

[54] **DISC TYPE REFINER WITH AUTOMATIC PLATE SPACING CONTROL**

[75] Inventors: William S. Holmes, South Williamsport; Chester Donald Fisher; Clifford D. Ihrig, both of Muncy, all of Pa.

[73] Assignee: Sprout, Waldron & Company, Inc., Muncy, Pa.

[22] Filed: Dec. 14, 1973

[21] Appl. No.: 424,688

[52] U.S. Cl. 241/34, 241/37

[51] Int. Cl. B02c 7/14

[58] Field of Search 241/33, 34, 35, 36, 37, 241/245, 246, 259.1, 259.2, 259.3

[56] **References Cited**

UNITED STATES PATENTS

3,077,007	2/1963	Coghill.....	241/34 X
3,654,075	4/1972	Keyes et al.	241/37 X
3,754,714	8/1973	Reinhall.....	241/34

Primary Examiner—Granville Y. Custer, Jr.
Attorney, Agent, or Firm—Howson and Howson

[57] **ABSTRACT**

A disc type refiner including a control mechanism for automatically adjusting the refiner plate spacing in response to variations in the rate of feed of input material. The material feed rate is gauged by measuring the power input to the material feed conveyors. An electric control circuit monitors the power requirements of the feed conveyors and automatically adjusts the refiner plate spacing by control of the hydraulic plate-positioning cylinders to insure a constant work input to the material regardless of fluctuations in material flow. In the preferred form of the invention, the refiner comprises an axially fixed rotor disposed between opposed axially adjustable non-rotating heads. The heads are axially positioned by hydraulic cylinders controlled so as to maintain an equal working force on each side of the rotor.

15 Claims, 7 Drawing Figures

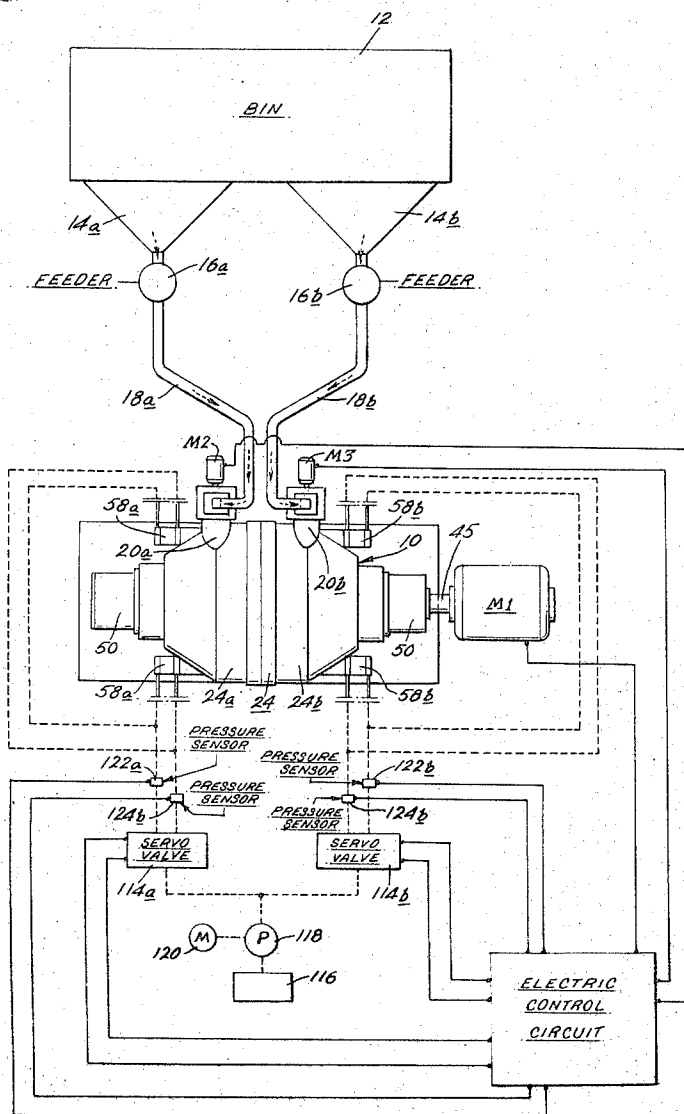


FIG. 1.

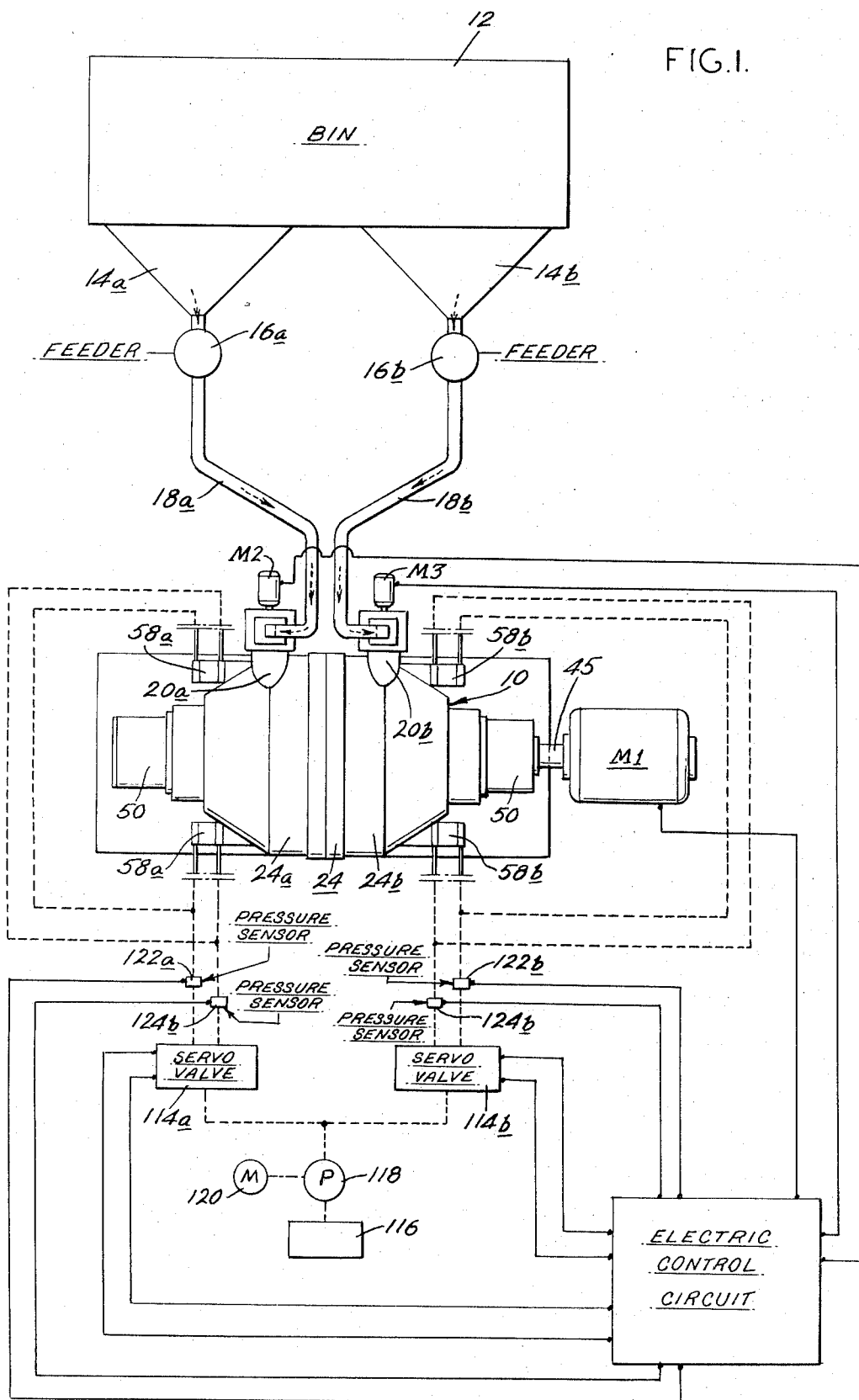


FIG. 2.

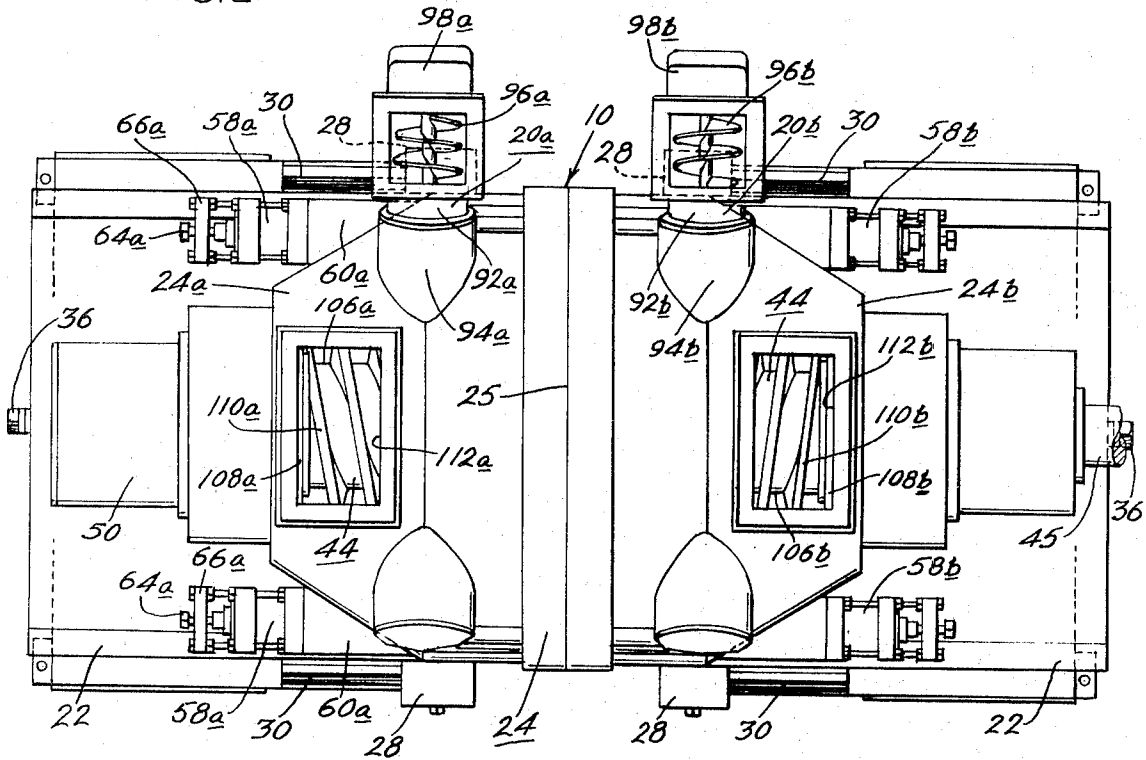


FIG. 3.

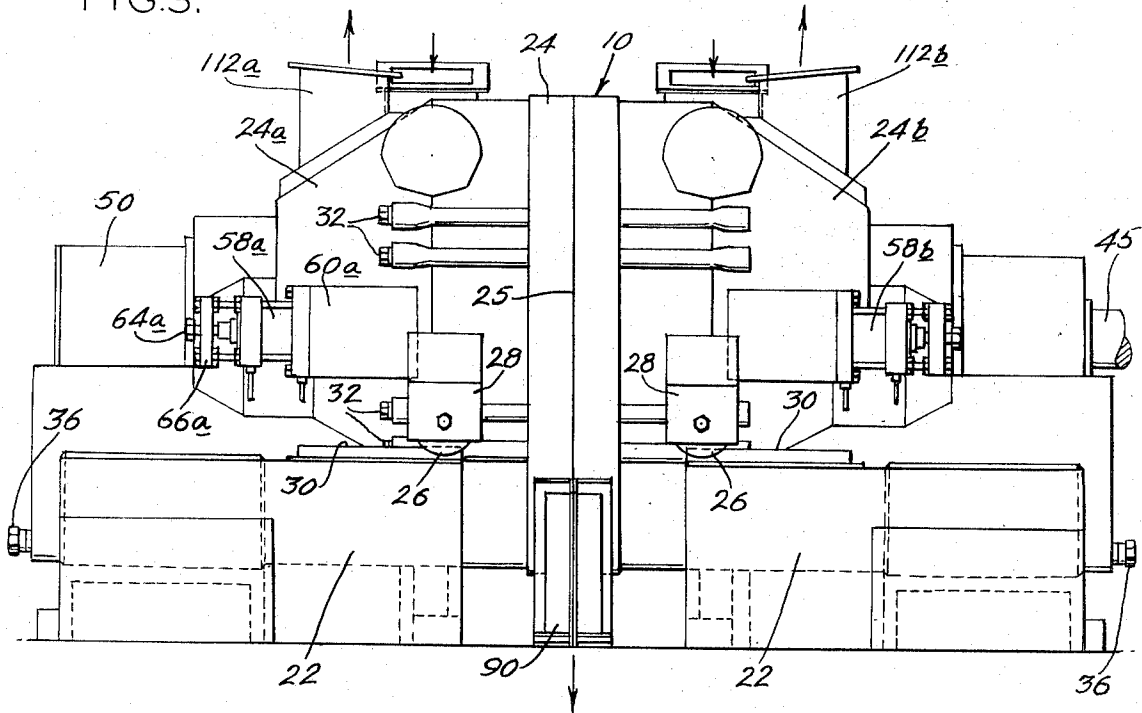


FIG. 4.

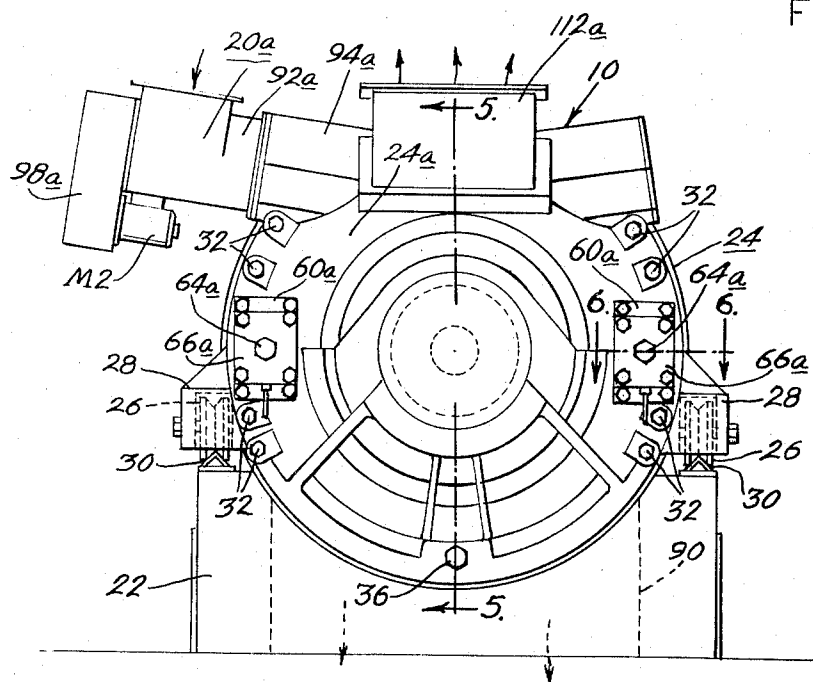
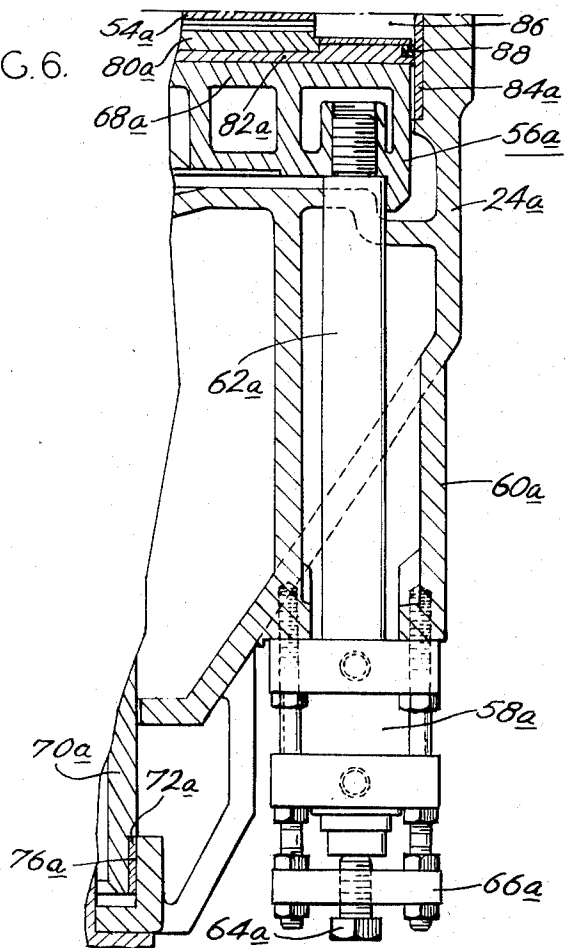


FIG. 6.



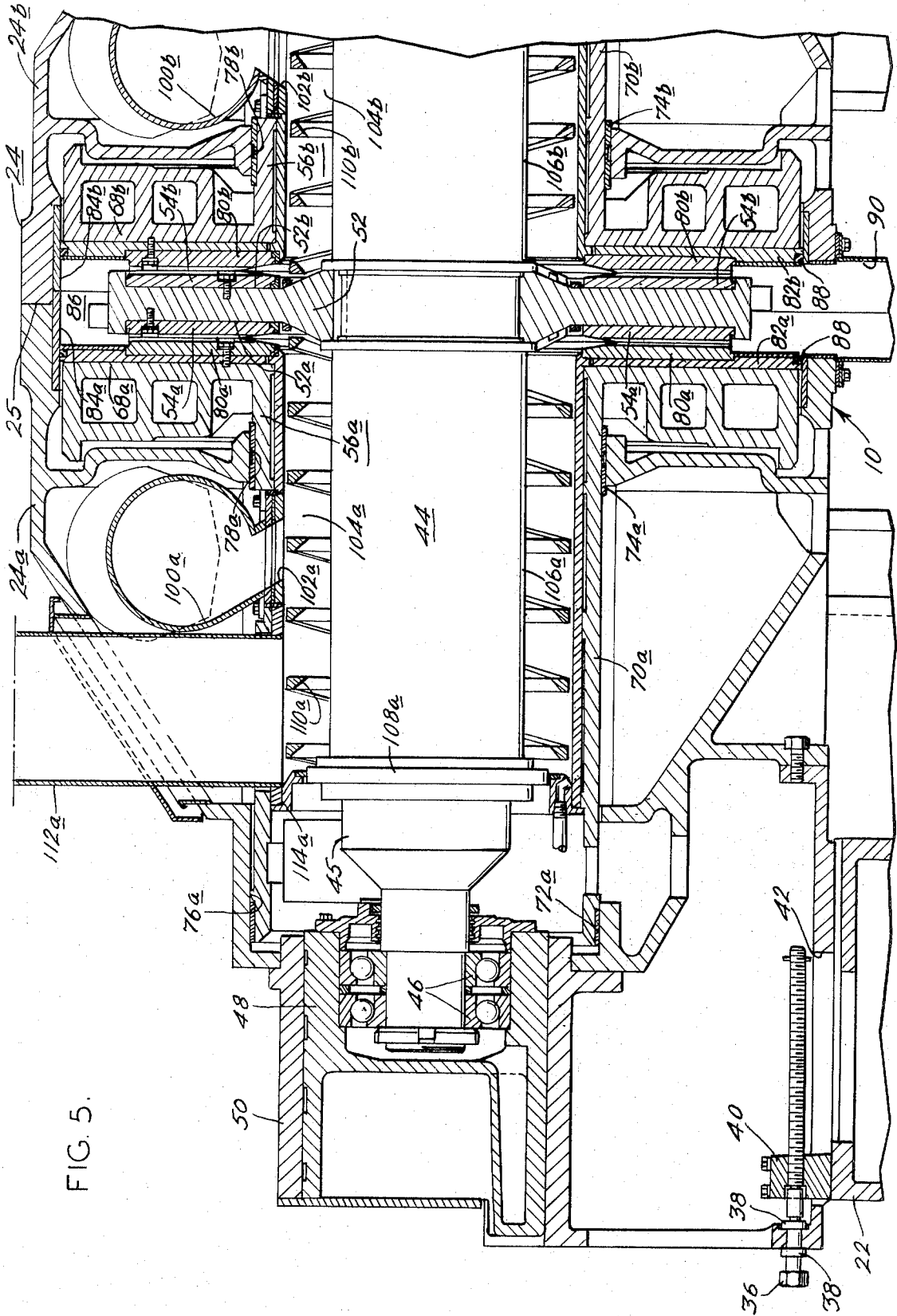
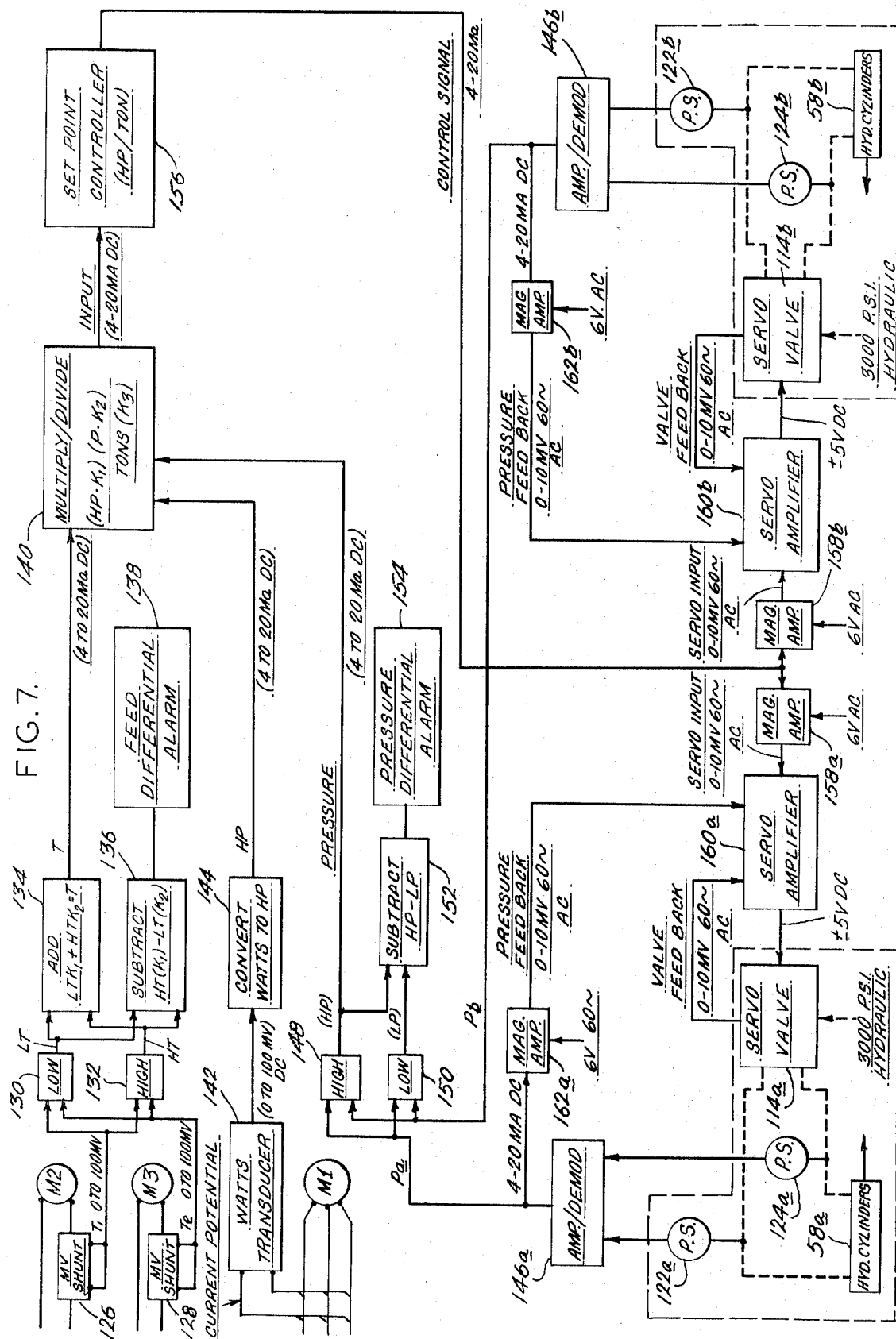


FIG. 7.



DISC TYPE REFINER WITH AUTOMATIC PLATE SPACING CONTROL

The present invention relates generally to disc type attrition mills and more particularly to a disc type refiner having load sensing control means for automatically regulating the refiner plate spacing in accordance with the rate of material input.

Disc type refiners are widely used in paper pulp processing and are characterized by a rotor supporting a concentric disc. A set of refining plates is mounted on one or both axial faces of the disc. The refining plates are available in a variety of patterns, and the plate faces typically are configured in a ribbed or toothed pattern. A stationary set of refining plates is disposed in juxtaposed relation to the plates of the rotor, and the material to be refined is introduced near the rotor axis between the opposing plate surfaces. The relative rotation of the plates centrifugally advances the material radially across the plate surfaces and the relatively close spacing of the plates breaks down the material fibers.

The refining effect produced by a disc type refiner is dependent on a number of factors, including the spacing of the plates, the rate of flow of material between the plates, the speed of rotation of the disc, the refining plate pattern, and the consistency of the stock. Since a uniform work input to the stock is essential to a high quality paper product, it is important that the variations in stock flow through the refiner be minimized to provide as constant a work input to the material as possible. Despite efforts to maintain a constant stock feed rate, fluctuations in the feed rate are for various reasons inevitable and it has thus heretofore been necessary to periodically manually adjust the plate spacing of the refiner after an examination of samples of refined materials.

Not only does this method fail to correct the plate spacing until after a change in work input to the stock has taken place, but in addition it requires the constant sampling of the refined material and an appropriate readjustment of the machine. Although such a procedure has been commercially accepted for want of a better alternative, it will be evident that a device which will automatically adjust the plate spacing in accordance with the material feed rate and which will accurately time the plate spacing adjustments to coincide with the fluctuations in feed rate will not only provide a superior product in terms of uniformity, but will in addition render substantial savings in manpower.

The present invention in addition to correlating the plate spacing with the material feed rate, also serves to prevent clashing of the opposed plates in the event that the feed rate should for some reason fall below that required to keep the plates apart. Devices have in the past been developed to open the plates and stop the machine in the event that the infeed rate falls below a certain value. Such a condition has been detected by monitoring the work input into the rotor required to maintain a certain rotation rate, or by monitoring the flow rate of stock into the refiner. The latter method has been carried out by several techniques including weight sensitive indicators and the gauging of the power input to a material feed conveyor as in the present system.

Although the present invention could be suitably employed with various types of disc refiners, it has primarily been developed for use with so-called "twin disc"

refiners for processing high consistency (low water content) stock. In a conventional twin disc refiner, such as that shown in U.S. Pat. No. 3,276,701 issued on Oct. 4, 1966, the rotor disc carries refining plates on each face thereof which cooperate with corresponding sets of stationary plates. The rotor is free floating between the stationary plates and it is accordingly necessary to axially adjust only one of the stationary plate supporting heads to vary the plate spacing and hence the work input into the processed material. The largest refiners of this type have had about a 2,000 horsepower capacity.

It is now proposed to build twin disc refiners of a size sufficient to handle 8,000-10,000 horsepower motors. It can be appreciated that with refiner motors of this size, and the consequently larger capacity of the refiners, the need for an effective automatic plate spacing control is even more important than with smaller units since even a small delay in adjusting the plate spacing for a change in stock feed would result in a relatively large quantity of non-uniform stock output. Similarly, the danger of plate clashing in refiners with such large motors would be so destructive that an extremely reliable safeguard must be provided to prevent its occurrence.

Since it is not believed that a freely floating rotor could be engineered within acceptable cost limits for such high horsepower applications, a preferred refiner for use in combination with the present plate spacing control, comprises a twin disc refiner having an axially fixed rotor and axially adjustable non-rotating heads supporting refiner plates in juxtaposed relation to the disc plates. The non-rotating heads are axially adjusted by hydraulic cylinders which permit the heads to float axially in response to the pressures generated between the juxtaposed refining plates, thereby insuring a constant work input to the material passing on both sides of the disc. The stock to be refined is introduced along the rotor shaft at each side of the disc by a ribbon screw and is fed to the ribbon screws by a pair of screw-type stock infeed conveyors driven by electric motors.

In the present control system, the refiner plate spacing is automatically adjusted in accordance with the stock feed rate by an electrical control circuit which monitors the power requirements of the input conveyors as a measure of the feed rate and correlates the plate spacing by control of the hydraulic cylinders of the non-rotating heads. Since the flow rate is monitored upstream of the refining plates, the plate adjustment can be made to coincide with the arrival of the changed material flow at the plates. There is accordingly no correction lag, and the uniformity of the work input to the stock can be assured.

It is accordingly a first object of the present invention to provide an automatic plate spacing control for a disc type refiner which will effectively insure a uniform work input to processed stock regardless of fluctuations in the stock feed rate.

A further object of the invention is to provide a refiner control as described which adjusts the plate spacing coincidentally with fluctuations in the stock feed rate to maintain a continuously uniform work input to the stock.

An additional object of the invention is to provide the combination of a disc type refiner and automatic control system which is particularly suited for large capacity installations.

Still another object of the invention is to provide a refiner control system as described which will automatically prevent plate clashing in the event of an interruption in the stock feed.

Additional objects and advantages of the invention will be more readily apparent from the following detailed description of an embodiment thereof when taken together with the accompanying drawings in which:

FIG. 1 is a schematic view illustrating a preferred embodiment of a disc type refiner having automatic plate spacing control in accordance with the present invention;

FIG. 2 is an enlarged plan view of the disc type refiner shown schematically in FIG. 1;

FIG. 3 is a side view of the refiner shown in FIGS. 1 and 2;

FIG. 4 is an end view of the refiner as seen from the left in FIGS. 1-3;

FIG. 5 is an enlarged partial sectional view of the refiner taken along lines 5-5 of FIG. 4 and showing the interior refiner details;

FIG. 6 is an enlarged partial sectional view taken along lines 6-6 of FIG. 4 showing details of one of the nonrotating head hydraulic positioning cylinders; and

FIG. 7 is a schematic view of the electric control circuit for adjusting the refiner plate spacing in response to variations in the stock infeed rate.

Referring to the drawings, FIG. 1 illustrates schematically a preferred embodiment of the present invention including a disc type refiner generally designated 10 and the hydraulic and electric circuits pertaining to the refiner automatic plate spacing control. The stock to be refined, which might typically comprise high consistency (low water content) wood chips, is stored in a bin 12 and drops from bin hoppers 14a and 14b into feeders 16a and 16b respectively. From the feeders, the stock passes through conduits 18a and 18b into infeed conveyors 20a and 20b respectively of the refiner. For convenience in illustrating the refiner circuits, the refiner is schematically shown in FIG. 1 in a top plan view although the bin, feeders and conduits are shown in side elevation.

The details of the preferred form of refiner for use in combination with the present plate spacing control system are shown in FIGS. 2-6. A refiner base 22 supports a casing 24 formed by casing sections 24a and 24b which are separable along vertical interface 25. Grooved wheels 26 journaled in members 28 extending from each side of the casing sections movably support the sections on parallel rails 30 on the base 22. The casing sections are secured together during operation of the refiner by a plurality of bolts 32 spaced along the sides of the sections perpendicular to and spanning the interface 25 therebetween. The casing sections are readily separable to change the refiner plates or for other servicing of the unit by opening the bolts 32 and by rotation of a positioning screw 36 at the bottom of the outer end of each casing section. Each screw 36 is rotationally secured to its casing section by the collars 38 thereon and threadably engages a block 40 on the base 22 which extends upwardly through a slot 42 in the casing section. The rotation of each screw 36 will accordingly move its casing section on the tracks 30 along the longitudinal axis of the refiner.

A rotor 44 passes centrally through the casing and includes a rotor shaft 45 which is supported at the outer end of each casing section. As shown in FIG. 5, one end of the rotor shaft is journaled in casing section 24a by bearings 46 which are mounted in a bearing support member 48 and secured therewithin by a clamping ring 50. Member 48 is slidably disposed within its casing section to permit axial displacement of the section without changing the position of the rotor with respect to the base. The rotor shaft extends through the casing section 24b as shown in FIGS. 2 and 3 and is journaled within this casing section in a manner similar to that shown in FIG. 5. Although bearings 46 are combined radial and thrust bearings, the bearing assembly in section 24b should include only radial bearings to allow for axial thermal expansion of the rotor. Refiner motor M1, as shown in FIG. 1, is connected to the extending rotor shaft, and for large size refiners, may be as large as 10,000 horsepower.

A radially extending rotor disc 52 is centrally mounted on the rotor shaft within the casing and is keyed to the shaft for rotation therewith. Sets of refining plates 54a and 54b of conventional construction are bolted to the opposite faces 52a and 52b of the disc 52. Disposed within the respective casing sections 24a and 24b and axially juxtaposed the faces 52a and 52b of the rotor disc are the nonrotating heads 56a and 56b which are axially slidable within the casing sections. As shown in FIG. 6, the non-rotating heads are each respectively connected to a pair of diametrically opposed hydraulic cylinder assemblies 58a and 58b which control the axial position of the heads.

The details on one of the hydraulic cylinders 58a are shown in FIG. 6. The cylinder 58a is bolted to a projecting portion 60a of the casing section and the cylinder piston rod 62a extends through the casing section into threaded attachment to the axially movable head 56a. The outward travel of the piston rod is controlled by the stop bolt 64a supported coaxially outwardly of the cylinder 58a by a supporting member 66a secured thereto. The range of allowable axial movement of the non-rotating head is small and in practice, the head adjustment required during operation is only a small fraction of an inch.

As shown in FIG. 5, the non-rotating heads 56a and 56b respectively include radially extending plate support portions 68a and 68b, and cylindrical portions 70a and 70b coaxial with the longitudinal axis of the refiner. Cylindrical outer bearing elements, only one of which is shown at 72a, and similar inner bearing elements 74a and 74b provided with seal rings (not shown) respectively support the movable heads in axially slidable sealed relation to the respective casing section surfaces, only one of which is shown at 76a, and 78a and 78b. Refiner plates 80a and 80b are bolted to the radial faces of the non-rotating heads in respectively juxtaposed relation to the plates 54a and 54b of the rotor 52. Ring shaped radially disposed elements 82a and 82b extending from between the head portions 68 and the plates 80 seal the non-rotating heads against cylindrical elements 84a and 84b secured to the casing sections and form an annular chamber 86 between the non-rotating heads and circumferentially of the rotor into which the refined products are discharged. Seal rings 88 between the members 82a and 82b and the plates 84 permit sliding movement of the heads with respect to the casing while maintaining the sealed condi-

tion of the annular chamber 86. A discharge chute 90 at the bottom of the refiner communicates with the annular chamber 86 for removal of the refined products from the machine.

The stock infeed to the refiner as indicated above is controlled by the infeed conveyors 20a and 20b which as shown in FIGS. 2 and 4 comprise respectively infeed conveyor housings 92a and 92b which are attached to the respective hollow casing section extensions 94a and 94b. Conveyor screws 96a and 96b are respectively mounted in the infeed conveyor housings 92a and 92b and driven by feed motors M2 and M3 which are connected to the feed screws by variable speed drive units 98a and 98b. The infeed stock passes from the infeed conveyors through the internal conduits 100a and 100b and exists through inclined openings 102a and 102b in the bottom thereof into the annular stock feed passages 104a and 104b coaxial with the rotor shaft. Sleeves 106a and 106b mounted on the rotor shaft abutting each side of the rotor disc and secured thereagainst by nuts 108a and 108b support ribbon type screw conveyor elements 110a and 110b within the chambers 104a and 104b respectively. The ribbon conveyor screws 110a and 110b have opposite pitch angles selected so that rotation of the rotor will advance infeed stock from the conduits 100a and 100b toward the rotor disc and into the throat regions of the refining plates. Similarly, the infeed conveyor screws 96a and 96b are of opposite pitch angles, the screw 96a being rotated counterclockwise and the screw 96b being rotated clockwise as viewed in FIG. 5 so that the feed stock will be thrown by the screws toward the rotor disc through the inclined openings 102a and 102b.

The ribbon screws 110a and 110b are utilized to permit steam generated during the refining of low consistency stock to escape along the surface of the tubes 106a and 106b into the steam discharge spouts 112a and 112b at the ends of the casing sections. A suitable seal assembly, only one of which is shown at 114a, is provided at each end of the ribbon screw conveyors 110a and 110b to prevent steam and stock particles from passing to the bearing region of the rotor.

The mechanical operation of the refiner should to a large extent be obvious from the above discussion of the refiner structure and will be summarized prior to considering the details of the electrical control circuit which provides the automatic plate spacing control. With the rotor driven at a predetermined speed by the refiner motor M1, the stock to be refined is fed from the bin 12 by feeders 16a and 16b into infeed conveyors 20a and 20b. The feed motors M2 and M3 drive the infeed conveyors at a predetermined rate and the power consumed by the feed motors will vary as a function of the weight of the stock passing through the conveyors at any given moment. As schematically indicated in FIG. 1, the power input to each of the infeed conveyors is monitored by the electric control circuit to gauge the input to each side of the refiner disc. Similarly, the power input to the refiner motor M1 is gauged to measure the work input to the stock for a given stock flow rate. Since the work done to the stock is commonly expressed in horsepower per ton, it can be appreciated that the work input can be readily determined by correlating the power required by the refiner main drive motor M1 and the infeed conveyors feed motors M2 and M3.

The stock passing from the infeed conveyors is advanced by the ribbon conveyors 110a and 110b into the throat of the refining plates on each side of the disc and is centrifugally passed between the opposed sets of refining plates 54a and 80a, and 54b and 80b. The close spacing of the plates, which may be only a few thousandths of an inch, causes a fibrillation of the stock and in the process generates a considerable amount of steam which escapes axially back through the ribbon conveyor and out through the steam discharge spouts 112a and 112b. The refined stock passes from the refiner through the exhaust conduit 90.

The power requirements of the motor M1 and hence the work input to the stock passing through the refiner will vary as a function of the spacing of the refiner plates and hence the control of the plate spacing, and the correlation of the change in plate spacing with a change in the stock flow rate is of critical importance to the production of a uniform product. Since the plate spacing is controlled by the movement of the non-rotating heads 56a and 56b by the hydraulic cylinders 58a and 58b, it is apparent that the coordination of the change in pressures to the hydraulic cylinders with the change in stock flow rate through the infeed conveyors must be accurate. This is automatically accomplished by the electric control circuit described below.

As shown in FIG. 1, the circuit includes servo valves 114a and 114b which respectively control the pressure inputs to the pairs of hydraulic cylinders 58a and 58b. The high pressure hydraulic fluid delivered to the servo valves is pumped from a reservoir 116 by a pump 118 driven by a motor 120. Pressure sensors 122a, 122b, 124a and 124b provide an indication to the electric control circuit of the hydraulic force exerted on the non-rotating heads by the hydraulic cylinders 58a and 58b.

Referring now to FIG. 7, the control circuit of the present invention is shown in some detail greater than appears in FIG. 1. More specifically, only the structure shown in the dashed boxes in the lower corners of FIG. 7 and motors M1, M2 and M3 appear in FIG. 1. The remainder of the system of FIG. 7 is represented in the block entitled "Electric Control Circuit" in the lower right hand corner of FIG. 1.

As discussed in connection with FIG. 1, the hydraulic cylinders 58a and 58b, respectively, provide pressure to move the fixed plates 80a and 80b in response to control by servo valves 114a and 114b, respectively. The controlling pressures within the hydraulic cylinder 58a and 58b are monitored by pressure sensors 122a and 124a, 122b and 124b respectively. The pressure sensors provide feedback information to the electrical control circuit, which may, when needed, cause readjustment of either or both servo valves 114a and 114b.

In operation of the circuit of FIG. 7, the current of the feed motors M2 and M3 providing the feed drive are monitored by standard d.c. millivolt shunts 126 and 128, which provide 0 to 100 millivolt output signals proportional to the power consumed, respectively, by d.c. feed motors M2 and M3. As previously explained, the weight of woodchips in the load sensing conveyors determines the amount of power consumed by their d.c. feed motors M2 and M3. The millivolt signal is proportional to the power consumed. Both feed motors M2 and M3, in turn, feed both lower signal selector 130 and high signal selector 132. These selectors respectively select the lower and higher level signals of the

two inputs and provide discrete signals of the lower signal from Low Signal Detector 130 and high signals from High Signal Detector 132. The low and high signals, now identified as such, are both fed into summing amplifiers 134 and 136, respectively, which apply constants to each signal and perform the addition or subtraction to produce signals which correspond to the sum and difference, respectively, of the signals applied to each of the two amplifiers. The difference output from amplifier 136 is fed through a feed differential alarm indicator 138 which may, for example, be a light, a buzzer or a bell, or both aural and visual means to attract the attention of the machine operator. In addition, the feed differential alarm indicator 138 may be used to shut down the machine automatically if the difference in loads of the two conveyor exceeds a predetermined dangerous condition. The subtract summing amplifier 136 produces an output, for example, between 4 and 20 milliamperes d.c. signal proportional to the difference in the tonage between the two conveyors.

The output of summing amplifier 134 is also a signal between 4 and 20 milliamperes d.c. which is proportional to the total tonage (T) on both conveyors. This signal is fed to suitable calculator means 140 for combining signals representative of tonage, horsepower of the main drive and fluid pressure in the hydraulic cylinders. This combination is done in accordance with the following mathematical expression: $(HP \cdot K_1) / (P \cdot K_2) / (T \cdot K_3)$, where HP is refiner motor horsepower, P is hydraulic fluid pressure, T is tons and K_1 , K_2 , and K_3 are predetermined constants expressing the desired relationship in terms peculiar to a given system. The HP signal is derived from the current and voltage of motor M1 through a watts transducer 142. Watts transducer 142 includes a wattmeter and means to translate the measured wattage into a d.c. voltage between 0 and 100 millivolts proportional to the wattage or power consumed by the motor M1. An electronic converter 144 performs the watts to horsepower conversion using the well known formula and derives a horsepower output signal of between 4 and 20 milliamperes d.c. to be fed as the HP signal to calculator 140.

The pressure (P) input to the calculator 140 is obtained from amplifier modulators 146a and 146b which, in turn, receive input signals from pressure sensors 122a and 124a or 122b and 124b, respectively. The outputs of these amplifier modulators are signals representative of pressure Pa and Pb, respectively, in the cylinders 58a and 58b. These output signals are fed to high voltage detector 148 and low voltage detector 150, respectively. The higher of the pressures Pa or Pb is taken to be the output pressure P applied to the calculator 140. The high and low pressure signals are combined by differential amplifier 152 to obtain a difference signal which, if it exceeds a predetermined, is sufficient to actuate a differential alarm 154 which may include a light and/or buzzer or other aural device. The difference output signal may also, or alternately, actuate means to shut down the machine should pressure differentials become too great.

The calculation accomplished by computer element 140 is an output of between 4 and 20 milliamperes d.c. which, in turn, feeds the setpoint controller 156. Setpoint controller produces an output signal representative of departure from a predetermined ratio of HP to T which constitutes the control signal of the system within a range of 4 to 20 milliamperes d.c. This signal

is fed to magnetic amplifiers 158a and 158b which, in turn, act upon the servo amplifiers 160a and 160b to produce output signals of ± 5 volts d.c. to operate the servo valves 114a and 114b respectively. Each servo amplifier is designed to respond to a 60 Hz 0 to 10 millivolt a.c. signal which the magnetic amplifiers 158a and 158b generate in conversion of the 4 to 20 milliamperes d.c. control signal. The servo valves have a built-in feedback to their associated servo amplifier, which feedback may range from a 0 to 10 millivolt 60 Hz a.c. signal. Servo valves 114a and 114b also receive a hydraulic supply of 3,000 pounds per square inch, as described in connection with FIG. 1. The hydraulic cylinders move in the direction determined by the setting of their servo valves.

The condition of the hydraulic cylinder determines the amount of power consumed by the main drive motor M1. The position of the hydraulic cylinder is a function of the pressure in its supply lines which, in turn, is sensed as previously indicated by pressure sensors 122a and 124a or 122b and 124b which, in turn, feed amplifier modulators 146a and 146b, respectively. The outputs from the amplifier modulators are also used as feedback to the servo amplifiers 160a and 160b, respectively, passing a 4 to 20 milliamperes d.c. signal through 6 volt 60 Hz a.c. magnetic amplifier 162a and 162b. The magnetic amplifier converts the d.c. signals to pressure feedback signals of from 0 to 10 millivolts 60 Hz a.c. which are applied as a feedback input to the servo amplifiers 160a and 160b, respectively.

It should be observed that the generation of control signals from the d.c. millivolts shunts 126 and 128 through the system to the servo valves 114a and 114b is so rapid as to be considered instantaneous. This is compared to the level of instantaneous output signal from signal converter 144 which lags behind the level of the output signal to the adjustable setpoint controller 156, due to the flexibility of the hydraulic system and the inertia of the adjustable portion of rotating heads 56a and 56b. By the nature of the conveyor flights the load generates a slight sine wave superimposed on the power consumed by motors M2 and M3. Thus, the control signals throughout the system will appear with a slight ripple that is proportional to this output. However, because of the great inertia of the rotating parts, the horsepower of motor M1 is unable to respond fast enough to the change in pressure on the cylinders 58a and 58b. This delay is taken advantage of by modifying the horsepower signal in the calculator unit 140 which performs the calculation previously described. The calculation then is effectively an error control or deviation from a set value of a ratio of horsepower to tons through-put of material by proper choice of constants K_1 , K_2 and K_3 and assuming that the factor $P \cdot K_2$ will be small. It will be seen that as tonage signal T is varied in response to the output of motors M2 and M3 through shunts 126 and 128, the signal selectors and summing amplifier 134, the modification due to $P \cdot K_3$ will be quite large as tonage T is rising and falling as a response to the ripple output because the calculation occurs before P has either increased or decreased. This factor will cause a slight overreaction of the signals applied to the servo amplifiers. Moreover, if the load on motors M2 and M3 is actually increasing due to a definite increase in which the overreaction will be increased or exaggerated, thus bringing the horsepower

HP into proper relationship with T more rapidly than would otherwise occur. The time lag allows the system to combine and compensate for its inertia and it also results in a much improved sine wave output in the T signal from summing amplifier 134. The effects tend to balance out so that the motor M1 consumes a steady amount of power for a given condition of flow and provides a correct amount of power at the proper times to produce a superior grade of wood pulp.

Manifestly, changes in details of construction can be effected by those skilled in the art without departing from the spirit and scope of the present invention.

We claim:

1. A disc type refiner for the refining of paper pulp and the like comprising a base, a rotor mounted for rotation on said base, a motor for driving said rotor in rotation, a refining surface on said rotor, a non-rotating head on said base, a refining surface on said non-rotating head in juxtaposed relation to said rotor refining surface, means for axially adjusting the position of said non-rotating head to control the spacing of said juxtaposed refining surfaces, feed means for introducing materials to be refined between said refining surfaces, means for monitoring the flow rate of material through said feed means, and means for continuously adjusting the spacing of said refining surfaces in accordance with the flow rate of material through said feed means to provide a constant work input to the material despite fluctuations in the material flow rate.

2. The invention as claimed in claim 1 wherein said feed means includes a drive motor and wherein said means for monitoring the flow rate through said feed means comprises means for monitoring the power consumed by said feed means drive motor.

3. The invention as claimed in claim 1 wherein said feed means comprises a screw conveyor, and an electric motor for driving said screw conveyor, said means for monitoring the flow rate of material through said feed means comprising means for monitoring the power consumed by said electric motor.

4. The invention as claimed in claim 3 wherein said means for axially adjusting the position of said non-rotating head comprises a hydraulic cylinder, a pressurized fluid source connected with said hydraulic cylinder, and electrically controlled valve means for controlling the flow of fluid from said source to said hydraulic cylinder.

5. A disc type refiner for the refining of paper pulp and the like comprising a base, a casing on said base, a rotor passing through said casing and mounted for rotation therewithin, said rotor being axially fixed with respect to said casing, a motor operatively connected to said rotor for driving said rotor in rotation, said rotor comprising a rotor disc extending radially therefrom, refining plates mounted on each axial face of said rotor disc, a non-rotating head mounted within said casing adjacent each axial face of said rotor disc, refining plates on each of said non-rotating heads juxtaposed said rotor refining plates, means for axially adjusting the position of each of said non-rotating heads to control the spacing of said juxtaposed refining plates, feed means for introducing materials to be refined between said refining plates, means for monitoring the flow rate of material passing through said feed means, and means for continuously adjusting the spacing of said refining plates in accordance with the flow rate of material through said feed means to provide a constant work

input to the material despite fluctuations in the material flow rate.

6. A disc type refiner as claimed in claim 5 wherein said feed means for introducing materials between said refining plates comprises a screw conveyor adjacent each said nonrotating head, and an electric drive motor for each said screw conveyor, said means for monitoring the flow rate through said feed means comprising means for monitoring the power input to each said conveyor drive motor.

7. The invention as claimed in claim 6 wherein said means for axially adjusting the position of each of said nonrotating head comprises a hydraulic cylinder, a pressurized fluid source connected with said hydraulic cylinder, and electrically controlled valve means for controlling the flow of fluid from said source to said hydraulic cylinder.

8. The invention as claimed in claim 7 wherein said rotor includes a ribbon conveyor extending axially therealong at each side of said rotor disc for feeding material from said screw conveyors toward said disc and into the space between said refiner plates.

9. The invention as claimed in claim 7 wherein said rotor motor comprises an electric motor, means for monitoring the power input to said rotor motor, pressure sensing means associated with said hydraulic cylinders for sensing the cylinder pressure and hence the force generated by said cylinders, said means for continuously adjusting the spacing of said refiner plates comprising an electric control circuit connected with said pressure sensing means, said valve means, and with said means for monitoring the input power to said refiner motor and screw conveyor motors.

10. Control means for a refiner having opposed relatively rotatable refining plates, hydraulic means for adjusting the spacing of said plates, refiner motor means for driving said plates, feed means for feeding process material through said plates for refinement, and feed motor means for the feed means, comprising means for sensing power of the feed motor means and means for producing a signal T proportional thereto representative of tons of feed, means for sensing power of the refiner motor means and means for producing a signal HP proportional thereto representative of drive horsepower, and means sensing pressure P in the hydraulic means controlling spacing of the opposed plates and producing a signal P proportional thereto, means to combine said signals in accordance with the formula $(HP \cdot K_1) (P \cdot K_2) / (T \cdot K_3)$, where K_1 , K_2 and K_3 are constants dependent upon individual refiner performance, and produce an output signal representative thereof, a setpoint controller for comparison of said output signal with a predetermined signal level, and adjust means for adjustment of the hydraulic means responsive to the setpoint controller to make correction of spacing of the plates of the refiner by the hydraulic means upon deviation from the setpoint by a predetermined amount.

11. The control system of claim 10 in which the control of the hydraulic means includes a servo valve regulating flow to the hydraulic means and a servo amplifier controlling the valve in response to the control signal.

12. The control system of claim 11 wherein said hydraulic means comprises a hydraulic cylinder, means sensing the pressure on the opposite sides of said cylinder, a differential amplifier for receiving and comparing the signals from said latter pressure sensing means

11

12

to generate a signal proportional to any difference in pressures, said servo valve and said differential amplifier providing feedback to said servo amplifier.

13. The control system of claim 10 in which said refiner comprises two pairs of opposed plates each adjustable relative to one another, and a pair of feed means, one for each pair of plates, and each having its own drive means, wherein the signals from the respective feed motors are summed, the outputs of the respective hydraulic cylinders are compared and the higher pressure used as the pressure signal.

14. The control system of claim 13 in which signals from feed motors are subtracted by differential amplifier means and are compared with a standard and an alarm is activated when the differential exceeds a predetermined amount.

15. The control system of claim 13 in which the pressure signals are applied to a differential amplifier and a difference signal derived such that, when the difference signal exceeds a predetermined amount, alarm means is activated.

* * * * *

15

20

25

30

35

40

45

50

55

60

65