Feb. 6, 1962 L. J. -M. GAMET 3,020,057

ROTARY FLUID SUPPLY SYSTEM

Filed Feb. 1, 1960 5 Sheets-Sheet 1

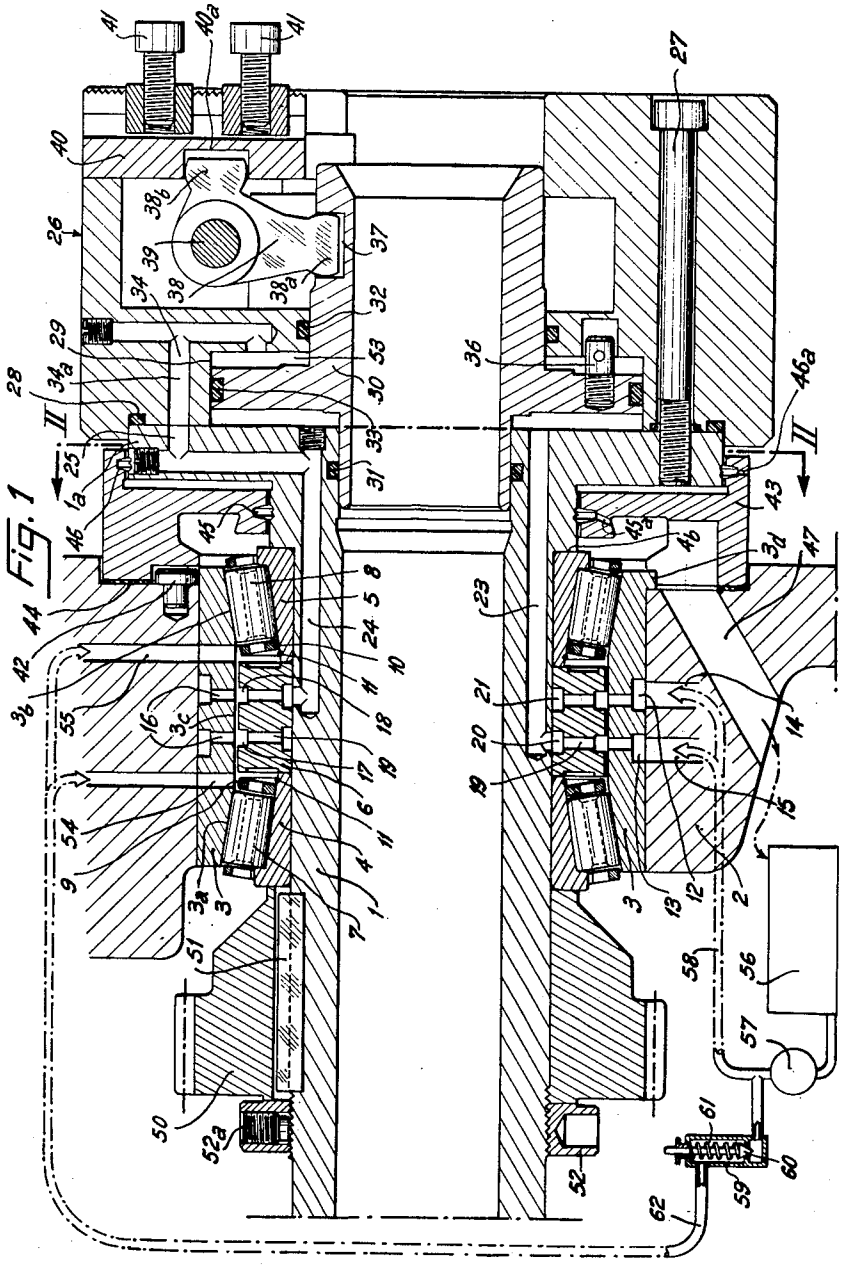


Fig. 1

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Fig. 2

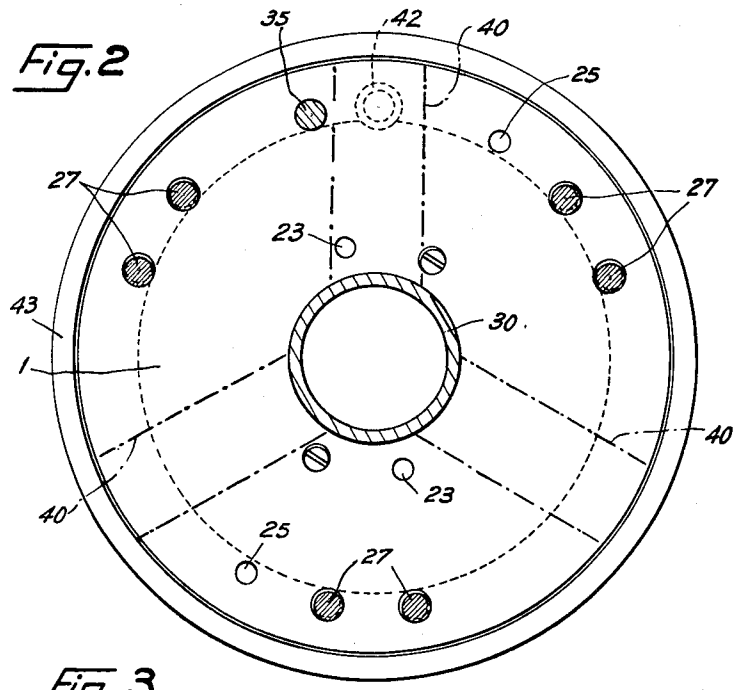


Fig. 3

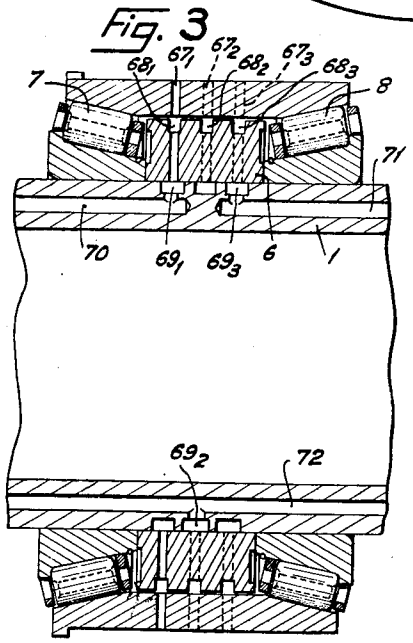
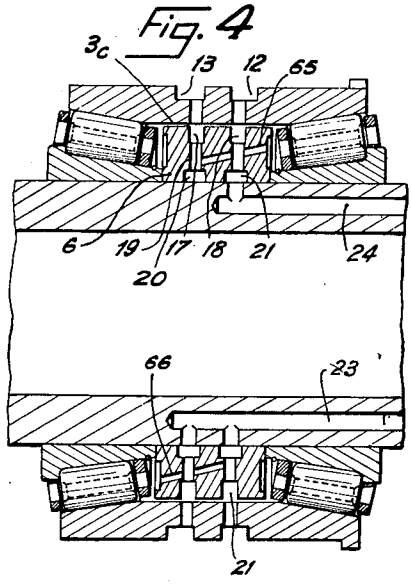


Fig. 4



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Fig. 5

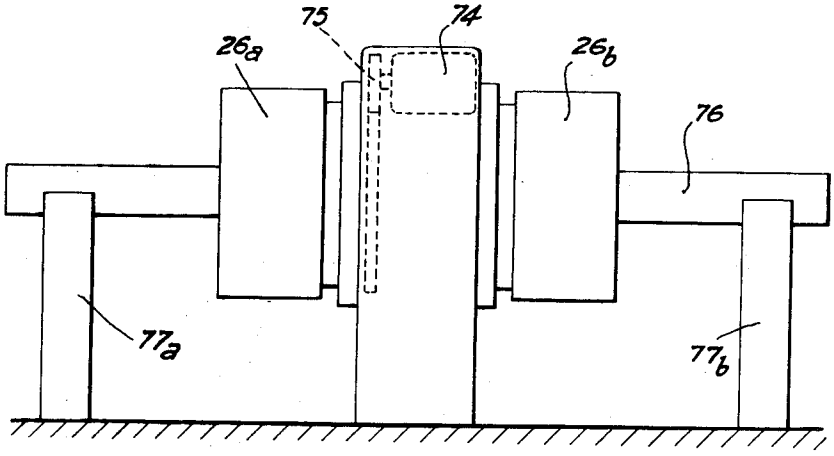


Fig. 6

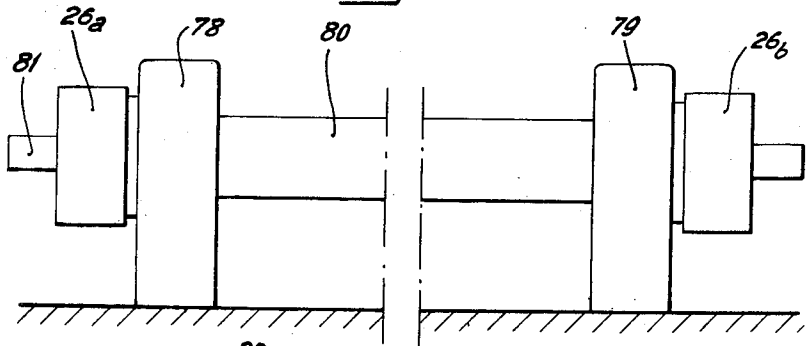
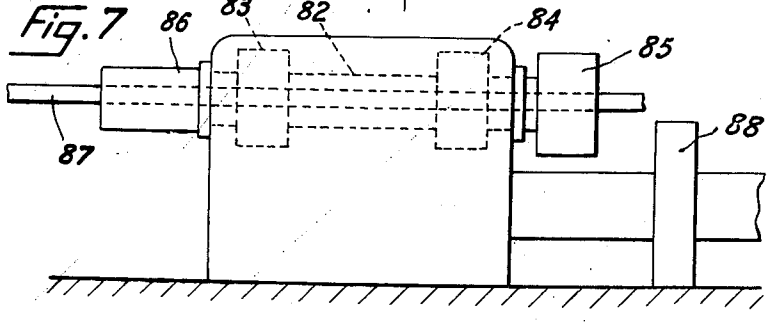


Fig. 7



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Fig. 10

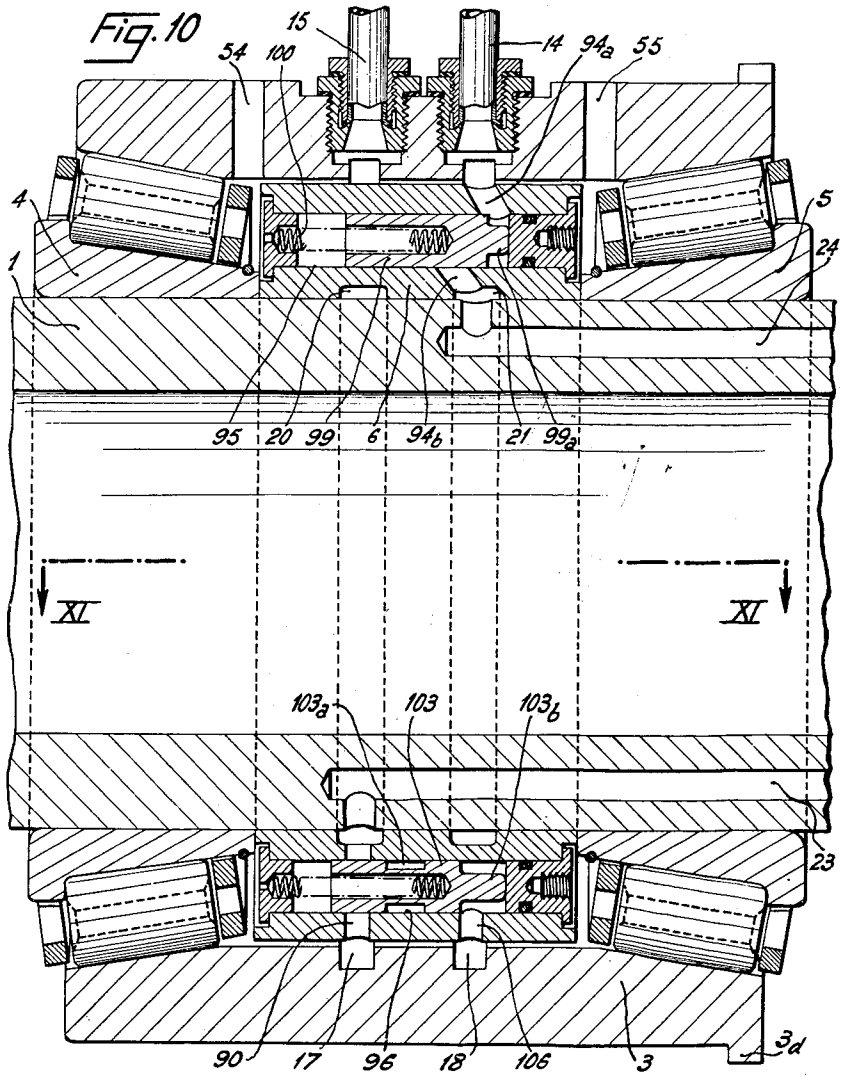
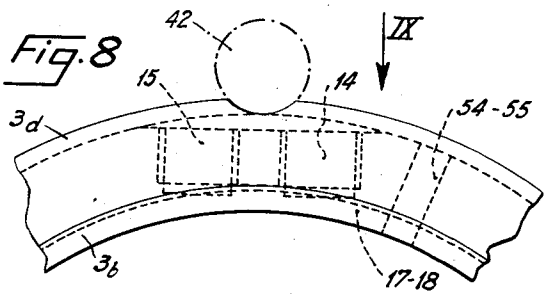


Fig. 8



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Fig. 9

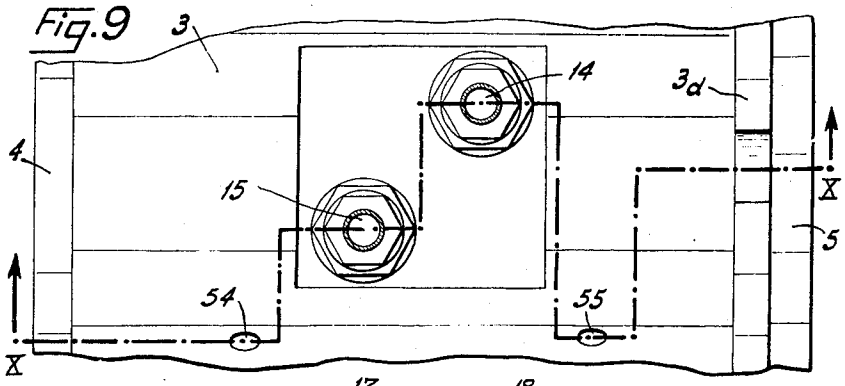
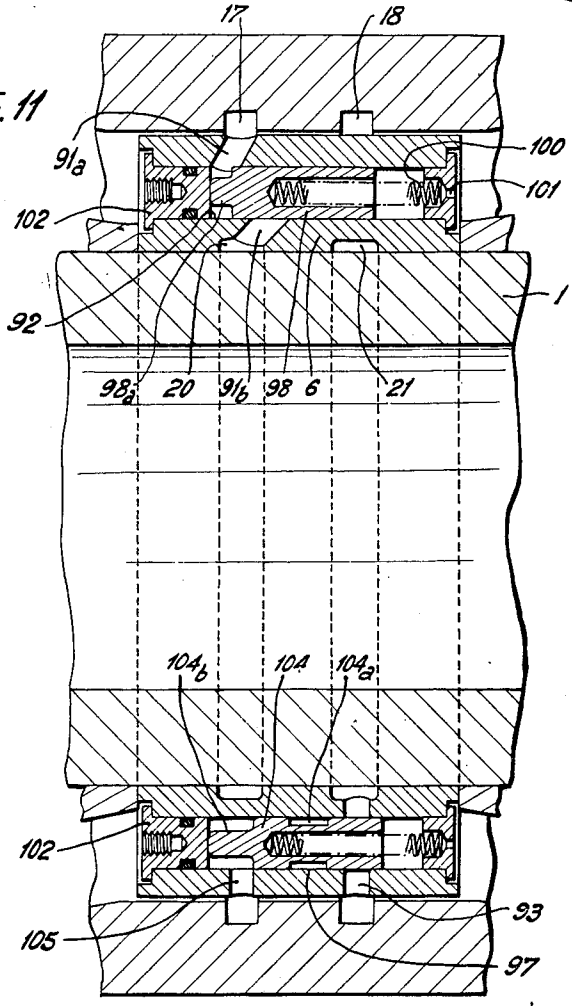


Fig. 11



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United States Patent Office 3,020,057 Patented Feb. 6, 1962

3,020,057

ROTARY FLUID SUPPLY SYSTEM

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Filed Feb. 1, 1960, Ser. No. 5,735
Claims priority, application France Feb. 12, 1959
11 Claims. (Cl. 279—4)

This invention relates to an improved means of distributing a hydraulic fluid to a revolving fluid motor. More especially though not exclusively, the invention relates to such a fluid distributing system for use in machine-tool headstocks of the type incorporating a hydraulic motor rotatable with the spindle of the headstock, and serving to actuate a work-carrying chuck, or other power-actuated mechanism, during rotation of the spindle.

In some types of machine-tools, e.g. lathes, there is provided a fluid motor, for example of the cylinder and piston type, incorporated in the headstock and rotatable with the spindle of the machine-tool. Such incorporated fluid motor may serve for various purposes including the actuation of a work-clamping chuck between its work engaging and disengaging conditions, the selective clamping and feeding of continuous bar stock through the machine, the machining of work in a jig carried by the spindle, or the like. The motor has to be supplied with fluid from a stationary source of pressure fluid outside the machine and a rotary distributor assembly has to be provided for permitting the flow of fluid from the stationary source to the revolving motor and back.

Thus, in my U.S. Patent No. 2,835,227, there was disclosed a hydraulic cylinder and piston motor mounted on one protruding end of the machine-tool spindle and including a two-part distributor having one part rotatable with said motor and spindle and another, stationary, part surrounding said rotatable part and defining therewith an annular clearance space of narrow radial width. This clearance is provided sufficient to prevent any mechanical friction between the rotary and fixed parts of the distributor while being small enough to introduce only an acceptably low pressure drop in the pressure of fluid supplied to the motor. The leakage flow accompanying such pressure drop in thereby maintained within controlled bounds so as to permit recovery of the fluid leaking out of the distributor, preferably after having caused the leakage flow to lubricate the distributor bearings. While the rotary distributor system disclosed in the said prior patent yielded satisfactory results in that it minimized wear on the distributor parts, certain disadvantages exist due to its design. It is the primary object of the present invention to eliminate those disadvantages.

It will be understood that in distributor systems of the type here contemplated it is essential to maintain extremely precise centering between the two parts of the distributor. This involves providing roller bearings to either side of the revolving distributor which bearings are then desirably lubricated by the leakage flow of fluid from the distributor as mentioned above. In the construction of the prior patent the provision of such special bearings in addition to the spindle bearings consumed considerable axial space for the distributor device, and the fluid motor had to be mounted beyond the headstock at the end of the spindle remote from the chuck, or other mechanism to be operated by the motor. This undesirably increased the length of the spindle. Further with the motor thus mounted it was necessary to provide a lengthy drive transmission from the motor to the chuck (or other mechanism) in the form of an axially reciprocating tube extending through the spindle. The

presence of such tube reduces the useful inner diameter of the spindle and the axial reciprocations of the tubes may interfere with certain operations, bar-feeding for instance.

Objects of the invention are to provide a fluid distributor system of the specified type which will be of considerably more compact construction than was heretofore possible; to provide such a system wherein no additional bearings will be required for the distributor, beyond those supporting the spindle in the headstock, or other frame in which the system of the invention may be used; to provide such a distributor system for a machine-tool headstock wherein the distributor and motor may be mounted in directly adjacent relation to the chuck or other mechanism to be actuated thereby, thus doing away with undesirable mechanical drive transmissions such as tubes extending through the spindle.

Another important object is to facilitate the construction of the distributor assembly for a given, extremely narrow width of the annular clearance space between its stationary and rotatable parts. An object is to apply such a distributor system to various machine-tool headstock configurations.

Further objects of the invention are to provide means in such a distributor for blocking the fluid within the motor under a prescribed operating pressure, thereby preventing unwanted de-activation of the mechanism, e.g. release of the work by the chuck, in case of defective fluid supply from the pressure source. Yet other objects will appear.

Exemplary embodiments of the invention will now be described for purposes of illustration but not of limitation with reference to the accompanying drawings, wherein:

FIG. 1 is an axial section of a machine-tool headstock incorporating a pressure fluid distributor, fluid motor and chuck mechanism in accordance with one embodiment of the invention;

FIG. 2 is a cross-section on line II—II of FIG. 1;

FIG. 3 is a partial axial section relating to a modified embodiment applicable to a headstock of the central, or double-ended type;

FIG. 4 is a similar view showing a further modification of the distributor arrangement of FIG. 1, providing for full lubrication of both roller bearings without separate oil delivery;

FIG. 5 is a simplified outline view, to a reduced scale, of a lathe embodying a central or double-ended headstock arrangement;

FIG. 6 is a similar view of a machine-tool for machining both ends of elongated stock, e.g. lengths of tubing;

FIG. 7 is a similar view of a lathe embodying bar feed mechanism; the last three figures being all illustrative of types of machine-tools to which the invention is supplied;

FIG. 8 is a partial end view of the outer ring member of a distributor system according to another embodiment of the invention;

FIG. 9 is a plan view of the embodiment of FIG. 8 as seen in the direction shown by the arrow IX in the latter figure;

FIG. 10 is a sectional view on the broken line X—X of FIG. 9 illustrating the last-mentioned embodiment; and

FIG. 11 is a partial view on line XI—XI of FIG. 10.

Before proceeding with the particular description with reference to the drawings, the main aspects of the invention will be explained in general terms.

In order to provide a more compact rotary distributor structure wherein a common set of bearings is used both for supporting the spindle and the distributor, the invention contemplates providing a pair of spaced roller bearings having a common stationary outer race ring member formed with a cylindrical internal surface intermediate

its length with at least one pressure fluid delivery orifice opening radially into said surface, said bearings having separate, axially spaced inner race ring members. Between such inner race rings is mounted a spacer ring member having an outer cylindrical surface defining an extremely narrow radial clearance space with an inner cylindrical surface of the stationary common outer ring. The said rotatable inner race ring members and the intermediate spacer ring member are mounted in a close fit on a cylindrical section of the spindle, and means are provided for exerting an axial clamping pressure on the three juxtaposed members for bodily rotation thereof with the spindle. A pair of interconnected circumferential grooves are defined respectively by the outer and inner cylindrical surfaces of the intermediate spacer ring member, respectively with the inner cylindrical surface of the stationary outer ring member, and with the cylindrical surface of the spindle, which grooves are both formed in a common axial plane corresponding with that in which said fluid delivery inlet orifice delivers, and both grooves being interconnected by at least one generally radial duct formed through the rotatable inner ring member. The inner one of said grooves communicates with at least one generally axial passage formed through the spindle and leading to the fluid motor carried by the spindle.

It will be clear that since both bearings embody a common outer stationary ball race, an accurate coaxial and concentric relationship can easily be provided between the outer circumferential roller race surfaces and the cylindrical intermediate surface constituting the stationary part of the distributor, by simply boring the inner surface of said common stationary outer ring. Similarly, an accurate coaxial and concentric relationship is readily had between the inner circumferential roller race surfaces and the cylindrical surface of the intermediate ring member constituting the rotatable part of the distributor, by simply grinding the surface of the cylindrical spindle section. This accurately centered relationship will in turn ensure that the intermediate spacer ring can revolve in very closely spaced relationship with the outer ring surface, a radial clearance distance on the order of one or two hundredths of one millimeter being readily achievable. The pressure fluid can thus be led from the inlet through the outer groove and through the inner groove and thence into the spindle passage leading to the motor at the cost of low controlled laminar leakage through the gap between the outer and inner ring members, the leakage flow as being collected and used for lubricating the roller bearings supporting the spindle.

Furthermore, owing to the steep axial pressure gradient, i.e. the high rate of pressure drop per unit axial length along the flow path through the gap between the coaxial ring surfaces, it becomes possible to provide parallel flowpaths side by side in axially spaced transverse planes, between the spaced roller bearings. Thus there can be provided two such parallel flowpaths leading to the opposite ends of a double-acting cylinder-and-piston motor or jack, without having to impart an excessive axial length to the intermediate spacer ring. Conversely, in view of the short axial length of the spacer ring, it becomes possible to position the bearings close to each other, thereby to minimize the objectionable effects of longitudinal expansion of the spindle on the centering thereof in its roller bearings. In practice, the spacing selected between the bearings would be just sufficient to ensure a firm support of the spindle without the provision of auxiliary bearings. The spindle may be provided comparatively very short, whereby machining of its inner bore from either end of the spindle as far as its central distributor-forming portion can very simply and conveniently be performed.

In carrying the invention into effect according to one embodiment, as shown in FIGS. 1 and 2 of the accompanying diagrammatic drawings, a spindle **1** of a headstock **2** of a machine-tool, such as a lathe, is supported

in two axially spaced taper roller bearings, the bearing assemblage advantageously being constructed in accordance with the prior Patent No. 2,499,640 and comprising a common external ring **3** having two internal rolling surfaces **3***a*, **3***b* of truncated conical form, and a central cylindrical surface **3***c*. Internal rings **4**, **5** of the two roller bearings and an intermediate or central spacer ring **6**, are fixedly positioned on the spindle **1**. The rollers **7**, **8** of the bearings are respectively enclosed in cages **9**, **10** which may be retained on the internal rings **4**, **5** by steel wire rings **11** during mounting of the spindle **1** in the headstock **2**. In assembling the headstock, the three internal rings **4**, **5** and **6**, with the cages **9**, **10** of the rollers **7**, **8** carried by the rings **4**, **5** are inserted in the external ring **3** and the whole assembly is placed on the spindle **1**. The assemblage is then inserted into the headstock **2**, the angular position of the external ring **3** being defined by a spigot **42** the head of which engages in a rounded recess in an external flange **3***d* of the ring **3**.

As shown in FIG. 1, the external ring **3** is provided with two circumferential grooves **12** and **13** for receiving oil under pressure from channels **14**, **15** respectively. Each of the grooves **12**, **13** communicates with the internal cylindrical surface **3***c* through a series of radial holes **16**. These holes **16** are opposite grooves **17** and **18** respectively, which are provided in the central ring **6**. Through the medium of the radial holes **19**, the external grooves **17** and **18** of the central ring **6** are connected to internal grooves **20** and **21** of the latter.

From opposite the groove **20** two channels **23** (see FIG. 2) in the spindle **1** extend to the front face of a back plate **1***a* on the nose of the spindle **1**. In the same manner, from opposite the groove **21** two opposite channels **24** extend in the spindle **1** and are offset so as to terminate at orifices **25** in the front face of the back plate **1***a*.

On the back plate **1***a* a chuck **26** is fixed by means of three pairs of screws **27** (see FIG. 2) which traverse the chuck from front to back. Peripheral packing **28** ensures oil-tightness of the joint.

The face of the chuck turned towards the spindle is provided with a cavity **29** which, with the front face of the back plate **1***a*, forms an annular ram cylinder in which an annular piston **30** of a hydraulic jack is slidable. Packings **31** and **32** hermetically seal an extending central part of this piston in the interior of the spindle **1** and an oppositely extending part in the interior of the chuck **26**. A packing **33** forms a seal at the periphery of the piston **30** in the cavity **29**.

In the body of the chuck **26** and diametrically opposed to one another, are arranged the two series of holes **34**, each forming a U-shaped channel, of which one of the terminal orifices **34***a* registers with one of the orifices **25** in the front face of the back plate **1***a*. A centering pin **35** (FIG. 2) projecting from the back plate **1***a* registers with a blind cavity in the rear face of the chuck to prevent incorrect positioning of the chuck on the spindle, which would not ensure registering of the orifices **25** and **34***a*.

The piston **30** is prevented from turning in its housing by a pin **36** which engages in a recess in the rear face of the cavity **29**.

The front part of the piston **30** comprises three cavities **37** spaced at 120° from one another and in which there are housed cylindrically faced ends **38***a* of bell-crank levers **38** movable around pivots **39** and carried by the chuck **26**. The other ends **38***b* of these levers **38** also are cylindrically faced and are engaged in the housings **40***a* of jaw-holders **40** which, by means of screws **41**, support the chuck jaws.

At the inner side of the chuck **26** leakage of oil towards the exterior is prevented by an annular connecting part **43** between the periphery of the back plate **1***a* and the headstock **2**. This part **43** is fixed to the headstock **2**, with the interposition of joint packing **44**, by screws

(not shown) suitably spaced. It is provided with an oil-retaining groove 45 around the spindle 1 opposite a peripheral slot therein. An orifice 45*a*, arranged at the lowest point of this groove 45, permits return of oil received by the groove 45 to a collecting reservoir through a cavity in the annular part 43 and a channel 47 connecting therewith. A similar groove 46 surrounds the back plate 1*a* to prevent infiltration of the cutting or coolant liquid towards the interior of the headstock, any infiltration being passed to the exterior through a channel 46*a*.

In the embodiment shown, the spindle 1 carries, on the left-hand in FIG. 1, a gearwheel 50 which is prevented from turning by a key 51. A threaded ring 52 with grub screw 52*a* presses against a shoulder 1*b* of the spindle 1, the assembly of the rings 4, 5 and 6 and the gearwheel 50, so as to prevent any longitudinal displacement of the assembled units and rotation of the rings 4, 5 and 6 on the spindle 1.

The gearwheel 50 (and the gearwheels which drive it) may be enclosed in an oil-tight gearcase.

The function of the assembly above described is as follows:

Owing to the precision of centering of the spindle 1, by the taper roller bearings, the internal diameter of the cylindrical surface 3*c*, and the external diameter of the ring 6 can be selected so that the annular space which separates them is very small, for example, of the order of one to two hundreths of a millimeter.

Thus, when oil is fed under pressure through one of the channels 14 or 15, and passes into the appropriate groove 12 or 13 and thence through the channels 16, the groove 17 or 18 is likewise placed under pressure and a small quantity of oil escapes from each side of this groove into the annular clearance space which separates the ring 6 from the surface 3*c*. This oil expands in this space and arrives at practically atmospheric pressure, either at the inner end of the right-hand roller bearing, or in the adjacent groove 17 or 18 where it is collected.

From the groove 18 oil under pressure passes through the radial holes 19 into the internal groove 21 and, from the latter, into the two diametrically opposite channels 24. From there the oil under pressure enters one of the compartments of the cavity 29 of the jack, in the example the right-hand compartment 53, which displaces the piston 30 towards the left and, in consequences, drives the jaw-carriers inwards for tightening the chuck jaws on a workpiece.

If oil under pressure is fed through the channel 15, the piston 30 is displaced towards the right.

It will be noticed that elimination of any mechanical connection between the operating jack and the tightening members renders it possible to leave the internal channel of the spindle 1 completely free for the passage of stock or parts to be machined.

By the peripheral grooves 12 and 13 the oil is distributed to all the radial channels 16 and an effect of concentric pressure on the periphery of the ring 3 is secured which prevents any distortion of the latter capable of causing seizing on contact with the ring 6 owing to the very small distance which separates the internal face of the ring 3 from the said ring 6. In the same way, the similar internal and external grooves of the ring 6 play a similar role in the uniform distribution of the flow of liquid and maintaining equilibrium of forces and, moreover, they render it possible to obtain without special precautions at the time of mounting, registering of the orifices in the wall of the spindle (ends of the channels 23 and 24) with the oil feed passages.

In the embodiment described, the right-hand bearing (rollers 8) is lubricated by leakage of oil only when the groove 12 is under pressure, and the left-hand bearing (rollers 7) only when the groove 13 is under pressure because, in both cases, the leakage of oil which escapes from one groove 17 or 18 towards the other is collected by the latter and evacuated. There is therefore a risk of inadequate lubrication of the roller bearings which support the entire load of the machine.

To overcome this drawback, the external ring 3 may be provided with oil feed orifices 54 and 55 which emerge adjacent the small diameter end of each of the conical bearings. Oil fed through these channels traverses the said conical bearings, and especially the internal holes of the rollers 7 and 8, to be expelled, after its lubricating and cooling effect. to the outside of the roller bearings.

The orifices 54 and 55 can be fed with oil from any source but preferably are fed with the same oil which is utilised in the hydraulic circuits, since the two oils are mixed in the collecting sump.

From the sump 56, which contains the reserve of oil, a pump 57 feeds oil under pressure through the channels 14 and 15 along the path shown by the dotted lines 58, and including a distributor which is not shown. However, for proper working of the jack, the pressure of oil supplied by the pump must be regulated to a predetermined value. For this purpose there is used in known manner a pressure-regulating or relief valve 59 comprising a flap 60 closed by a spring 61, the thrust of which is adjustable. When the pressure of oil delivered by the pump 57 exceeds the thrust of the spring 61, the oil passes into the upper compartment of the valve 59. It is then led through a pipe 62 to the orifices 54 and 55.

As the rate of discharge of the valve 59 is practically constant (except at the moment when the piston 30 is displaced), the roller bearings are permanently lubricated.

Moreover, as long as a part to be machined is gripped by the jaws of the chuck, this grip is maintained by the pressure of the liquid in the jack, so that, in contradistinction to chucks incorporating irreversible tightening, there is no risk of inadvertent release of the part through loosening of the jaws.

It is, however, possible to ensure correct lubrication of the bearings which carry the spindle 1 without making use of a supplementary oil feed. As shown, for example, in FIG. 4, a bleed passage 65 communicates with the groove 17 and extends to the inner end of the right-hand roller bearing. Similarly, a bleed passage 66 communicates with the groove 18 and extends to the inner end of the left-hand roller bearing. The grooves 17, 18 are placed under pressure alternately and when, for example, the left-hand groove 17 is under pressure, leakage which takes place towards the left-hand bearing, in the space between the rings 3 and 6, is sufficient to lubricate this bearing, and a certain quantity of oil passing through the bleed passage 65 lubricates the right-hand bearing. Conversely, when the right-hand groove 18 is under pressure, the right-hand bearing is lubricated by leakage and the left-hand bearing by oil which flows through the bleed passage 66.

The leakage paths and the bleed passages 65, 66 can be calibrated in such a way that, for normal working of the machine, lubrication of the two bearings is substantially equal and adequate.

The arrangement of bearing assembly to provide a rotary distributor for oil under pressure is particularly appropriate in a lathe having a central or double-ended headstock, that is, a headstock provided with a chuck at each end of the spindle for engaging parts which traverse the spindle.

The two jacks of the chucks can be connected by suitable holes in the spindle extending in opposite directions, as is shown in FIG. 3. In the embodiment, the ring 3 has three series of holes 67_1, 67_2 and 67_3, which feed grooves 68_1, 68_2 and 68_3 of the ring 6. The grooves 68_1 and 68_3 correspond to grooves 69_1 and 69_3 provided in the spindle 1, which communicate with the holes 70 and 71 for feeding oil under pressure for operation of jacks at the ends of the spindle 1 in the direction of the tightening of the corresponding chuck. The central groove 69_2 of the spindle, which communicates with the groove 68_2

of the ring 6, also communicates with a channel 72 common to the two jacks which permits loosening.

In this case, while the jacks are being put under pressure for tightening, that is, during the working phase of the machine, leakages towards the left from the groove 68_1 and towards the right from the groove 68_3, are sufficient for lubrication of the roller bearings 7 and 8. During the phase of loosening, when the load on the machine is less, the lubrication is momentarily interrupted because the grooves 68_1 and 68_3 are released from pressure and only the groove 68_2 is under pressure.

FIGS. 5 to 7 illustrate other possible modifications.

In FIG. 5, a central headstock 74 incorporates a driving motor which, through a transmission 75 by belt, chain or gearwheel, drives the internal spindle 1 which is provided, at its two ends, with chucks 26*a* and 26*b*. A part 76, which traverses the headstock, can thus be machined at both ends by tools mounted in tool-holders 77*a* and 77*b*.

In the modification shown in FIG. 6, each of two headstocks 78 and 79 carries a spindle provided at its ends with a chuck 26*a* and 26*b* respectively. The two spindles are connected by a tube 80 which ensures their being driven simultaneously. In this tube 80, a length 81 of bar or tube may be accommodated while being machined at both ends which are held in the chucks 26*a* and 26*b*.

As shown in FIG. 7, a tubular spindle 82 is carried by two bearing assemblies 83 and 84, such as described with regard to FIGS. 1, 3 and 4. The rotary distributor of the bearing assembly 84 feeds oil under pressure for operating the chuck 85, while the rotary distributor of the bearing assembly 83 feeds oil under pressure for operating a bar-feed device 86 of known type comprising automatic tightening clamps for a bar or tube 87 and hydraulic jacks for the longitudinal displacement of these clamps. During the operation of the tool mounted in the hool-holder 88, the bar or tube is held by the chuck 85. When the part after machining is cut from the tube of bar 87, the chuck is loosened and the bar-feed device comes into action to move the bar forward by the length of a new part to be machined. The combination of the operation of a hydraulic motor of a chuck and of a bar-feed device can be obtained by means of end-of-stroke limit switches actuating electro-vales or by manual contacts or by a combination of these two methods.

In those applications of the invention where the fluid motor incorporated with the rotatable spindle is used to operate a work clamping chuck or the like, it is usually important that the supply pressure to the motor be permanent. In case of an accidental failure in the pressure fluid supply circuit, it would be very undesirable for the chuck to be suddenly disengaged and release the work during rotation of the machine-tool.

According to an important feature of the invention, this possibility is averted by providing means for blocking the pressure fluid in the motor in an engaged condition of the mechanism operated thereby.

For this purpose, the invention contemplates providing valve means for selectively controlling the communication between an outer and an inner annular groove of the intermediate ring member. The valve may assume the form of an axially displaceable plunger spring-biassed in a direction to seal such communication, and exposed to the pressure of the fluid in the outer groove to move in the opposite direction for opening said communication. A further plunger member may be provided for similarly controlling a communication between the grooves for blocking the return flow of fluid from the motor to exhaust.

FIGS. 8 to 11, illustrating a form of the invention in which the last described features are embodied will now be described.

The construction shown in the figures is generally similar to that previously described with reference to FIGS. 1 and 2, and corresponding components have been designated by similar reference numerals. It will be seen from FIG. 10 that the stationary common outer ring member 3 is provided with two threaded fluid connections 14 and 15 to the fluid supply line. Cooperating with an arcuate recess formed in the peripheral end flange 3*d* of said ring member is a spigot 42 serving to locate the angular position of the ring 3 in the headstock frame. Just as in the embodiment shown in FIG. 1, lubricating orifices 54 and 55 are formed through the ring member 3 which discharge into proximity with the taper rollers of the bearings.

The connections 14 and 15 respectively open into the two axially spaced grooves 17 and 18 formed in the inner periphery of ring member 3. The spacer member 6 interposed between the inner race ring members 4 and 5 of the bearings and rotatable with the spindle is formed with internal annular grooves 20 and 21 which respectively register in axial position with the grooves 17 and 18. The inner grooves 20 and 21 respectively communicate with the axial passages 23 and 24 formed in the spindle and leading to the opposite ends of an hydraulic motor carried by the spindle, such as a cylinder and piston motor similar to that shown in FIG. 1.

The groove 17 is connected with groove 20 through a first radial duct 90, which is intersected by an axial bore 96 formed through the member 6, and is further connected with said groove 20 through a zigzag passageway comprising the oblique passages 91*a* and 91*b* and the intersecting axial bore 92 (see FIG. 11) formed in the ring member 6. In a similar way the groove 18 is connected with the groove 21 through a first radial duct 93 intersected by axial bore 97 (see FIG. 11), and by a second zigzag passageway comprising the oblique passages 94*a* and 94*b* intersected by axial bore 95 formed in ring member 6 (FIG. 10).

Slidably positioned in the axial bores 92 and 95 forming part of the zigzag passageways are valve plungers 98 and 99 respectively. The plungers 98 and 99 are biassed in opposite directions by compression springs 100. These bias springs are seated at one end against perforated plugging members 101, and at their opposite ends against imperforate plugging members 102.

The plungers 98 and 99 are formed with stop pins 98*a* and 99*a* respectively, preventing the plunger end faces directed towards the imperforate plugs from engaging the latter.

Similarly, slidably positioned in the bores 96 and 97 are the valve plungers 103 and 104 mounted between an imperforate and perforate plugging members 102 and 101 and urged by bias springs 100 towards the imperforate plugs 102. Each of these plungers 103 and 104 includes a reduced diameter intermediate section, 103*a* and 104*a* respectively, and an end stop member, 103*b* and 104*b* respectively, serving to provide pressure chambers betwen the imperforate plugging members and the corresponding end faces of the plungers. The pressure chambers are pressurized through the bores 105 and 106 with fluid from the respective outer grooves 17 and 18.

It will be noted that the plungers 98 and 104 associated with groove 17 are both oriented in one common direction while the plungers 99 and 103 associated with groove 18 are both oriented in a common direction reverse from the first.

In operation, when pressure fluid is delivered by way of inlet connection 14, the fluid flows from groove 18 through bores 94*a* and 106 and acts on both plungers 99 and 103 to urge them leftward (as in FIG. 10), so that a communication is established from said inlet groove through zigzag passageway 94*a*, 95, 94*b* to the passage 94, and another communication is established from the passage 93 for the exhaust flow of fluid, to groove 18, by way of the passage defined by the reduced-diameter section 103*a* and radial duct 90. Thus, the hydraulic motor will be operative even in the event that the exhaust pres-

sure in groove 17 should be insufficient to urge the plungers 98 and 104 to the right (as in FIG. 11).

Conversely, when pressure is supplied through the connection 15, then the plungers 98 and 104 are urged rightward and the operation of the motor is permitted even if plungers 99 and 103 are not urged leftward due to inadequate pressure from the groove 18.

Thus, the bias springs 100 may be calibrated so as to determine some suitable intermediate pressure in the range between the inlet pressure and the exhaust pressure. In such an arrangement, should there occur an accidental drop in pressure in that part of the external fluid circuit supplying the particular connection 14 or 15 that was pressurized, the pair of plungers associated with that connection are both moved to sealing position; since the other pair of plungers (asociated with the other fluid connection) were both moved to sealing positions due to the effect of the return or exhaust pressure, it is seen that all the connections between the motor and the external supply circuit are effectively blocked, and the motor is thus positively locked in its present condition. Thus the arrangement just described provides for a fully positive and irreversible operation of the motor.

However, in a desirable modification of this aspect of the invention, it is contemplated that the bias springs may be calibrated so as to move the plungers to their sealing positions only for a pressure lower than the return or exhaust pressure. In such case, the motor is rendered irreversible (or self-locking) only as concerns such faults in the supply system as may involve the part of the system between the pump and the distributor. In such case the plungers will at all times assume their open positions during normal operation and will only be moved to sealing condition in the event of a pressure drop simultaneously occuring in both sides of the pressure system connected to the respective lines 14 and 15.

Such a modification offers a number of advantages. For in the first place, it is for all practical purposes substantially as safe as the arrangement first described, since faults are much more liable to occur in the hydraulic lines between the pump and distributor (including the latter) than in the portion of the circuit beyond the distributor and to the exhaust sump. On the other hand, with such an arrangement the plungers will normally remain stationary throughout operation of the system and will only be actuated in the relatively unfrequent event of a defect. Wear is thus reduced and service life is greatly extended. Moreover, since a permanent communication is now present between the opposite sides of the motor and the distributor and the pressure source (just as in the absence of any locking valve arrangement), a possibility is made available which was not provided in the "closed-center" type of operation previously described. This possibility, now to be described, is of especial advantage in connection with lathes and similar machine-tools on which work may have to be exposed to more than one roughing and finishing passes.

In such cases, during the roughing pass or passes the work required to be clamped under extremely high pressure in the chuck to prevent any displacement of it by the tool, and such high clamping pressure may tend to cause some amount of distortion in the workpiece. It then becomes imperative, before commencing a finishing pass, to relieve the workpiece of any such distortion, and experience shows that for such purpose it is not sufficient just to relieve the supply pressure to the chuck actuating motor, but it is necessary to exert a brief yet relatively thorough disengaging action to permit a complete relaxation of the strains present in the workpiece before reengaging the chuck for the finishing pass. Such an effect can be obtained by imparting a sudden to-and-fro action to the distributor to reserve momentarily the fluid pressure acting in the opposite sides of the motor. However, such an operation cannot be successfully performed if the oil pressure has to displace the valve plungers, since the inertia opposed by the plungers to such displacement would introduce a substantial lag into the response time of the motor thereby preventing the desired rapid reciprocation of said motor and the chuck actuated thereby. There are, of course, various other circumstances in which one or the other of the two arrangements described above may be found more convenient, depending on the particular application.

It will be noted that in those forms of the invention described with reference to FIGS. 8 to 11, the plungers are arranged for displacement in axial directions so that their displacements are not affected by centrifugal forces which might otherwise undesirably modify the biassing forces applied to the plungers.

In all cases, when the valve plungers have been moved to their sealing positions, the pressure fluid is entrapped in the compartments of the fluid motor due to the sealing of the spindle passages such as 23 and 24, so that the piston or other displaceable element of the motor is positively locked in position.

While only two such passages 23 and 24 have been shown for connecting the distributor with the motor, it will be understood that more than two may be provided in parallel in order to increase, if desired, the total flow section and thus permit higher rates of oil flow for increasing the rate of response of the motor.

Various other modifications will occur to those familiar with the art, in particular, various features of the embodiments shown may be combined in other ways than those specifically disclosed, without exceeding the scope of the invention.

What I claim is:

1. In a machine-tool including a stationary frame, a rotatable spindle having a cylindrical portion and at least one axially directed bore, and a fluid motor bodily rotatable with said spindle and operatively connected with the opening of said bore, the combination of a pair of axially spaced roller cages, each including a set of regularly distributed taper rollers, said cages having their smaller diameter ends facing one another; a stationary outer-ring fitted in said frame, secured against rotation therein and surrounding said cages and rollers, and having spaced outer-race ways for said sets of rollers and a cylindrical inner surface therebetween; a set of three coaxially juxtaposed annular members having cylindrical inner faces fitted about said cylindrical portion of said spindle, both outer members being each provided with an inner-race way respectively for said sets of rollers and the middle member having a cylindrical outer surface defining with said cylindrical inner surface an annular gap of narrow radial width; axial tightening means carried by said spindle for securing said set of members to said spindle for rotation therewith; an external pressure fluid system; orifice means in said outer ring connectable with said fluid system; first annular groove means defined between said cylindrical inner and outer surfaces and registering with said orifice means; second annular groove means coplanar with the former between said cylindrical inner face of said middle annular member and the outer surface of said cylindrical portion; radial duct means connecting respectively first and second annular groove means and said second groove means with said axially directed bore, whereby when under pressure fluid of said system feeds said fluid motor and a minor portion of said fluid passing through said annular gap cools and lubricates said roller bearings; and means supplied by said fluid system for further cooling and lubricating said roller bearings.

2. The combination according to claim 1 wherein said taper rollers are axially bored for providing fluid passages therethrough.

3. The combination according to claim 1 wherein said stationary outer-ring is provided with substantially radial bores at the inner limit of the outer-race way

thereof, wherein said external fluid system comprises a sump for said fluid, a fluid pump, duct means connecting said pump to said orifice means and relief valve means for regulating the fluid pressure and wherein the means for further cooling and lubricating said roller bearings comprise further duct means connecting the outlet of said relief valve means with said radial bores.

4. The combination claimed in claim 1, wherein said fluid motor comprises a coaxial ram motor carried by the spindle, and further comprising a chuck assembly carried by the spindle adjacent to said ram motor and connected thereto for operation thereby between an engaging and a disengaging condition.

5. In a machine-tool including a rotatable spindle having a fluid ram bodily rotatable therewith, the combination of a pair of axially spaced roller bearings supporting the spindle; a common stationary outer ring surrounding the rollers of said bearings and defining outer-race surfaces therefor; an inner ring between the bearings and rotatable with the spindle, and having an annular cylindrical outer surface defining with an annular inner surface of said outer ring a narrow annular clearance gap; an external pressure fluid system; first orifice means in said outer ring connectable with said system to operate said ram to a selected condition; second orifice means in the spindle connected with said ram; groove means defined between said annular surfaces and registering with said first orifice means; radial duct means in the inner ring connecting said groove means with said second orifice means; and valve means interposed in said duct means and movable in response to pressure in said orifice means in a direction to permit flow through said duct means and means biassing said valve means in a direction to block the flow to thereby lock said motor in a selected condition thereof in the event of a drop in said pressure.

6. In a machine-tool including a rotatable spindle having a fluid ram bodily rotatable therewith, the combination of a pair of axially spaced roller bearings supporting the spindle; a common stationary outer ring surrounding the rollers of said bearings and defining outer race surfaces therefor; a recessed inner ring between the bearings and rotatable with the spindle and having an annular outer surface defining with an annular inner surface of said outer ring a narrow annular clearance; an external pressure fluid system; a pair of first orifices in said outer ring selectively connectable with a high- and a low-pressure side of said system to operate said ram to a selected condition; a pair of second orifices in the spindle connected with opposite sides of said ram; a pair of grooves defined between said annular surfaces and registering with said first orifices; ducts in the inner ring connecting said grooves with the respective second orifices; and valve means movably positioned in said recessed inner ring so as to be moved to a first position in response to pressure in one of said first orifices for permitting flow through a related duct to permit operation of said motor to a selected condition, and means biassing said valve means to another position for preventing flow through said duct to lock said ram in its selected condition in case of a drop in said orifice pressure.

7. The combination claimed in claim 6, wherein said valve means comprise a first and a second pair of valve members, the members of each said pair being respectively responsive to high pressure present in a corresponding one of said first orifices to be moved to positions permitting flow from the groove relating to said one first orifice towards the related second orifice, and from the other second orifice towards the groove relating to said other first orifice.

8. The combination claimed in claim 7, wherein said valve members comprise plungers movable in bores formed in said inner ring in directions parallel to the spindle axis.

9. The combination claimed in claim 6 wherein the means biassing said valve means are adjusted for preventing flow through said duct for a fluid pressure in said first orifices lower than the low pressure of said pressure fluid system.

10. A head-stock assembly for a machine-tool comprising a head-stock frame; a stationary outer-ring fitted into said frame and secured against rotation therein, said outer-ring having two outwardly flaring outer-race ways for taper rollers and a cylindrical inner surface therebetween in the middle part of said outer-ring and further having a pair of axially spaced apart first recessed duct means in said middle part; a pair of roller bearing cages, each including a set of taper rollers respectively cooperating with said outer-race ways; an axially bored rotatable spindle having a cylindrical outer portion and a pair of axially directed blind bores starting from one end of said spindle; a set of three coaxially juxtaposed annular members having cylindrical inner faces fitted about said cylindrical portion of said spindle, both outer members having each an inner-race way respectively for said sets of rollers and the middle member having a cylindrical outer surface defining with said cylindrical inner surface of the said outer-ring an annular gap of narrow radial width having a pair of recessed grooves respectively registering with said pair of first recessed duct means, said middle part further having a pair of second radially directed duct means respectively issuing from said grooves; axial tightening means carried by said spindle for securing said set of members to said spindle for rotation therewith; a chuck assembly including an annular double-acting ram motor secured to said end of the spindle; further duct means for respectively connecting each compartment of said double-acting ram to one of said blind bores; an external pressure fluid system having high and low-pressure sides; pipe means connecting respectively said sides to said first duct means; third duct means including peripheral groove means connecting respectively said second duct means to said blind bores; gearing means for rotating said spindle cooperating with the part thereof which is, with respect to the bearings, opposite to that which carries the chuck assembly and duct means supplied by said fluid system and issuing towards the smaller diameter of said roller bearings for cooling and lubricating thereof.

11. A head-stock assembly for a machine-tool comprising a head-stock frame; a stationary outer-ring fitted into said frame and secured against rotation therein, said outer-ring having two outwardly flaring outer-race ways for taper rollers and a cylindrical inner surface therebetween in the middle part of said outer-ring; a pair of roller bearing cages, each including a set of taper rollers respectively cooperating with said outer-race ways; an axially bored rotatable spindle having a cylindrical outer portion and two pairs of axially directed bores, each pair issuing at one end of said spindle; a set of three coaxially juxtaposed annular members having cylindrical inner faces fitted about said cylindrical portion of said spindle, both outer members having each an inner-race way respectively for said sets of rollers and the middle member having a cylindrical outer surface defining with said cylindrical inner surface of the said outer-ring an annular gap of narrow radial width; axial tightening means carried by said spindle for securing said set of members to said spindle for rotation therewith; a chuck assembly including an annular double-acting ram motor secured to each end of the spindle; duct means for respectively connecting one axial bore of the spindle to one compartment of said motor rams; an external pressure fluid system having high and low pressure sides; a set of at least two spaced apart first duct means in the middle part of said outer-ring connectable with said sides of said system; first annular axially spaced apart groove means defined between said cylindrical inner and outer surfaces and registering with said first

duct means; second radially directed duct means in said middle member issuing from said first groove means; and third duct means including second groove means and respectively coplanar with the first groove means for connecting one of said first groove means with one of said axial bores of the spindle; gearing means located between one chuck assembly and the nearer roller bearing for rotating said spindle and duct means supplied by said fluid system and issuing towards the smaller diameter of said roller bearings for cooling and lubricating thereof.

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