(12) UK Patent Application (19) GB (11) 2 342 695 (13) A

(43) Date of A Publication 19.04.2000

(21) Application No 0001829.1

(22) Date of Filing 27.05.1997

Date Lodged 26.01.2000

(30) Priority Data
(31) 677085 (32) 09.07.1996 (33) US

(62) Divided from Application No 9710889.8 under Section 15(4) of the Patents Act 1977

(71) Applicant(s)
Simpson Industries Inc
(Incorporated in USA - Michigan)
47603 Halyard Drive, Plymouth,
Michigan 48170-2429, United States of America

(72) Inventor(s)
David L Killion
Michael D Hendrian
Robert J Crist

(51) INT CL^7
F16F 15/26 , F02B 75/06

(52) UK CL (Edition R)
F1M M18

(56) Documents Cited
WO 92/02720 A1 US 5483932 A US 5375571 A

(58) Field of Search
UK CL (Edition R) F1M M18 , F2U U1
INT CL^7 F02B 75/06 , F16F 15/26
online: EPODOC, JAPIO, WPI

(74) Agent and/or Address for Service
J A Kemp & Co.
14 South Square, Gray's Inn, LONDON, WC1R 5LX, United Kingdom

(54) Abstract Title

Balance shaft, eg for engines, with recessed cross-section

(57) The balance shaft, of the "single unbalance" type (figs. 1-4), or the "rotating couple" type (figs. 5-21), has a connector portion 32 with a cross-section, eg of I-beam shape, with recesses 140 to improve rigidity and minimise material usage. The connector portion 32 of the single unbalance shaft may also taper longitudinally. The rotating couple shaft has a pair of opposed weights (113, 114) and may have surfaces (128, 130) of hyperbolic shape.

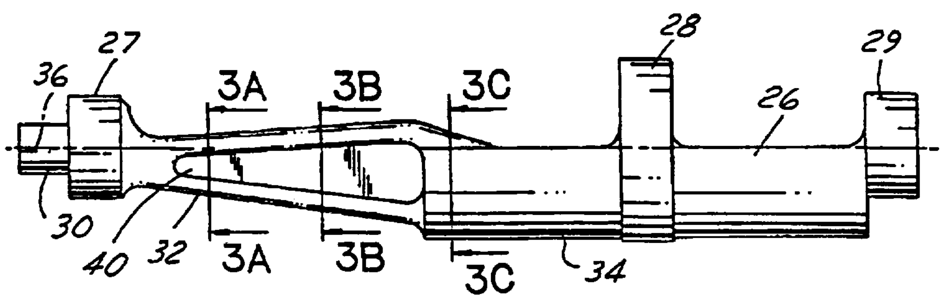

FIG.3

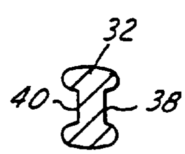

FIG.3A

GB 2 342 695 A

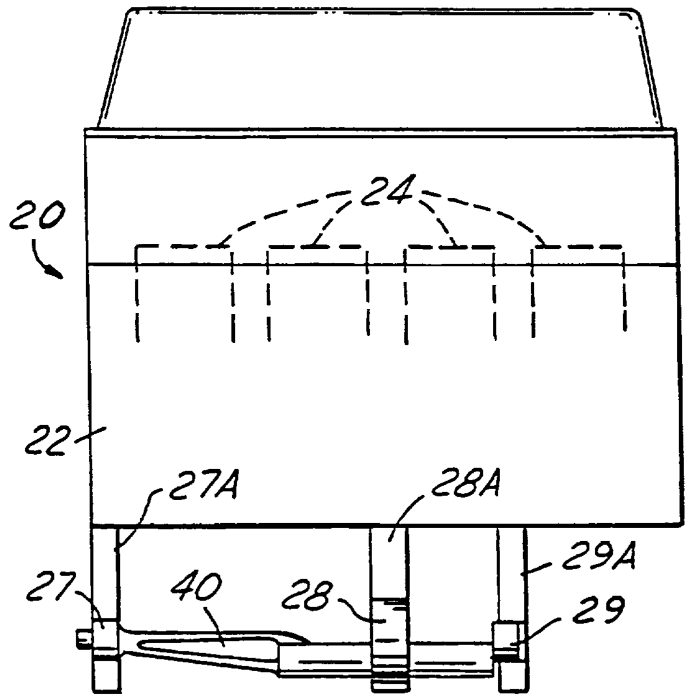

FIG.1

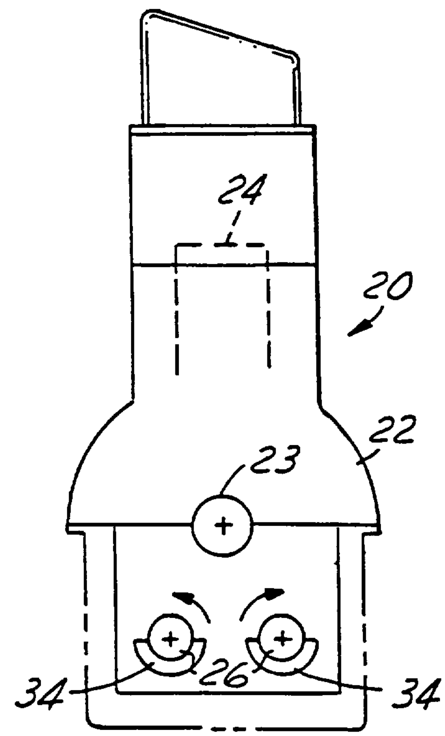

FIG.2

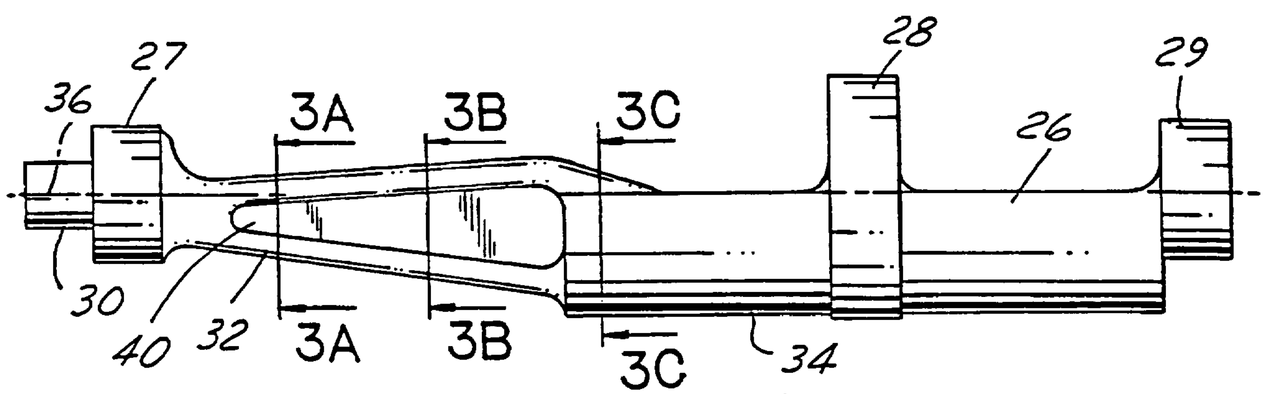

FIG.3

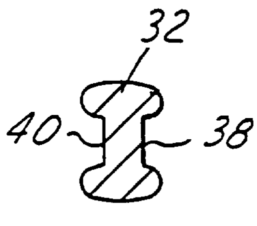

FIG.3A

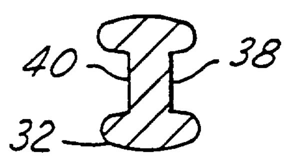

FIG.3B

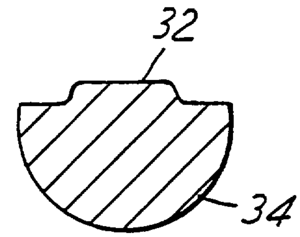

FIG.3C

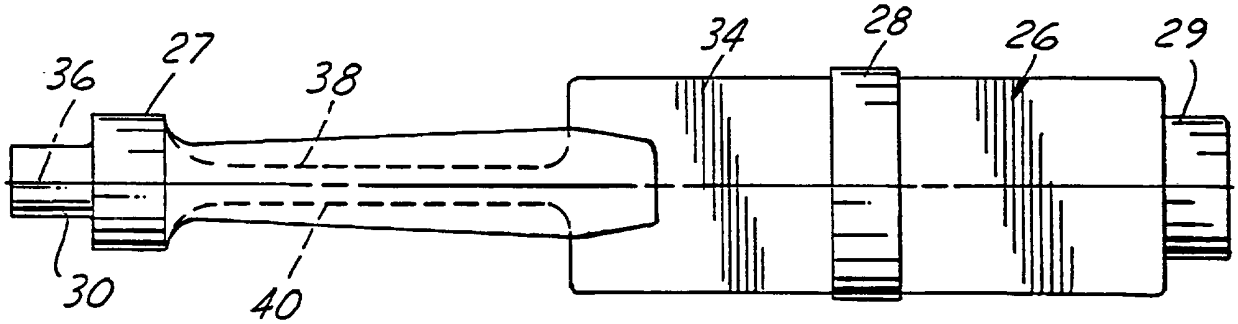

FIG.4

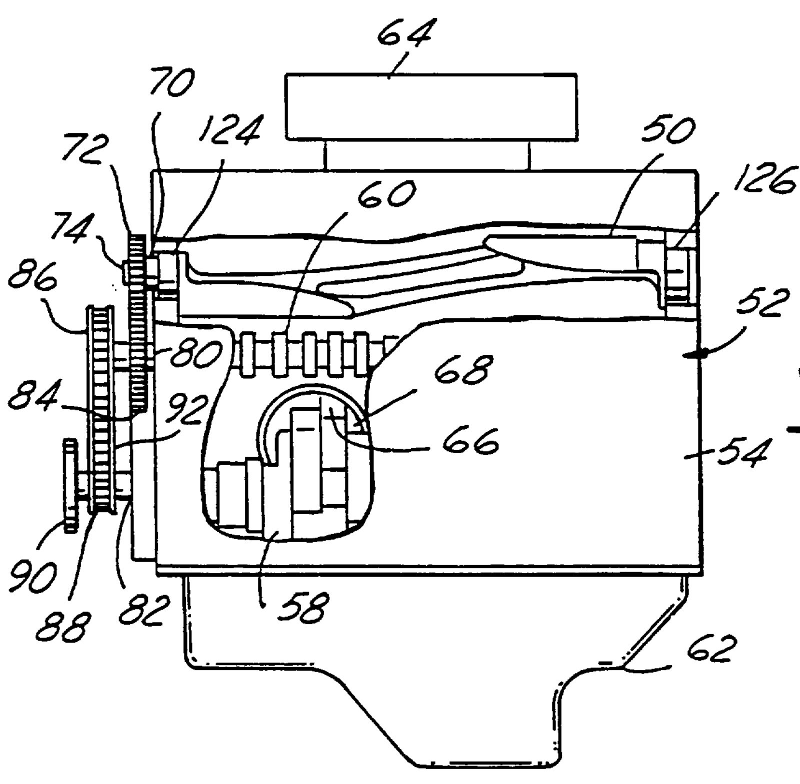

FIG.5

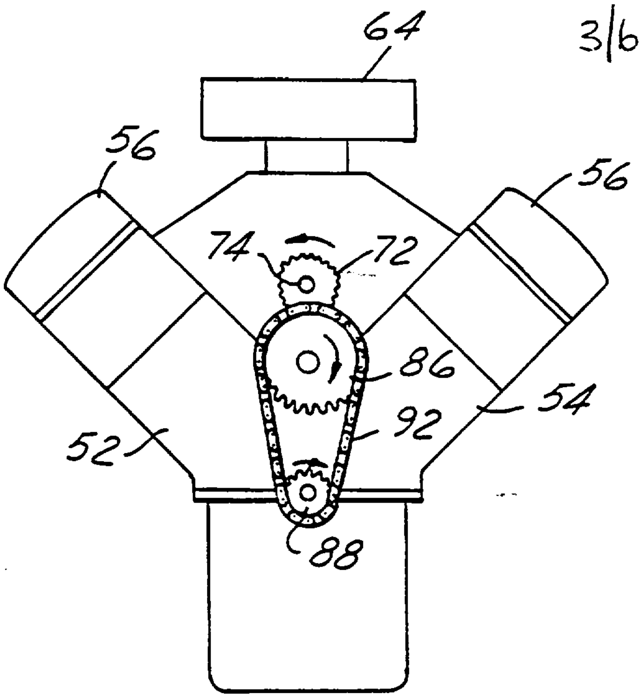

FIG.6

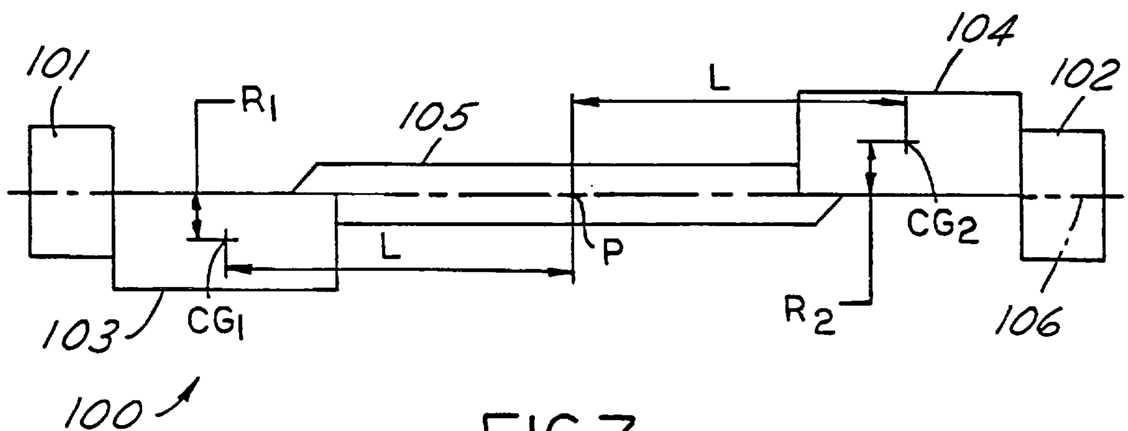

FIG.7

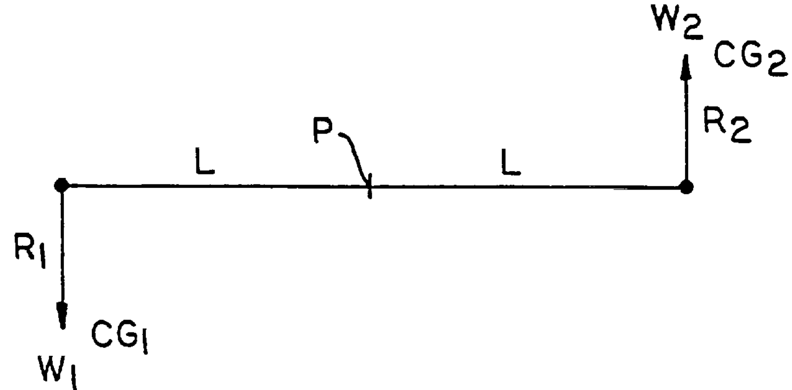

FIG.8

4/6

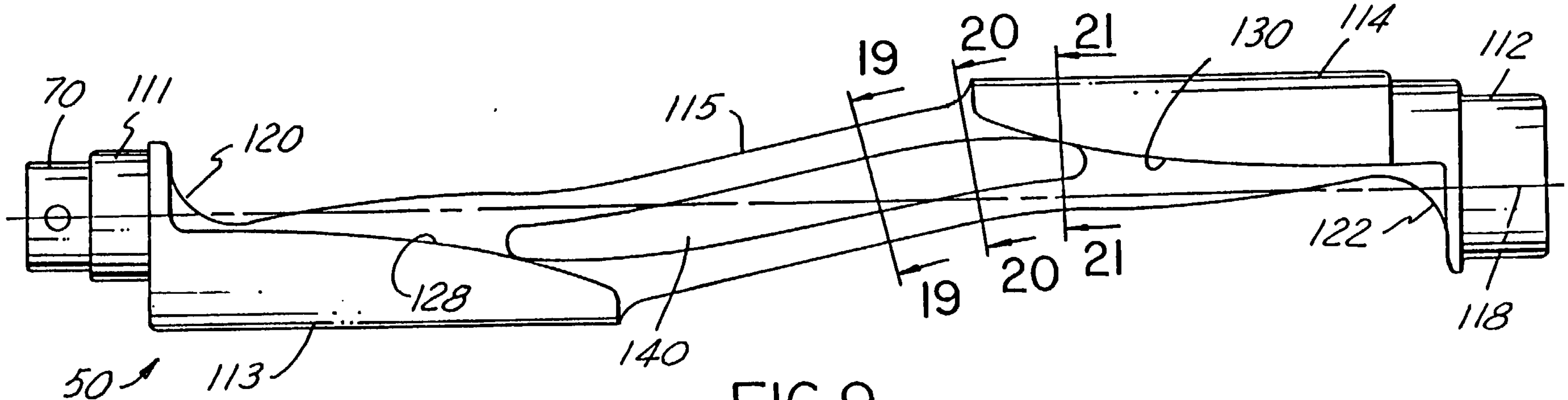

FIG.9

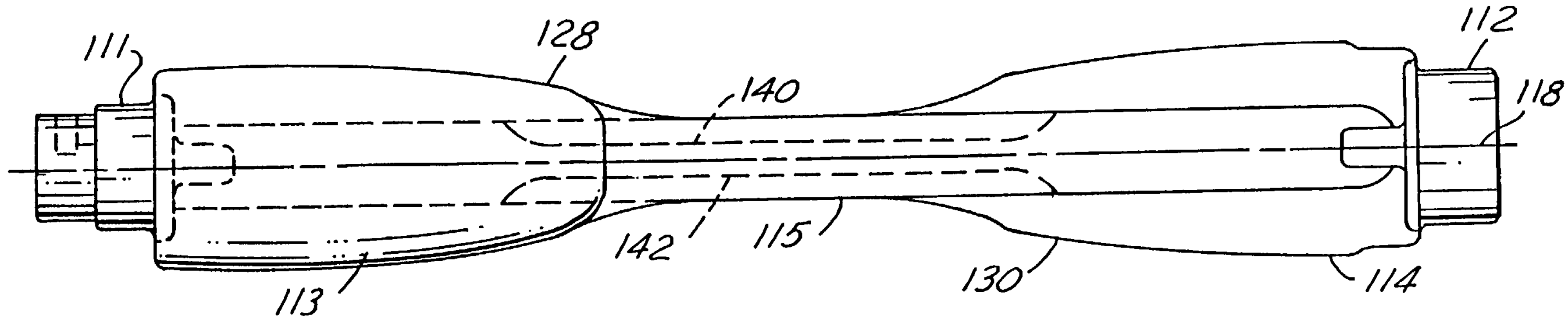

FIG.10

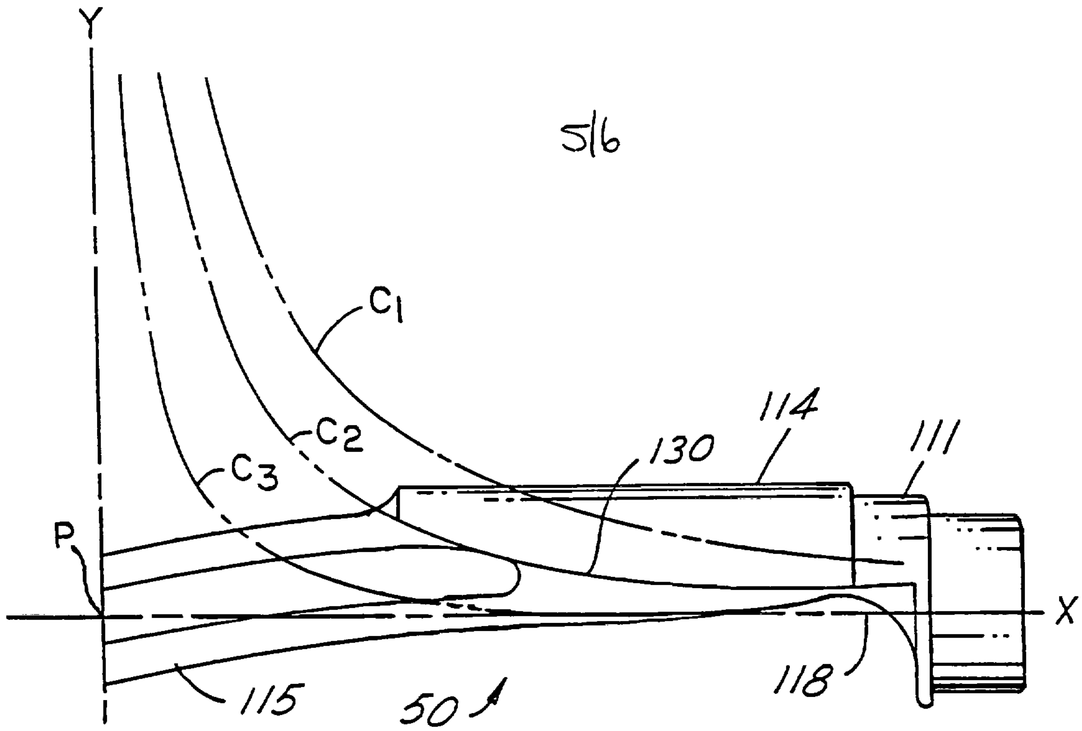

FIG.11

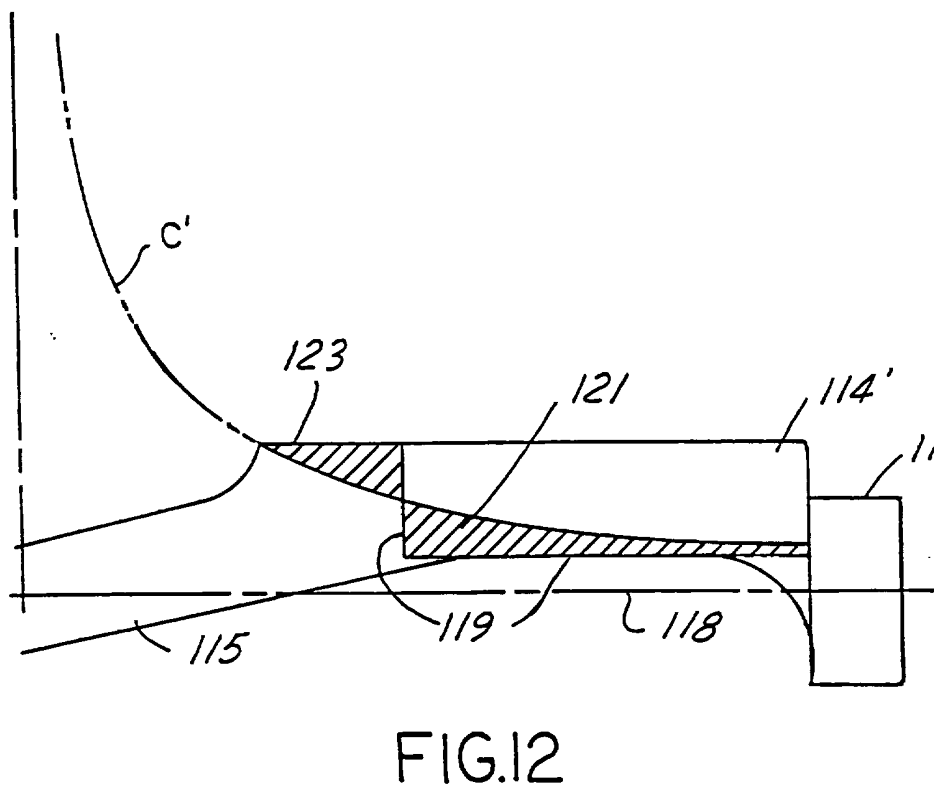

FIG.12

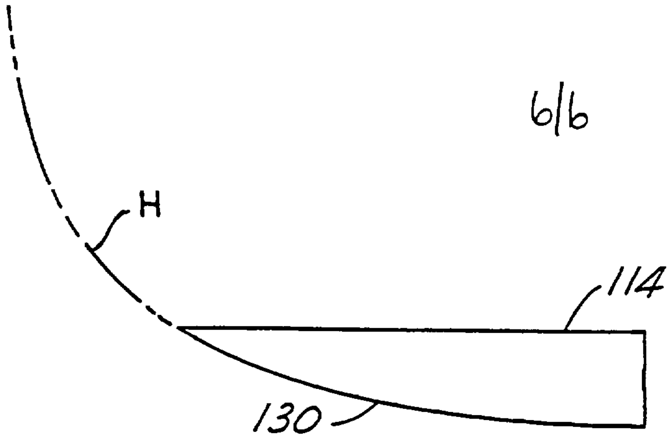

FIG.13

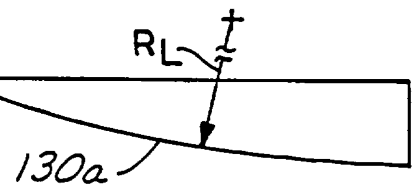

FIG.14

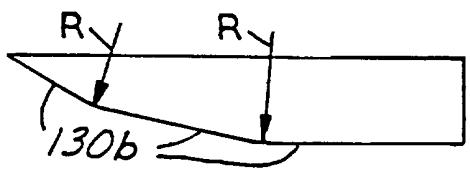

FIG.15

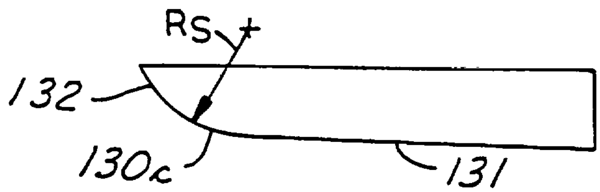

FIG.16

FIG.17

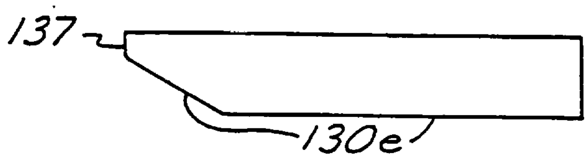

FIG.18

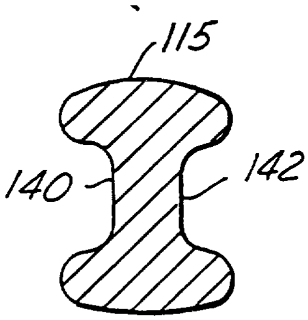

FIG.19

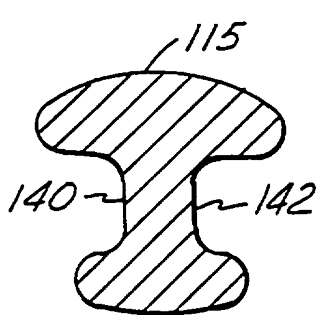

FIG.20

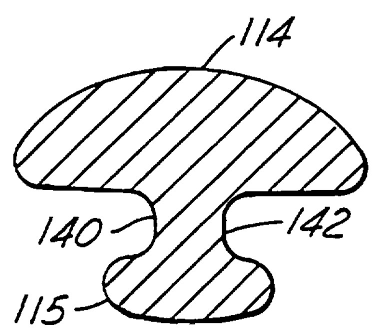

FIG.21

2342695

BALANCE SHAFTS HAVING MINIMAL MASS

Technical Field

The present invention relates to balance mechanisms for rotating machinery, particularly balance shafts for multicylinder internal combustion engines which exhibit shaking forces and/or rotating imbalance couples.

Background Art

Balance shafts are commonly used to reduce or cancel shaking forces and/or vibrations which result from residual imbalances inherent in the design architecture of machinery with rotating parts or mechanisms, such as motors. These balance shafts are sometimes called "counterbalance" shafts.

Balance shafts are particularly valuable when operator or passenger comfort and freedom from noise and vibration-related fatigue or distraction are desired, as in the case of motor vehicles such as automobiles, motorcycles, and the like. It is also advantageous to minimize vibration from the standpoint of equipment reliability. Where vibrations are reduced, the size, mass and/or complexity of the mounting structures can often also be reliably reduced, thus potentially reducing cost.

With multicylinder motor vehicle engines, the inline four-cylinder engines and 90-degree V-6 engines are favored in automotive use today due to their space efficiency and cost. Both of these engine architectures benefit from balance shafts, although for different reasons and vibratory characteristics, and thus require distinctly different balance shaft arrangements.

Balance shafts for inline four-cylinder engines typically are paired to rotate in opposite directions at twice the engine speed. The two balance shafts cancel each other's lateral shaking forces while opposing the vertical secondary shaking forces that are typical with this type of engine. Each shaft produces a single unbalance force, which taken together with its mating shaft's unbalance force, produces a resultant vertical shaking force located centrally among the bank of cylinders. These "single unbalance" type shafts are shown, for example, in U.S. Patent No. 4,819,505.

Other engines, such as 90-degree V-6 engines (i.e., six-cylinder engine with two sets of three cylinders spaced 90-degrees apart), produce resultant imbalance forces in the form of a crankshaft-speed rotating couple. These engines benefit from a single balance shaft with two balance "weights", or masses, on opposite sides of its axis of rotation, but spaced apart axially so as to have a dynamic imbalance providing a rotating couple. The couple produced by the balance shaft is designed to oppose or cancel that of the engine when the shaft is rotated at crankshaft speed and in the opposite direction to the crankshaft. The axial location of this "rotating couple"-type shaft relative to the engine is not critical since the output of the balance shaft is a pure couple or torque on the crankcase.

Balance shafts of both types frequently incorporate an elongated support member, or shaft, which provides a structural connection between the balance weights, in the case of the rotating couple-type shaft, or between the centrally located balance weight(s) and a driving member, in the case of the single unbalance-type shaft. The elongated support member is subjected to both torsion and bending forces, and thus must be substantial enough to fulfill structural requirements. Since the mass of the elongated support member is largely "dead weight" and has little, if any, contribution to unbalance, its mass can be reduced in applications where overall mass is a factor in product cost and/or operating efficiency. These elongated support members or shafts typically have a circular cross-section. This circular section represents a structurally inefficient distribution of material that causes the components and their support structures to be more massive and often more costly than necessary.

The room or space for placement of balance shafts in the engine is typically small or limited. Balance shafts usually are constrained to operate within specified radii, whether to clear mating parts or to enable installation. Thus, efficient material usage typically motivates a balance weight cross-sectional shape that is, except for elongated support member intersection areas, "circular segment" in shape, i.e. the area between a radius and a chord. The radius of such a shape represents the clearance boundary beyond which the balance shaft cannot extend without risk of unwanted contact. The chord represents a locus of constant contribution to unbalance within the section, placing elements of mass equidistant from the axis of rotation, with regard to the ability of the mass element to generate centrifugal force in a particular direction, i.e., when viewed from a direction normal to the desired direction of unbalance force.

Typically, the "circular segment" shape of the balance weights are constant along their lengths. This enables easy calculation of their unbalance value from a design standpoint. However, this shape also results in inefficient distribution of material in the case of a rotating couple-type shaft, thus causing components and their support structures to be more massive and thus also often more costly than necessary.

Summary of the Invention

It is the object of the present invention to provide improved balance shafts for rotating machinery such as motor vehicle engines by enabling balance shaft design configurations which:

1.) result in lighter weight, and thus also potentially lower cost, by means of improved utilization of material in the elongated support member areas of the component for given load conditions;

2.) are stronger, having greater factor of safety for a given material usage, by means of improved utilization of material in the elongated support member areas of the component;

3.) contribute to increased bearing life due to the reduced bearing journal tilt angles that result from increased stiffness (resistance to bending under centrifugal loads) for a given material usage, by means of improved utilization of material in the elongated support member areas of the component;

4.) exhibit increased stiffness (resistance to bending under centrifugal loads) by means of improved utilization of material in the elongated support member areas of the component, with the associated benefit of reduced bearing journal tilt and thus potentially increased operating efficiency by means of smaller, and thus lower drag, bearing sizes;

5.) result in lighter weight and thus also potentially lower cost by means of improved utilization of material in the balance weight areas of the rotating couple type component; and/or

6.) reduce parasitic power loss by means of reduced "windage", or drag from air resistance, due to the reduced "frontal area" and bluntness of smaller, more efficiently shaped balance weights of the rotating couple type component.

The present invention enables the above object to be achieved by providing design methods and structures which result in improved balance shaft configurations having increased stiffness to centrifugal bending loads, increased bearing life, and/or reduced weight, with potential attendant benefits of reduced cost and friction drag. The reduced weight can allow for subsequent weight reductions in associated support structures of the engine or vehicle.

Accordingly, the present invention provides a balance shaft for rotating machinery, said shaft having a first bearing surface adjacent one end, a second bearing surface adjacent the other end, a pair of opposed balance weights and a connector portion positioned between said balance weights; said connector portion having a cross-section with at least one recessed surface.

The present invention also provides a single unbalanced-type shaft having a first bearing surface adjacent a first end, a second bearing surface adjacent the other end, a balance weight adjacent said first end, and a connector portion connecting said balance weight to said other end, said connector portion having a cross-section with at least one recessed surface.

The present invention further provides a method of optimizing the mass of a single unbalanced-type shaft for a vehicle engine, said shaft having a first bearing surface adjacent a first end, a second bearing surface adjacent the other end, a balance weight adjacent said first end, and a connector portion connecting said balance weight to said other end, said method comprising the steps of:

forming at least one recessed surface on said connector portion.

The present invention further provides a method of optimizing the mass of a balance shaft for a vehicle engine, said balance shaft having a pair of bearing surfaces at the ends thereof, a pair of opposed balance weights and a connector portion positioned between said balance weights, said method comprising the step of forming said connector portion with at least one recessed surface.

In accordance with one embodiment of the present invention, the cross-sectional shape of the elongated support member or shaft, hereafter referred to as the "connector portion", between the balance weight(s) and the driving means of the single unbalance-type balance shaft, is formed in an optimized manner to minimize material usage while maintaining required bending stiffness, torsional stiffness, and safe levels of mechanical stress. The cross-section of the connector portion is shaped substantially like an "I-beam" with recessed or concave portions. This improves the ratio of section modulus to mass in the direction of the centrifugal loads, which in turn reduces the peak stress for a given material usage. Optimization of the connector portion may involve tapering, such that the "I-beam" varies in section along its length to address the variation in bending moment along its length.

Other benefits, features and advantages of the present invention will become apparent from the following written description of the invention, when taken in accordance with the appended claims and accompanying drawings.

FIGURE 1 is a side view of an inline four-cylinder engine incorporating two single unbalance-type shafts;

FIGURE 2 is a front view of the engine shown in Figure 1;

FIGURE 3 is a side view of a single unbalance-type shaft for use in an inline four-cylinder engine;

FIGURES 3A, 3B and 3C are cross-sectional views of the balance shaft shown in Figure 3, the cross-sectional views being taken along lines A-A, B-B and C-C, respectively, in Figure 3 and in the direction of the arrows;

FIGURE 4 is a top view of the single unbalance-type shaft shown in Figure 3;

FIGURE 5 is a side elevational view of an automobile engine incorporating a rotating couple-type balance shaft in accordance with the present invention;

FIGURE 6 is a front elevational view of the engine shown in Figure 5;

FIGURES 7 and 8 depict schematic diagrams of a typical rotating couple-type balance shaft illustrating the weights, forces and moments associated therewith;

FIGURE 9 is a side elevational view of a rotating couple-type balance shaft in accordance with the present invention;

FIGURE 10 is a bottom elevational view of the rotating couple-type balance shaft as shown in Figure 9;

FIGURE 11 illustrates a manner in which the hyperbolic shape of the curved surfaces can be determined for the balance weights for a rotating couple-type balance shaft in accordance with the present invention;

FIGURE 12 illustrates the relocation of inefficient mass on a balance shaft to make it efficient in accordance with the present invention;

FIGURES 13-18 illustrate alternate embodiments of balance weights; and

FIGURES 19-21 are cross-sectional views of the balance shaft shown in Figure 9, the cross-sectional views being taken along the lines 19-19, 20-20 and 21-21, respectively, in Figure 9 and in the direction of the arrows.

Preferred embodiments of the present invention are shown in the drawings. The present invention particularly relates to improved "single unbalance"-type balance shafts, which are shown in Figures 1-4 of the drawings, and rotating couple-type balance shafts, which are shown in Figures 5-21 of the drawings.

Figures 12 to 18 and accompanying description are provided purely to aid in the understanding of the invention and the hyperbolic surface is not claimed in the present application.

Figures 1 and 2 show the side and front views, respectively, of an inline four-cylinder automobile engine 20. The engine has an engine block 22 and a crankshaft 23 which is rotated by the rods connected to the pistons 24 in the engine. A pair of balance shafts 26 is used to reduce or cancel shaking forces and/or vibration caused by the movement of the reciprocating components in the engine 20. The balance shafts 26 are "single unbalance"-type shafts and each produces a single unbalanced force. The two balance shafts 26 cancel each others' lateral shaking forces, while opposing the vertical secondary shaking forces that are caused by the engine 20.

Each of the balance shafts is typically held in position by bearings 27, 28 and 29. These bearings are held in bearing seats 27a, 28a and 29a, respectively, as shown in Figure 1. Although the location and support for only one of the two balance shafts 26 are shown in Figure 1, the second balance shaft of the pair of balance shafts for the engine 20 is positioned and held in place in substantially the same manner.

Figures 3 and 4 show the side and top views, respectively, of one of the two single unbalance-type balance shafts 26. Each of the balance shafts 26 has a nose or drive shaft 30 at one end, a connector portion 32 and a balance weight 34. The connector portion 32 is positioned between bearing surface 27 and one end of the balance weight 34, while bearing surface 29 is positioned at the opposite end of the balance weight. Bearing surface 28 is positioned in approximately the middle of the length of the balance weight 34. The balance shaft 26 rotates around its central axis 36. The balance weight 34 is semi-circular in shape, which is shown more clearly in Figures 2 and 3C.

The connector portion 32 has a pair of recesses or channels 38 and 40 on opposite sides thereof. The recesses 38 and 40 significantly reduce the overall weight of the balance shaft 26 without significantly sacrificing strength or stiffness of the balance shaft. Figures 3A, 3B and 3C show the cross-sectional size and shape of the connector portion 32 at various positions along its length. Alternatively, if desired, only one recess could be provided in the connector portion.

Another embodiment of the invention relates to rotating couple-type balance shafts which are used to reduce or cancel vibration and/or shaking forces caused by certain engines, such as the V-6 engine 52 shown in Figures 5 and 6. Engine 52 is a 90-degree V-6 engine. These engines, due to their structure and geometry, produce an imbalance couple which rotates in the opposite direction of the crankshaft, and can thus significantly benefit from a counter-rotating balance shaft of the rotating couple-type. The couple produced by the balance shaft is designed to oppose or cancel that of the engine when the balance shaft is rotating at crankshaft speed and in the opposite direction.

The inventive balance shaft is generally indicated by the numeral 50 in the drawings. The engine 52, in which the balance shaft 50 is situated, generally comprises a cylinder block 54, a pair of cylinder heads 56, a crankshaft 58, a cam shaft 60, an oil pan 62 and an air cleaner 64. A plurality of pistons 66 are positioned in cylinders 68 and connected to the crankshaft.

A nose or drive shaft 70 on the balance shaft 50 protrudes outside the front of the cylinder block 54 and has a drive gear or sprocket 72 attached to it. The gear 72 is attached in any conventional manner, such as bolt 74. Gear 72 is also oriented to the drive shaft 70 by a slot and key mechanism (not shown) or by any other conventional means.

The camshaft 60 and crankshaft 58 also have noses or drive shafts 80 and 82, respectively, which protrude outside the front of the cylinder block 54. Nose 80 of camshaft 60 is secured to drive gear 84 and sprocket 86. The nose 82 of crankshaft 58 is secured to drive sprocket 88. A vibration damper 90 is also preferably attached to the nose 82 of the crankshaft 58. Sprockets 86 and 88 are connected by a conventional drive chain or toothed timing belt 92. Drive gear 84 is meshed with gear 72 on the balance shaft 50.

Sprockets 86 and 88 are both rotated in the same direction by the drive chain or toothed timing belt 92, as shown in Figure 6. The respective sizes and diameters of sprockets 86 and 88 are such that the crankshaft 58 rotates at twice the speed of the camshaft 60.

The meshing of gears 72 and 84 causes the balance shaft 50 to rotate in a direction opposite to that of the crankshaft and thus counterbalance the vibrations caused by the engine 52. The size and diameters of the gears 84 and 72 determine the rotational speed of the balance shaft 50. Typically, shaft 50 is rotated at twice the speed of the camshaft 60, and the same speed as the crankshaft 58.

The shape and characteristics of a conventional rotating couple-type balance shaft are shown schematically in Figures 7 and 8. As shown in Figure 7, the balance shaft 100 has a pair of bearing surfaces 101 and 102, a pair of balance weights 103 and 104 and a connector portion 105. The balance weights 103 and 104 have centers of gravity "CG_1" and "CG_2, respectively, at the points shown. The balance shaft 100 rotates about a central longitudinal axis 106. As shown, the balance weights 103 and 104 are on opposite sides of the axis 106. The cross-sectional shapes of the balance weights 103 and 104 can be of any cross-section, but typically are "circular segment" shaped, where the straight inside edge of the weight represents constant contribution to unbalance within the section.

The balance shaft's unbalance couple "C_u", required to offset that of the engine is based on the masses and geometry of the engine. This is calculated by conventional methods known in the art. The unbalance couple can be expressed by the equation :

$$C_u = LR_1W_1 + LR_2W_2 \quad (1)$$

where L is one-half the length or distance between the centers of gravity CG_1-CG_2 of the balance weights, R_1 and R_2 are the distances from the axis of rotation 106 to the centers of gravity of the balance weights, and W_1 and W_2 are the masses or weights of the balance weights. These distances and weights are expressed in the diagram shown in Figure 8.

When the engine is designed, the dimensions of the cavity for placement of the balance shaft are determined. In this regard, the length between the bearings which house the bearing surfaces 101 and 102 is determined, together with the clearance boundary radius/radii of the balance shaft. The shape and configuration of the balance shaft is constrained within these boundaries. As a result, in accordance with equation (1) set forth above, if it is desired to decrease the weights W of the balance weights, then the distances L or R can vary to the extent permitted by the boundary conditions in order to meet the requisite couple C_u for the engine.

A rotating couple-type balance shaft 50 made in accordance with the present invention is shown in Figures 9 and 10. The balance shaft 50 has a pair of bearing surfaces 111 and 112, a pair of balance weights 113 and 114 and a central connector portion 115 which extends between the balance weights. The balance shaft rotates about a longitudinal axis 118.

The balance weights 113 and 114 may have curved or straight gusset portions 120 and 122 which are used to integrally connect the balance weights to the bearing surfaces 111 and 112, respectively. These add strength to the structure.

Surfaces 111 and 112 on the ends of the balance shaft are manufactured in order to allow proper fitting in bearings 124 and 126, respectively, in the engine (as shown in Figure 5). When the balance shaft 50 is mounted in the engine 52, bearings 124 and 126 are positioned to allow the balance shaft to rotate freely. The nose 70 of the balance shaft 50 is positioned at one end of the balance shaft and is configured to extend outside the cylinder block 54 and be connected to the drive gear 72, as discussed above. As indicated earlier, the drive gear 72 rotates the balance shaft 50 in the direction and at the speed desired for the engine.

Although the drawings and above description disclose that the balance shaft is mounted in the engine by bearings positioned at the two ends of the balance shaft, it is also possible to position the bearings at intermediate positions spaced from the ends of the shaft, for example within the length of the balance weights. Further, more or less than two bearings can be provided.

Surface 128 of balance weight 113 and surface 130 of balance weight 114 are manufactured to have a curved surface. As shown in Figure 10, the curves of the surfaces 128 and 130 also allow the sides of the balance weights 113 and 114 to form curves which taper from the bearing surfaces 111 and 112 toward the connector portion 115.

In Figure 9, the surfaces 128 and 130 are formed as hyperbolic curves, or approximations of hyperbolic curves. This feature is better shown in Figure 11. In that Figure, one half of balance shaft 50 is shown superimposed on an X-Y grid. The axis of rotation 118 of the balance shaft is aligned along the X-axis, and the intersection of the X and Y axes is positioned at the center P of the couple. As shown, the curve of the surface 130 of balance weight 114 is formed along a hyperbola in accordance with the equation:

$$(X) \times (Y) = C \qquad (2)$$

The desired output of the rotating couple-type shaft is a pure couple of specific magnitude. This output requires that both unbalances $(R_1) \times (W_1)$ and $(R_2) \times (W_2)$ be equal, or a couple plus a residual unbalance will result. Thus the "half moment" distance L can be defined, in simplification (for purposes of discussion and as shown in Figures 7 and 8) of the more general equations summing forces and moments, as also equal for each side, namely the axial distance from one CG to point P midway between the CG's.

Disregarding, also for purposes of simplifying the discussion, the unbalance contributions due to connector portions, gussets, and the like, it may be seen that the contribution to the magnitude of the rotating unbalance couple made by any element of mass within the balance weight is a function of that element's location, specifically the product of its axial distance from the centerline of the unbalance couple and its radial distance from the shaft's rotational centerline, when viewed normal to the plane of the unbalance couple as in Figures 7, 9 and 11. From this, it can be seen that locations with an (X)x(Y) product greater than a reference value "C" represent more efficient use of material than locations having lesser products. Therefore, in order to secure mass reduction for balance shafts of the rotating couple-type in accordance with the present invention, mass (balance weight material not dedicated to structural purposes such as connector portions, gussets and the like) is relocated from low (X)x(Y) product locations to more efficient locations having products greater than or equal to a reference value "C".

A general representation of this relocation is shown in Figure 12. In that Figure, the profile of a typical rectangular counterweight 114' is indicated by the reference numeral 119. The balance shaft rotates around axis 118 and has a connector portion 115. The inefficient portion 121 of the counterweight mass is situated below the envelope or area defined by hyperbolic curve C'. In accordance with the present invention, the inefficient mass portion 121 is effectively relocated to position 123 above the hyperbolic curve C′ on the balance shaft in order to provide the required unbalance moment with less material.

The preferred mode for the present invention for rotating couple-type shafts is to add or subtract material uniformly along the full length of the side elevation hyperbolic surfaces defined by the equation (X)x(Y)=C, or Y=C/X. The value of C is adjusted until the target unbalance couple magnitude is reached and after a full utilization of the clearance boundary radius/radii has been made.

In cases where a single radius defines the clearance boundary envelope, the balance shaft will be symmetrical (except for the effects of differences in features dedicated to structural purposes), having common C value for both of the balance weights. Where clearance boundary conditions differ, i.e., where multiple radii define different envelope sizes or shapes for the two balance weights, mass optimization will involve use of differing values for C in order to equate (R)x(W) unbalances between the two balance weights. The differing C values will result in differing CG locations, thus influencing the distance between CG′s, and hence the value of distance L, which is a determinant in the unbalance moment′s magnitude. In this case of differing boundary conditions, it will be necessary to determine the distinctly different C values that will provide for the target unbalance couple magnitude while fully utilizing the clearance boundary envelope, in order to avoid any residual unbalance which would result from unequal (R)x(W) unbalance values.

It is preferred that the shape of the surface 130 be a curve of a true hyperbola (as shown in Figures 11 and 13). In Figure 13, the hyperbolic shape is shown by phantom line H which is a continuation of the curve which forms surface 130 on balance weight 114.

It is also possible, however, for the surface 130 to have a shape that is a reasonable approximation of a hyperbolic curve. Examples of these are shown in Figures 14-18. For example, as shown in Figure 14, the surface 130a has a generally curved surface. Surface 130a is formed as part of a large circle having radius R_L. In Figure 15, a series of straight line segments 130b are used to approximate the hyperbolic shape. In this regard, although three straight line segments are shown in Figure 15 approximating a hyperbolic curve, it is understood that any number of straight line segments could be utilized.

In Figure 16, the curved surface 130c is formed from a combination of a straight line 131 and a curved line 132. In this regard, the curved portion 132 is formed as a part of a small circle having radius R_s. As shown in Figure 17, the curved surface 130d is formed as a truncated hyperbola 133 with a blunt end portion 135. Also, in Figure 18, the hyperbolic curve is approximated by a series of straight lines 130e and has a truncated or blunt end 137. It is understood that the blunt end portion 137 can be used with any of the previous contour variations. A blunt end 137 can be provided, for example, due to manufacturing and/or design considerations.

Moreover, it is also possible that the shape of surface 130 could be a portion of another geometric figure, such as a portion of a parabola or an ellipse, and still constitute a reasonable approximation of a hyperbolic curve or shape.

As stated above, the curved shape of the balance weight allows the product of the length L which extends from couple midpoint P to the centers of gravity CG_1 and CG_2 of the balance weights and the radii R_1 and R_2 to the CG's (see Figures 7 and 8), to be maximized by means of material distribution along the hyperbolic surface 130, thus avoiding inefficiently located material which would fall below the threshold of "equal efficiency," i.e., having constant contribution to unbalance. (This is shown in Figure 12 where the curve is designated by the letter C'.) This in turn allows the mass or weight W of the balance weights to be minimized.

The particular curve actually utilized for the balance weights of the balance shaft, such as curve C_2 which forms surface 130 in Figure 11, is selected in accordance with the length and weight parameters afforded by the engine's clearance envelope and the correcting couple needed. In this regard, as shown in Figure 11, a balance weight having a curved surface along curve C_1 would provide a lower unbalance moment, while curve C_3 a greater unbalance moment, than curve C_2. The needed unbalance couple is thus obtained by means of the appropriate value(s) for constant C, thus avoiding unnecessary weight or mass.

In accordance with the present invention, the cross-sectional size and shape of the connector portion 115 is optimized for given load conditions in order to minimize its mass and thus the weight of the balance shaft 50. Figures 19, 20 and 21 illustrate a preferred shape of the connector portion 115 of the balance shaft 50 shown in Figures 9 and 10.

As shown in Figures 19-21, the sides 140 and 142 of the connector portion 115 are recessed or shaped in a concave manner. This lightens or reduces the weight of the balance shaft without significantly reducing its resistance to bending in the plane of balance weight centrifugal loading. Essentially, the cross-sectional shape of the connector portion 115 has a generally "I-beam" shape. This maximizes the section modulus in the direction of the centrifugal loads. This in turn minimizes the peak stress for a given amount of material usage. Alternatively, only one recess could be provided in the connector portion.

Figure 21 shows a cross-sectional view of the balance shaft 50 including a portion of the connector portion 115 and a portion of the balance weight 114. As shown, the weight or mass of the balance shaft is distributed over a wider area to maintain section modulus and avoid stress concentrations. It is clear that other cross-sectional shapes and proportions for the connector portions 115 and transition areas to the balance weights 114 can be utilized in accordance with the present invention.

Although particular embodiments of the present invention have been illustrated in the accompanying drawings and described in the foregoing detailed description, it is to be understood that the present invention is not to be limited to just the embodiments disclosed, but that they are capable of numerous rearrangements, modifications and substitutions without departing from the scope of the claims hereafter.

CLAIMS

1. A balance shaft for rotating machinery, said shaft having a first bearing surface adjacent one end, a second bearing surface adjacent the other end, a pair of opposed balance weights and a connector portion positioned between said balance weights; said connector portion having a cross-section with at least one recessed surface.

2. A single unbalanced-type shaft having a first bearing surface adjacent a first end, a second bearing surface adjacent the other end, a balance weight adjacent said first end, and a connector portion connecting said balance weight to said other end, said connector portion having a cross-section with at least one recessed surface.

3. The balance shaft as set forth in claim 1 or 2, wherein said cross-section has substantially an I-beam shape.

4. A balance shaft as set forth in any one of claims 1 to 3, wherein said cross-section is substantially an I-beam shape with two recessed surfaces.

5. A balance shaft as set forth in claim 2, 3, or 4 (when dependent on claim 2), wherein said connector portion tapers in cross-section along the longitudinal length thereof.

6. A method of optimizing the mass of a single unbalanced-type shaft for a vehicle engine, said shaft having a first bearing surface adjacent a first end, a second bearing surface adjacent the other end, a balance weight adjacent said first end, and a connector portion connecting said balance weight to said other end, said method comprising the steps of:

forming at least one recessed surface on said connector portion.

7. A method of optimizing the mass of a balance shaft for a vehicle engine, said balance shaft having a pair of bearing surfaces at the ends thereof, a pair of opposed balance weights and a connector portion positioned between said balance weights, said method comprising the step of forming said connector portion with at least one recessed surface.

8. The method of claim 6 or 7, further comprising the step of forming said connector portion in a substantially I-beam shape with two recessed surfaces.

9. The method of claim 6, further comprising the step of forming said connector portion to have a longitudinally tapered cross-section.

The Patent Office

Application No:	GB 0001829.1	**Examiner:**	John Twin
Claims searched:	1 to 9	**Date of search:**	11 February 2000

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.R): F1M (M18); F2U (U1)

Int Cl (Ed.7): F02B 75/06; F16F 15/26

Other: online: EPODOC, JAPIO, WPI

Documents considered to be relevant:

Category	Identity of document and relevant passage		Relevant to claims
X	WO 92/02720 A1	(Audi) - see eg figure 12	1,2,7,6
X	US 5483932	(Simpson) - see the recesses 66,68 in fig.3	1,2,7,6
X	US 5375571	(Ford) - see eg figure 11	1,2,7,6

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
Y	Document indicating lack of inventive step if combined with one or more other documents of same category.	P	Document published on or after the declared priority date but before the filing date of this invention.
		E	Patent document published on or after, but with priority date earlier than, the filing date of this application.
&	Member of the same patent family		