

July 7, 1970

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3,519,006

LOAD INDICATING MEANS

Filed June 21, 1968

5 Sheets-Sheet 1

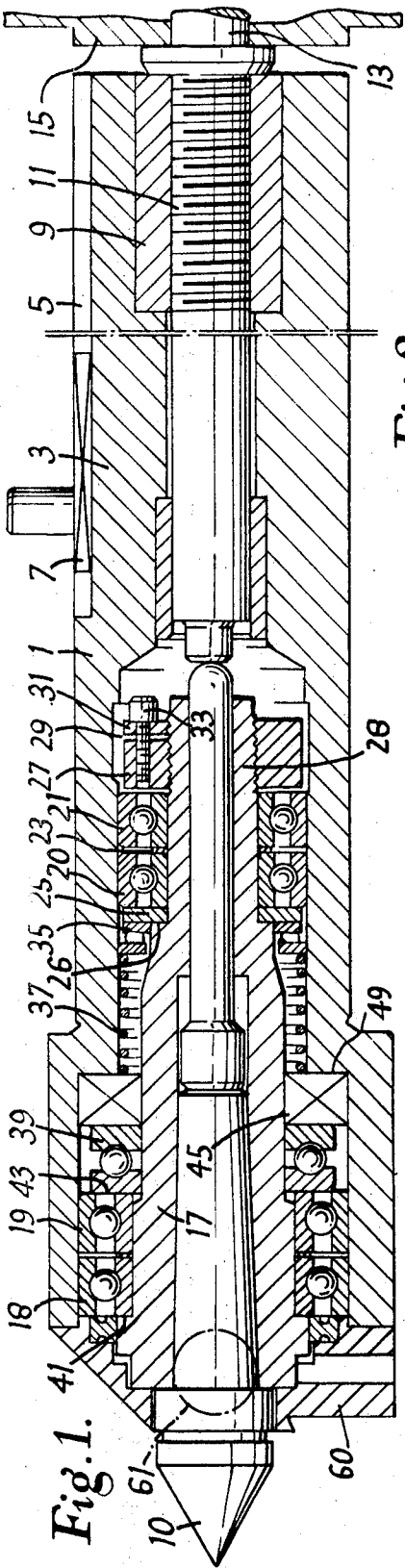


Fig. 3.

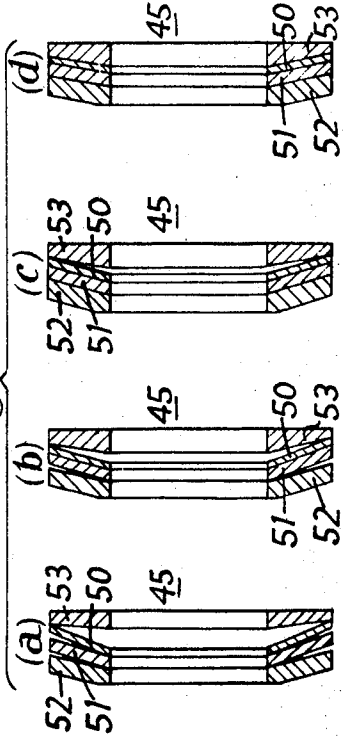
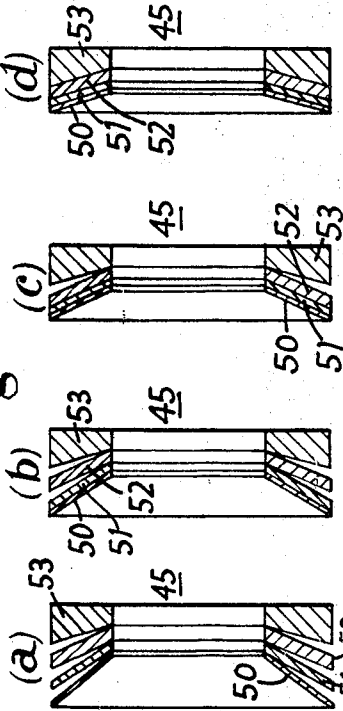


Fig. 2.



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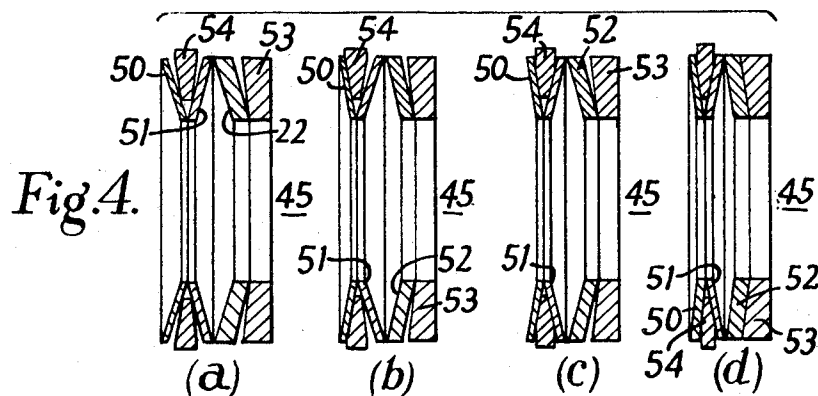
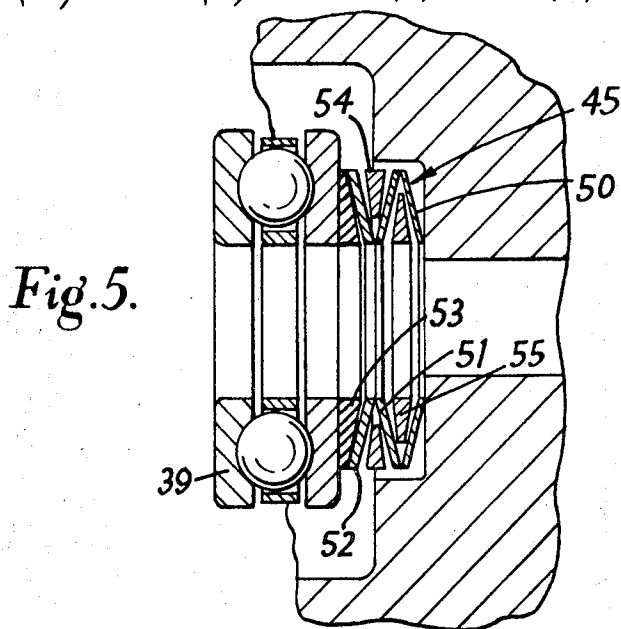
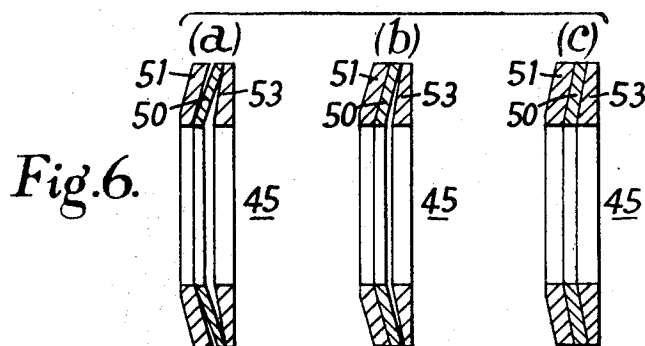
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5 Sheets-Sheet 2



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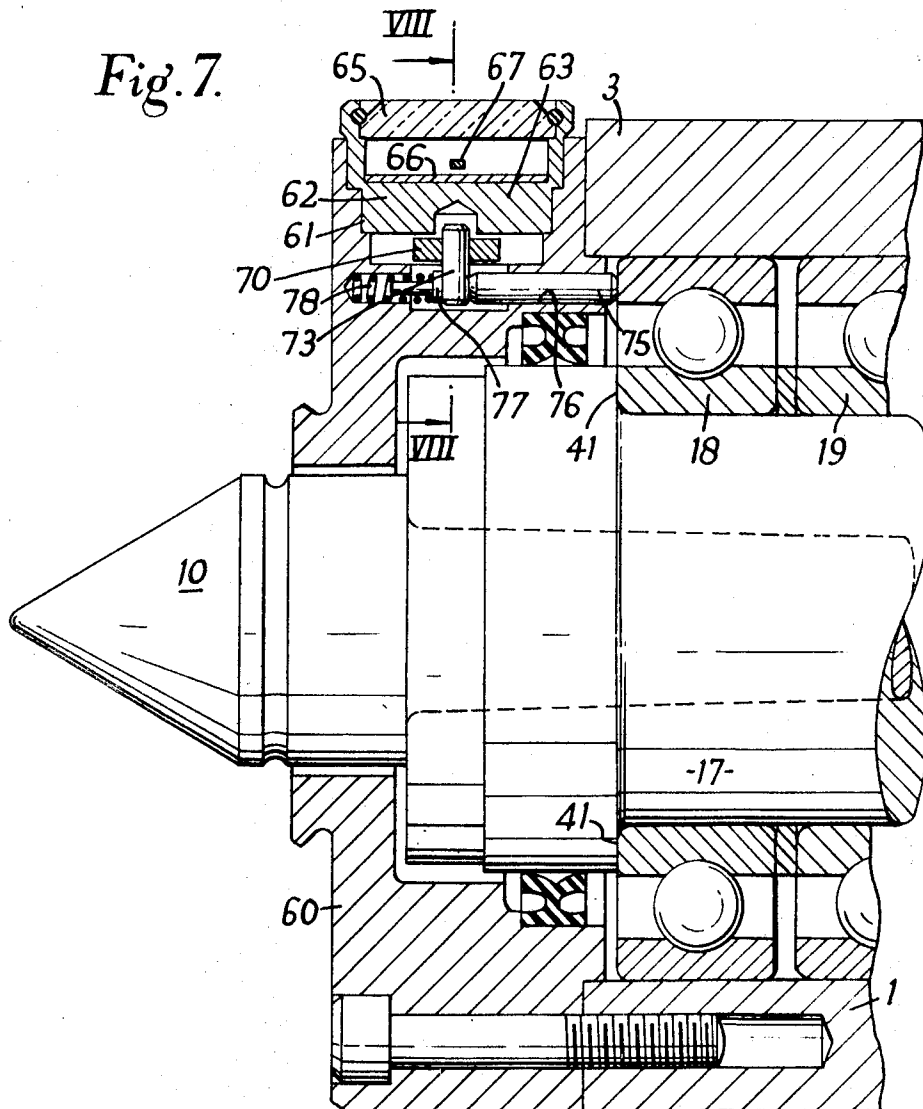
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LOAD INDICATING MEANS

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5 Sheets-Sheet 3



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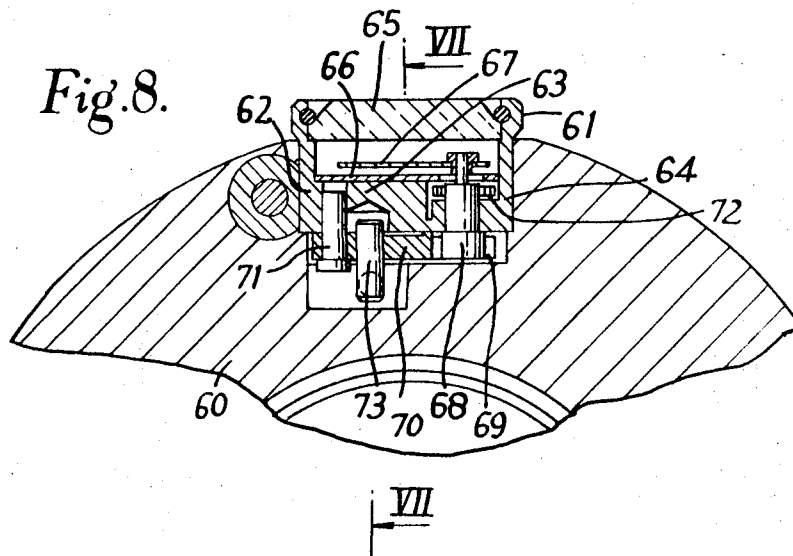
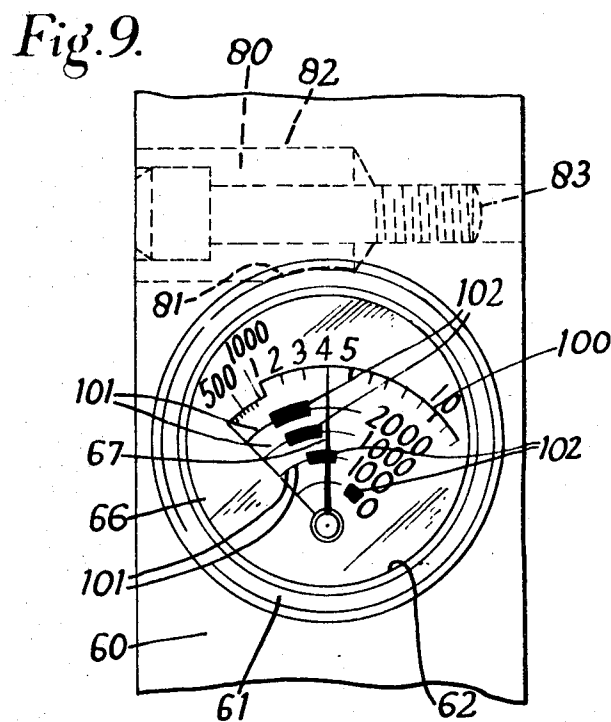
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LOAD INDICATING MEANS

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5 Sheets-Sheet 4



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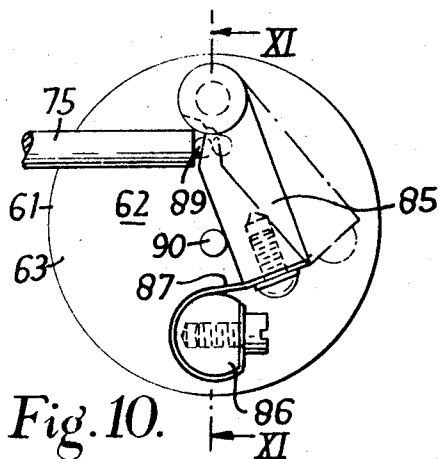


Fig. 10.

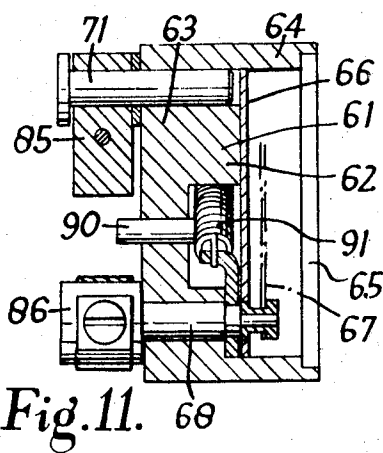


Fig. 11.

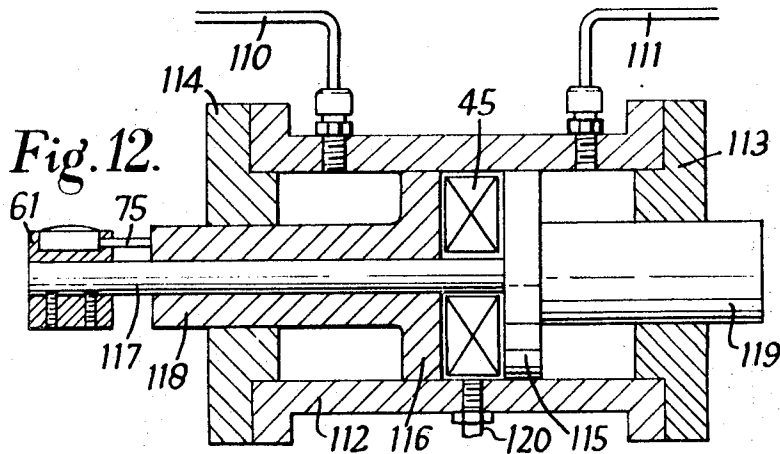


Fig. 12.

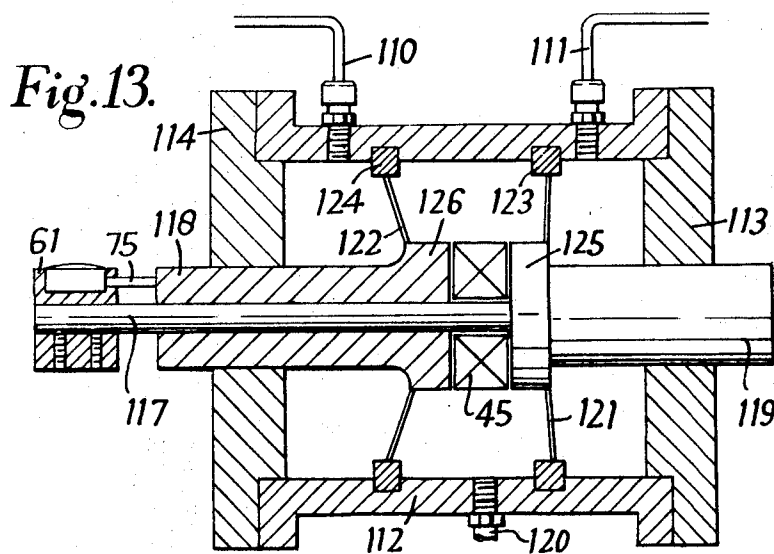


Fig. 13.

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LOAD INDICATING MEANS

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20 Claims

ABSTRACT OF THE DISCLOSURE

Load indicating means comprising a stack of springs having different flexural rigidities, the springs in the stack being arranged in sequence and in order of their flexural rigidities, load applying means for applying load to the spring stack thereby successively to deflect the springs of the stack, and, spring deflection responsive means which move in dependence upon the deflection under applied load of the spring stack.

This invention relates to load indicating means.

One important application of the invention is concerned with machine tools. It is well known in rotating centres for machine tools whether of the type in which the rotatable workpiece supporting part is mounted in the forward end of a parallel or taper shank housing or of the kind in which the rotating part is removably mounted directly into the tailstock barrel of the machine tool, to provide rearward of a thrust bearing of the rotating centre a flexible member which deflects under thrust load imposed in adjustment on the part of the machine operator and by heat expansion during operation of the machine tool, of the workpiece.

Accurate control by the machine tool operator of the thrust load imposed on the rotating centre is frequently desired. For example when the component being produced in a lathe is of small diameter the axial thrust must be sufficient to support the cut but not so great as to bend the component. Also, when a face driver is used, the axial force on the rotating centre in the tailstock of the machine must be high enough during loading of the workpiece so that the driving teeth on the face driver indent the end face of the workpiece. Once loaded it is necessary, before driving the workpiece, appreciably to reduce the thrust load on the rotating centre to prevent bending of the workpiece when the latter is reduced to a much smaller diameter. Furthermore, it is essential to maintain adequate but not excessive axial thrust when using the full power of the machine and the cutting capacity of the tool. In certain operations requiring very heavy radially directed tool pressures, such as plunge forming, maximum axial thrust must be applied to the workpiece compatible with the capacity of the thrust bearing calculated for an acceptable bearing life of the thrust bearing required at the particular speed of operation. An object of the invention is to provide an improved load indicating means suitable for use, inter alia, with a rotating centre of a machine tool. It is emphasized however that though the load indicating means of the invention is of particular interest in connection with rotating centres of machine tools, it is also useful in other contexts.

The present invention consists in load indicating means comprising a stack of springs having different flexural rigidities, the springs in the stack being arranged in sequence and in order of their flexural rigidities, load applying means for applying load to the spring stack thereby successively to deflect the springs of the stack, and, spring deflection responsive means which move in dependence upon the deflection under applied load of the spring stack.

Suitably the spring deflection responsive means actuate a meter having a scale and a pointer which traverses the scale, the scale being divided into sections which are

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traversed by the pointer as deflection of respective springs of the stack takes place so that a scale section traversed by the pointer on deflection of a spring of the stack of relatively lower flexural rigidity has greater resolution than a scale section traversed by the pointer on deflection of a spring of higher flexural rigidity.

Preferably, the spring stack comprises a plurality of springs each having the form of an annular dished washer.

In one form the invention consists in load indicating means disposed in a rotating centre of a machine tool wherein the load applying means comprise the rotatable workpiece supporting member of the rotating centre which is subject to thrust load from the workpiece and the spring stack is disposed rearwardly of a main thrust bearing of the rotating centre and the spring deflection response means comprise a rod spring biased so as to move on deflection of the spring stack and adapted to actuate a meter for measuring the load on the rotatable workpiece supporting member of the rotating centre.

It will be apparent that when a rotating centre in accordance with the invention is employed in operation of a machine tool at any given speed in conjunction with any known means for measuring the thrust load occasioned by heat expansion of the workpiece giving rise to movement of the workpiece supporting member or by initial loading the measuring means are more sensitive in the lower than in the higher axial thrust load range. The operator knowing the thrust load for the desired life expectancy of the thrust bearing at the particular speed of operation can then determine whether the actual thrust load as measured is too high or too low and take appropriate action to adjust the actual thrust load.

Suitably the meter has a scale and a pointer which is adapted to traverse the scale, the scale being divided into sections and the sections having respective resolutions which progressively diminish from one to the next scale section, said scale sections each providing a measure of the loading on the spring stack as the deflection increases.

Advantageously, adjacent the scale are a plurality of markings which can be aligned by the pointer with respective points on the scale, the scale reading for each marking indicating a critical load at a particular speed of rotation of the workpiece supporting part of the running centre, the critical load being maximum load which can safely be applied at the speed of operation under continuous working conditions.

Suitably, further markings are provided adjacent the scale which can be aligned by the pointer with respective points of the scale the scale reading for each further marking representing the maximum load at one of said particular speed of rotation under which the running centre can be operated for a period much less than the rated life of the main thrust bearing.

The invention will now be described, by way of example, with reference to the accompanying drawings, in which:

FIG. 1 is a longitudinal sectional view of a rotating centre of a machine tool;

FIGS. 2 to 6 is a longitudinal sectional view of a spring stack employed in the rotating centre of FIG. 1;

FIG. 7 is a sectional view to a larger scale of a forward end part of the rotating centre of FIG. 1, this view being at right angles to that of FIG. 1 and taken on the line VII—VII of FIG. 8;

FIG. 8 is a scrap sectional view on the line VIII—VIII of FIG. 7;

FIG. 9 is a fragmentary plan view of a part of FIG. 7 and illustrates a meter for indicating thrust load on the rotating centre;

FIG. 10 is a rear elevation of the interior of a meter similar to that of FIGS. 7 and 8 but having a different actuating mechanism;

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FIG. 11 is a sectional view taken on the line XI—XI of FIG. 10;

FIG. 12 is a longitudinal sectional view of another embodiment of the invention, and

FIG. 13 is a longitudinal sectional view of a further embodiment of the invention.

In the drawings like parts have been designated with the same reference numerals.

Referring to FIG. 1 the tailstock barrel 1 fits within a tailstock (not shown) bolted to the base of the machine tool. The main or outer barrel 3 is formed in known manner with an external keyway 5 in which is slidably engaged a key 7 mounted on the tailstock body so as to be movable from a withdrawn position to an operative position in which it projects within a bore in the body in which the outer barrel 1 slidably fits. The key engages the keyway of the barrel thus limiting the barrel to axial movement only relatively to the body. Within the barrel 1 at the forward end thereof is rotatably mounted a workpiece supporting member 10.

A rear part 9 of the outer barrel is internally threaded and is engaged by an externally threaded shaft 11 a rear unthreaded end 13 of which projects through a plate 15 secured to the rear of the body of the tailstock. On the end of the shaft 11 which projects through the plate is a hand wheel (not shown) rotation of which effects axial movement of the outer barrel.

A forward part of the barrel has rotatably mounted therein an inner barrel part 17 of annular section. Towards forward and rear ends of the inner barrel part and disposed in known manner between the inner barrel part and the inner surface of the outer barrel are respective pairs of ball bearings 18, 19 and 20, 21 which are pre-loaded. The bearings 20 and 21 at the rear of the inner barrel part have their inner races and their associated annular spacer 23 located between a first ring 25 which engages a shoulder 26 on the inner member forwardly of the inner races of the bearings and a second ring 27 which is threaded and engaged on a threaded rear end portion of the inner barrel part. The ring 27 is formed with a cut 29 extending normal to the axis of the ring to provide a rear part 31 of the ring which is clamped to a forward part thereof by a bolt 33 tightening of which locks the ring 27 to the barrel part 17. Against a forwardly facing surface of the first ring 25 is a further thrust bearing 35 forwardly of which is a light helical spring 37 which pre-loads a main thrust bearing 39.

The inner races of the two ball bearings 18 and 19 engage a shoulder 41 on the inner barrel part and a forward face 43 of the main thrust bearing 39 a rear race of which is engaged by the forward end of a stack 45 of annular dished spring members, the rear end of the stack engaging a shoulder 49 on the main or outer part of the barrel. The rear face of the hardened annular plate is also engaged by the helical spring but since the stack of dished annular spring members is much stiffer than the light helical spring the annular plate acts as a substantially fixed abutment for the latter.

The stack 45 of dished spring members suitably Belleville washers, may take a variety of forms. In one form illustrated in FIG. 2 there are three frusto-conical spring washers 50, 51, 52 the forward washer 50 having the least and the rear washer the greatest flexural rigidity. The conical angle of the washers increases from the forward to the rear washer so that the inner edges of the first and second and of the second and third washers are in contact. Rearwardly of the rear washer 52 is a hardened annular plate 53 the rear surface of which is flat and flatwise engages shoulder 49 whilst the forward surface is frusto-conical having a larger conical angle than the washer 52 so that the inner edge thereof contacts the inner edge of washer 52. As the axial thrust increases on the workpiece supporting member 10 which fits within the inner barrel part 17, which barrel part is retained endwise during ejection of the workpiece supporting

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member when replacement of the latter is required by a centrally apertured end cap bolted 60 to the forward end of the outer barrel, the first dished washer 50 flexes and flattens against the forwardly directed surface of the second washer 51 (FIG. 2(b)). Under increasing load the first and second washers 50 and 51 together deflect and flatten against the inner surface of the third washer 52 (FIG. 2(c)). The three washers then deflect together until the maximum thrust is reached at which stage the rear face of washer 52 (see FIG. 2(d)) engages the forward face of the plate 53 so that stressing of the spring washers beyond their elastic limit is prevented.

The stack 45 of spring washers 50, 51 and 52 and plate 53 may be arranged as shown in FIG. 3 in the reverse order from that described in which case the least stiff washer 50 is disposed at the rear and the stiffest washer 52 at the front of the stack. It will be appreciated that the plate 53 engages shoulder 49 and washer 52 engages the rear race of thrust bearing 39.

In another arrangement shown in FIG. 4 the least stiff washer 50 is disposed at the front of the stack 45 with its inner conical face forwardly directed, its outer edge contacting thrust bearing 39 and its inner edge contacting the inner edge of the next stiffer or intermediate washer 51 the outer conical surface of which is forwardly directed and the outer edge of which contacts the outer edge of the rear or stiffest washer 52 of the stack which has its inner conical face forwardly directed. The inner edge of washer 52 contacts the conical face of plate 53 which is located rearwardly of washer 52. Between the outer conical surfaces of the front and intermediate washers 50 and 51 is a rigid ring 54 of inwardly tapering wedge shaped cross section. As the axial thrust increases the front washer 50 first flattens against the ring 54 (see FIG. 4(b)). The front washer 50 and the ring 54 then move together against the intermediate washer 51 which flattens against the rear face of the ring (see FIG. 4(c)) and the ring then moves together with the front and intermediate washer to deflect the rear washer 52 till maximum thrust load is reached at which stage the rear face of washer 52 engages the forward face of plate 53 (see FIG. 4(d)).

In another arrangement shown in FIG. 5 there are three annular spring washers 50, 51 and 52 the forward washer being the stiffest and the rearward washer the least stiff. The hardened annular plate 53 is at the front of washer 52. The inner edge of the forward washer 52, which has its inner conical face directed forwards, contacts the inner edge of the intermediate washer 51 which has its outer conical face forwardly directed. The outer edge of the intermediate washer 51 contacts the outer edge of the rear washer 50 which has its inner conical face directed forwards. Between the front and intermediate washers is the rigid ring 54 of inwardly tapering wedge like section whilst a rigid ring 55 of outwardly tapering section is disposed between the intermediate and rear washers 51 and 50. The operation is similar to the previously described arrangement of FIG. 4 with the extra stage that the rear and intermediate washers 50 and 51 together with the ring 55 therebetween flatten under increasing load against the rear surface of the inwardly tapering ring 54 which then flattens against the rear surface of the forward washer 52 after which the entire stack deflects until maximum thrust load is reached at which stage the washer 52 is flattened against the rear surface of plate 53.

The deflection caused by movement of the workpiece supporting member 10 is a measure of the thrust load and is metered in the following manner.

In the arrangement shown in FIG. 6 there are provided only two spring washers 50 and 51 of which the washer 50 has lesser flexural rigidity than the washer 51 and is disposed rearwardly with respect to the washer 51. The washer 50 has a lesser conical angle than washer 51.

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Accordingly, the inner edge of washer 50 engages the inner edge of washer 51. Rearwardly of washer 50 and in engagement with the outer edge thereof is the plate 53. As the axial thrust on the workpiece supporting member 10 increases the washer 50 first flattens against the washer 51 (see FIG. 6(b)) and then the washers 50 and 51 together move rearwardly until the rear surface of washer 50 engages the plate 53 (see FIG. 6(c)). At this stage maximum thrust load is applied.

Referring now to FIGS. 7 to 9, the end cap 60 adjacent to the inner barrel 17 incorporates a meter 61 having a body 62 including a flat base 63 and a cylindrical side wall 64. The end of the side wall 64 remote from base 63 is closed by a transparent window 65. Against the base 63 is located a dial 66 swept by a pointer 67 which is secured to a spindle 68 passing through the base 63 and formed at its rear end with a pinion 69. A geared segment 70 meshes with the pinion 69 the segment being pivotally mounted on a shaft 71 supported in the base 63. It will be apparent that rocking movement of the segment 70 rotates pinion 69 and thus causes the pointer 67 to sweep the dial 66. A hair spring 72 anchored at its end to the base 63 and the spindle 68 biases the pointer towards zero on the dial. Carried in the segment between the pinion 69 and the shaft 71 is a pin 73 one end of which engages in a slot 74 in the base 63 which enables transverse movement of the pin between limiting positions whilst the other end of the pin abuts the forward end of a rod 75 extending axially in a bore 76 in the end cap 60. The rear end of the rod 75 engages the forward face of the outer race of ball bearing 18. Contact between the rod 76 and the pin 73 is maintained by a plunger 77 on the side of pin 73 opposite rod 76 the plunger being biased by a spring 78 seated in the end cap 60. It will be apparent that the rod 75 is moved to the right in FIG. 7 by spring plunger 77 when load increases.

The meter 61 is held by friction in the cap 60. To this end a pad-bolt 80 formed with an arcuate surface 81 fits in a bore 82 in the cap 60. The arcuate surface 81 of the pad-bolt frictionally clamps the meter 61 in the cap 60, a bolt 83 securing the pad-bolt in the cap 60. On release of the bolt 83 the meter 61 can be rotated in the cap and when so rotated the pin 73 rotates about the axis of the meter 61 and thereby enables zero adjustment of the pointer 67 to be effected. After zero correction of the pointer the meter is clamped to the cap 60 by tightening the bolt 83.

In the alternative form of meter 61 shown in FIGS. 10 and 11, instead of a geared segment and pinion there is provided on the shaft 71 and spindle 68 an arm 85 and a cylindrical boss 86. Secured at opposite ends thereof to the arm 85 and boss 86 is a band 87 which is wrapped round part of the circumference of the boss 86. The arm 85 is movable by rod 75, which engages a protuberance 89 on the arm, between the positions shown in solid and broken lines. A stop peg 90 prevents return of the arm 85 from the position shown in broken lines beyond that shown in solid lines. The spindle 68 is biased to urge the pointer 67 towards zero by a spring 91 mounted at one end on a spring anchor 92 carried on the spindle 68 and at the other end at a point (not shown) the body of the meter. When load is applied the arm 85 moves from the position shown in broken lines to that shown in solid lines, that is to say under the influence of the biasing spring causing the band 87 to be wrapped round the boss 86. Thus on relief of load the arm 85 is moved by rod 75 and the band 87 rotates spindle 68 against the action of the biasing spring 91 and the pointer traverses the dial 66. This form of the meter employs the same clamping and zero adjusting arrangements as shown in FIG. 9.

The dial 66 as shown in FIG. 9 which is also appropriate for the meter of FIGS. 10 and 11, consists of an arcuate scale 100 centred on the axis of spindle 68 on which the pointer 67 is mounted. Concentric with scale 100 are four

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lines 101 each of which represents a speed of rotation of the machine tool. The innermost line represents a speed of zero revolutions per minute, the remaining lines 101 representing speeds of 100, 1000 and 2000 revolutions per minute in the case of a machine with maximum speed 2000 r.p.m. On each line 101 is a marking 102 the left hand end of which represents when extended radially outwards a reading on scale 100 of a critical load i.e. an axial thrust which at the relevant speed of operation is the maximum which can be applied under continuous working conditions to the workpiece supporting part whilst ensuring that the main thrust bearing 39 will operate for its rated number of hours. The right hand end of the marking 102 when extended radially outwards gives a scale reading which represents the maximum thrust load that is permissible over periods of operation of short duration that is to say of duration very much less than the rated life of the main thrust bearing. It will be seen that the scale 100 is made up of sections and in the left hand section which represents loads up to one ton, change in load per unit of deflection of the spring stack 45 is given by a wider pointer movement than that obtaining in the next section of the scale representing loads of one to five tons and in this latter range the change in load per unit of deflection of spring stack 45 gives a greater pointer movement than that applying in the right hand section of the scale which represents loads of five to ten tons. In short, therefore, the scale in the lower load ranges is more extensive per unit load that is to say has greater resolution than is the case in the upper load ranges.

The spring washer stack 45 and associated meter 61 thus afford the machine operator with a sensitive means for ascertaining the actual loading at a given operating speed relative to the critical load at a given operating speed. In the event of the critical load being exceeded the operator can adjust the hand wheel at the rear of the plate 15 to bring the actual load below the critical value.

It will be apparent to those skilled in the art that other forms of bearing arrangement apart from those namely 18, 19, 20, 21 and 39 described can be employed. Also short taper roller bearings can replace the ball bearings 18 to 21.

When a face driver is used the hand wheel of the tail-stock is rotated to move the workpiece supporting part 10 of the rotating centre first into engagement with the workpiece and then to move the workpiece against the face driver until the force is sufficient for the teeth on the face driver to bite into the forward end of the workpiece to give a driving engagement therewith. The force indicated on the meter at this stage will probably greatly exceed the critical load for the intended speed of operation. Therefore, before driving is commenced, the thrust load is reduced to or below the critical value.

Those skilled in the art will appreciate that the invention is applicable also where the part 10 is supported in a parallel or taper shank housing.

FIG. 12 illustrates another embodiment of the invention. In this embodiment a load indicating means is adapted to indicate the difference in pressure between a first chamber and a second chamber respectively containing fluid at different pressures. The chambers are connected by conduits 110 and 111 which open into a cylinder 112 near opposite ends thereof. One end of the cylinder is closed by a plate 113 whilst the other end of the cylinder is closed by a plate 114. Disposed within the cylinder 112 are two pistons 115 and 116 having associated piston rods 117 and 118. The rod 117 passes axially through a bore in the centre of the piston 116 and its associated rod 118, the rod 118 together with the rod 117 passing through a central aperture in the cylinder end plate 114.

On the side of piston 115 remote from piston rod 117 is a further rod 119 which is of the same diameter as rod 118 and passes through the cylinder end plate 113. The

purpose of this rod 119 is to equalize the area of piston 115 subject to fluid pressure with the area of piston 116 which is subject to fluid pressure.

Between the pistons 115 and 116 is disposed the stack 45 of Belleville washers including the hardened plate 53, the arrangement of the washers and the plate being as described in connection with any one of FIGS. 2 to 6. On the sides of the pistons 115 and 116 remote from the stack 45 of spring washers there is applied pressure fluid from the respective high and low pressure sources.

Passing through the sidewall of the cylinder in the region of the cylinder occupied by the spring stack 45 is a connection 120 for exhausting a partial vacuum in that part of the cylinder. When the appropriate condition of partial vacuum has been achieved the connection 120 is sealed. In this way the pressure in the part of the cylinder occupied by the spring stack 45 never exceeds a predetermined value.

The meter 61 previously described is carried on the end of piston rod 117 remote from piston 115 whilst the meter actuating rod 75 is carried on the corresponding end of piston rod 118. It will be appreciated that as the differential pressure of fluid acting on the pistons 115 and 116 changes (provided the sense thereof does not reverse) the pointer of the meter 61 scans the scale. It will be apparent that changes in the differential pressure which causes flexure of the most flexible of the Belleville washers in the stack 45 result in larger movements of the pointer of the meter 61 than changes in differential pressure which result in flexure of the less flexible washer or washers of the spring stack 45.

FIG. 13 illustrates a load indicating means of a similar character to that of FIG. 12 and in which like parts have been accorded the same reference numerals as in FIG. 12. In the arrangement of FIG. 13 instead of pistons 115 and 116 there are provided diaphragms 121 and 122 connected to rings 123 and 124, each of the rings being secured to the cylinder sidewall. The inner edges of diaphragms 121 and 122 are secured to bosses 125 and 126 from which the rods 117 and 118 project. The arrangement of FIG. 13 is more sensitive to changes of differential pressure than the arrangement of FIG. 12 owing to the reduced inertia of the moving parts.

Meters with a dial and pointer, as meter 61, may be replaced by mechanical tips or micro-switches or electronic devices where it is desired to use these sensitive means of detecting pressure difference to control a process or to regulate the pressure difference or in any context where visible load indication is not required.

I claim:

1. Load indicating means comprising a stack of springs having different flexural rigidities, the springs in the stack being arranged in sequence and in order of their flexural rigidities, load applying means for applying load to the spring stack thereby successively to deflect the springs of the stack, and, spring deflection responsive means which move in dependence upon the deflection under applied load of the spring stack, wherein the spring deflection responsive means actuate a meter having a scale and a pointer which traverses the scale, the scale being divided into sections which are traversed by the pointer as deflection of respective springs of the stack takes place so that a scale section traversed by the pointer on deflection of a spring of the stack of relatively lower flexural rigidity has greater resolution than a scale section traversed by the pointer on deflection of a spring of higher flexural rigidity.

2. Load indicating means as claimed in claim 1, wherein the load applying means comprise two fluid pressure responsive members between which is disposed the spring stack the members being arranged to be subject to fluid pressure on the sides thereof remote from the spring stack, and wherein the spring deflection responsive means comprise a meter fixed relative to one of said fluid pressure

responsive members and a meter actuating element fixed relative to said other fluid pressure responsive member.

3. Load indicating means as claimed in claim 2, wherein the fluid pressure responsive members are pistons disposed in a common cylinder.

4. Load indicating means as claimed in claim 2, wherein the fluid pressure responsive members comprises diaphragms secured respectively to central members between which the spring stack is located.

5. Load indicating means as claimed in claim 1 and disposed in a rotating centre of a machine tool wherein the load applying means comprise the rotatable workpiece supporting member of the rotating centre which is subject to thrust load from the workpiece and the spring stack is disposed rearwardly of a main thrust bearing of the rotating centre and the spring deflection responsive means comprise a rod spring biased so as to move on deflection of the spring stack and adapted to actuate a meter for measuring the load on the rotatable workpiece supporting member of the rotating centre.

6. For use in load indicating means as claimed in claim 5, a meter having a scale and a pointer which is adapted to traverse the scale, the scale being divided into sections and the sections having respective resolutions which progressively diminish from one to the next scale section, said scale sections each providing a measure of the loading on a spring as the spring deflection increases.

7. A meter as claimed in claim 6, wherein the pointer is mounted on a rotatable spindle provided with a return spring and on which is carried a pinion which is in mesh with a gear rotation of which to effect consequential movement of the pointer by or against the return spring bias is effected by transverse movement of a pin secured to the gear.

8. A meter as claimed in claim 6, wherein the pointer is mounted on a spindle provided with a return spring connected to a pivotally mounted arm by a band wrapped around part at least of the spindle and secured at its ends to the spindle and the arm, rotational movement of the arm effecting rotation of the spindle by or against the return spring bias and consequent sweeping of the scale by the pointer.

9. A meter as claimed in claim 6, wherein adjacent the scale are a plurality of markings which can be aligned by the pointer with respective points on the scale, the scale reading for each marking indicating a critical load at a particular speed of rotation of the workpiece supporting part of the running centre driven by the machine tool, the critical load being maximum load which can safely be applied at the speed of operation under continuous working conditions.

10. A meter as claimed in claim 15, wherein further markings are provided adjacent the scale which can be aligned by the pointer with respective points of the scale, the scale reading for each further marking representing the maximum load at a one of said particular speeds of rotation under which the running centre can be operated for a period much less than the rated life of the main thrust bearing.

11. Load indicating means as claimed in claim 1, wherein the spring stack comprises a plurality of springs each having the form of an annular dished washer.

12. Load indicating means as claimed in claim 11, wherein the spring washer at one end of the stack is disposed adjacent an annular plate having a surface facing the washer shaped to prevent stressing of the washer under load beyond the elastic limit thereof.

13. Load indicating means as claimed in claim 11, wherein the spring washers are each of frusto conical form and arranged in nested array.

14. Load indicating means as claimed in claim 13, wherein the conical angle of the spring of the stack having the least flexural rigidity is greater than the conical angle of the spring of the stack having the greatest flexural rigidity.

15. Load indicating means as claimed in claim 13, wherein the conical angle of the spring of the stack having the least flexural rigidity is less than the conical angle of the spring of the stack having the greatest flexural rigidity.

16. Load indicating means as claimed in claim 11, wherein the springs of the stack are each of frusto-conical form and arranged so that alternate springs face in opposite directions.

17. Load indicating means as claimed in claim 16, wherein between one pair of successive springs is a ring disposed outwardly of the springs and of inwardly tapering cross-section against which under increasing load the said pair of springs successively flatten in order of the increasing magnitude of the flexural rigidity of the springs, whilst between another pair of successive springs is a ring of outwardly tapering cross-section disposed inwardly of the springs of the said other pair and against which the springs of the said other pair flatten in order of the increasing magnitude of the flexural rigidity of the springs of the said other pair.

18. Load indicating means as claimed in claim 16, wherein between a pair of successive springs is a ring of tapering radial cross-section against which under increasing load the springs successively flatten in order of their flexural rigidities, the spring of lower flexural rigidity flattening first against the ring.

19. Load indicating means as claimed in claim 18, wherein the ring is of radial cross-section tapering inwardly and is disposed between oppositely facing springs outwardly thereof.

20. Load indicating means as claimed in claim 18, wherein the ring is of radial cross-section tapering outwardly and is disposed between oppositely facing springs inwardly thereof.

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