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Automatic wheel balancer for automobile wheel

Frederick Eugene Nevin

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AUTOMATIC WHEEL BALANCER
FOR AUTOMOBILE WHEEL

BY
FREDERICK E. KEVIN

A
THESIS

Submitted to the faculty of the
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Automobiles of today have been so constructed that greater speeds under safer conditions now exist. One of the improvements has been the low pressure tire. These tires carrying less pressure necessitates a casing of greater volume. This has created a tire of greater thickness in the axial direction but at the same time has decreased the diameter. Because of these innovations in present-day tires the author believes the wheel assembly (tire, tube, wheel, and brake drum) should be considered as a rotating cylinder not as a rotating disk. This being the case, the condition of dynamic unbalance is more critical.

This investigation deals with the application of a balancing head to be attached permanently to a wheel assembly to determine the feasibility of dynamic balancing of the unit under operating conditions.
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The investigation of an automatic balancing device to maintain an automobile wheel in balance under operating conditions is of importance. A tire mounted on an automobile wheel which travels at today's high speeds becomes a part of a rapidly turning mass. Wheel, brake drum, hub cap, hub, wheel mounting studs or bolts, and tire revolve as a unit. This turning mass revolves at a speed sometimes reaching 1000 revolutions per minute. At these higher speeds centrifugal forces increase, and any condition of unbalance of the revolving mass of steel and rubber creates dangerous and destructive vibrations.

Due to the fact that the two front wheels are interconnected by the steering linkages, it is of vital importance that the front wheels' assemblies be maintained in smooth running balance. If they are not in balance the undue shock on the linkage will shorten their life, to say nothing of the tiresome affect on the driver of the car, thus causing unsafe driving conditions.

The tires have a vital part in the unbalance of the wheel assembly. Most tires when in the process of manufacture may become warped or out of round due to improper curing. This warped or out-of-round condition creates a static unbalance. This unbalance, and also the out of round of brake drums with no other factors interfering will cause one flat spot to be worn in the tire tread.

1 Wheel and Steering Alignment Technical Manual 2nd Ed. Lansing Michigan, John Bean Mfg. Co. 1946 p. 91
Several attempts have been made to make automatic balancing machines. One was devised by the Wm. Sellers & Co. Inc., Philadelphia, Pa. sometime previous to 1904. This balancer employed three eccentric disks indicated by 1, 2, and 3 as shown in figure 1. These disks are eccentrically mounted on a shaft as is also disk D, the disk to be balanced. The fit between the eccentric disc 1, 2, and 3 is such that there is some friction between them and the shaft but not enough to prevent the eccentrics from moving under the influence of the dynamic forces produced by rotation. As the disk D rotates the eccentrics 1, 2, and 3 move from the original position of 120 degrees apart to a position where they dampen out any unbalance.

In 1913 the French engineer, LeBlanc, proposed the use of a hollow cylinder partially filled with mercury. He thought that when the shaft was whirling the mercury would take the form of a cylinder within a cylinder. Below the critical speed the mercury caused greater unbalance, but above the critical speed the light side moved away from the axis of rotation and the mercury moved toward the light side thus equalizing the forces and balancing the rotor. Theoretically it seemed that it should work, but practically it did not.
Regardless of the original causes of spotty tread wear on either front tire no alignment or balance job, however perfect, can prevent future excessive wear of the spots since once a front tire acquires a flat or cupped spot accelerated wear will result.

In no manufacturing process can the distribution of rubber be uniform throughout the tread. Nonuniformity necessitates balancing the tire after manufacture. This is accomplished by the insertion of a rubber balancing strip on the inside of the casing. This does not necessarily mean the tire is in balance, because of the human element on the part of the worker. With the balancing strip in place, static balance is generally accomplished, but does not necessarily mean that dynamic balance is also accomplished.

As stated above out of round drums produce an unbalance force. This unbalance with the inherent unbalance in the tire and tube creates a condition which must be corrected in order to have a smooth rotating unit.

The study of an automatic wheel-balancing device was chosen as a subject for a graduate thesis while the author was employed as a tire salesman. During that period as a salesman many problems concerning tire wear confronted the author. Abnormal wear of the tires still remained even though the alignment of the wheels was found to be in proper adjustment. This led to determining the effects of unbalance. It was found that balancing, for both static and dynamic conditions, removed the abnormal wear on the tires. At that time the author became interested in a dynamic-balancing machine that could be built into or attached permanently to an automobile wheel.
Figure 1

Excentric disks dampner
Others have tried using low melting point metals, such as solder and babbitt, in suitable balancing grooves. The liquid metal supposedly automatically finds its proper place under actual operating conditions in the grooves, and solidifies thus balancing the unit.

The use of slip washers or rings have also been used in attempts of achieving automatic balancing.

The above mentioned methods of purely mechanical balance have heretofore proven unsatisfactory.

In 1930 E. L. Thearl of General Electric Companies Research Laboratory devised a mechanical balancer which has proven quite efficient. The author has applied this balancer to this research problem.

A rotor can be statically in perfect balance, but still be dynamically out. It can be in perfect dynamic balance, but still be statically out. Or it can be out both statically and dynamically.¹

¹ Davey, Peter. Portable Dynamic-Balancing Outfit Iron Age, Vol. 123 No. 9 pp. 610-12 (1929)

It can be shown that any rotor may be balanced statically as well as dynamically by the addition or subtraction of two weights one in each of two arbitrary transverse planes.

In any given rotor the size and location of unbalance is unknown. They can be determined by placing the rotor in a dynamic balancing machine such as is shown in figure 2. The rotor is placed in two bearing rigidly attached to the table T supported on springs. The table may be rotated about either one of two fulcrum axis F₁ or F₂ located in the correction planes A and B.
Figure 2
This removes the mass of the driving motor from the unit thus eliminating undesirable weight on the flexible mounting.

In the process of balancing, the rotor is placed in the bearings and a fulcrum made at $F_1$ by releasing $F_2$. The rotor is driven by belt or flexible shaft until it together with the table vibrates in resonance with the springs.

The simplest method of determining the location of the two correction weights is by the phase-angle relationship. A pencil, chalk, or scribe, is held close to the whirling rotor shaft. When the shaft is rotating, below resonance the heavy side of the shaft will be marked. Above resonance the light side will be marked, but at resonance the mark trails the unbalance force by approximately 90 degrees. By the addition of a proper weight in the proper place as indicated by the scribe in plane $E$, the rotor may be rotated without vibrating the table.

The fulcrum at $F_1$ is then released and $F_2$ is made the fulcrum. The above procedure is then repeated to determine the balance weight necessary in plane $A$. 5


Figure 3 shows the suspension of the front wheel of an automobile. The wheel assembly (tire, tube, wheel, hub cap, and brake drum) is to be considered the rotor to be balanced. The fulcrum is at pin $A$. The rotation of the unit is caused by the contact of the wheel with the road, or in case the car is stationary the wheel is jacked up and rotated with a spinner motor.
Front wheel suspension

with pivot at A

Figure 3
\[
\frac{\omega r \omega^*}{g}
\]
For the purpose of analysis the wheel assembly to be balanced may be replaced by the approximately equivalent system shown in figure 1. It is known that any general case of unbalance may be corrected by the application of two weights, in two arbitrary correction planes. These two weights determine the magnitude of the unbalance.

In figure 4 the pivot point O is placed in the unbalance plane u. The unbalance weights u and w are independent of the rotor, but their respective actions on the rotor are shown as external forces. Consider the rotor in figure 4 as being in a general position displaced from the rest positions through the angles $\alpha$ and $\beta$.

$\alpha$ being measured in the x or horizontal direction, $\beta$ in the y or vertical direction. The rotation of the rotor is assumed to be constant. The path of the point S at the center of the shaft is described by the equations $x = L\alpha$ and $y = L\beta$.

Let $\mu$ = a damping factor

$g$ = acceleration due to gravity

$I$ = moment of inertia of rotor and suspension unit about either the x or y axis, assumed to be the same through any diametral axis through O

$j$ = polar moment of inertia about the axis of rotation

$k$ = spring constant of equivalent supporting springs.

They are assumed to be equal

$\omega$ = rotative speed of rotor, assumed to be constant

$\rho$ = radius of gyration of rotor

$\phi$ = phase angle by which the unbalance force trails unbalance mass
All external forces exerted on the rotor, except that due to gravity, are represented by vectors in figure 4 (a). The omission of the forces of gravity is due to the fact that it can be shown to have no effect on the motion of the rotor.

The forces exerted on the rotor by the masses u and w are represented by \( \frac{urw}{g} \) and \( \frac{wrw}{g} \). The reactions at O are indicated by the vectors Ax and Ay. The resultants of all spring reactions and damping forces, or friction forces are shown by Fx and Fy. The magnitude of these vectors may be expressed by

\[
Fx = k_1 \alpha + \mu l \frac{d\alpha}{dt} 
\]

\[
Fy = k_2 \beta + \mu l \frac{d\beta}{dt} 
\]

Looking at figure 5 the three components of angular momentum of the suspension unit, are shown with respect to their corresponding axis. They are \( I \frac{d\alpha}{dt} \), \( I \frac{d\beta}{dt} \), and \( J\omega \), when the axis lies along axis O-P. As the rotor moves to a new position the angular displacements are \( \alpha + \Delta \alpha \) and \( \beta + \Delta \beta \) then the component of angular momentum are \( I \frac{d\alpha}{dt} \), \( I \frac{d\beta}{dt} \), \( J\omega \), and \( J\omega \). The rates of change of angular momentum then becomes \( I \frac{d^2\alpha}{dt^2} \), \( I \frac{d^2\beta}{dt^2} \), \( J\omega \), and \( J\omega \). These rates of change are in the same direction as the changes. These are shown as vectors in figure 6. They are couple vectors while in fig. 5 force vectors are shown. These force vectors as shown in 5 may be equated to the rates of change of angular momentum with respect to the axis. This must hold true because for every reaction there must be an equal and opposite reaction. (Newtons 3rd Law) Therefore,
Coupled vectors acting on system

Figure 6
Substituting (1) and (2) in (3) and (4) and rearranging

\[ I \frac{d^2 x}{dt^2} + \omega^2 \beta \frac{dx}{dt} + k \frac{dx}{dt} + \omega^2 x = \frac{\text{ar} \omega^2 \sin \omega t}{g}, \quad (5) \]

\[ I \frac{d^2 \theta}{dt^2} + \omega^2 \alpha \frac{d\theta}{dt} + k \frac{d\theta}{dt} + \omega^2 \theta = \frac{\text{ar} \omega^2 \cos \omega t}{g}, \quad (6) \]

Taking only the forced vibrations into consideration, this pair of simultaneous equations may be solved where

\[ \alpha = A \cos \omega t + B \sin \omega t, \quad (7) \]

\[ \beta = A \sin \omega t - B \cos \omega t, \quad (8) \]

solving for A and B

\[ A = \frac{\text{ar} \omega^2 \left[k \ell^2 - (I - J) \omega^2 \right]}{g \left[k \ell^2 - (I - J) \omega^2 \right] + m^2 \ell^2 \omega^2}, \quad (9) \]

and

\[ B = \frac{\text{ar} \omega^2 \ell \alpha}{g \left[k \ell^2 - (I - J) \omega^2 \right] + m^2 \ell^2 \omega^2}, \quad (10) \]

At any instant of time the resultant angular displacement of the rotor shaft is given by
\[ \gamma = \sqrt{x^2 + \beta^2} \quad \ldots \ldots \ldots \ldots \quad (11) \]

\[ \gamma = \sqrt{A^2 + \beta^2} \quad \ldots \ldots \ldots \ldots \quad (12) \]

Substituting (9) and (10) in (12)

\[ \gamma = \frac{x \omega_0}{\sqrt{\left[k(t^2 - \omega^2) + \omega^4 \right]}} \quad \ldots \ldots \ldots \ldots \quad (13) \]

Since \( \gamma \) is constant with respect to time the axis of the rotor shaft describes a cone of revolution with the vertex at O. The point S moves on the circumference of the base describing a circle of radius \( \rho \).

Figure 4 (a) shows

\[ \rho = \sqrt{x^2 + \gamma^2} \quad \ldots \ldots \ldots \ldots \quad (14) \]

or

\[ \rho = \gamma \ell \quad \ldots \ldots \ldots \ldots \quad (15) \]

then

\[ \rho = \frac{x \omega_0 \ell}{\sqrt{\left[k(t^2 - \omega^2) + \omega^4 \right]}} \quad \ldots \ldots \ldots \ldots \quad (16) \]

The radius \( \rho \) is the amplitude of vibration since it is proportional to the amplitude of vibration of any point along the system.

Figure 4 (b) shows that the displacement lags the disturbing force by the angle \( \phi \). When \( \gamma = 0 \), \( \beta = 0 \) and \( \sin \omega t = \sin \phi \) and \( \cos \omega t = \cos \phi \) then substitution in (8)
When a system is acted upon by an external periodic force having the same frequency as the natural frequency of the system, the amplitude of the system becomes a maximum, and the system is said to be in resonance.

In order to obtain maximum displacement at resonance there must be no damping. This maximum displacement occurs when the denominator of equation (15) disappears. The resonance frequency is

\[ \omega_r = \sqrt{\frac{k}{(1 - \beta)}} \]  

(19)

Figure 7 shows the relationship between displacement and the ratio of forced vibration frequency (\( \frac{\omega'}{\omega} \)) to natural vibration frequency, or resonance frequency, \( \omega_r \). The curve C shows that \( \frac{\omega'}{\omega} \) increases towards 1 with no damping the displacement approaches infinity. As the ratio increases beyond 1, the displacement approaches zero. When \( \frac{\omega'}{\omega} \) equal 1 the displacement will be a maximum.
The phase angle $\phi$ between the force and displacement as given by equation (17) is shown in Figure 8, with respect to the frequency ratio $\nu/\omega$. Theoretically curve C will hold true, but not in practice as no practical setup for vibrations can approach the theoretical setup because there is always some friction present, such as the friction of air surrounding the system. Therefore curve D will more nearly approximate a curve drawn from experimental data for a system with no damping, other than air. If the speed of the rotor is constant, the radius of gyration $\rho$ and the phase angle $\phi$ are constant with respect to time. When the radius of gyration is constant with respect to time the rotor motion is one of pure rotation. At speeds below resonance the phase angle $\phi$ is small. At resonance the phase angle is 90 degrees, and above resonance the phase angle approaches 180 degrees. Figure 9 (a) shows the instantaneous view of the rotor rotating below resonance. The weight $W$ is in the same direction as $\rho$. The rotor whirls with the heavy side out. Figure 9 (b) shows the rotor rotating above resonance. The weight $W$ acts in the opposite direction to $\rho$. The light side tends to rotate farthest from the axis of rotation. This is the induced force for the dynamic balancer to be described.
Figure 7

Velocity ratio \( \frac{\nu}{\omega_R} \)

\( \nu \) frequency of forced vibration
\( \omega_R \) frequency of resonance
Figure 8

Phase Angle $\phi$ Degrees

Velocity Ratio $\frac{v}{\omega_R}$
below resonance rotor whirls with heavy side out
φ is small

above resonance rotor whirls with light side out φ approaches 180 deg.
BALANCING HEAD

Figure 10 shows the cross section of a balancing head used in this investigation. By means of the back plate mounted on the wheel studs the head is centered over the axle. The hub protrudes through the hole in the back plate and housing and into the empty space in the clutch plate. The ball race is held in the housing by three screws. To the ball race is attached the front plate which carries the control mechanism. The two fly-balls actuate the adjustment nut which in turn depresses the clutch plate against the force of the spring, thus allowing the two balls, of equal size, to roll in the race.

The two steel balls in the balancing head provide the necessary variables necessary and sufficient to satisfy any condition of unbalance required of the head within its capacity, when used in conjunction with a pivoted rotor. For any steady speed of rotation, the head may impart a motion or a force to the rotating system. This force or motion lies in the plane of the head. It may be in any radial direction and of any amount, depending on the position of the balls. If they are diametrically opposite the force is zero, if they touch each other the force is that equal to the full capacity of the head.

Consider the balancing head attached to the rotor shaft in the plane of the point S in figure 4. The system pivoted at O as in figure 4 has an unbalanced moment equal to wra (w) is the unbalance weight on the periphery of the rotor at radius (r) from axis of rotor and (a) distance from pivot point O.
AUTOMATIC BALANCING HEAD

figure 10
Balancing head mounted on the wheel in position for operation

Figure 11
Disassembled balance head

Figure 12
Figure 13 is a view parallel to the axis of the head and rotor showing the balls (B) in any general position within the race (R). The force of the combined unbalance of the head and rotor is represented by the vector (U).

If the speed of rotation is above that of resonance for the rotor, the phase angle will be slightly less than 180 degrees, and the axis of rotation of the system will be about the point Q, with the point S moving in a circle of radius ρ.

If the motion and equilibrium of the balls, are to be considered relative to the race they must be treated as existing within centrifugal force fields determined by the forces \( F_i = \frac{W_i \omega^2}{g} \) and \( F_i = \frac{W_i \omega^2}{g} \) exerted on the balls. The direction of these forces is determined by the axis of rotation Q of the system and the balls centers.

The rolling resistance λ of the ball in the race is considered negligible, the reaction of the race on the ball will be normal to the race or towards the center of the balancing head point S. From figure 13 the centrifugal force F on the ball about Q and the normal N to the race will urge the balls in the direction e. The displacement of the balls in the direction e will decrease the moment of unbalance. The balls will remain unstable relative to the races until the forces F and N act equal and opposite to each other. This condition exists when Q coincides with S, thus ρ becomes zero, and the moment of unbalance of the head is equal and opposite that of the rotor. Above resonance speed, the balls, if free to roll, will automatically assume the correct position to eliminate oscillation of the system. The moment of unbalance introduced by the balls must be equal and opposite to that of the rotor.
Bisecting the angle between the balls will determine the direction of countering force created by the balls.

Let figure 14 represent the rotor and balancing head at equilibrium. The resultant force from the position of the balls is equal and opposite to the force set up by the unbalance weight \( w \). Rotation of the system is now about \( S \). Assume that one of the balls \( B \) is displaced through the angle \( \Delta \theta \) to a new position \( B' \). The unbalance force vector \( U \) resulting from the displaced ball is approximately 90 degrees to balanced position of the displaced ball. The axis of rotation for the system is now at \( Q \), \( \rho \) distance from \( S \). The direction \( Q \) moves from \( S \) is determined by the phase angle \( \phi \). In the unbalance position the force on the ball of weight \( \bar{w} \) is

\[
F = \frac{\bar{w} \cdot B'Q \cdot \omega^2}{g} \quad \ldots \ldots \ldots \ldots (20)
\]

The component of this force necessary to restore the ball to equilibrium position

\[
F \sin \phi = \frac{\bar{w} \cdot B'Q \cdot \omega^2 \sin \phi}{g} \quad \ldots \ldots \ldots \ldots (21)
\]

From the geometry of figure 14

\[
\sin \phi = \frac{\bar{g} \cdot \bar{n}}{\bar{B'Q}} \quad \ldots \ldots \ldots \ldots (22)
\]

and

\[
\bar{g} \cdot \bar{n} = \rho \cos (180^\circ - \phi) \quad \ldots \ldots \ldots \ldots (23)
\]
Therefore the restoring force on the ball is

\[ F \sin \theta = \frac{W}{g} \rho \omega^2 \cos (180^\circ - \phi) \]  \hspace{1cm} (24)

\( \cos (180^\circ - \phi) \) varies slightly with comparatively large variations of the phase angle \( \phi \). From equation (24) indications that a not-too-great difference of the phase angle from the value 180 degrees will influence the stability of the head very little, so little it may be neglected. The departure of the phase angle \( \phi \) from 180 degrees in no way effects the accuracy of the indications established by the balancing head.

The rolling resistance, a linear distance \( \lambda \), is important in determining the sensitivity of the head. Figure 15 shows the balls B in the race R in the state of impending rolling. The axis of rotation of the head is at Q. \( \theta \) or the amplitude of whirl is just enough to start the balls B rolling. The rolling resistance is constant. The equilibrium of the ball relative to the race establishes forces \( F \) and \( N \) are equal, opposite, and collinear.

Let

- \( d \) = diameter of ball
- \( D \) = diameter of race
- \( \Omega \) = half the angle between the balls
- \( \lambda \) = coefficient of rolling resistance for ball and race
- \( R \) = ratio of race diameter to ball diameter = \( \frac{D}{d} \)
- \( \eta \) = sensitivity of balancing head

From the geometry of figure (15)

\[ \sin \theta = \frac{5 \eta}{\lambda} \]  \hspace{1cm} (25)
also

\[ \sin d = \frac{2\lambda}{d} \] (26)

equating (25) and (26)

\[ \frac{sn}{r} = \frac{2\lambda}{d} \] (27)

but

\[ sn = R \sin (\Omega - d) \] (28)

The value of \( R \) is small compared to the radius \( r \), thus making the angle \( d \) small as compared to \( \Omega \) and may be neglected. (28) then becomes

\[ sn = R \sin \Omega \] (29)

Substituting (29) into (27)

\[ \frac{R \sin \Omega}{r} = \frac{2\lambda}{d} \] (30)

\[ R = \frac{2\lambda}{d \sin \Omega} \] (31)

but

\[ 2r = D - d \] (32)
therefore

\[ p = \frac{\lambda}{\sin \Omega} \left( \frac{D-d}{d} \right) = \frac{\lambda}{\sin \Omega} (R-1) \quad \ldots \ldots \ldots \ldots \ldots (33) \]

The reciprocal of the amplitude of the impending ball motion is the sensitivity of the balancing head

\[ \eta = \frac{1}{\rho} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (34) \]

The maximum sensitivity of the head is realized when determining the smallest unbalance. This occurs when the balls are almost diametrically opposite each other, then

\[ \sin \Omega \approx 1 \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (35) \]

and

\[ \eta_{\text{max}} = \frac{1}{\lambda (R-1)} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (36) \]

When the two balls are together the maximum output of the balancing head is obtained. This gives the minimum sensitivity as

\[ \sin \Omega = \frac{d}{2r} = \frac{d}{D-d} = \frac{1}{R-1} \quad \ldots \ldots \ldots \ldots \ldots (37) \]

then

\[ \eta_{\text{min}} = \frac{1}{\lambda (R-1)} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (38) \]
The balancing head is attached to the wheel by means of the back plate mounted on the wheel mounting studs. Employing a portable spinner motor, the wheel and head are brought to speed 30 to 40 percent above that of resonance. The fly balls acting through the lever arm exert a force on the adjustment nut depressing the clutch, thus allowing the balls to assume positions which suppress oscillation. The wheel is allowed to come to rest, the spring force, now greater than the force exerted by the fly balls, moves the adjustment nut out actuating the clutch which clamps the balls in position.

The fly ball governor may be adjusted to cover normal ranges of driving speeds.

Using a spinner motor and some method of determining revolutions per minute, the resonance frequency of the wheel is determined, while the automobile wheel is jacked up. Converting rpm to miles per hour the speed of the automobile, where resonance exist, may be determined. Setting the adjustment nut for 20 miles per hour above the resonance road speed, the wheel may be balanced, under running conditions, by driving above the setting on the adjustment nut, allowing the balls to assume position of balance.
Before the balancing head was attached to the automobile it was first tested on a Bean wheel balancer. The first run was to balance the wheel and tire. Then adding a one ounce weight to the rim of the wheel, and mounting the balancing head over the spindle, the assembly was rotated by the driving motor on the balancing machine. On the second run the clutch was released before the critical speed of the system was attained. Because the balls assumed a position on the heavy side of the system, the out of balance was greater. The magnitude of amplitude of vibration progressively grew larger. This necessitated stopping the rotation of the system. By removing the automatic control from the balancing head and operating the clutch manually, the third run was above the critical speed before the clutch was depressed to release the balls. The position assumed by the balls removed some of the vibrations.

After this third run the balancing head was mounted on the left front wheel of an automobile. The clutch spring was replaced by a heavier spring. This raised the operating speed up to the desired range. At speeds between 50 and 65 miles per hour, during acceleration, the wheel vibrated so that it could be felt through the steering mechanism. As 75 miles per hour was approached the vibrations decreased. During deceleration through the transient speed, 50 to 65 miles per hour, there appeared to be no increase in the unbalance reactions. The automobile was brought to a stop with the balls in the balancing head in a position of balance. Then the clutch was depressed releasing the balls allowing them to assume positions of unbalance. On repeated acceleration test the results were similar except for one run. This one run gave results just
the opposite. The vibration still remained. It did not decrease or increase as determined through the steering wheel. It was thought that surface conditions of the road had some effect on the vibration. Tests were run in two directions, over each different section of road. The balancing head continued to function as before except for the one run.

The vibrations were not eliminated entirely but were decreased a notable amount. The permanently attaching of a dynamic-balancing machine to the wheel of an automobile appears to have possibilities.
SUMMARY

The public demand for safer automobiles capable of sustained high speeds with greater comfort have led the engineers to design the low pressure tires. These tires are larger in cross-section and have a greater volumetric capacity. The larger volume of the tires carries the automobile on less pressure. The larger volume results from the widening of the tire in the axial direction. This increase in width affects the wearing characteristics of the tread.

Greater width demands more accuracy in alignment of the front wheels. But even so the tread may wear excessively. This may be due to unbalance.

The question of unbalance brings forth the inquiry of whether the wheel should be considered as a rotating disk, or a rotation cylinder. If considered as a disk any unbalance would be static unbalance. By the addition of a balance weight opposite the heavy side the unit would be in balance. This does not hold true for an unbalanced wheel. The wheel is dynamically unbalanced, due to the unequal distribution of rubber throughout the casing walls, and tread. The wheel may be dynamically balanced by the addition or subtraction of two weights, one in each of two arbitrary transverse planes. That is on the inside and outside rims of the wheels.

An automatic dynamic-balancing head was designed to be permanently attached to an automobile wheel. Under operating conditions of the automobile, the wheels may be balanced.

Two steel balls in the balancing head provide the necessary variables necessary and sufficient to satisfy any condition of
unbalance required of the head within its capacity, when used within or in conjunction with a pivoted rotor. The steel balls are held in place in the race by a spring loaded conical clutch. Two fly-balls on the governor actuate the clutch. When the speed of rotation is great enough to exert a force on the fly balls sufficient to overcome the clutch spring force the clutch is depressed allowing the balls to roll in the race. The balls will assume positions which tend to decrease oscillation.

The speed of rotation at which the fly balls operate is ten to forty percent above that of resonance. If they operated below resonance the steel balls would move towards the heavy side, thus increasing the unbalance moment. Above resonance the balls will seek the light side, which rotates away from the axis of rotation, thereby damping the vibration.

The test under operating conditions indicate the possibilities of a dynamic-balancing head being attached permanently to a wheel of an automobile.
BIBLIOGRAPHY


VITA

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Entered primary school at Woodstock, Illinois in 1927. Graduated from Woodstock Community High School with the class of 39. Entered the Missouri School of Mines and Metallurgy 1940. Received a B. S. degree in Mechanical Engineering January 25, 1944. Upon graduation he entered the Army and served in the China, Burma and India theater; Discharged April 1946; Re-entered Missouri School of Mines and Metallurgy in the Fall semester and served as graduate assistant in Drawing Department. February 1947 joined MSM faculty as instructor in Engineering Drawing and descriptive Geometry. Served in that capacity to present time.