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Lung-Wen Tsai
E.R. Maki
R. L. Jacques

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Lung-Wen Tsai and E.R. Maki
Power Systems Research Dept.
General Motors Research Laboratories
Warren, MI
R.L. Jacques
Buick-Oldsmobile-Cadillac Group
General Motors Corp.
Detroit, MI

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Lung-Wen Tsai and E.R. Maki**
Power Systems Research Dept.
General Motors Research Laboratories
Warren, MI

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Buick-Oldsmobile-Cadillac Group
General Motors Corp.
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ABSTRACT

This paper describes the effectiveness of an Oldham-coupling-type balancer in reducing the first- and second-harmonic rotating unbalance couples of a 90° V6 even-firing engine. It is shown that the prototype balancer can be configured to eliminate the residual, first-harmonic rocking couple usually remaining in a conventionally balanced 90° V6 engine. Further, the Oldham-coupling-type balancer can be used to reduce the elliptical second-harmonic rotating couple to a horizontal rocking couple. The power consumption of the balancer was measured to be approximately 0.35 kW at 2000 r/min and 1.5 kW at 4500 r/min.

A 90° V6, FOUR-STROKE, INTERNAL-COMBUSTION ENGINE with even firing intervals has first- and second-harmonic rotating unbalance couples of variable magnitude. A portion of the first-harmonic unbalance may be compensated for by the addition of crankshaft counterweights [1-3].* However, the remaining unbalance represents a substantial challenge to the designer in order to isolate it adequately in the vehicle. To achieve perfect balance would require the addition of one first-harmonic and two second-harmonic auxiliary balance shafts. The complexity of such a solution is difficult to justify in current engines with the emphasis on mass and friction reduction and compactness.

Recently, Tsai [4] described the feature of a mechanical balancer based on the Oldham-type coupling for reducing or eliminating out-of-balance in mechanical systems. It was shown that by proper arrangement of two Oldham couplings, a balancer can be obtained for the elimination of second-harmonic shaking forces, second-harmonic shaking moments, or a combination of both shaking forces and moments. A unique feature of this balancer mechanism is that it runs at the primary speed of the device to be balanced. The development of a second-harmonic Oldham-coupling-type balancer to counterbalance the shaking force in a four-cylinder engine is given in Reference 5.

This paper describes the design of an Oldham-coupling-type balancer for 90° V6 engines with even firing intervals and discusses its effectiveness in reducing engine vibration.

THE BALANCER CONCEPT

The features of the Oldham-coupling-type balancer are given in Reference 4. However, for the convenience of the reader, the principle of operation will be reviewed briefly. Generic features of the Oldham coupling are shown schematically in Figure 1. It consists of a driving clevis, a coupler used as the balance mass, and a

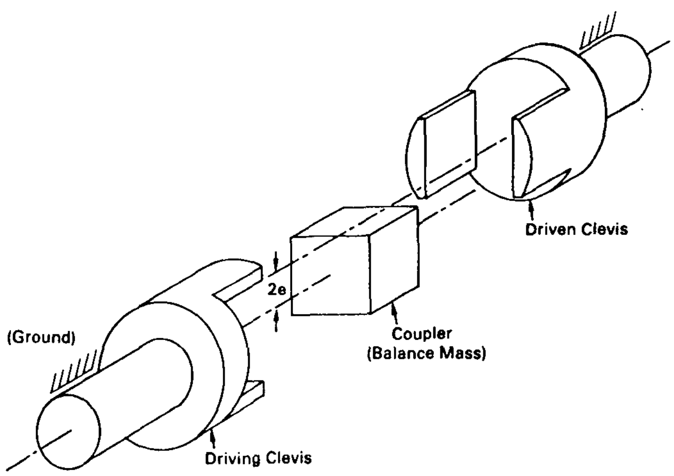

Figure 1. An Oldham Coupling.

driven clevis. The driving and driven clevises are supported in journal bearings that are fixed to the machine frame. The rotating axis of the driven clevis is parallel to that of the driving clevis but offset by a distance of 2e. Two pairs of opposing projections attached to the respec-

*Numbers in brackets designate references at end of paper.

**L.-W. Tsai is currently with the Department of Mechanical Engineering and Systems Research Center, University of Maryland, College Park, Maryland.

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tive clevises are positioned at right angles to one another to constrain the coupler.

Rotation of the driving clevis at a constant angular velocity, ω, causes the coupler and driven clevis to rotate at the same angular velocity. The center of the coupler, however, revolves in a circular path (whose diameter is determined by the offset 2e of the two parallel shafts) at two times the angular velocity of the driving clevis. Hence, the coupler generates a rotating centrifugal force with a magnitude of $4\ me\omega^2$, where m is the mass of the coupler. The total kinetic energy of the coupling is constant for all time. Therefore, to drive the coupling at any given angular velocity, the only input torque required is that to overcome friction.

Figure 2 illustrates schematically how two Oldham couplings can be arranged along a driveshaft axis, or an axis parallel to the driveshaft axis, to balance a second-harmonic rotating couple. In Figure 2, A_i and B_i, i = 1,2, denote the axes of rotation for the driving and/or driven clevises; C_i denotes the midpoint between A_i and B_i; 2ϕ denotes the angular displacement of the center of the couplers; ⊕ denotes the location of the center of the coupler; and r denotes the axial distance between the couplers. The couplings rotate in the same direction but are

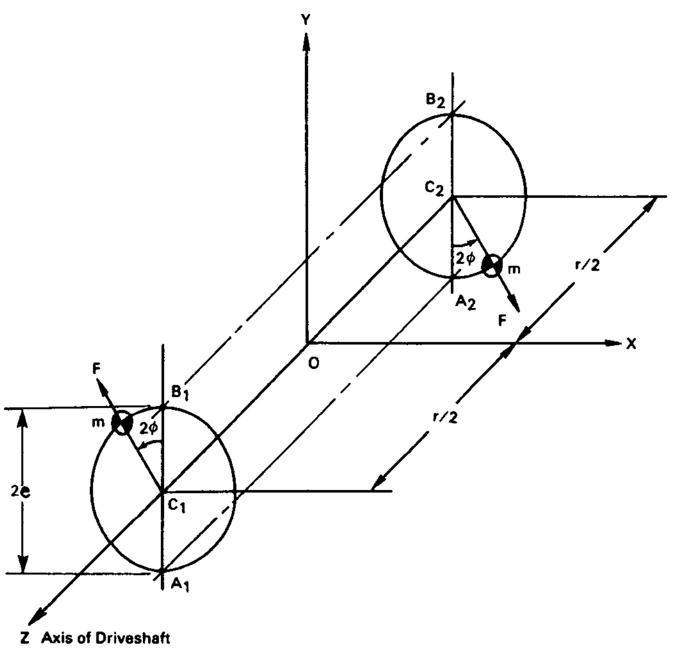

Figure 2. Two Oldham Couplings Arranged for Balancing a Second-Harmonic Rotating Couple.

phased 180° apart. This arrangement results in a second-harmonic rotating couple whose x and y components are given as follows:

$$M_x = -4\ me\omega^2 r\ \cos(2\phi)$$

$$M_y = -4\ me\omega^2 r\ \sin(2\phi)$$

In addition, eccentric masses can be added to the clevis end-discs or the attached shafts to provide a first-harmonic balance couple effective at the shaft-rotating frequency.

BALANCER IMPLEMENTATION

The first- and second-harmonic rotating unbalance couples acting in the 90° V6 even-firing engine are shown graphically in Figure 3, where C_h denotes the horizontal magnitude of the rotating couple and C_v the vertical magnitude of the rotating couple. There is an elliptical first-harmonic rotating couple rotating in the

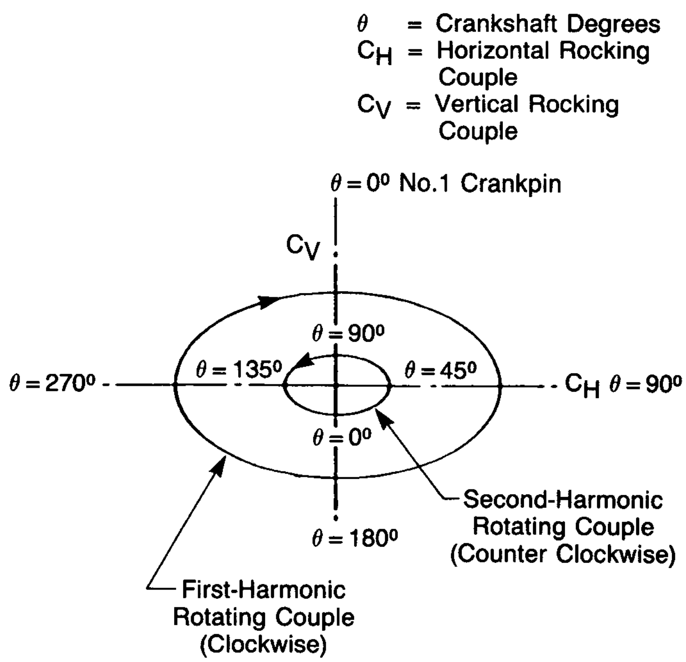

Figure 3. 90° V6 First- and Second-Harmonic Rotating Couples.

direction of crankshaft rotation, illustrated as clockwise rotation, with a horizontal major axis. The second-harmonic unbalance couple is also elliptical with its major axis horizontal, but its rotational direction is opposite to the primary unbalance at a rotational velocity twice that of the crankshaft velocity, illustrated here as counterclockwise rotation.

Normal balancing procedures usually apply crankshaft compensation to modify the first-harmonic elliptical rotating couple into a horizontal rocking couple. This remaining first-harmonic unbalance and the total second-harmonic unbalance are then absorbed, to the extent possible, by providing suitable vibration-absorbing engine mounts.

In the case of the Oldham-coupling balancer, the crankshaft counterweight compensation is adjusted to provide compensation equal to that of the magnitude of the minor moment axis of the rotating couple plus one-half the magnitude difference between the major and minor elliptical axes. This produces a resultant counterclockwise rotating couple of constant magnitude which can then be completely compensated for by the counterweights attached to the Oldham-coupling clevises or end-discs. The masses of the coupler blocks are then selected to produce a constant-magnitude, compensating rotating couple equal to the magnitude of the minor moment axis of the

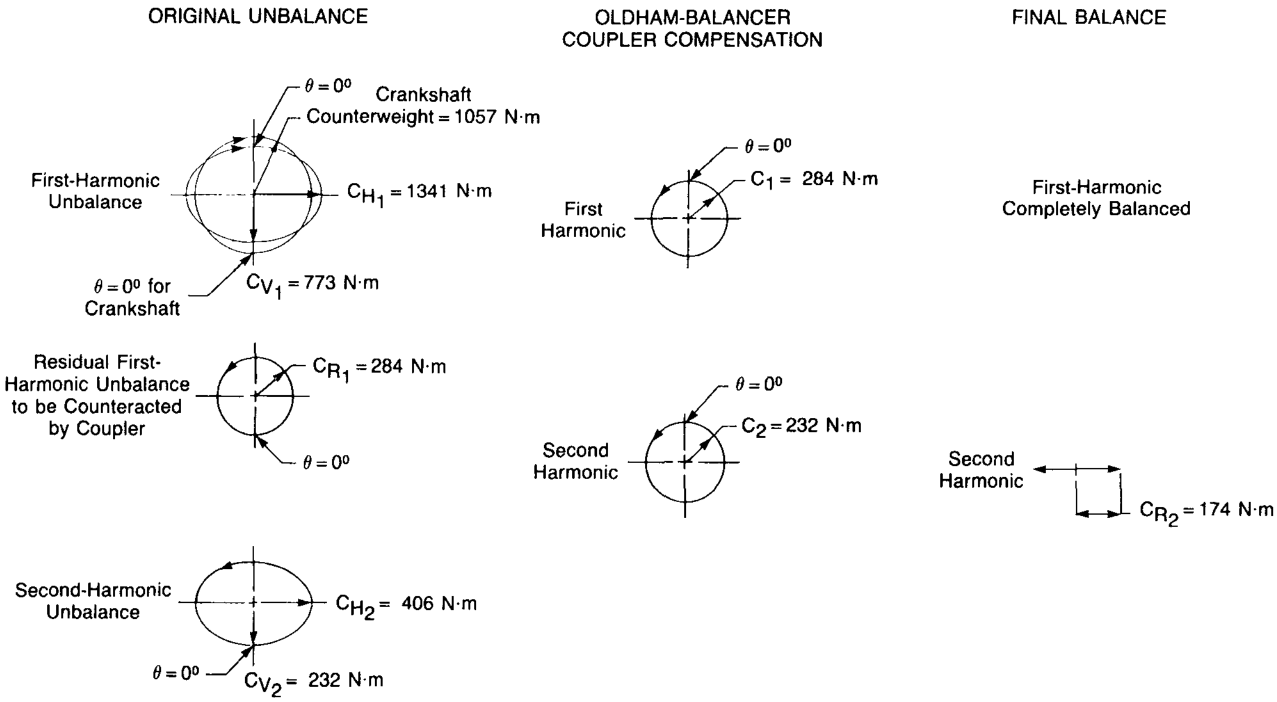

Figure 4. Original Unbalance, Compensation, and Final Balance. Speed = 4000 r/min.

second-harmonic unbalance couple. This produces a resultant second-harmonic horizontal rocking couple.

The engine to be balanced has the following characteristics: reciprocating mass = 0.763 kg, rotating mass = 0.41 kg, connecting rod length = 140 mm, and stroke = 84 mm. This results in the necessity to balance the following first- and second-harmonic rotating couples.

At 4000 r/min, the first-harmonic rotating couple to be balanced has a magnitude of 1341 N•m (989 ft lb) along the major axis and 773 N•m (570 ft lb) along the minor axis, and the second-harmonic rotating couple has a magnitude of 406 N•m (299 ft lb) along the major axis and 232 N•m (171 ft lb) along the minor axis.

The first-harmonic elliptical rotating couple is partially compensated by the addition of mass on the crankshaft, resulting in a residual 284 N•m (209 ft lb) circular rotating couple. Complete balance of this residual rotating couple is accomplished by incorporating balance masses on the outboard Oldham-coupling balancer clevis ends.

Compensation of the second-harmonic rotating couple is accomplished by selecting the values of e and m of the Oldham-coupling sliding block and the distance between the two couplings to obtain the desired result. In this case, the second-harmonic rotating couple was reduced to a residual second-harmonic rocking couple of 174 N•m (128 ft lb).

Figure 4 summarizes the results of balancing this 90° V6 engine using the Oldham-coupling balancer.

BALANCER DESCRIPTION

The conceptual implementation of an Oldham-coupling-type balancer for a 90° V6 even-firing engine is shown in Figure 5. Illustrated are the front and rear first- and second-harmonic balance

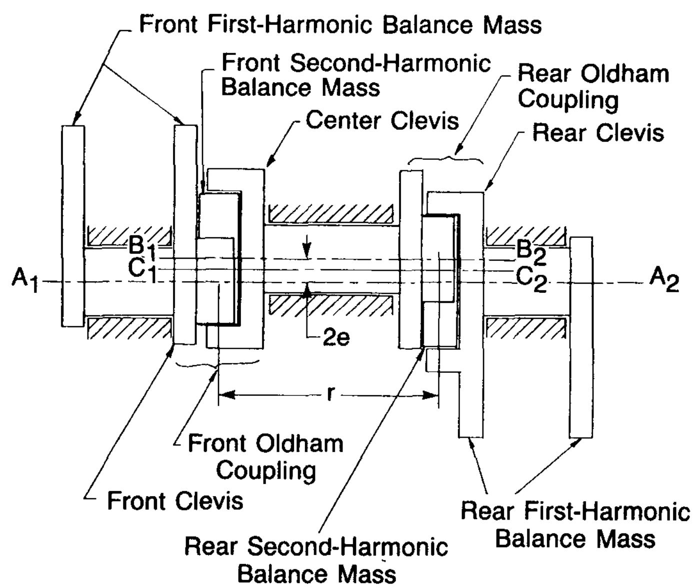

Figure 5. Schematic Cross-Section of Oldham-Coupling Balancer for 90° V6 Engine.

masses and their positions relative to one another. Note that the first-harmonic balance masses rotate about the main axis and the Oldham-coupling balance masses rotate about their respective centers of mass and, at the same time, revolve about an axis half-way between the main axis and the parallel offset axis.

Demonstration of the Oldham-coupling balancer concept was performed on a 3.1L 90° V6 engine. In order to implement the installation with minimum modification to the engine, it was designed to be completely enclosed in the oil pan, directly below the crankshaft.

Figure 6 shows the assembled balancer mounted on a sub-base prior to installation in the engine. It is supported on an A-frame-like structure which is attached to the engine main-bearing caps. As illustrated in Figure 6, the balancer is inverted relative to the way it would be installed in the engine.

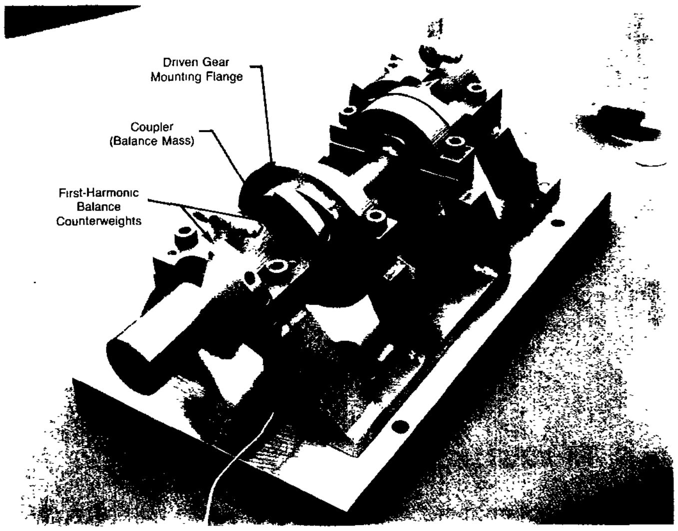

Figure 6. Oldham-Coupling Balancer for 90° V6 Engine.

The rotating components are all supported on cylindrical journal bearings. There are two bearings located at each end to support the end clevises and primary balance masses. These bearings are approximately 25 mm in diameter by 45 mm long and are center-fed by supplying lubricant to a 5-mm-wide undercut in the shaft at its midpoint. The location of the lubricant feed is shown by the position of the fittings located at each end of the balancer in Figure 6. Two additional journal bearings are used to support the intermediate shaft which incorporates the remaining clevises. These bearings are approximately 25 mm in diameter by 20 mm long. They are lubricated through drilled passages that connect to the clevis faces. Lubricant to this journal bearing is introduced to the clearance region via a flat on the unloaded side of the shaft. The coupler blocks are cross-drilled to ensure that lubricant is supplied to all sliding surfaces, and pick-up aperatures are located such that positive lubrication is ensured for all positions of the block. Thus, lubricant enters at the outboard journal bearing locations, is channeled to the driving clevis face, lubricates the coupler block, and also passes through the coupler block to lubricate the bearing supporting the intermediate shaft at that end.

Diametral clearances between the shafts of the clevises and journal bearings is in the range of 0.03±0.01 mm, and the clearances between the clevises and coupler block are 0.06±0.015 mm. The overall axial clearance between a coupler block and its respective clevis faces is 0.10 to 0.30 mm. No particular effort was made to optimize the design of the bearings or the supporting structure since the primary objective of this investigation was to demonstrate the feasibility of the concept in a 90° V6 engine.

The first-harmonic balance masses are located on either side of the journal bearings on the A_{1-2} axis. This places a near-symmetrical rotating load on the bearing which minimizes edge loading. The inner pair of the first-harmonic balance masses are integral with the clevises that constrain the second-harmonic balance masses. The counterweights are at a 171-mm spacing center-to-center and provide a counterbalance effect of 3.528 kg mm, the product of the mass and its distance from the axis of rotation. The outer pair of counterweights are 285 mm apart and provide a counterbalance effect of 3.609 kg mm. All of the first harmonic counterweights are made of steel. The second-harmonic balance mass is the coupler block, and it has a mass of 319 g and orbits on a circular path with radius e = 7 mm. The two coupler blocks are placed 140 mm apart center-to-center. They are made of a leaded bronze selected for its high density and good bearing characteristics.

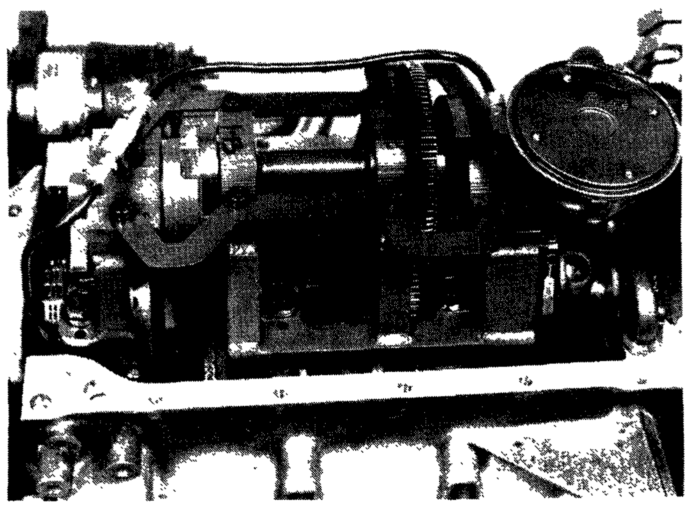

Figure 7. Oldham-Coupling Balancer Installed on 3.1L 90° V6 Engine.

In order to accommodate the installation of the balancer on the engine, it was necessary to make a number of modifications. A 14-mm wide split gear was attached to the crankshaft on the face of the number 2 cheek. This allowed the balancer drive to be completely within the oil pan and precluded the need to make a gear integral with the crankshaft. Main-bearing cap bolts were modified to permit attaching the balancer frame to the block, and lubrication lines were added for the balancer. Finally, a slightly enlarged oil pan was installed. Figure 7 shows this prototype balancer installed on the engine and illustrates the location of the various elements. Additional descriptions of the Oldham-coupling balancer may be found in Reference 6.

It was not possible to remove the drive gear from the crankshaft and disengage the balancer to make comparative measurements for friction and vibration level. This was due to the fact that the balancer rotating component positions were phase-matched to crankshaft counterweight positions with the minimum distance possible between their respective axes of rotation. If the balancer was not rotating, it would be struck by the piston connecting rods. Therefore, the friction measurements discussed in the following paragraph were obtained from two engines which were prepared to be as identical as possible in all respects. It is recognized that there probably were frictional differences between the two engines. However, the friction measurements reflect a consistent pattern and do correlate with the results reported in Reference 5. Thus, the variability introduced by using the two separate engines for these measurements was judged to be minimal.

FRICTION MEASUREMENTS

Motoring torque requirements (without the engine firing) are plotted against speed in Figure 8 for engines with and without the balancer. Multiplying the difference in motoring

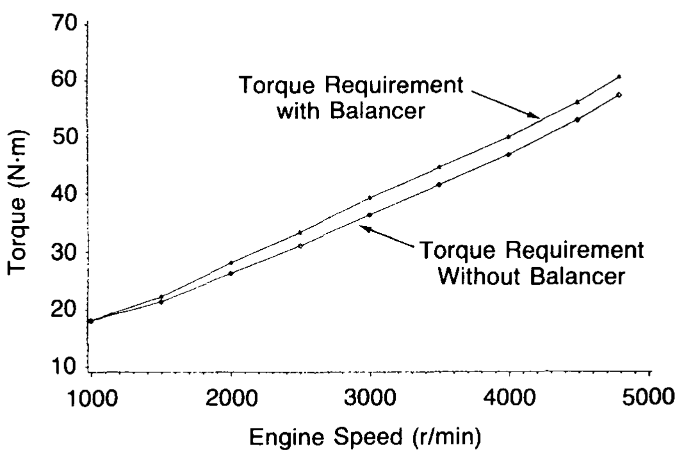

Figure 8. Engine Motoring Friction.

torque by the speed gives the power loss as shown in Figure 9. The power loss is approximately 0.35 kW (0.47 hp) at 2000 r/min and 1.5 kW (2.01 hp) at 4500 r/min. These results indicate that the power loss is less than that reported for a Lanchester-type balancer operating at twice the engine speed to balance the second-harmonic shaking force in a four-cylinder engine [7]. The power loss is very close to that measured for the Oldham-coupling balancer described in Reference 5, which had a similar total number of bearings and sliding surfaces. Overall, the Oldham-coupling balancer is a more efficient mechanism since it operates at engine speed.

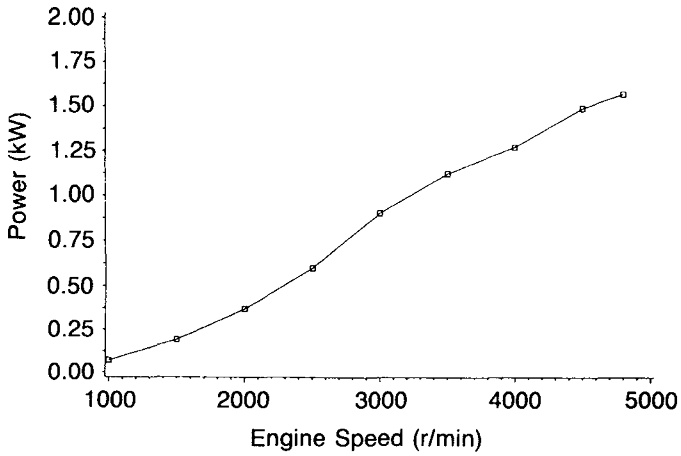

Figure 9. Engine Balancer Power Consumption.

BALANCER PERFORMANCE

The performance of the balancer was verified by measuring its vibrational characteristics on a firing test stand used specifically for that purpose. The engine was supported on flexible cable mounts, and engine vibration was detected with an accelerometer. A spectrum analyzer was used to observe and record the data. In order to obtain the truest perspective of the balancer's performance, there were no accessories mounted on the engine. However, the water pump did remain and was rotated during the tests.

Data were obtained for three engines. These were a 4.1L V8, a standard 3.1L 90° V6, and the Oldham-balancer-equipped 90° V6. The engines were prepared to be as identical as possible relative to reciprocating and rotating masses by ensuring that these parts were mass matched to within 0.01 g for all applicable pieces. Tests were performed over a speed range of 800 to 4200 r/min. The accelerometer was placed in various locations on the engine to measure the vibration level in order to determine the magnitude of the change relative to the V8 baseline. Front and rear, horizontal and vertical engine acceleration was measured.

Figure 10 illustrates typical results obtained from these tests. The data are for an engine speed of 1500 r/min and show the acceleration values as Δ dB vibration, where 20 dB reflects a tenfold increase in vibration over the reference V8 engine. These results are for a transducer located at the front of the engine.

The data are given relative to the V8 engine as a reference for both first- and second-harmonic vibration levels in the horizontal and vertical directions. In the horizontal direction for first-harmonic vibration, the Oldham-coupling-equipped engine was 7 dB greater than the V8, while the non-balancer-equipped engine was 29 dB greater. Vibration in the vertical direction was 5 dB lower for the balancer engine, while a 1 dB increase was measured for the standard V6 engine. Although it was expected that the standard engine would have vertical acceleration relative to the V8 baseline value, the

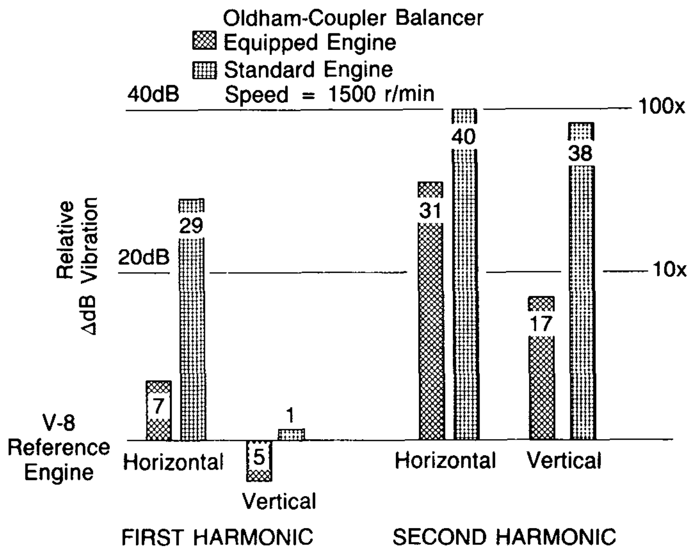

Figure 10. Engine Vibrations Measured at Front of Engine.

residual vibration detected in this case was small. A factor which may have contributed to this situation is the engine mounting geometry. The V6 engine mounts were not located symmetrically and therefore it was possible to get some cross-axis coupling effects of the engine motion.

Similar measurements were made for the second-harmonic vibration excitation. In the case of the horizontal acceleration, the Oldham-balancer-equipped engine was 31 dB higher than the V8, while the standard V6 was 40 dB greater. The horizontal acceleration was expected since balance compensation was designed to result in a residual second-harmonic rocking couple in this plane. Although there should have been no significant vertical second-harmonic vibration measured relative to the V8 baseline, a 17 dB increase was observed. This may also be in part due to the non-symmetrical engine mounting geometry permitting some cross-coupling motion to occur. The standard V6 vertical acceleration was 38 dB greater than the V8 reference engine.

CONCLUSION

The Oldham-coupling balancer has been shown to be an effective method for reducing the out-of-balance in a 90° V6 engine. Residual first-harmonic out-of-balance can be eliminated completely by incorporating counterweighting on the Oldham-coupling main axis. Second-harmonic rotating couples can be reduced to a minimal horizontal rocking couple that can be isolated by the engine mounts. The balancer friction is low since it runs at engine crankshaft speed.

While the installation of this prototype balancer in the oil pan of the engine served to demonstrate its functional characteristics satisfactorily, it is expected that a more realistic production design would be placed in the valve-lifter cavity and be driven by a 1:2 speed-increasing gear from the camshaft.

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