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Pettigrew

[11] **3,762,231**

[45] **Oct. 2, 1973**

[54] **VARIABLE DIAMETER PULLEY**

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[73] Assignee: **Rockwell International Corporation**, Pittsburgh, Pa.

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[21] Appl. No.: **228,064**

Related U.S. Application Data

[62] Division of Ser. No. 27,786, April 13, 1970.

[52] **U.S. Cl.** .... **74/230.17 F**
[51] **Int. Cl.** **F16h 9/00**
[58] **Field of Search** .... 74/230.17 F

[56] **References Cited**

UNITED STATES PATENTS

2,868,027	1/1959	Oberholtz et al.	74/230.17 FT
2,198,940	4/1940	Heyer et al.	74/230.17 F
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FOREIGN PATENTS OR APPLICATIONS

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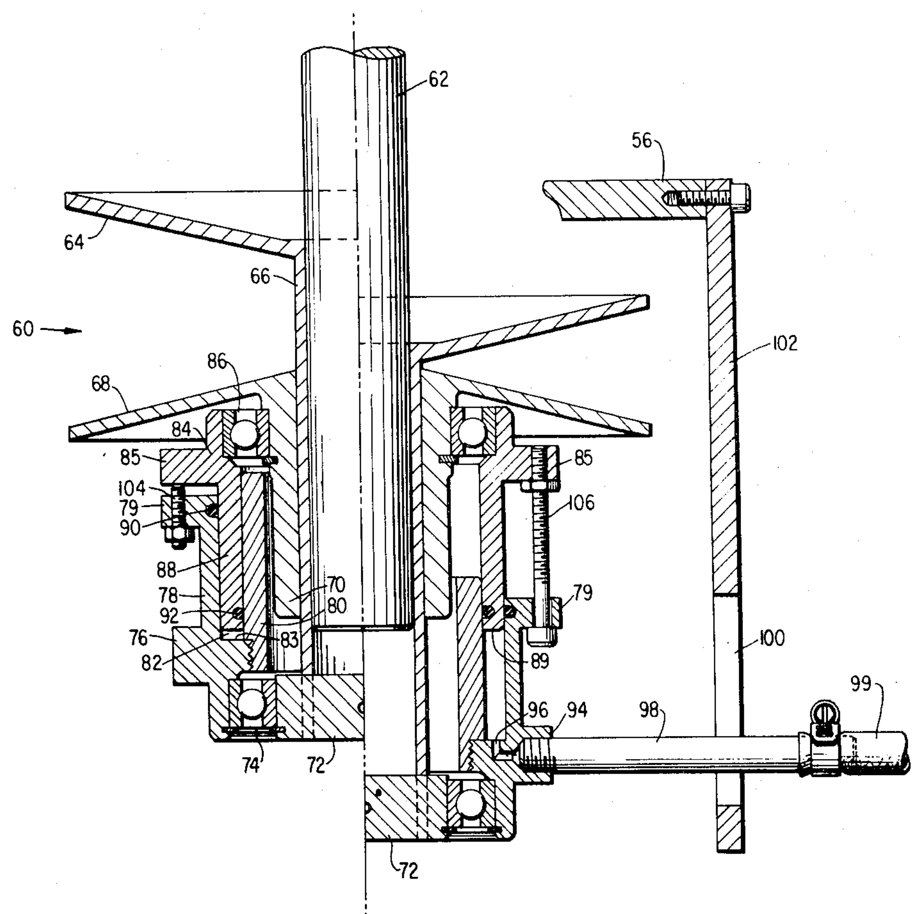
Primary Examiner—C. J. Husar
Attorney—Strauch, Nolan, Neale, Nies & Kurz

[57] **ABSTRACT**

A variable diameter pulley hydraulically shiftable to a selected effective diameter, the pulley including stop means for determining its maximum and minimum effective diameters.

2 Claims, 2 Drawing Figures



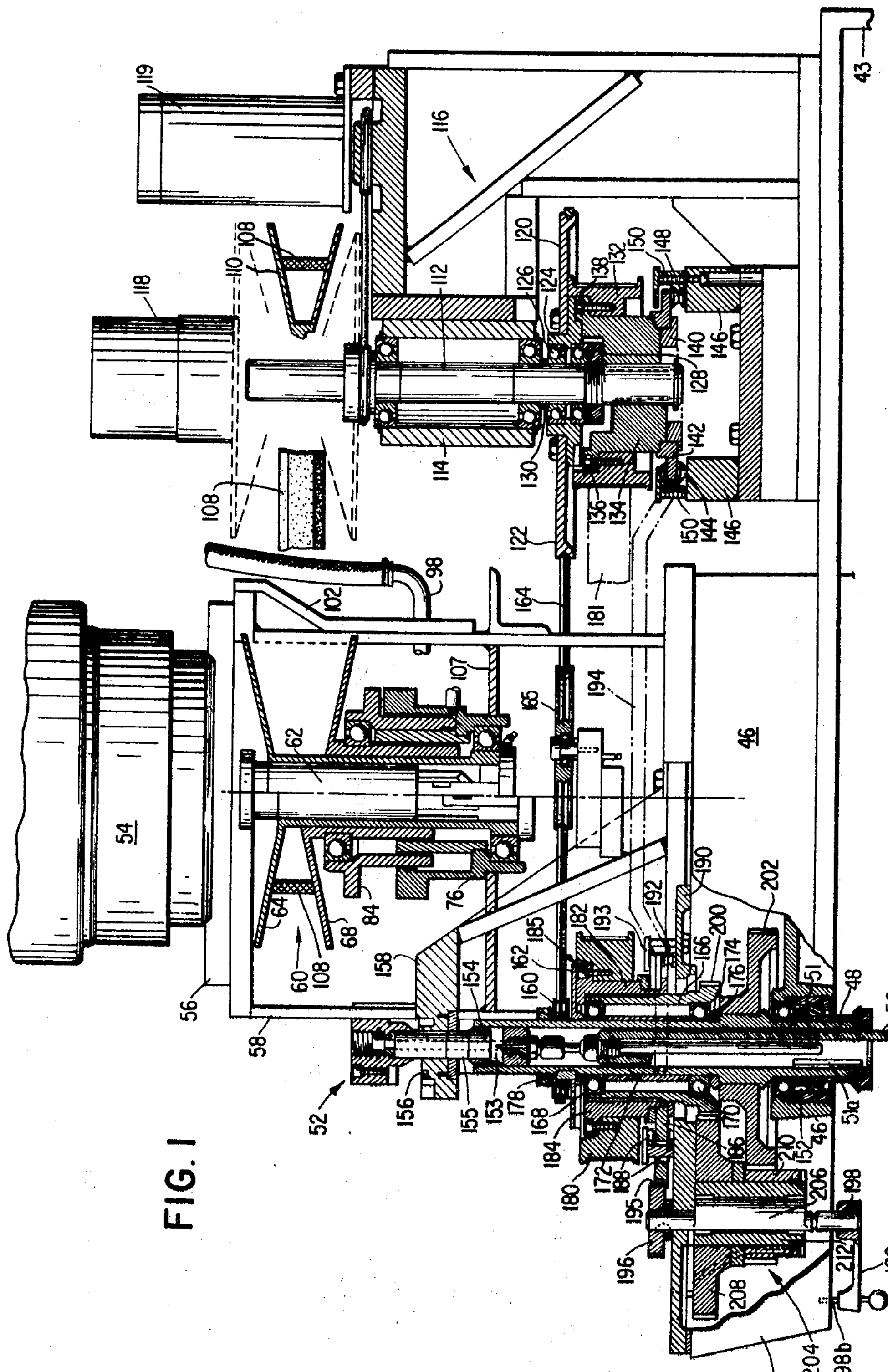
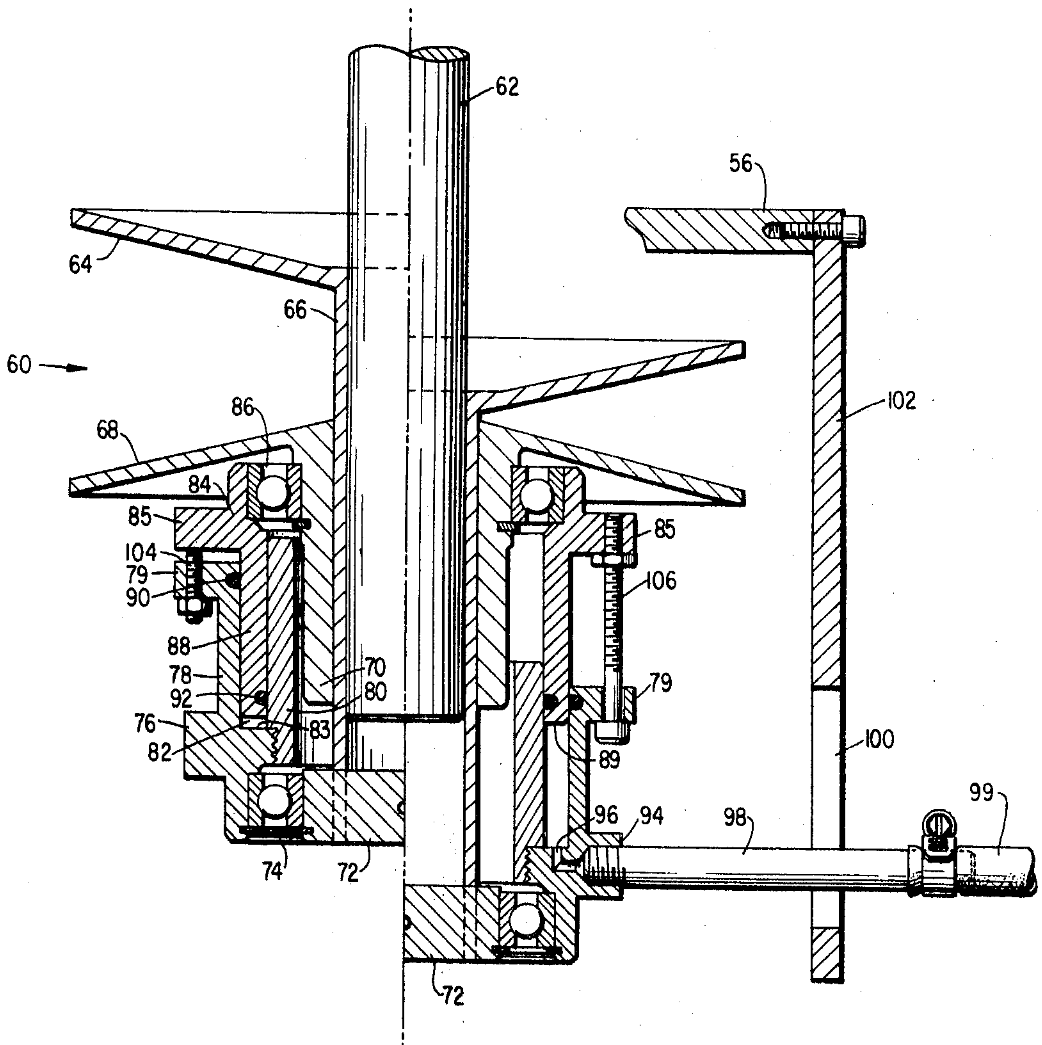


FIG. I

FIG. 2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VARIABLE DIAMETER PULLEY

This is a division, of application Ser. No. 27,786 filed Apr, 13, 1970.

BACKGROUND OF THE INVENTION

The invention described in original application Ser. No. 27,786, now U.S. Pat. No. 3,702,740, relates generally to drilling machines and more particularly to a novel, improved, coolant or oil-hole drilling machine capable of drilling accurate, true holes in a wide variety of metals at the maximum determinable cutting and penetration rates for the metal workpiece with acceptable tool life not heretofore attainable with conventional solid tool drilling machines and/or known oil-hole drilling machines.

The concept and process of oil-hole drilling has been known for about forty years, but has generally been confined to automatic screw machine and turret lathes and used primarily in speciality applications for drilling deep holes, e.g., up to about 20 diameters in depth. The limited usage and acceptance of the process for more conventional high production shop applications has been attributed to the high cost of oil-hole drills which generally were considered custom items, the unavailability of suitable oil-hole drilling machines, and the resulting attitude that relatively few advantages were gained from its usage. Available oil-hole drilling machines used in the past were either specially-built custom machines or mere adaptations of conventional solid tool, dry drilling machines modified to pump coolant through the drill.

In more recent years, industry has awakened to the potential of the oil-hole drilling process and its application to more conventional, high production drilling operations. It was pointed out at an American Society of Tool and Manufacturing Engineers' Conference held in April, 1967, that a solid tool drill press built years ago, e.g., in 1918, if it had adequate speed and power, would probably drill a hole as fast as any modern solid tool drill press, due largely to the fact that there has been no major change in the condition existing at that point where the metal is cut by the solid drill. When a drill enters the metal, it is completely enclosed on all sides but one, and that one is partially blocked by escaping chips. This then creates ideal conditions for overheating and wearing or breaking a cutting edge. Consequently, the speed of the drill and rate at which it is fed into the metal is very much limited by these existing conditions.

On the other hand, oil-hole drills direct a high pressure coolant through holes in the flutes of the oil-hole drill from the shank end down to the cutting end of the drill, with the coolant serving not only to cool the drill point but also to wash the chips upwardly out of the hole.

The conferees noted that recent advances in and availability of oil-hole drill equipment and associated coolant pumping equipment such as that illustrated in U.S. Pat. No. 3,342,086 have increased penetration rates of oil-hole drills substantially over penetration rates of conventional solid drills. While such an increase was considered impressive, it was generally agreed that in order to obtain the maximum benefit and savings from oil-hole drilling, new commercially available oil-hole drilling machines would have to be built, since existing machines simply did not have the rigidity, variable speed and feed capacities, and/or effective coolant system necessary to provide the maximum determinable cutting and penetration rates for metal workpieces with acceptable tool life desirable for production operations. It was pointed out that this inadequacy of conventional machines has been emphasized by the development of the carbide-tipped drill, since there is no commercial standard machine capable of utilizing the carbide-tipped drill to its fullest advantage in any but the smallest drill sizes.

At the conference, it was recognized that the most prevalent type of oil-hole drill press such as that illustrated by U.S. Pat. No. 2,977,827 was merely a modification of a conventional drill press with an attachment type inductor gland attached to the lower end of the spindle and receiving the shank end of the oil-hole drill. Coolant is fed into the inductor and thence through the passageways of the drill down to the cutting edge. However, it was concluded that this type of machine would not be acceptable for general application, since experiments showed that the length of the extension of the drill point from the drill spindle and spindle bearing had an appreciable effect on drill life which was improved by minimizing the total extension of the drill from the spindle. Thus, in a machine employing an inductor gland, the gland adds to this total extension and thereby decreases tool life. It was generally concluded that the shortest total extension will be obtained by feeding coolant directly through the spindle rather than an external attachment type inductor.

In addition, it was also agreed that no conventional available drill press could be readily converted since none possesses the stiffness, horsepower, variable feed and speed requirements necessary for a versatile, general purpose, oil-hole drill press.

SUMMARY OF THE INVENTION

The object of this invention resides in the provision of a novel variable diameter pulley included in a novel variable speed drive assembly capable of rotating a drill spindle within an infinitely variable speed range, e.g., 125 rpm to 10,000 rpm. The drive assembly includes a relatively high horsepower motor driving a variable diameter pulley arrangement which is connected to the drill spindle through an intermediate gear assembly which may be shifted to one of three possible speed ranges. Thus, a desired spindle speed may be selected by setting the gear assembly within one of the speed ranges and then adjusting the diameter of the pulleys to select a specific speed within the set range.

Other objects and advantages will become more apparent from reading the following detailed description of the invention with reference to the accompanying drawings in which like numerals indicate like parts and from the appended claims.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a fragmentary, partially sectioned side elevation view of the variable speed spindle drive system drivingly connected to the upper spindle of the two piece spindle structure mounted on the head of the drill press described in original application Ser. No. 27,786;

FIG. 2 is a fragmentary side elevation section view of the primary hydraulically operated, variable diameter pulley which is part of the drive system shown in FIG. 1, the right and left sides of FIG. 2 showing the pulley in its maximum and minimum diameter positions, respectively;

DETAILED DESCRIPTION OF THE INVENTION

As shown in FIGS. **1**, the spindle structure for rotating and vertically moving a drill is a two-piece structure including a hollow upper spindle **48** rotatably mounted in housing **46** and a lower spindle **50**, the upper end **51** of which is vertically reciprocable within upper spindle **48** and rotatable therewith through a spline connection **51***a*.

THE VARIABLE SPEED DRIVE SYSTEM AND SPINDLE STRUCTURE

Referring to the drawings, the drive assembly **52** for rotating upper spindle **48** includes a motor **54** mounted on a support platform **56**, the corner legs **58** of which are fixed to the top of housing **46**. A hydraulically operated, variable diameter pulley **60** (FIG. **2**) includes a first inner conical disc **64** having a hub **66** slidably mounted on motor shaft **62** by a spline connection and a second outer conical disc **68** having a hub **70** slidable on hub **66** and rotatable therewith. The lower end of hub **66** is closed by an end plate **72** which engages the inner race of a lower roller bearing **74**.

A lower non-rotative sleeve member **76** has a lower annular face against which the outer race of bearing **74** seats and spaced outer and inner cylindrical sleeve sections **78** and **80** which form an annular recess **82** therebetween. The inner surface of section **80** is spaced from hub **70**. An upper non-rotative sleeve member **84** has an annular face against which the outer race of upper roller bearing **86** seats, the inner race of which seats against outer disc hub **70**. Member **84** has a cylindrical sleeve section **88** which closely fits and slides within annular recess **82**. O-ring **90** provides a seal between the sliding surfaces of sleeve sections **78** and **88** and O-ring **92** provides a seal between the sliding surfaces of sleeve sections **80** and **88**.

A fitting **94** is provided on sleeve member **76** and has a fluid passageway **96** communicating with the base **83** of recess **82**. A rigid pipe section **98** threads into fitting **94** and is movable vertically within a slot **100** of a guide bracket **102** fixed to the platform **56**. Pipe **98** is connected via flexible tube **99** to a suit able fluid source. The effective diameter of pulley **60** is adjusted by introducing into or withdrawing fluid from recess **82** which acts between base **83** and end face **89** of sleeve section **88** to move cones **64** and **68** relative to each other. The minimum effective diameter of the pulley is set by an adjusting screw **104** which threads through the flange **79** of sleeve section **78** and abuts against flange **85** of sleeve member **84**. The maximum effective diameter is established by a screw rod **106** which threads into flange **85** of sleeve member **84** and freely passes through an opening in flange **79**. Rotation of sleeve members **76** and **84** is prevented by conduit **98** fixed against rotation within slot **100** and guide **102**.

Pulley **60** through a belt **108** drives a second variable diameter pulley **110** which is fixed against rotation on the upper end of a jackshaft **112** mounted parallel to motor shaft **62** and upper spindle **48**. Jackshaft **112** rotates within a bearing housing **114** that is supported from bracket assembly **116** fixed to the top of drill head casing **43**. Puley **110** is a conventional follower type in which the relatively movable pulley cones are spring biased by a spring assembly in hub casing **118** to a maximum effective diameter position shown in full in FIG. **1**.

With no fluid in recess **82** of pulley **60**, the spring force in pulley **110** will position pulley **110** in its maximum diameter position and, through the tension in belt **108**, pulley **60** in its minimum diameter position. As fluid is introduced into recess **82**, the effective diameter of pulley **60** is increased and the diameter of pulley **110** is decreased against the spring force in hub assembly **118**.

An electrical tachometer **119** mounted on bracket assembly **116** is belt connected to shaft **112** and has a readout device located on the central control panel **41** to indicate the speed at which shaft **112** is being driven.

A large diameter pulley **120** includes a disc **122** connected to a central hub **124** which is rotatable on the lower output end of shaft **112** through the roller bearing assembly **126** vertically positioned on the shaft between a threaded nut **128** and collar **130**.

A smaller diameter timing pulley **132** includes a hub portion **134** which is keyed on shaft **112** against rotation but is slidable thereon to permit a limited amount of axial movement along the shaft. The upper edge of hub **134** is formed with a plurality of clutch teeth **136** which cooperate with a plurality of mating clutch teeth formed on the periphery of hub **124**, the mating teeth forming a clutch **138** which couples pulleys **122** and **132** for rotation together when pulley **132** is shifted upwardly as shown in the right pulley position in FIG. **8**A.

The shifting mechanism for pulley **132** includes an annular ring cam member **140** connected to the bottom of hub **134**, with ring **140** having radially extending flanges **143** formed with a plurality of dimples **144** at predetermined radial positions which rest on support blocks **146**. Blocks **146** also have dimples **148** at selected radial positions so that, when ring **140** is rotated through a predetermined angular distance from the position shown in the left hand section of FIG. **1** in which dimples **144** rest on the upper surface of blocks **146** to a position shown at the righthand section of FIG. **1** in which lower dimples **144** ride up on dimples **148**, pulley **132** will be raised to engage clutch **138** and thereby cause pulley **120** to be rotated with pulley **132** and shaft **112**. A pair of retainer dogs **150** is mounted on blocks **146** to suitably guide and retain ring **140** in its selected clutch engaged or disengaged position.

The upper hollow spindle **48** is rotatably supported in housing **46** by lower high-speed bearing assembly **152**. A fluid conduit fitting **153** is retained in the upper end of spindle **48** by a threaded lock nut **154** and includes an outer shaft section **155** which is rotatably supported by upper high-speed, bearing assembly **156** mounted on a bracket **158** that is fixed to the top of housing **46**.

A small diameter pulley **160** and toothed clutch plate **162** are keyed on spindle **48**, with pulley **160** being driven from pulley **122** by a polyflex V-belt **164**, the tension of which is adjusted by an idler pulley **165** supported from bracket **158**. A sleeve **166** is rotatably mounted on spindle **48** by spaced bearings **168** and **170**, the inner races of which are separated by a spacer collar **172**. A ring **172** is positioned between the inner race of bearing **170** and an outer annular shoulder **176** on spindle **48**. A nut **178** threads onto spindle **48** against pulley **160** and, along with shoulder **176**, retains the elements in operative position.

A larger diameter timing pulley **180** is driven from pulley **132** by a timing belt **181** and includes a hub **182** which is splined on the upper portions of sleeve **166** so as to be axially slidable on the sleeve but rotatable on spindle **48.** Hub **182** is formed with a plurality of clutch teeth **184** which engages clutch plate **162** when pulley **180** is shifted axially, with teeth **184** and plate **162** forming a clutch **185** by which a drive connection may be established between pulley **180** and spindle **48.**

An annular gear shifting cam plate **186** rotatably slidably engages the lower portion of hub **182** and includes a plurality of upper and lower dimples **188,** the latter of which rest on the upper face of a plate **190** when clutch **185** is disengaged. Plate **190** also has a plurality of dimples **192** formed on its upper surface at predetermined angular positions so that when plate **186** is rotated to a selected position lower dimples **188** ride up on dimples **192** to raise pulley **180** and engage clutch **185.** A plurality of guide and retaining ears **193** engage the upper dimples **188** and retain gear plate **186** in a set position. As shown in FIG. **1,** a shift rod **194** extends between and connects plate **186** and ring **140** for rotation together. Gear plate **186** has a forward gear segment **195** which is moved by a pinion **196** fixed on the upper end of a shaft **198** which is part of a backgear assembly **204** and is rotatably mounted in gear housing **46.**

Pinion **196** may be rotated to position gear plate **186** and ring **140** in one of three possible set positions, each representing a specific speed range. The gear plate **186** and ring **140** are so designed that clutches **138** and **185** cannot be both engaged at the same time, i.e., the angular position of dimples **144** of plate **140** is different than that of dimples **188** on gear plate **186** so that pulleys **132** and **180** cannot be raised together.

As is evident from the description thus far, in one set position representing the high speed range, clutch **138** will be engaged and clutch **185** disengaged and spindle **48** is driven from shaft **112** through pulley **132,** clutch **138,** pulley **120,** belt **164,** and pulley **160.** Pulley **180** and sleeve **166** will simply freely rotate on the spindle.

In a second set position representing an intermediate speed range, only clutch **185** will be engaged and spindle **48** is driven from shaft **112** through pulley **132,** timing belt **181,** pulley **180,** and clutch plate **162.** Pulley **120** will be freely rotating on shaft **112.**

In a third set position representing a low speed range, neither clutch **138** nor clutch **185** will be engaged and a low speed drive for spindle **48** is established through a first smaller spur gear **200** formed on the lower end of sleeve **166,** a second larger spur gear **202** keyed on spindle **48,** and a back gear assembly **204** by which gears **200** and **202** are drivingly connected. Gear assembly **204** includes the shaft **198,** an eccentric hub portion **206** formed on the shaft, and upper and lower gears **208** and **210** which are keyed on a sleeve **212** that is rotatably mounted on hub **206.**

The lower end of shaft **198** extends below housing **46** and may be rotated by a suitable hand lever **198a** which has a pin **198b** that fits in suitable recesses in housing **46** to lock shaft **198** and pinion **196** in a set position. To operate spindle **48** within the low speed range, the back gear assembly **204** will be positioned as shown in FIG. **8A** with gear **208** engaging gear **200** and gear **210** engaging gear **202.** Both clutches **138** and **185** will be disengaged and spindle **48** will be driven from shaft **112**

through pulleys **132** and **180,** sleeve **166,** and gears **200, 208, 210** and **202.**

In a prototype drill press constructed according to the overall description set forth in original application Ser. No. 27,786, the various components of the drive system were designed so that the overall speed range was 125 to 10,500 rpm, with the low speed range being 125 to 625 rpm, the intermediate range being 550 to 2,700 rpm, and the high speed range being 2,100 to 10,500 rpm. Motor **54** was a 7 ½ hp, 1,725 rpm, 220 volt three phase motor and variable diameter pulleys **60** and **110** were capable of providing a speed ratio of 5:1 within each of the ranges and of varying the speed of jackshaft **112** from 600 to 3,000 rpm.

For a more complete description of the overall drive system as it is incorporated in a drill press, reference may be made to original application Ser. No. 27,786.

While the variable diameter pulley **60,** which constitutes the subject matter of this application, has been described for particular utility in the drive system for a drill press, it is to be understood that the pulley has a general utility and may be used in any suitable drive system.

The invention may be embodied in other specific forms without departing from the spirit or essential characteristics thereof. The present embodiment is therefore to be considered in all respects as illustrative and not restrictive, the scope of the invention being indicated by the appended claims rather than by the foregoing description, and all changes which come within the meaning and range of equivalency of the claims are therefore intended to be embraced therein.

What is claimed and desired to be secured by letters Patent is:

1. A hydraulically operated variable diameter pulley comprising a first cone having an elongated hub adapted to be slidably mounted on a shaft, a second cone having a hub slidably mounted on said first cone hub, an axially expansible, non-rotatable sleeve assembly surrounding said hubs and including a first sleeve and a first bearing assembly connecting said first sleeve and said first cone hub for axial movement together but permitting relative rotation therebetween, said first sleeve having spaced inner and outer annular sleeve sections defining an annular recess therebetween, a second sleeve having an annular sleeve section which slidingly fits within said recess, a second bearing assembly connecting said second sleeve and said second cone hub for axial movement together but permitting relative rotation therebetween, means for conducting pressurized fluid to and from said recess to produce relative axial movement between said first and second sleeves and thereby change the effective diameter defined by said cones, and sealing means between the sliding surfaces of said sleeve sections to confine said fluid within said recess.

2. A hydraulically operated, variable diameter pulley as defined in claim **1,** said sleeve assembly including means provided between said first and second sleeves for adjusting the maximum and minimum diameters formed by said cones.

* * * * *

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,762,231 Dated October 2, 1973

Inventor(s) David D. Pettigrew

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 3, line 63, delete "Puley" and substitute "Pulley".

Column 4, line 31, delete "143" and substitute "142".

Signed and sealed this 27th day of August 1974.

(SEAL)
Attest:

McCOY M. GIBSON, JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents