PATENT SPECIFICATION

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Complete Specification Accepted: Aug. 15, 1939.

PROVISIONAL SPECIFICATION

Improvements in or relating to Rotary Valves for Internal Combustion Engines

I, FRANK METCALF ASPIN, a British subject, of Walmer Place, 149, Walmersley Road, Bury, Lancashire, do hereby declare the nature of this invention to be as follows:—

This invention relates to rotary valves for internal combustion engines of the kind having a rotary valve of conical or other tapered external shape of a volume of revolution about the axis of rotation, which external surface coacts with a correspondingly shaped internal surface in a housing in which it rotates the said coacting surfaces together forming a gas seal and the principal pressure on the valve being axial and in the direction to load such tapered surfaces if acting as bearing surfaces.

An example of such kind of engine is described in the Specification of my Patent No. 463,412 to which type of engine this invention is particularly applicable, though it must be clearly understood that this invention is not restricted to such application.

One of the problems in connection with the rotary valve of internal combustion engines of the kind described is the provision of an effective gas seal whilst at the same time avoiding undue wear and friction. For that purpose it has been proposed, for example in my aforesaid Patent Specification, to provide anti-friction thrust bearings, adjustable so that the clearance at the tapered surfaces can be set to reduce the gap to a minimum whilst the load is taken by the anti-friction bearing. Such an arrangement is not however without disadvantages, especially when the axial pressures are high as space limitations tends to limit the bearing size and the life of such anti-friction bearings is therefore liable to be short, and the rate of wear very high, making the maintenance of the clearance adjustment almost impossible.

The object of the invention is a new method of and means for mounting such valve member and is based upon experimental research and theory. As regards theory it is well known that every substance has some degree of elasticity so that even an anti-friction ball or roller bearing of hard steel has a measurable resilient yield under safe working loads, even if such yield is only about one ten-thousandth of an inch. An anti-friction ball or roller bearing will withstand considerable loading in ordinary use without excessive wear taking place but there is a limit though not sharply defined, as is well known and accepted, beyond which the rate of wear increases rapidly and below which the loads may be termed safe working loads.

In practical design and construction of valve gear of the kind described and especially in the case of the engine of my aforesaid patent in which unusually high compression and explosive pressures obtain, it becomes difficult, if not sometimes impossible to provide an anti-friction bearing of such dimensions as to avoid overloading, but even assuming that were possible, the present invention constitutes an appreciation that a bearing of such large proportions is unnecessary for a rotary valve of the kind to which this invention relates.

Having appreciated that overloading of such kind of valve may be considered as temporary and not continuous, as it will occur only during moments of maximum pressure in the cylinder, and having further appreciated that any axial loading on the anti-friction bearing is accompanied by axial movement of the valve due to yield of the bearing, however small, and further bearing in mind that, as in the Specification of my aforesaid Patent, an oil film may be maintained at the tapered surfaces of the valve as an effective means of providing a gas seal, I have conceived the idea that, the total axial load could be distributed between an anti-friction bearing and such lubricated tapered surfaces whereby overloading of the anti-friction bearing can be avoided.

According to the invention the valve of the kind described is constructed and adjusted as to its anti-friction bearing and its lubricated tapered surface so that the yield of the anti-friction bearing within the limits of its safe working loads permits such axial movement of the valve as will enable an increasing proportion of the load to be taken on the said lubricated tapered surfaces.

According to a further feature of the invention, an anti-friction bearing is selected, such as a conical roller bearing having an axial movement owing to the inclination of its bearing surfaces greater than the actual compression of its bearing surfaces measured in a direction normal thereto.

In one example of the invention, the rotary valve has a conical head of an angle of 60 degrees (included angle) such head being substantially only a steel shell about 3/32 inches thick filled with a core of aluminium in which the valve chamber is formed in accordance with my aforesaid Patent No. 463,412. This construction is adopted partly to reduce the weight of the eccentric mass but it also has other advantages. This conical head is approximately $3\frac{5}{8}$ inches diameter at its lower or larger end and $1\frac{1}{4}$ inches diameter at its upper or smaller end and at each end merges into a short cylindrical portion about $\frac{1}{8}$th inch long. The shaft, about which the valve rotates is formed integral with the head part above described and extending coaxially from the smaller end thereof and adjacent to such end it is ground to receive the inner races of a pair of conical roller bearings. These roller bearings are selected from manufacturers standard sizes and are arranged so that the inner race of the lower bearings is conical in the same direction as the head of the valve, whilst the other is oppositely arranged. They are selected for size, so as to be capable of taking about one third of the maximum calculated load during combustion, estimated from the calculated explosion pressures. The inner race of the lower bearing is made a tighter push fit on the shaft than that of the upper bearing, and such lower inner race rests against the shoulder formed at the junction of the shaft with the smaller cylindrical end of the head part. This difference of fit is for a definite purpose, as explained hereafter. The outer races are supported on a shoulder of the housing in which the valve rotates, there being a spacing ring between the race and such shoulder and also a similar ring between the two outer races. These bearings are standard commercial articles having a specified angle measured in the accepted manner on the axes of their rollers of about 20 degrees and the outer races are located in the end of a steel shell, fitted into a recess in the upper end of the valve housing, such shell having a shoulder adapted to engage the upper face of the outer race of the upper bearing so as to position and finally secure the outer races against end movement, the shell being secured to the housing by a suitable plate, engaging an outer shoulder on the shell and bolted to the top of the housing. At its upper end this steel shell is adapted to hold the outer race of a ball bearing. On the shaft above the inner race of the upper conical bearing is a collar, which is an easy sliding fit on the shaft and has a flange engaging the said upper race and extending in diameter almost to the wall of the steel shell so as to enable oil pressure to be established within the region of these bearings. Oil ducts are provided in the shaft comprising a diametrical through hole located between the inner races, an axial hole therefrom downwards towards the head part, and a second diametrical through hole, the end of which emerge at the small end of the head cone. Oil through these passages reaches grooves in the conical face of the housing in which the head fits and rotates. On the shaft above the collar aforesaid is keyed or splined a skew gear wheel whilst above that is the inner race of the upper ball race, above which is a washer and a strong coil spring secured by a locking washer and nut on the extreme end of the shaft. The skew gear wheel, inner ball race and washer are all made a good but easy sliding fit on the shaft, so that the pressure obtainable from the spring which may conveniently be 100 lbs. may act to pull the valve head resiliently up into its housing.

In assembly, the valve is assembled dry, that is, without lubricant, into its housing and the lower spacing ring at the outer races of the conical bearing is selected of such thickness that appreciable rubbing contact at the conical faces can be felt to occur when loaded axially to approximate maximum working loads. After assembly, lubricant can easily be induced to enter between the conical faces.

The lubricating oil grooves for the conical surfaces of the rotary valve are preferably of the known type, having an inclined trailing edge or corner so that some of the lubricant is drawn in between the surfaces to be lubricated in known manner with such grooves and at high pressure.

Thus, in operation, there is an effective oil film between the coating conical surfaces of the valve head and its housing and the normal working pressures are carried partly on such oil film but mainly on the lower conical roller bearing. During peak pressures, the normal yield of the roller bearings, though extremely small, is greater than the yield of the oil film between the conical valve surfaces and thus as the load on the valve increases a progressively greater proportion of such increase is taken by the oil film so that the roller bearing is protected from being overloaded. In this manner, frictional losses are reduced to a minimum as the anti-friction roller bearings support the greater part of the normal loads whilst the higher frictional losses during the periods of maximum pressures are only during short periods.

Such combination of oil film and anti-friction bearings has been proved in practice to be entirely satisfactory, especially with conical roller bearings and it has been found that no adjustment for wear is necessary. During one prolonged test it was found that even where the wear was sufficient to produce axial lifting of the valve head as much as 10 thousandths of an inch, both the conical valve surfaces and the roller bearing surfaces were in excellent condition. Furthermore, it must be appreciated that the spring, on the end of the valve shaft ensures that the conical roller bearings are always maintained in mutual adjustment to take up wear, because, as the valve shaft rises with wear so the inner race of the upper conical roller bearing is forced down the shaft to take up any but the necessary working clearance.

The invention is obviously not limited to the details of the example above given as obviously the proportions, size and angles are capable of variation without departing from the nature of the invention. Clearly, the effective combined action of the two kinds of bearings, plain, oil film and anti-friction roller or ball, is very dependent upon many features of construction and design, and particularly upon the resilience of the materials used in the construction of the valve and valve seating, the distance or length of valve stem between the valve head and the anti-friction bearing, the angle of the conical valve faces, and the natural resiliency of the anti-friction bearing under normal load, which, of course, will depend upon the materials of its construction and the angles of its conical race, of which standard types are already obtainable with angles between 15 and 45 degrees. It is probable that deep groove and such types of anti-friction bearings could be employed instead of conical roller bearings though there is an obvious natural preference for the latter type. Probably a steeper angle of conical valve surface would be required in such case if the latter types of bearing permit less axial movement under normal loads. It is obvious that proper initial adjustment as well as design are both necessary for full and efficient advantage to be obtained from the combination of the two types of bearings.

It is to be appreciated that during the induction period, very low-working loads occur and when the throttle is closed, negative pressures may occur and the coil spring is required to resist downward pull on the valve.

The strength of the spring on the end of the shaft is therefore determined, not only by the sliding fit resistance of the upper inner conical roller bearing race which it should be able to overcome in order to keep the valve against its other bearings, but also upon the downward forces which it must resist under working conditions. The partial vacuum for example in a high compression engine, say 12 to 1 compression ratio, and with a valve of the dimensions given above, has been found to require a spring giving about 100 lbs. pressure. This spring may be short and of high rating, such as to be practically a spring washer, adjusted until almost flattened.

The upper conical roller race may be replaced by a ball thrust bearing to take the pull of the coil spring. Conveniently a plain bearing may also be combined with the ball thrust bearing to act as a steady for the valve shaft and having also the advantage of providing an oil seal replacing the flange previously provided between the races for maintaining oil pressure on the lower conical race and cone surfaces.

Dated this 28th day of February, 1938.
For the Applicant,
WILSON, GUNN & ELLIS,
Chartered Patent Agents,
54/56, Market Street, Manchester, 1.

COMPLETE SPECIFICATION

Improvements in or relating to Rotary Valves

I, FRANK METCALF ASPIN, a British subject, of Walmer Place, 149, Walmersley Road, Bury, Lancashire, do hereby declare the nature of this invention, and in what manner the same is to be performed, to be particularly described and ascertained in and by the following statement:—

This invention relates to rotary valves, for example for internal combustion engines, of the kind having a conical or other tapered external shape of a volume of revolution about the axis of rotation, which external surface coacts with a correspondingly shaped internal surface in a housing in which it rotates the said coacting surfaces together forming a gas seal and the principal pressure on the valve being axial and in the direction to load such tapered surfaces if acting as bearing surfaces. The rotary valve improvements of this invention are also applicable to compressors, pumps, steam engines and other apparatus.

An example of the kind of engine to which the valve above referred to is applicable is described in the Specification of my Patent No. 463,412 to which type of engine this invention is particularly applicable, though it must be clearly understood that this invention is not restricted to such application.

One of the problems in connection with a rotary valve of the kind described for an internal combustion engine is the provision of an effective gas seal whilst at the same time avoiding undue wear and friction. For that purpose it has been proposed, for example, in my aforesaid Patent Specification, to provide anti-friction thrust bearings, adjustable so that the clearance at the tapered surfaces can be set to reduce the gap to a minimum whilst the load is taken by the anti-friction bearing. Such an arrangement is not however without disadvantages, especially when the axial pressures are high as space limitations tend to limit the bearing size and the life of such anti-friction bearing is, therefore, liable to be short, and the rate of wear very high, making the maintenance of the clearance adjustment almost impossible.

The object of the invention is a new method of and means for mounting such valve member and is based upon experimental research and theory. As regards theory it is well known that every substance has some degree of elasticity so that even an anti-friction ball or roller bearing of hard steel has a measurable resilient yield under safe working loads, even if such yield is only about one ten-thousandth of an inch. An anti-friction ball or roller bearing will withstand considerable loading in ordinary use without excessive wear taking place but there is a limit though not sharply defined, as is well known and accepted, beyond which the rate of wear increases rapidly and below which the loads may be termed safe working loads.

In practical design and construction of valve gear of the kind described and especially in the case of the engine of my aforesaid Patent in which unusually high compression and explosive pressure obtain, it becomes difficult, if not sometimes impossible to provide an anti-friction bearing of such dimensions as to avoid overloading, but even assuming that were possible, the present invention constitutes an appreciation that a bearing of such large proportions is unnecessary for a rotary valve of the kind to which this invention relates.

Having appreciated that overloading of such kind of valve may be considered as temporary and not continuous, as it will occur only during moments of maximum pressure in the cylinder, and having further appreciated that any axial loading on the anti-friction bearing is accompanied by axial movement of the valve due to yield of the bearing, however small, and further bearing in mind that, as in the Specification of my aforesaid Patent, an oil film may be maintained at the tapered surfaces of the valve as an effective means of providing a gas seal, I have conceived the idea that, the total axial load could be distributed between an anti-friction bearing and such lubricated tapered surfaces whereby overloading of the anti-friction bearing can be avoided.

According to the invention the valve of the kind described is constructed and adjusted as to its anti-friction bearing and its lubricated tapered surface so that the yield of the anti-friction bearing within the limits of its safe working loads permits with increase of load such axial movement of the valve as will enable an increasing proportion of the load to be taken on the said lubricated tapered surfaces.

According to a further feature of the invention, an anti-friction bearing is selected, such as a conical roller bearing, having an axial movement owing to the inclination of its bearing surfaces greater than the actual compression of its bearing surfaces measured in a direction normal thereto.

In the accompanying drawings:—

Fig. 1 is a longitudinal part sectional view showing a rotary valve assembly made in accordance with one example of the invention.

Fig. 2 shows a modified construction of the valve shown in Fig. 1.

Fig. 3 shows a further modified construction.

As illustrated in Fig. 1, the rotary valve has a conical head *a* of an angle of 60 degrees (included angle) such head being substantially only a steel shell *b* about 3/32 inches thick filled with a core of aluminium c in which an eccentric valve chamber d is formed in accordance with my aforesaid Patent, No. 463,412. This construction is adopted partly to reduce the weight of the eccentric mass, but it also has other advantages. This conical head is approximately $3\frac{5}{8}$ inches diameter at its lower or larger end and $1\frac{1}{4}$ inches diameter at its upper or smaller end and at each end merges into a short cylindrical portion a^1 and a^2 about $\frac{1}{8}$ inch long. The shaft or stem e about which the valve rotates is formed integral with the shell b of the head part a above described and extending coaxially from the smaller end thereof and adjacent to such end it is ground to receive the inner races f and g of a pair of conical roller bearings. These roller bearings are selected from manufacturers standard sizes and are arranged so that the inner race of the lower bearings is conical in the same direction as the head of the valve, whilst the other is oppositely arranged. They are selected for size, so as to be capable of taking about one-third of the maximum calculated load during combustion, estimated from the calculated explosion pressures. The inner race f of the lower bearing is made a tighter push fit on the shaft e than that g of the upper bearing, and such lower inner race rests against the shoulder formed at the junction of the shaft e with the smaller cylindrical end a^1 of the head part. This difference of fit is for a definite purpose, as explained hereafter. The outer races f^1 and g^1 are supported on a shoulder h of the housing in which the valve rotates, there being a spacing ring f^2 between the race f^1 and such shoulder and also a similar ring g^2 between the two outer races. These bearings are standard commercial articles having a specified angle measured in the accepted manner on the axis of their rollers of about 20 degrees, and the outer races are located in the end of a steel shell i fitted into a recess in the upper end of the valve housing,, such shell having a shoulder i^1 adapted to engage the upper face of the outer race g^1 of the upper bearing so as to position and finally secure the outer races against end movement, the shell i being secured to the housing by a suitable plate j, engaging an outer shoulder on the shell and bolted to the top of the housing. At its upper end this steel shell is adapted to hold the outer race k of a ball bearing. On the shaft e above the inner race g of the upper conical bearing is a collar l, which is an easy sliding fit on the shaft and has a flange l^1 engaging the said upper race and extending in diameter almost to the wall of the steel shell i so as to enable oil pressure to be established within the region of these bearings. Oil ducts are provided in the shaft comprising a diametrical through hole m located between the inner races, an axial hole n therefrom downwards towards the head part, and a second diametrical through hole o, the ends of which emerge at the small end of the head cone. Oil through these passages reaches grooves in the coacting conical bearing face p of the housing in which the head fits and rotates. On the shaft e above the collar l aforesaid is keyed or splined a skew gear wheel q whilst above that is the inner race k^1 of the upper ball race, above which is a washer r and a strong coil spring s secured by a locking member t and nut t^1 on the extreme end of the shaft e. The skew gear wheel q, inner ball race k^1 and washer r are all made a good but easy sliding fit on the shaft e, so that the pressure obtainable from the spring s which may conveniently be 100 lbs, may act to pull the valve head a resiliently up into its coacting conical bearing surface p.

In assembly, the valve is assembled dry, that is, without lubricant on the head a, into its housing and the lower spacing ring f^2 at the outer races of the conical bearing is selected of such thickness that appreciable rubbing contact at the conical faces of b against the coacting conical face p can be felt to occur when loading axially to approximate maximum working loads, which can be obtained by tightening the nut t^1. After the correct ring has been found the nut can be reset for the required spring loading. After assembly, lubricant can easily be induced to enter between the conical faces.

Lubricating oil grooves may be provided for the conical surfaces of the rotary valve and are preferably of the known type, having an inclined trailing edge or corner so that some of the lubricant is drawn in between the surfaces to be lubricated in known manner with such grooves and at high pressure.

Thus, in operation, there is an effective oil film between the coacting conical surfaces of the valve head a and its conical housing surface p and the normal working pressures are carried partly on such oil film but mainly on the races f and f^1 of the lower conical roller gearing. During peak pressures, the normal yield of the races f and f^1 and of the roller bearings between them, though extremely small, is collectively greater than the yield of the oil film between the conical valve surfaces and thus as the load on the valve increases the valve head moves axially and a progressively greater proportion of such load increase is taken by the oil film at the coacting conical bearing surfaces so that the roller bearing is protected from being overloaded. In this manner, frictional losses are reduced to a minimum as the anti-friction roller bearings support the greater part of the normal loads whilst the higher frictional losses during the periods of maximum pressures are only during short periods.

Such combination of tapered bearing surface with oil film and of anti-friction bearing has been proved by me in practice to be entirely satisfactory, especially with conical roller bearings and it has been found that if properly designed and assembled in the first place no adjustment for wear is necessary. During one prolonged test it was found for instance that even where the wear was sufficient to produce axial lifting of the valve head as much as 10 thousandths of an inch, both the conical valve surfaces and the roller bearing surfaces were in excellent condition. Furthermore, it must be appreciated that the spring *s* on the end of the valve shaft ensures that the conical roller bearings are always maintained in mutual adjustment to take up wear, because, as the valve shaft rises with wear, the inner race *g* of the upper conical roller bearing is forced down the shaft to take up any but the necessary working clearance.

As shown in Fig. 2, the valve head 10 is rotatably mounted in a coacting conical face 11 whilst on its stem 12 are carried the inner races 13 and 14 of two conical roller bearings arranged in the same, instead of in opposite, directions. Axial pressure on the bearings is maintained by a spring 15 secured by a nut 16 acting through an upper ball bearing 17 carried in a shell 18 secured to the housing 19 and having a shoulder 20 to support the ball bearing. In this arrangement, some small lateral movement or flutter of the valve is possible.

As shown in Fig. 3, the valve 21 is rotatably mounted in a coacting conical face 22 and on its stem 23, it carries inner races 24, 25 and 26 of a roller bearing, conical roller bearing and ball bearing respectively. All the outer races of these bearings are carried in a shell 27. Axial pressure on the conical thrust bearing and the coacting conical faces is maintained by means of the spring 28 acting through the bearing race 26 through which the shaft 23 is slidable. In this arrangement, the valve is very firmly located against flutter or loss of axial alignment within its housing.

The invention is obviously not limited to the details of the examples above given, as obviously the proportions, size and angles are capable of variation without departing from the nature of the invention. Clearly, the effective combined action of the two kinds of bearings, plain oil film and anti-friction roller or ball, is very dependent upon many features of construction and design, and particularly upon the resilience of the materials used in the construction of the valve and valve seating, the distance or length of valve stem between the valve head and the anti-friction bearing, the angle of the conical valve faces, and the natural resiliency of the anti-friction bearing under normal load, which, of course, will depend upon the materials of its construction and the angles of its conical races, of which standard types are already obtainable with angles between 15 and 45 degrees. More important still is probably the return axial movement of the valve plug by the thrust bearing when the peak load falls as this enables the lubrication film between the tapered bearing surfaces to be restored to normal thickness and allows the lubricant to be distributed and losses made up. Probably a different angle of conical valve surface would be required in some cases to permit more or less axial movement under normal loads. It is obvious that proper initial adjustment as well as design are both necessary for full and efficient advantage to be obtained from the combination of the two types of bearings, namely anti-friction and lubricant film bearings respectively.

It is to be appreciated that during the induction period, very low-working loads occur and when the throttle is closed, negative pressures may occur and the coil spring is required to resist downward pull on the valve.

The strength of the spring on the end of the shaft is therefore determined, not only by the sliding fit resistance of the upper inner conical roller bearing race which it should be able to overcome in order to keep the valve against its other bearings, but also upon the downward forces which it must resist under working conditions. The partial vacuum for example in a high compression engine, say 12 to 1 compression ratio, and with a valve of the dimensions given above, has been found to require a spring giving about 100 lbs. pressure. This spring may be short and of high rating, such as to be practically a spring washer, adjusted until almost flattened.

The upper conical roller race may be replaced by a ball thrust bearing to take the pull of the coil spring. Conveniently a plain bearing may also be combined with the ball thrust bearing to act as a steady for the valve shaft and having also the advantage of providing an oil seal replacing the flange previously provided between the races for maintaining oil pressure on the lower conical race and cone surfaces.

Having now particularly described and ascertained the nature of my said invention, and in what manner the same is to be performed, I declare that what I claim is:—

1. A rotary valve assembly comprising a rotary tapered valve plug, and bearings therefor comprising a coacting tapered plain bearing surface on the valve and valve seating and an anti-friction thrust bearing, the said bearings being relatively set so that the load is always divided between them and the anti-friction thrust bearing being more yielding than the tapered plain bearing surface so that as the valve plug moves axially with increase of load a progressively increasing proportion of such load increase is taken by the tapered plain bearing surface.

2. A rotary valve assembly according to claim 1, characterised in that the axial movement of the valve on maximum load is less than a predetermined maximum safe axial compression of the thrust bearing.

3. A rotary valve assembly according to claim 1 or 2, characterised in that the anti-friction thrust bearing is a conical roller bearing.

4. A rotary valve assembly according to claim 1 or 2, characterised in that the anti-friction bearing provides an axial movement greater than the actual compression of its bearing surfaces measured in a direction normal thereto.

5. A rotary valve assembly according to any of the preceding claims, characterised in that the bearings are set so that the anti-friction bearings take the greater portion of the normal load.

6. A rotary valve assembly according to any of the preceding claims, characterised by one or more further bearings arranged to maintain axial alignment of the tapered plug in its coacting tapered bearing surface whilst permitting axial movement.

7. A rotary valve assembly according to any of the preceding claims 1 to 5, characterised in that the rotary valve plug has a stem on which is mounted the anti-friction thrust bearing.

8. A rotary valve assembly according to claim 6, characterised in that the rotary valve plug is formed with a stem and the anti-friction bearing and the one or more further bearings are all arranged on such stem.

9. A rotary valve assembly according to claim 6 or 8, characterised by two further bearings, one on each side of the anti-friction conical thrust bearing.

10. A rotary valve assembly according to claim 6, 8 or 9, characterised in that each further bearing is an anti-friction ball or roller bearing.

11. A rotary valve assembly, constructed, arranged and operating substantially as herein described with reference to and as illustrated in the accompanying drawings.

Dated this 28th day of February, 1939.
For the Applicant,
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Chartered Patent Agents,
54/56, Market Street, Manchester, 1.

Leamington Spa: Printed for His Majesty's Stationery Office, by the Courier Press.—1939.

[This Drawing is a reproduction of the Original on a reduced scale.]

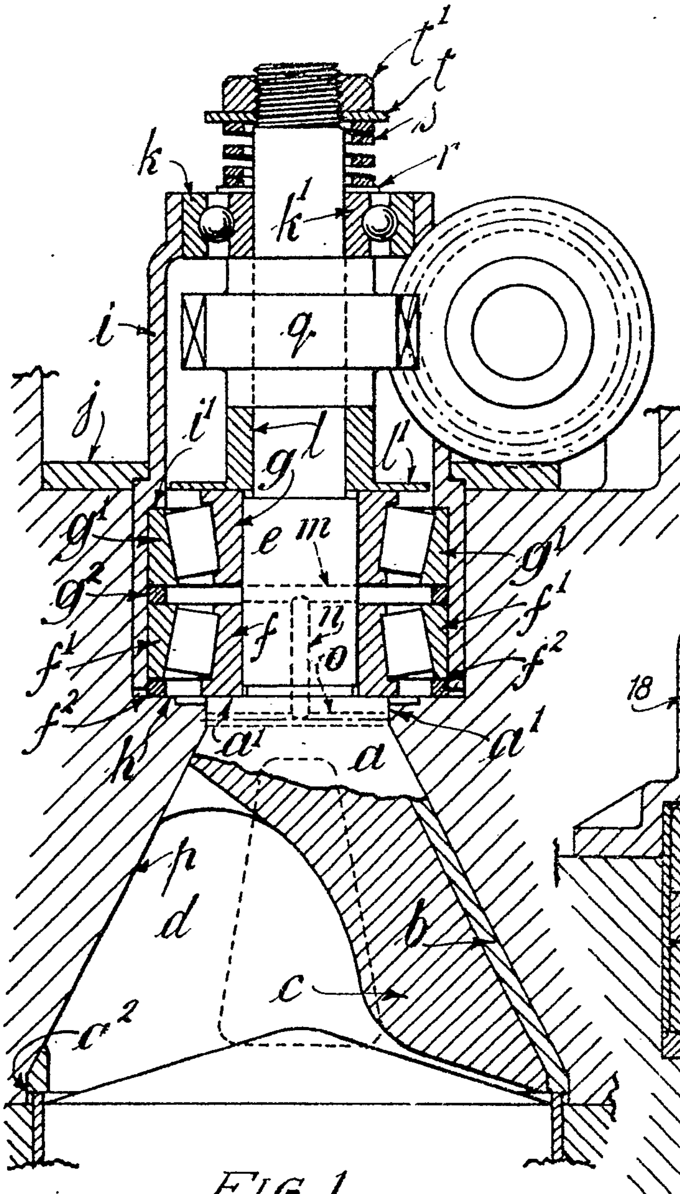


FIG.1.

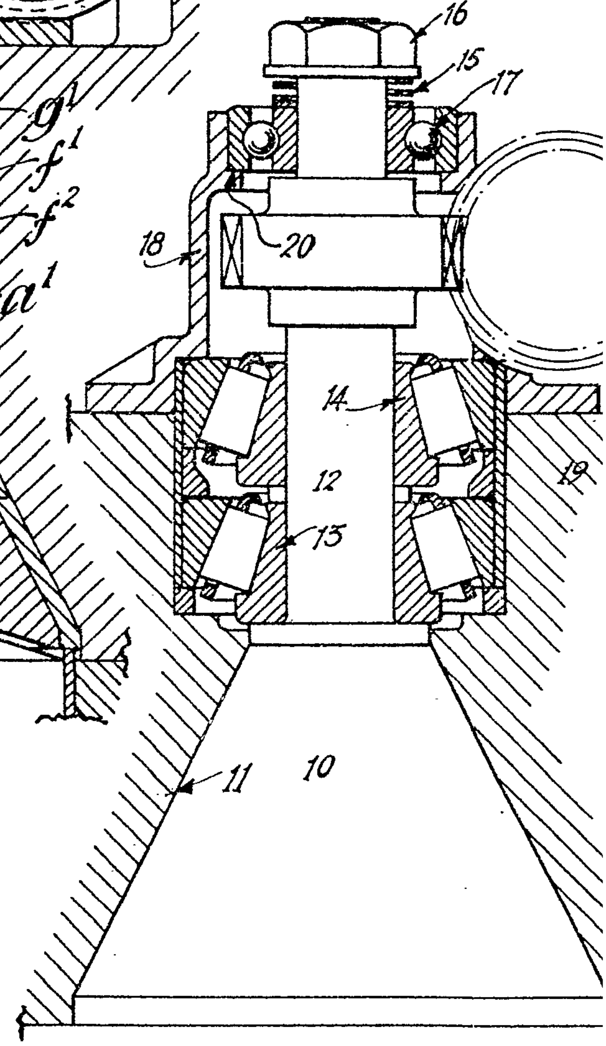
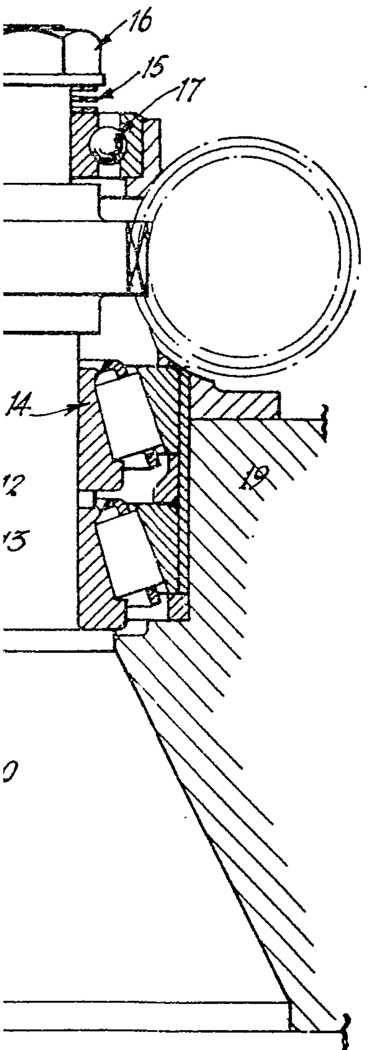


FIG.2.

2 SHEETS

SHEET 1



2.

SHEET 2

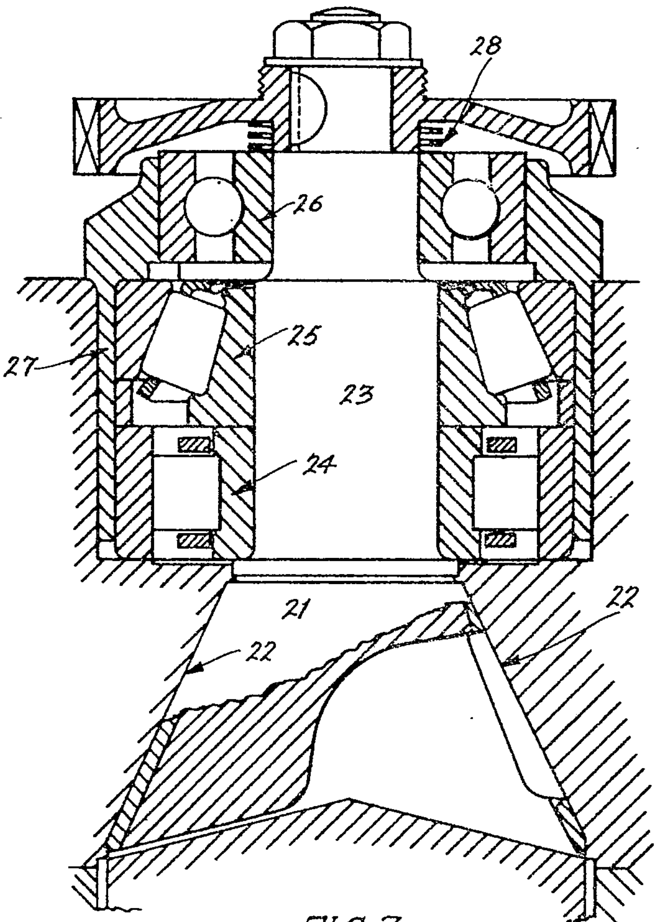


FIG. 3.

Malby & Sons, Photo-Litho.

SHEET 1

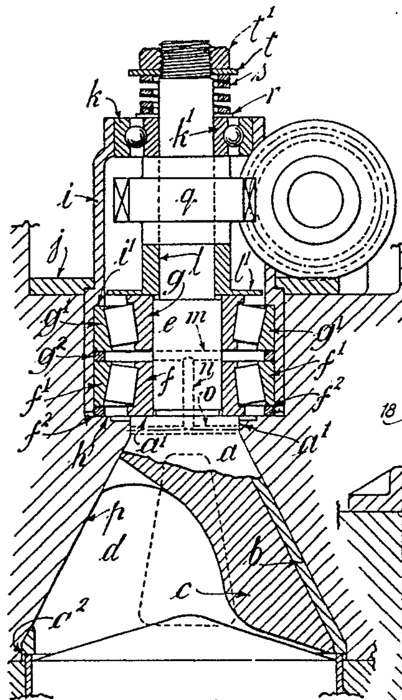


FIG. 1.

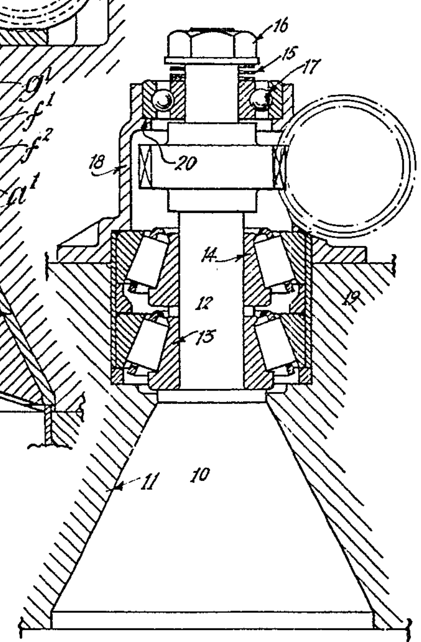


FIG. 2.

SHEET 2

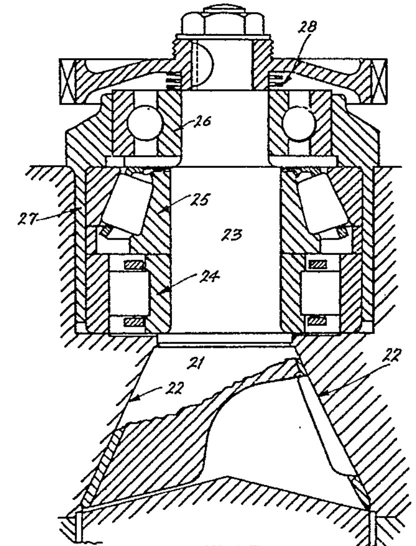


FIG. 3.

[This Drawing is a reproduction of the Original on a reduced scale.]