MACHINE TOOL HYDRAULIC SYSTEM

Filed Aug. 28, 1963 4 Sheets-Sheet 1

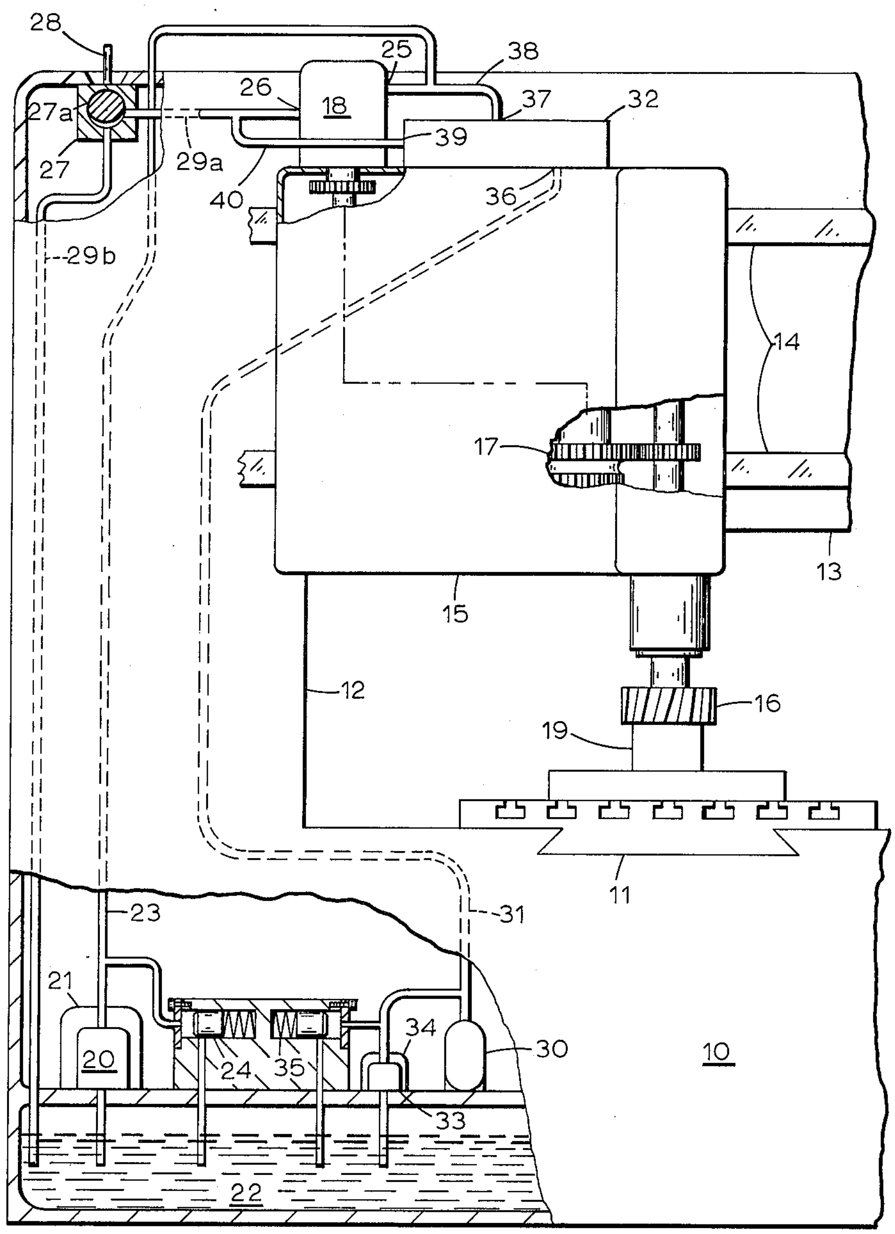

Fig. 1

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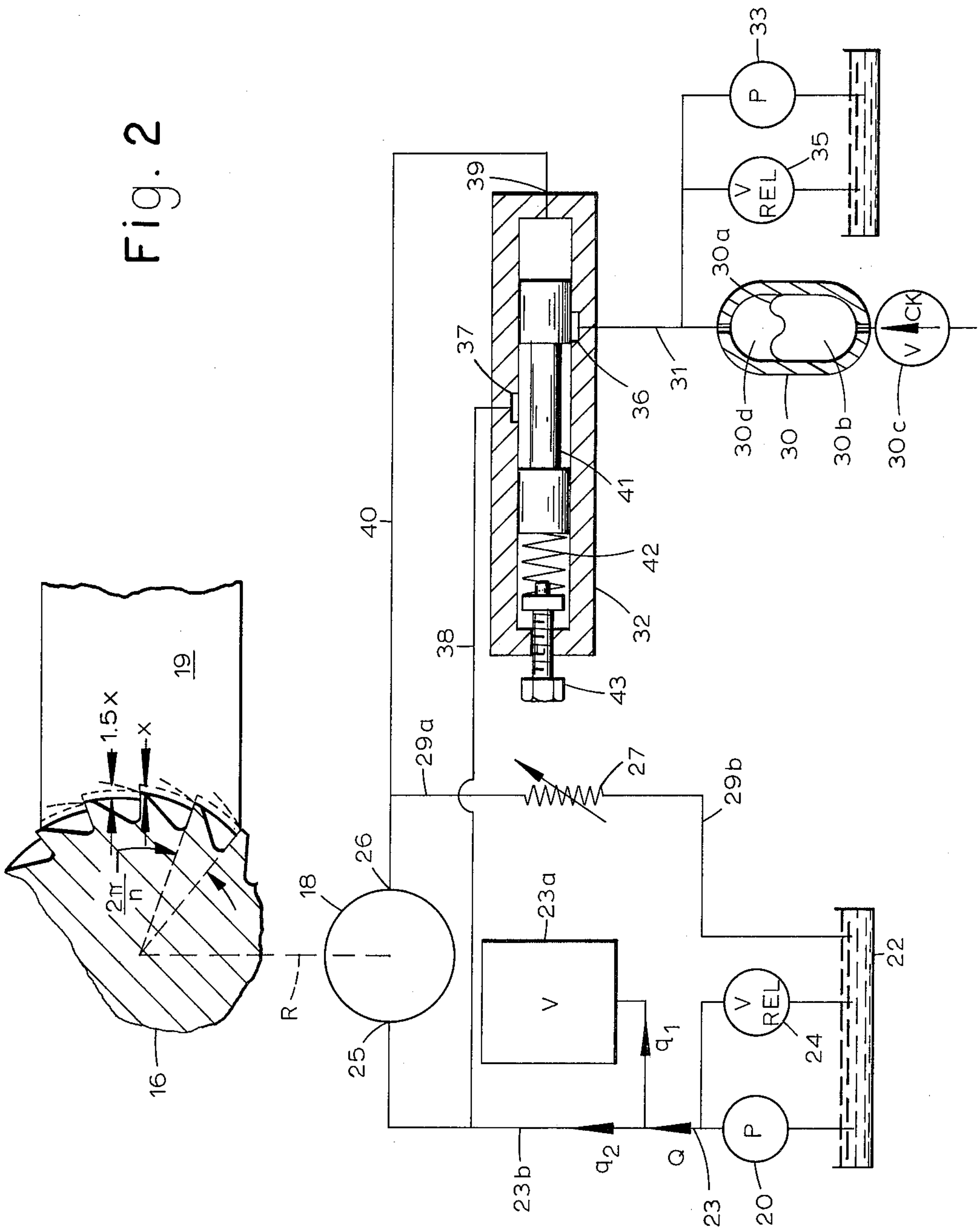

1.5x
x
19
16
R
18
26
25
23a
V
q_1
q_2
Q
23b
23
20
P
V REL
24
22
29a
27
29b
38
40
39
37
36
41
42
32
43
31
30a
30d
30
30b
30c
V CK
P
33
V REL
35

Fig. 2

MACHINE TOOL HYDRAULIC SYSTEM

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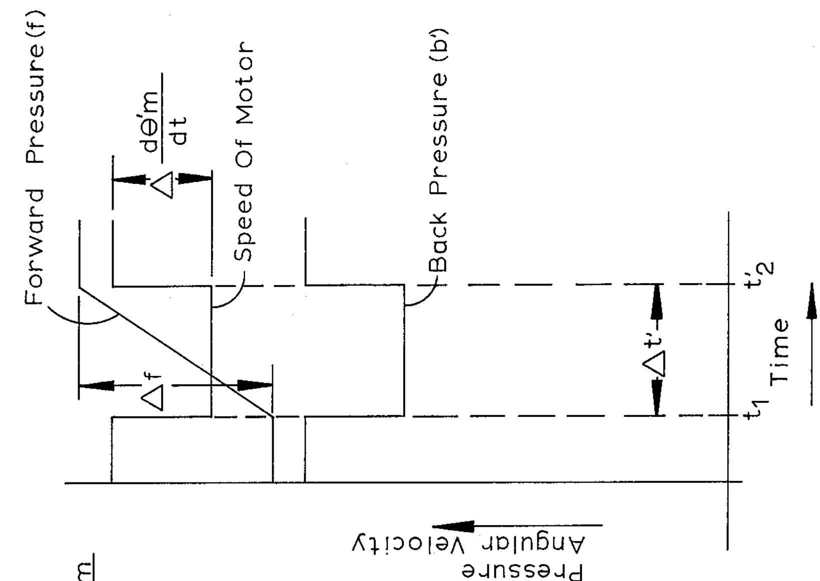


Fig. 4

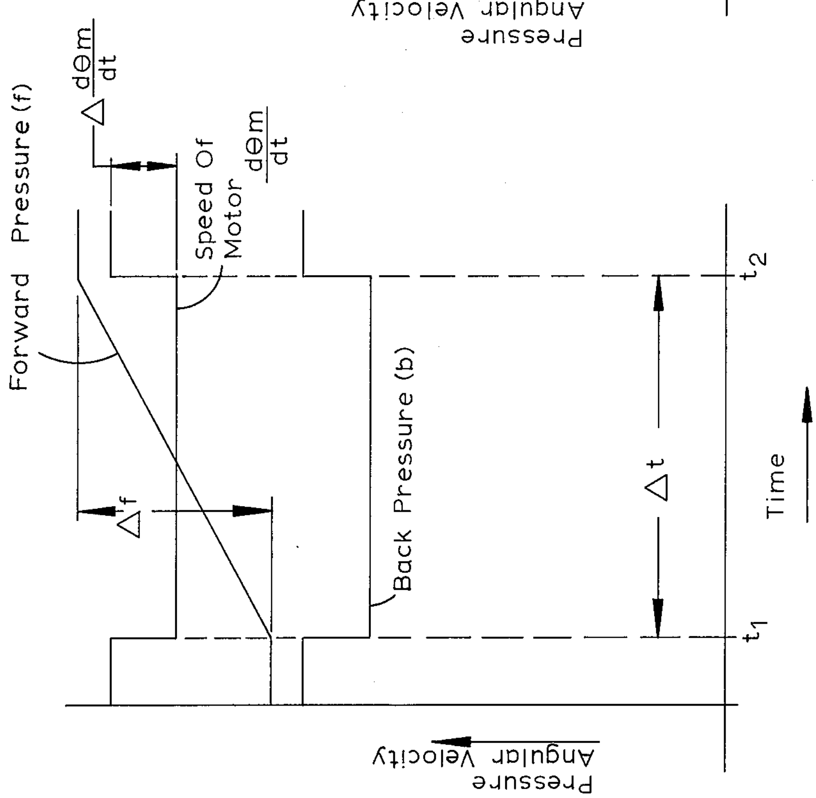
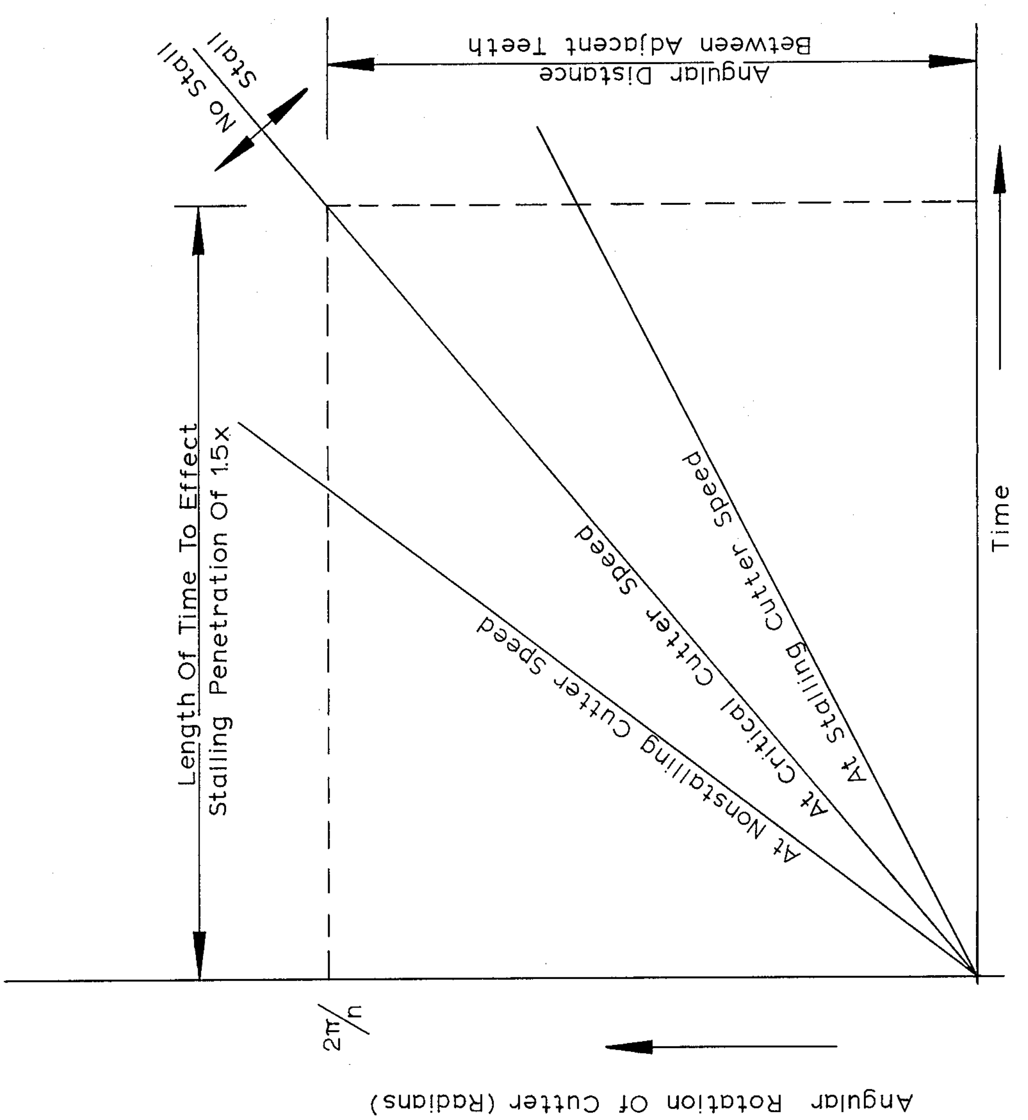


Fig. 3

Fig. 5


Angular Distance
Between Adjacent Teeth
No Stall
Stall
Length Of Time To Effect
Stalling Penetration Of 1.5x
At Stalling Cutter Speed
At Critical Cutter Speed
At Nonstalling Cutter Speed
Time
2π/n
Angular Rotation Of Cutter (Radians)

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3,220,312

MACHINE TOOL HYDRAULIC SYSTEM

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8 Claims. (Cl. 90—11)

The present invention relates to a hydraulic system in a machine, particularly suitable for the drive of a member, such as a machine tool cutter, which may be subjected to a suddenly increasing load.

When the load on a hydraulically driven machine tool member, such as a slide or rotating tool, increases, the load on the hydraulic motor which drives the member increases accordingly. With this increased load, the pressure of the hydraulic fluid supplied through a pressure line by a pump to the inlet side of the motor increases, and the fluid undergoes a slight compression. To provide continued operation of the motor without a drop in speed, the pump must not only supply the fluid required to keep the motor operating at the desired speed, but, because of the pressure increase resulting from the increased load, must also supply additional fluid to make up for the compression of fluid in the pressure line.

If the increase of load on the motor is gradual, the compression of fluid in the pressure line is gradual and, under these conditions, the pump can usually supply fluid at a rate to compensate for the gradual compression of fluid in the pressure line while, at the same time, supplying sufficient fluid at the rate required to keep the motor operating without a significant drop in speed. However, a sudden increase of the load on the motor results in a sudden compression of the fluid in the pressure line, and fluid for continued operation of the motor is not available until the pump has had time to supply the additional fluid in the pressure line required because of the compression of the fluid therein.

In some instances, particularly where the pressure line between the pump and the motor is relatively long, the momentary hesitation resulting from the application of a sudden load can have serious consequences. For example, in a large milling machine, where a cutter mounted on a carriage is rotated by a hydraulic motor mounted on the carriage remote from the pump in the base, a sudden increase in the load on the motor due, for example, to full sudden engagement of the cutter with a workpiece, will cause the motor to hesitate until the long pressure line between the pump and the motor can be filled by the pump as the fluid in the line compresses under the increased pressure therein caused by the increased load. However, during the time required for the pump to supply the fluid necessary to compensate for compression in the pressure line fluid, relative feed movement between the carrier and workpiece continues, increasing the load on the motor. This can cause a complete stalling of the cutter and fracture thereof as the feed movement continues.

One solution to this problem would be the installation of a large flywheel on the cutter spindle. With the cutter spindle rotating at the desired speed before the cutter is subjected to a sudden increase in load, the flywheel would define a source of stored kinetic energy which would be instantly available to keep the cutter spindle and cutter rotating while the pump is supplying fluid to the pressure line to compensate for the compression of fluid therein. However, the weight of the flywheel increases wear on the spindle bearings, and the increased inertia of the spindle cutter slows down starting and stopping of the cutter. Moreover, one advantage in using a hydraulic motor on a tool carriage to drive the tool instead of an

electric motor lies in its lighter weight, which is more easily supported by the carriage, and the use of a flywheel on the spindle cutter would substantially diminish this advantage. Finally, the use of a flywheel is applicable only with rotary driven members.

Another possible solution would be to increase the capacity of the pump supplying the hydraulic motor, or provide additional pump capacity available to the hydraulic motor, so that the demands of the hydraulic motor can be met at the same time the pump is supplying additional fluid to the pressure line to make up for the compression of the fluid therein when the load on the cutter increases sharply. Despite the fact that only a relatively small amount of fluid is required in the pressure line to compensate for compression of fluid therein, there is, in the case of a cutter being continuously fed into the work, only a very short time available to supply this fluid, and any additional pump capacity provided to meet this infrequent demand would have to be impractically large to be capable of supplying fluid at the high rate needed.

In the present invention there is provided a hydraulic system which, like the flywheel, provides a source of energy instantly available when a sudden load in encountered, but which offers many advantages over the flywheel. With the present invention, fluid can be supplied to the pressure line leading to the inlet side of the motor when needed at a high rate without the expense of large pump capacity. In brief, an auxiliary source of stored fluid under high pressure is provided for the hydraulic motor. This source is normally isolated from the motor, and is connected to the inlet side of the motor only when a sudden increase in load on the motor is encountered. Since only a small amount of fluid is required at a high rate, the auxiliary source of fluid under pressure is connected only briefly to the inlet side of the motor and when disconnected therefrom, is recharged. It is important, in the system of the present invention, that the auxiliary source of fluid under pressure be normally isolated from the pressure line leading to the inlet side of the motor. If such a source were continuously connected to this line, it would take fluid from the line when the pressure in the line was high and deliver fluid to the line when the pressure in the line was low. To prevent stalling of a hydraulic motor encountering a sudden load in accordance with the present invention, however, it is necessary that fluid be supplied at a high rate to the line leading to the inlet side of the motor when the pressure in that line is rising.

In the preferred form of the invention, the auxiliary source of stored hydraulic fluid under pressure comprises a hydraulic pressure accumulator which may be conveniently located in the base of the machine, a blocking valve which is preferably located near the motor, and a line connecting the accumulator to the blocking valve. The discharge line from the motor contains a restriction to establish a back pressure in the discharge line between the motor and the restriction which varies as the speed of the motor varies. The blocking valve has an operating port connected to the discharge line between the motor and the restriction, and has a discharge port connected to the inlet side of the motor. The instant the member, such as a rotary tool, driven by the hydraulic motor encounters a load sufficiently sudden to slow the hydraulic motor significantly, the back pressure drops to operate the blocking valve for the release of a surge of fluid under high pressure from the auxiliary source to the inlet side of the motor. This surge of fluid compensates for compression of the fluid in the line between the pump and the inlet side of the motor so that the motor can instantly resume normal speed despite the increase in load on the motor. As soon as the motor resumes speed, the accumulator is again isolated from the motor.

Unlike the mechanical flywheel, this system does not increase wear of the spindle bearings, does not prolong starting and stopping of the spindle, and can be utilized to overcome sudden large loads applied to a slide as well as those applied to a rotary member.

In a machine tool having a pump in the base, and having a movable carriage with a hydraulic drive motor thereon to drive, for example, a rotary tool mounted on the carriage, the accumulator, which is relatively heavy, is preferably mounted in the base, and the blocking valve is preferably mounted on the carriage. With this construction, the major portion of the weight of the system utilized to provide the auxiliary source of stored fluid under pressure is kept off the carriage, but, at the same time, the high pressure stored fluid is available right at the motor on the carriage where it is needed without travel through a long line which inevitably produces resistance to flow with a resultant pressure drop.

It is therefore one object of the present invention to provide a hydraulic system operable to prevent stalling of a hydraulic drive motor when subjected to a sudden load.

It is another object of the present invention to provide a source of stored fluid under pressure at a hydraulic drive motor on a carriage without the necessity of mounting heavy components on the carriage.

It is still another object of the present invention to provide an improved hydraulic system for maintaining the speed of a hydraulic drive motor.

It is yet another object of the present invention to provide a system for maintaining the speed of a hydraulic drive motor by providing a source of stored fluid under pressure which is released automatically to the inlet side of the hydraulic motor when the speed of the motor drops.

It is a further object of the present invention to provide a hydraulic system operable to keep a cutter rotating as it feeds into a workpiece without significant loss of speed when resistance to the cutter suddenly increases.

Other objects and advantages of the present invention should be readily apparent by reference to the following specification, considered in conjunction with the accompanying drawings forming a part thereof, and it is to be understood that any modifications may be made in the exact structural details there shown and described, within the scope of the appended claims, without departing from or exceeding the spirit of the invention.

In the drawings:

FIG. 1 is a fragmentary view in elevation, with parts broken away for clarity, of a milling machine having a hydraulic system constructed in accordance with the present invention;

FIG. 2 is a schematic diagram of the hydraulic system shown in FIG. 1;

FIGS. 3 and 4 are graphs showing the forward pressure, back pressure, and speed of the hydraulic drive motor plotted against time before the source of stored fluid under pressure is connected to the motor; and

FIG. 5 is a graph showing the relationship between rotation of the cutter and time for cutter speeds below, at, and above stalling speed.

There is shown in FIG. 1 part of a milling machine having a base **10** on which an elongated table **11** is mounted for longitudinal movement. Two columns **12** (only one of which is shown) extend upwardly from the base, one on each side of the table, and support a bridge portion **13** extending over the table. The bridge portion **13** has ways **14** on which a carriage **15** is mounted for cross movement with respect to the table. The longitudinal movement of the table on the base and the cross movement of the carriage on the bridge are effected by conventional hydraulic power means (not shown) independent of the hydraulic system shown.

A cutting tool **16** is rotatably mounted on the carriage **15** and is connected through a transmission **17** to a rotary hydraulic motor **18** also mounted on the carriage. Relative feed movement between a workpiece **19** mounted on the table and the cutting tool **16** is effected by movement of the table **11**, or movement of the carriage **15**, or both.

The hydraulic system for rotating the cutter **16** is shown in FIG. 1. A hydraulic pump **20**, and an electric motor **21** to drive the pump, are mounted in the base. The pump takes fluid from a sump **22** and delivers it under pressure to a pressure line **23**. A relief valve **24**, connected to line **23** and discharging to the sump, is provided to prevent overload and rupture of the pressure line. Pressure line **23** is connected to the inlet port **25** of the cutter motor **18** and, since the pump **20** is located in the base and the cutter motor is located on the carriage, the line **23** must be relatively long. The return, or discharge, line from the outlet port **26** of the motor to the sump contains a throttle valve **27** which defines a restriction adjustable in value by adjustment of throttle valve handle **28** to rotate the valve core **27*a*** connected thereto. The throttle valve, which may be secured in any convenient location, divides the return line into two sections: a secion **29*a*** extending between the motor **18** and the throttle valve and a section **29*a*** extending between the throttle valve and the sump. The pressure in return line section **29*b***, because it is in contimous communication with the sump, will always be approximately atmospheric pressure. The pressure in return line section **29*a*** will be at some higher value whenever the motor is running and therefore discharging hydraulic fluid. The pressure in return line section **29*a*** will vary in accordance with the rate of discharge of fluid from the motor (and hence will vary in accordance with the speed of rotation of the motor and the speed of rotation of the cutting tool).

A hydraulic pressure accumulator **30** is mounted in the base and is connected by line **31** to a blocking valve **32** mounted on the carriage close to the hydraulic motor **18**. A small, high pressure, hydraulic pump **33**, driven by electric motor **34**, takes fluid from sump **22** and delivers it under pressure to line **31**. A relief valve **35**, connected on the outlet side of the pump and discharging to the sump, establishes the maximum pressure in line **31**. The blocking valve **32** has an inlet port **36** connected to line **31**, and an outlet, or discharge, port **37** connected through the relatively short line **38** to the inlet port **25** of the hydraulic motor **18**. The blocking valve **32** also has an operating port **39** connected by line **40** to return line section **29*a*** and operates in response to the pressure in line section **29*a***, opening to connect the accumulator **30** to inlet port **25** of the motor when the pressure is below a predetermined value and closing to isolate the accumulator from the motor when the pressure is above that predetermined value.

During a normal cutting operation (assuming a constant feed rate between the rotating cutter and the workpiece with a normal cutter tooth penetration into the workpiece of, say, 0.13 inch) the forward pressure f on the motor (that is, the pressure in line **23**) will be constant under steady state conditions (that is, when there is no variation of the load defined by the engagement of the cutter with the workpiece) and will be at a value of, say, 2000 pounds per square inch. The relief valve **24**, which is provided merely to prevent rupture in the system, is set to operate at a relatively high pressure of, say, 5000 pounds per square inch. Under normal cutting conditions the back pressure b on the motor (that is, the pressure in line **29*a***) will be constant under steady state conditions (that is, when there is no variation in the speed of the motor) and will be at a value of, say, 150 pounds per square inch. Under these conditions, blocking valve **32** will be closed to isolate the accumulator **30** and line **31** from motor **18**.

If there is a gradual change in load on the motor (because, for example, of a gradual increase in the feed rate and a deeper penetration of the teeth of the cutter in the workpiece) of the forward pressure f will rise gradually. This pressure rise compresses the fluid in line **23**, but since the pressure rise is gradual, the constant volume pump **20** can supply the small amount of additional oil required to fill line **23** without any significant drop in speed of the motor. Since the back pressure b is directly related to the speed of the motor, there is no significant drop in the back pressure and blocking valve **32** remains closed.

If, however, there is a sudden change in the load on the motor, the forward pressure will rise sharply and, in an instant, the fluid in the line **23** will be compressed. Since the pump cannot in this instant, both supply fluid to compensate for the sudden compression of the fluid and, at the same time, supply sufficient fluid to keep the motor operating at a constant speed, the motor will slow down and, consequently, the back pressure b will drop. This drop in back pressure will open the blocking valve to connect the accumulator **30** to the motor.

Although the accumulator **30** may be of any suitable conventional type, a bladder-type accumulator is illustrated. As shown in FIG. 2, the accumulator has a bladder **30***a* dividing the accumulator into two chambers: a chamber **30***b* containing a fixed mass of air introduced through check valve **30***c*, and a chamber **30***d* in communication with line **31** and containing hydraulic fluid. With the valve **32** closed and the pump **33** supplying fluid, the air in chamber **30***b* is compressed to form a reservoir of high pressure. The instant valve **32** is opened, this pressure produces a surge of fluid out discharge port **37** of the valve **32** at a substantially higher rate than could be produced by pump **33**. Thus, the accumulator **30**, line **31**, and valve **32** define an auxiliary source of stored fluid under high pressure for the motor **18**. This source is continuously charged by the small, high pressure, low volume pump **33**. If, for example, relief valve **35** is set to open at a pressure of 5000 pounds per square inch, this pressure will be available at the discharge port **37** of the blocking valve **32** on the carriage despite the fact that the accumulator **30** is located in the base. When the blocking valve **32** is opened, a surge of fluid is communicated through the short line **38**, without significant pressure drop, to the inlet port **25** of motor **18** to increase the speed of the motor and the cutter. As the motor speed increases, the back pressure b increases to close the blocking valve **32**. After the blocking valve **32** is closed, any loss of pressure in line **31** due to the surge of fluid delivered to the motor is made up by pump **33**.

For a more complete understanding of the operation of the present invention, reference is made to the schematic diagram of FIG. 2. For convenience, the effect of compression of the hydraulic fluid in line **23** before value **32** is opened can be considered separately from the effect of the fluid on the motor **18**. In the diagram of FIG. 2, the line **23** between the pump **20** and the motor **18** is represented by a container **23***a* of a volume equal to the volume of line **23** between the pump and the motor in which all compression of the fluid between the pump and the motor occurs, and by a line **23***b* of zero volume in which all fluid available to the motor flows. In other words, the flow rate, or output, Q of the pump **20** is made up of a flow q_1 (into container **23***a*) which merely compensates for compression of fluid in line **23** and a flow q_2 (in line **23***b*) which is available to the motor. At all times $Q=q_1+q_2$.

The flow q_1 in cubic inches per second equals (V/E) times (df/dt) where V is the volume of line **23** between the pump and the motor in cubic inches, E is the bulk modulus of the fluid in pounds per square inch, and df/dt is the rate of increase of the forward pressure f in the system (before valve **32** is opened) in pounds per square inch per second.

If d is the motor displacement in cubic inches per revolution, the motor displacement in cubic inches per radian may be expressed as $d/2\pi$. If $d\theta(m)/dt$ is the speed of the motor in radians per second, the flow q_2 to the motor must equal $(d/2\pi)$ $(d\theta m/dt)$ in cubic inches per second. Thus

$$Q\text{in.}^3/\text{sec.}=(V/E)(df/dt)+(d/2\pi)d\theta m/dt)$$

or

$$(d\theta(m)/dt)\text{rad./sec.}=Q(2\pi/d)-(\text{V/E})(2\pi/d)(df/dt)$$

Thus, with a pump of constant volume output Q, the speed of the motor will equal $Q(2\pi/d)$ for any given steady state period when the pressure in the system is constant. However, the speed of the motor will be diminished in any given period (before valve **32** is opened) when the pressure is rising. The greater the rate of pressure increase, the slower the speed of the motor. This relationship is shown best in FIGS. 3 and 4. FIG. 3 shows how a given rise Δf in the forward pressure f, occurring in time Δt, causes a drop $\Delta d\theta(m)/dt$ in the speed of the motor. FIG. 4 shows that the same rise Δf in the forward pressure f, if occurring in a shorter time interval $\Delta t'$, will cause a larger drop $\Delta d\theta(m)/dt$ in the speed of the motor.

Since the restriction defined by throttle valve **27**, although adjustable, does not vary during operation, the back pressure b (that is, the pressure between the outlet side of the motor and the restriction) will vary in accordance with the speed of the motor, as indicated FIGS. 3 and 4. It is this back pressure which is in communication with the operating port **39** of the blocking valve **32**. As shown in FIG. 2, the operating port **39** of the valve is in continuous communication with one end of valve plunger **41** of blocking valve **32** to urge the valve plunger to the left (as viewed in FIG. 2) in opposition to a spring **42** in the valve, and closing the valve when the back pressure rises above some predetermined value. The valve member **41** shifts to the right as the back pressure drops and, when the back pressure drops below the predetermined value, the valve opens, connecting the accumulator line **31** to the inlet side of the motor. The spring is adjustable by screw **43** so that the pressure at which the valve opens and closes can be selected.

The speed of the motor

$$\frac{d\theta(m)}{dt} \text{ equals } R\frac{d\theta(c)}{dt}$$

before valve **32** is opened, where R is the ratio of the gear train in the transmission and $d\theta(c)/dt$ is the speed of the cutter so that

$$R(d\theta(c)dt)\text{rad./sec.}=Q(2\pi/d)-(V/E)(2\pi/d)(df/dt)$$

or

$$(d\theta(c)/dt)\text{rad./sec.}=Q(2\pi/Rd) -(V/E)(2\ /Rd)(df/dt)$$

Integrating this expression yields, when the limits of integration are f at $t=0$ and $(f+\Delta f)$ at $t=t$

$$\theta(c)\text{rad.}=Q(2\pi Rd)t-(V/E)(2\pi/Rd)\Delta f$$

where t is time in seconds and Δf is the increase of pressure in time t.

When a workpiece is fed into a rotating cutter (or a rotating cutter into a workpiece) each tooth, in traversing the span $2\pi/n$ radians between adjacent teeth (where n is the number of teeth on the cutter), will penetrate the workpiece some distance depending on the relation between the feed rate and the rate of rotation of the cutter. Assuming a constant feed rate, the penetration of the teeth of the cutter on each rotation of $2\pi/n$ radians will vary inversely with the speed of the cutter motor. In other words, the slower the cutter motor rotates, the deeper the penetration of the cutter teeth into the work, and the greater the load on, or resistance to continued rotation of, the motor.

If, in a normal cut with a constant feed rate between the tool and the work, each tooth of the tool penetrates the work a distance x equal to .013 inch (as shown in FIG. 2) as it traverses the span $2\pi/n$ radians, the resistance to cutter movement will be great enough to stall the cutter motor (if no source of auxiliary fluid under pressure were available) at some greater tooth penetration, say, $1.5x$ (or .0195 inch). It is therefore necessary for each tooth to complete a rotation of $2\pi/n$ radians in order to encounter the cut of the preceding tooth) before a relative feed movement of $1.5x$ occurs between the work and the cutter. This relationship is shown in FIG. 5, assuming a constant feed rate. Each line shown indicates the relationship between angular rotation of the cutter and time so that, in each case, the slope of the line represents a cutter speed. FIG. 5 shows that at cutter speeds below a critical cutter speed, each tooth will fail to reach the cut of the preceding tooth before a feed movement of $1.5x$ distance occurs so that, at these speeds, the cutter and cutter motor will stall. Since cutter speed is proportional to motor speed and motor speed is proportional to back pressure, the back pressure provides a reliable signal for release of the auxiliary supply of stored fluid under pressure.

Screw **43** of blocking valve **32** is set so that the valve opens (that is, the valve plunger **41** moves to the right of the position shown in FIG. 2), connecting the accumulator to the inlet side of the motor, when the back pressure (which is determined by the cutter motor speed) drops to a value, say 100 pounds per square inch, in response to a drop in cutter motor speed which drops the cutter speed to a value approaching the critical cutter speed. The opening of valve **32** introduces high pressure fluid to the forward pressure line **23** and raises the speed of the cutter motor before it stalls. As the cutter motor speed increases, the back pressure increases, and the blocking valve member **41** is shifted to the left (to the position shown in FIG. 2) to again isolate the accumulator **30** from the cutter motor. While the accumulator is isolated from the cutter motor, it is recharged by the high pressure, low volume, pump **33**.

If the power available at the pump motor **21** is incapable of rotating the cutter motor (even if these is no compression of oil in the system) when the teeth of the cutter engaged with the work penetrate .0195 inch or deeper into the work, and if, for example, the relative feed rate between the work and the cutter is 10 inches per minute, or 0.17 inch per second, a stalling penetration of .0195 inch will be realized in .0195/.167 or .117 second. If it be assumed that the cutter has 12 teeth, the cutter must rotate one tooth span, or $2\pi/12$ radians in .117 second or less to prevent stalling. Therefore, the critical speed of the cutter is $2\pi/12(.117)$, or 4.47 radians per second.

It has been shown that the speed of cutter may be expressed as

$$d\theta(c)/dt=(2\ \ /Rd)(Q)-(V/E)(2\pi/Rd)(df/dt)$$

Assume, for illustrative purposes, that the ratio R of motor rotation to cutter rotation is 17.5, that motor displacement d is 4.07 cubic inches per revolution, that the constant output of the pump Q is 76 cubic inches per second, that the volume of the line between the pump and the motor is 300 cubic inches, and that the bulk modulus E of the oil utilized as the hydraulic fluid is 238,000 pounds per square inch. With no change of pressure, and hence no compression of oil in line **23**, the speed of the cutter will be 6.70 radians per second. The rate of change of pressure df/dt at the critical cutter velocity of 4.47 radians per second, however, is 20,100 pounds per square inch per second, producing a change of pressure Δf of 20,100 (.117), or 2350 pounds per square inch in the time required for stalling penetration. The change in volume ΔV of the oil is equal to $\Delta f V/E$ that is, (2350) (300)/(238,000), or 2.96 cubic inches of fluid. To replace this fluid in .117 second or less requires a minimum flow of 2.96/.117, or 25.3 cubic inches per second, which the accumulator can easily supply since only a small volume is required at infrequent intervals. It will be noted that pump **33** can have a flow rate substantially below this value, say 3.0 cubic inches per second, since it has ample time after valve 32 closes to recharge the accumulator.

What is claimed is:

1. In a machine tool having a rotary cutting tool,
(a) a hydraulic drive motor to rotate the tool,
(b) a hydraulic pump to supply fluid under pressure to the motor for operation thereof,
(c) an accumulator normally isolated from the motor,
(d) and means responsive to a sudden increase in load on the cutting tool to connect said accumulator momentarily to the motor.

2. In a machine tool having a rotary cutting tool,
(a) a rotary hydraulic drive motor to rotate the tool,
(b) a hydraulic pump to supply fluid under pressure to the inlet side of the motor for operation of the motor,
(c) an accumulator normally isolated from the inlet side of the motor,
(d) means responsive to a sudden increase in load on the cutting tool to connect said accumulator momentarily to the inlet side of the motor,
(e) and means to charge the accumulator.

3. In a machine tool having a rotary cutting tool,
(a) a rotary hydraulic drive motor to rotate the tool,
(b) a first hydraulic pump to supply fluid under pressure to the inlet side of the motor for operation of the motor,
(c) an accumulator normally isolated from the inlet side of the motor,
(d) a valve operable in response to the speed of the motor to connect the accumulator to the inlet side of the motor when the speed of the motor drops below a predetermined value and to disconnect the accumulator from the inlet side of the motor when the speed of the motor rises above said predetermined value,
(e) and a second pump to charge the accumulator.

4. In a machine tool having a rotary cutting tool,
(a) a hydraulic drive motor connected to said cutting tool to effect rotation thereof,
(b) a hydraulic pump connected to the inlet side of the motor for operation thereof,
(c) a discharge line connected to the outlet side of the motor, said discharge line having a restriction therein to establish a back pressure in the discharge line between the motor and the restriction,
(d) a hydraulic accumulator normally isolated from the inlet side of the motor,
(e) and means to connect said accumulator to the inlet side of the motor in response to a drop in said back pressure.

5. In a machine tool having a rotatable cutting tool,
(a) a rotary hydraulic motor connected to said cutting tool for rotation thereof,
(b) a first hydraulic pump connected to the inlet side of the motor for operation thereof,
(c) a discharge line connected to the outlet side of the motor, said discharge line having a restriction therein to establish a back pressure in the discharge line between the motor and the restriction varying as the speed of the motor varies,
(d) a hydraulic accumulator normally isolated from the inlet side of the motor,
(e) means including a blocking valve operable in response to variation of said back pressure to isolate the accumulator from the inlet side of the motor when the back pressure is above a predetermined value and to connect the accumulator to the inlet side of the motor when the back pressure drops below said predetermined value,

(f) and a second hydraulic pump to charge said accumulator.

6. In a machine tool having a rotatable cutting tool,

(a) a rotary hydraulic drive motor connectable to said cutting tool to effect rotation thereof,

(b) a hydraulic pump,

(c) a pressure line connecting the pump to the motor,

(d) a hydraulic accumulator, a blocking valve, and a line connecting the accumulator to the blocking valve to define a source of stored fluid under pressure, said blocking valve located close to the motor,

(e) a line connected between the blocking valve and the inlet side of the motor,

(f) and means responsive to an increase in load transmitted to the motor through said cutting tool to operate the blocking valve and release stored fluid under pressure to the inlet side of the motor.

7. In a machine tool having a base and having a carriage movable relative to the base, said carriage having a tool rotatably mounted thereon, the combination comprising:

(a) a rotary hydraulic motor supported by the carriage and connected to rotate the tool,

(b) a hydraulic pump in the base,

(c) a pressure line connecting the pump to the inlet side of the motor,

(d) a discharge line connected to the outlet side of the motor, said discharge line having a restriction therein to establish a back pressure in the discharge line between the motor and the restriction,

(e) a hydraulic accumulator mounted in the base and a blocking valve mounted on the carriage in communication with the accumulator to define a source of stored fluid under pressure, said blocking valve having a discharge port connected to thet inlet side of the hydraulic motor and having an operating port connected to the discharge line between the motor and the restriction, said blocking valve operable in response to the pressure at said operating port to close said discharge port when said back pressure is above a predetermined value and to open said discharge port when said back pressure drops below said predetermined value.

8. In a machine tool having a base and having a carriage movable relative to the base, said carriage having a tool rotatably mounted thereon, the combination comprising:

(a) a rotary hydraulic motor supported by the carriage and connectable to rotatably drive the tool,

(b) a first hydraulic pump in the base,

(c) a pressure line to connect the first pump to the inlet side of the motor,

(d) a discharge line connected to the outlet side of the motor, said discharge line having a restriction therein to establish a back pressure in the discharge line between the motor and the restriction, varying as the speed of the motor varies,

(e) a source of stored fluid under pressure comprising a hydraulic accumulator mounted in the base, a blocking valve mounted on the carriage, and a line connecting the blocking valve to the accumulator,

(1) said blocking valve having a discharge port connected to the inlet side of the hydraulic motor and having an operating port connected to the discharge line between the motor and the restriction,

(2) said blocking valve operable in response to the pressure at said operating port to close said discharge port when the speed of said motor is above a predetermined value and to open said discharge port when the speed of said motor drops below said predetermined value,

(f) and a second hydraulic pump in the base connected to said source of stored fluid under pressure to charge said source.

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