using accelerometer data from a smartphone installed on the dashboard. While the car is being driven normally, a system that monitors wheel balance, damper state, and tire pressure is suggested. Each wheel-axle station's vertical vibration is measured as the foundation of the system. The driver can view dashboard indications of tire pressures, damper effectiveness, and wheel unbalance by means of subsequent signal processing [10]. For the purpose of high-speed machine tool spindle balancing, a new pneumatic double-face online dynamic device together with a control system have been developed [11]. Aneesh Nabar [5] studied several operational factors that could influence the wheel balance measurement. Uneven wheel geometry and rim and tire imbalance are typical issues in cars. The consequences can range from a small inconvenience for the driver because of a poorer ride to serious problems with the vehicle's operation and a shorter lifespan for expensive or safety-critical parts [12,13]. Unintentional vibration on suspension components is caused by bent or unbalanced wheels, which can rattle a car and its contents, put loads on vehicle components in unexpected ways. These factors make wheel tire assembly balance monitoring essential to the safe operation and appropriate long-term maintenance plan of any wheeled vehicle, especially those that are lightweight and run at high sustained speeds, like many present passenger cars.

2 Tire/wheel dynamic balancing machine

Even though wheel tire assembly balancing is a relatively straightforward service. With numerous automated and computer-generated features intended to provide outstanding balance, modern wheel tire assembly balancers are far simpler to operate than older models. Direct drive motors, several balancing modes, laser guidance, automated starting with a cycle of second, weight storage bins, and automatic static balancing are features found in many contemporary wheel tire assembly balancers. The following are the three fundamental instances to balance: When a balance weight is moved or comes loose; When a tire is changed or fixed; When new tires are bought.

A rotating shaft of a wheel tire assembly balancer as shown in Figure 1, which has a forces transducer attached to reliantly supported bearings for the shaft, is usually used to mount the combined wheel tire assembly. These forces on the shaft bearings measured by the force transducers are used to determines the amount of unbalance in the wheel and tire assembly. To determine the position of the unbalance force, encoders are incorporated; the machine uses this data to determine where the correction weight should be placed. Typically, wheel balancing machines use piezoelectric quartz load sensors or strain gauge-based force transducers to measure the forces and moments induced by unbalance, and rotary encoders, also known as shaft encoders, to monitor wheel speed and angular position. By attaching corrective balancing weights to the wheel tire assembly, a wheel balancing system lowers vibrations in out-of-balance wheel tire assembly.

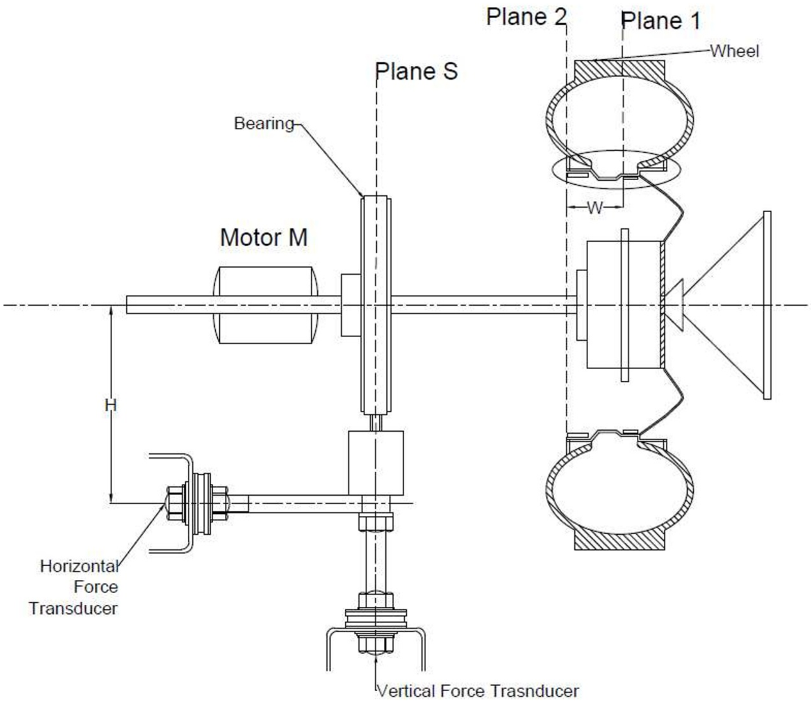

Fig.1. Schematic diagram of wheel balancer machine

3 Calibration of force transducer of balancer machine

The free body diagram of the force operating on the wheel balancer's spindle shaft is shown in Figure 2. When a wheel tire assembly has unbalanced mass and rotates around a fixed axis, centrifugal force is generated. In order to dynamically balance the wheel tire assembly, two planes are identified as plane1 and plane 2, to place the correction masses on the rim of wheel. The forces owing to unbalance masses in wheel tire assembly produces the forces and moment in the plane S. The vertical force transducer is mounted in plane S in the radial direction for measurement of the resultant force due to unbalance masses in the radial direction. The horizontal force transducer is generally mounted in the direction perpendicular to plane S and at an offset distance H from the shaft axis as shown in Figure 1. The horizontal force transducer measures the moment due to centrifugal forces generated by the unbalance masses.

It is suitable to calibrate the force transducers in the balancer machine during the manufacturing of the machine. After mounting the force transducers in the balancer machine, the known masses m1 and m2 can be placed in the plane 1 and plane 2. These known masses will introduce forces in the vertical and horizontal force transducer. The force magnitude and direction can be estimated mathematically as follows :

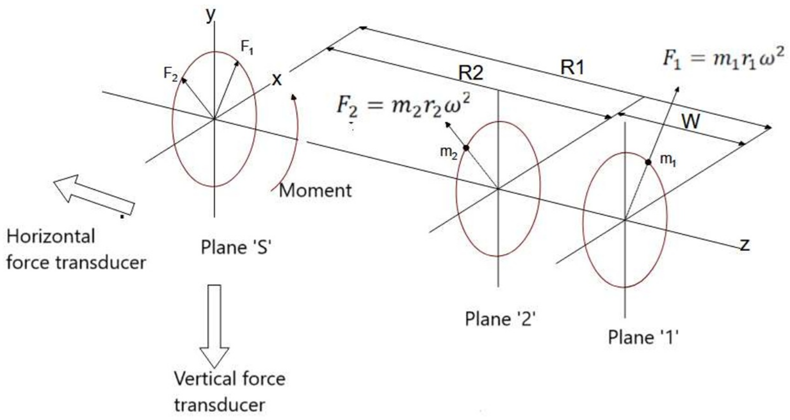

Fig. 2. Free body diagram of spindle shaft for determination of forces in transducer

The centrifugal forces due to known masses placed in plane 1 and plane 2 are $F_1 = m_1 r_1 \omega^2 \angle\theta_1$ and $F_2 = m_2 r_2 \omega^2 \angle\theta_2$ respectively. When these forces are transferred in the plane 'S'give force and moment. The force components in the *x* and *y*-direction in the plane 'S'are $F_x = m_1 r_1 \omega^2 \cos\theta_1 + m_2 r_2 \omega^2 \cos\theta_2$ and $F_y = m_1 r_1 \omega^2 \sin\theta_1 + m_2 r_2 \omega^2 \sin\theta_2$ respectively. The resultant radial force in the plane 'S'due to known masses in plane 1 and plane 2 is as mentioned in equation (1):

$$F_V = [(m_1 r_1 \omega^2 \cos\theta_1 + m_2 r_2 \omega^2 \cos\theta_2)^2 + (m_1 r_1 \omega^2 \sin\theta_1 + m_2 r_2 \omega^2 \sin\theta_2)^2]^{\frac{1}{2}} \text{.......(1)}$$

The resultant radial force is denoted as F_V as it being measured by the vertical force transducer. The moment due to the known masses placed in plane 1 and plane 2 give the moment as mentioned by equations 2 and 3 as follows :

$$M_1 = [-(m_1 r_1 \omega^2 \sin\theta_1)(R_1)]\hat{\imath} + [(m_1 r_1 \omega^2 \cos\theta_1)(R_1)]\hat{\jmath} \text{ ....(2)}$$

$$M_2 = [-(m_2 r_2 \omega^2 \sin\theta_2)(R_2)]\hat{\imath} + [(m_2 r_2 \omega^2 \cos\theta_2)(R_2)]\hat{\jmath} \text{ ....(3)}$$

The resultant moment due to above moments M_1 and M_2 is given in equation (4) as follows:

$$M = \left[\left\{\left(-(m_1 r_1 \omega^2 \sin\theta_1)(R_1)\right) + \left(-(m_2 r_2 \omega^2 \sin\theta_2)(R_2)\right)\right\}^2 + \left\{\left((m_1 r_1 \omega^2 \cos\theta_1)(R_1)\right) + \left((m_2 r_2 \omega^2 \cos\theta_2)(R_2)\right)\right\}^2\right]^{\frac{1}{2}} \text{..(4)}$$

The moment is measure by the force transducer placed in horizontal position at a distance H from the spindle shaft axis. Hence, the force F_H measured by the horizontal force transducer equals to

$$F_H = \frac{M}{H} \qquad (5)$$

The angle θ_H at which this force F_H acts can be determined as follows:

$$\theta_H = \frac{\{((m_1 r_1 \omega^2 \cos\theta_1)(R_1)) + ((m_2 r_2 \omega^2 \cos\theta_2)(R_2))\}}{\{(-(m_1 r_1 \omega^2 \sin\theta_1)(R_1)) + (-(m_2 r_2 \omega^2 \sin\theta_2)(R_2))\}} \tag{6}$$

Further, as the force and the moment will be at 90°, the actual angle as indicated by the transducer will be $\theta_H + 90°$. Based on the above mathematical formulation a case is taken as shown below in table 1 to explain the calibration of the both transducer at the time of manufacturing of balancer machine. Table 2 and 3 shows the calculated value of the F_V and F_H as discussed.

Table 1. Input parameter for calibration of transducers

Rotation speed (ω) in rpm	250
Tyre width (W) in meter	0.12
Distance from shaft axis to horizontal sensor (H) in meter	0.15
Location of known mass in Plane 1 for calibration of transducer	
Mass (m_1) in kg	0.05
Angle (θ_1) in degrees	60°
Radius at which known mass is attached (r_1) in meter	0.18
Distance from plane S to plane 1 (R_1) in meter	0.42
Location of known mass in Plane 2 for calibration of transducer	
Mass (m_2) kg	0.03
Angle (θ_2) in degrees	130°
Radius at which known mass is attached (r_2) in meter	0.2
Distance from plane S to plane 2 (R_2) in meter	0.3

Table 2. Calculation of force F_V

Force	Magnitude (N)	X-component	Y-component	Angle (Degree)
F_1	6.173469388	3.086734694	5.346381319	60°
F_2	4.115646259	-2.645486421	3.15	130°
F_V	8.510595647	0.441248273	8.499149265	87.03°

Table 3. Circulation of force F_H

Moment/Force	Magnitude	X component	Y- component	Angle (Degree)
Moment M1	2.592857143	-2.245480154	1.296428571	150°
Moment M2	1.234693878	-0.945830384	-0.793645926	-140°
Moment M	3.230673821	-3.191310538	0.502782645	171.04°
Force in horizontal sensor F_H (in Newton)				
Force F_H=M/H	21.53782547	-3.351884301	4.55612602	261.04°

Hence, the transducers can be calibrated by calculating F_H and F_V with the different value of m_1 and m_2. These known forces will produce the voltage in force transducer. The calibration plot can be obtained between the force (N) and voltage (mv).

4 Dynamic balancing of wheel-tire assembly

In order to find the correction mass required for balancing the wheel tire assembly, the input is from the measured value of forces from the force transducers that are F_H and F_V as discussed in previous section. Using the condition of equilibrium for the forces and moment the correction mass required in plane 1 (m_{1c}) and plane 2 (m_{2c}) can be obtained (refer fig.3).

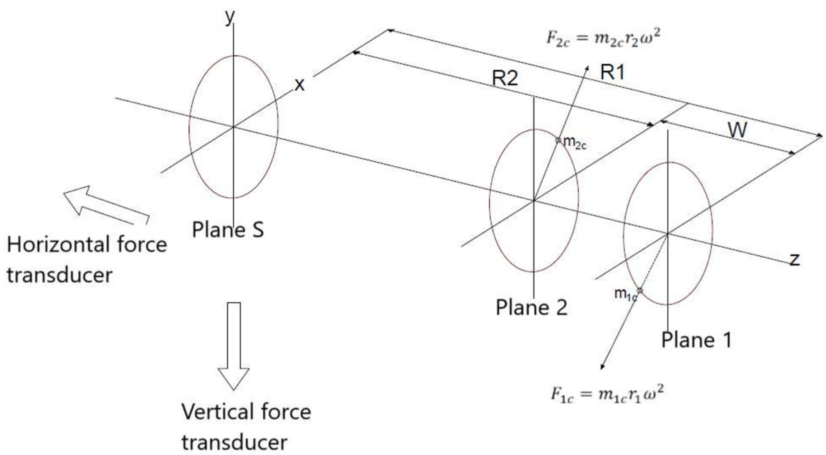

Figure 3. Free body diagram of spindle shaft for determination of correction masses

The spindle shaft force balance equation (7) is as follows :

$$F_{1c}\angle\theta_{1c} + F_{2c}\angle\theta_{2c} + F_H\angle\theta_H + F_V\angle\theta_V = 0 \tag{7}$$

In equation (7), there are four unknowns $F_{1c}, \theta_{1c}, F_{2c}$ and θ_{2c} which cannot be determined the from equation (7). The moment balance equation considering the plane 1 as reference plane can be written as follows :

$$(R_1 - R_2)XF_{2c} + r_1XF_V + HXF_H = 0 \tag{8}$$

In the equations above, the encoder provides the angular position values θ_H and θ_V, while the force transducers provide the values F_H and F_V. The aforementioned equation (8) is used to first compute the force F_{2c} and then using equation (7) F_{1c} can be obtained. From F_{1c} and F_{2c} the masses m_{1c} and m_{2c} required for balancing the wheel tire assembly can be obtained.

Table 4. Input parameter considering a specific case of wheel tire assembly unbalance

Reading from Vertical sensor	
Force magnitude (N)	8.510595647
Angle (Degrees)	87.02805712
Reading from Horizontal sensor	
Force magnitude (N)	21.53782547
Angle (Degrees)	261.0467919

Considering the particular case when $R_1 = 0.42$m and $R_2 = 0.3$m. Considering values of $H = 0.15$ m, $r_1 = 0.18$ m, $r_2 = 0.2$ m and tire width $W = 0.12$ m. The Angular velocity of wheel balancer shaft is 26.19 rad/s. The input parameter values of forces F_H and F_V as obtain based method discussed in previous section is depicted in Table 4.

Table 5. Determination of unbalance for the wheel tire assembly

					Distance from reference plane 1			Moment in N-m (Reference plane1)	
Part	Angle in Degree	Unbalance (N)	X Comp. force (N)	Y Comp. force (N)	Rx (m)	Ry (m)	Rz (m)	X-component	Y-component
Plane1	-120	6.17	-3.08	-5.34	0	0	0	0	0
Vertical sensor	87.02	8.51	0.44	8.49	0	0	-0.42	3.57	-0.18
Horizontal sensor	261.0	21.5	0	0	-0.02	-0.15	0	-3.19	0.50
Plane2	-50	4.1	2.64	-3.15	0	0	-0.12	-0.37	-0.31
			3.08	5.34				0	0
	Unbalance (kg.m)		Correction Angle (Degree)		Radius at which correction to be done (m_c)				Correction mass (kg)
At plane 1	0.009007246		-120		0.18				0.05
At plane 2	0.006004831		310		0.20				0.03

In order to verify the accuracy of the procedure, the wheel tire assembly is first unbalanced by purposefully placing the known masses at particular angular positions in planes 1 and 2. The magnitude of known masses placed in plane 1 and plane 2 are 0.05 kg and 0.03 kg as mentioned in Table 1. In actual balancer machine the encoder records the angular position and the force transducer records the forces acting on the bearings. The correction masses m_{1c} and m_{2c}, as well as their locations, are then calculated using the aforementioned equations (7) and (8). The calculated value of m_{1c} and m_{2c} as observed from the table 5 are 0.05 and 0.03 respectively which exactly matches with the known masses placed in the plane 1 and 2 to unbalance the tire-wheel assemblies. This shows that the method discussed in the present work accurately determines the correction masses.

5 Conclusions

A simplified method of balancing the wheel tire assembly on dynamic wheel balancer machine is discussed in the current work. The mathematical formulation for calibration of force transducer in the balancer machine and determination of unbalance is discussed in detail. A specific case is taken as an example to validate the mathematical formulation. It is observed that correction masses required for dynamic balance of wheel tire assembly exactly matches with the known mass placed in the same plane to unbalance the assembly. This shows the robustness of the method to dynamically balance the wheel tire assembly. The vehicle wheel balancer machine producer can easily implement the approach discussed in the present investigation.

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