United States Patent [19]

Bogdanov et al.

[11] **3,802,803**

[45] **Apr. 9, 1974**

[54] **SUBMERSIBLE SCREW PUMP**

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[21] Appl. No.: **321,280**

Related U.S. Application Data

[63] Continuation of Ser. No. 188,729, Oct. 13, 1971, abandoned.

[52] **U.S. Cl.** **417/211.5,** 418/48

[51] **Int. Cl.** **F01c 5/04,** F04c 11/00, F04c 17/12

[58] **Field of Search** 417/200, 435, 211.5; 418/48

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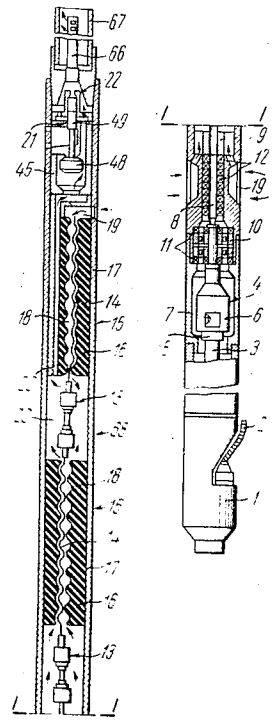
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Primary Examiner—Carlton R. Croyle
Assistant Examiner—Richard Sher
Attorney, Agent, or Firm—Holman & Stern

[57] **ABSTRACT**

Submersible screw pumps for transferring liquids, substantially viscous liquid containing gas under a high pressure, for example in the process of oil pumpage from deep oil wells or when pumping oil products from reservoirs in which there is provided a drive with a seal and a starting device, coaxially secured stators formed by rigid casings, housing coaxially disposed elastic bushings having inner threads and protected by a safety valve; twin screw rotors of which one has a right-hand thread and the other a left-hand thread and which are connected to the drive shaft and to each other through a flexible coupling with, each screw rotor being eccentrically disposed in the corresponding stator having an extra thread start as compared with the rotor, so that between the threads of the rotor and the stator there are defined closed spaces disposed along the axis of the stator from the intake chamber of each rotor to the common delivery chamber; the length of the elastic bushing of each stator being at least 20 times the cross sectional diameter of the corresponding screw rotor; the oblong elastic bushings being protected from dry friction and excess pressure owing to the fact that the delivery chamber is automatically connected to and disconnected from the intake chambers of the screw rotors by means of a gate and a pilot valve, disposed coaxially in the safety valve, depending on the output of the screw rotors and the pressure within the delivery chamber, in which case the centers of the extreme closed spaces of each rotor adjoining the delivery chamber are located on a common line extending through the axis of the stators and the axis of the rigid member connecting the two heads of universal joints serving as a flexible coupling between the rotors and transmitting torque from one rotor to the other.

16 Claims, 16 Drawing Figures



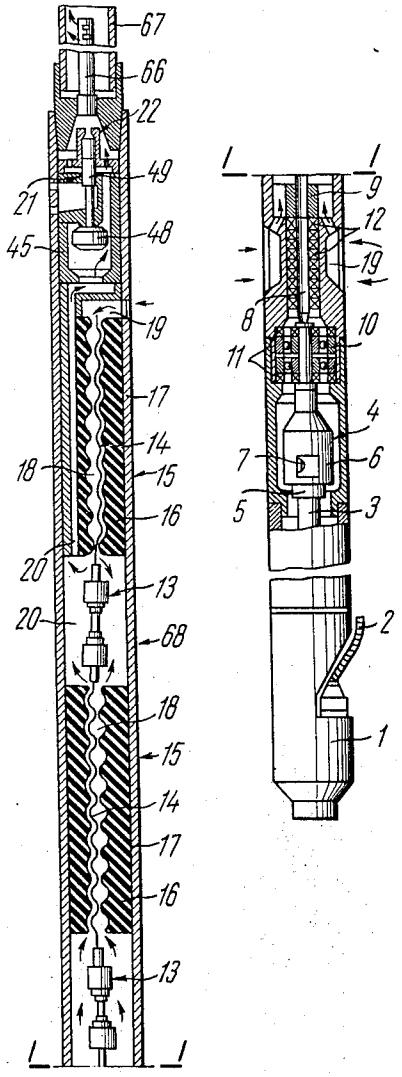

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FIG.1

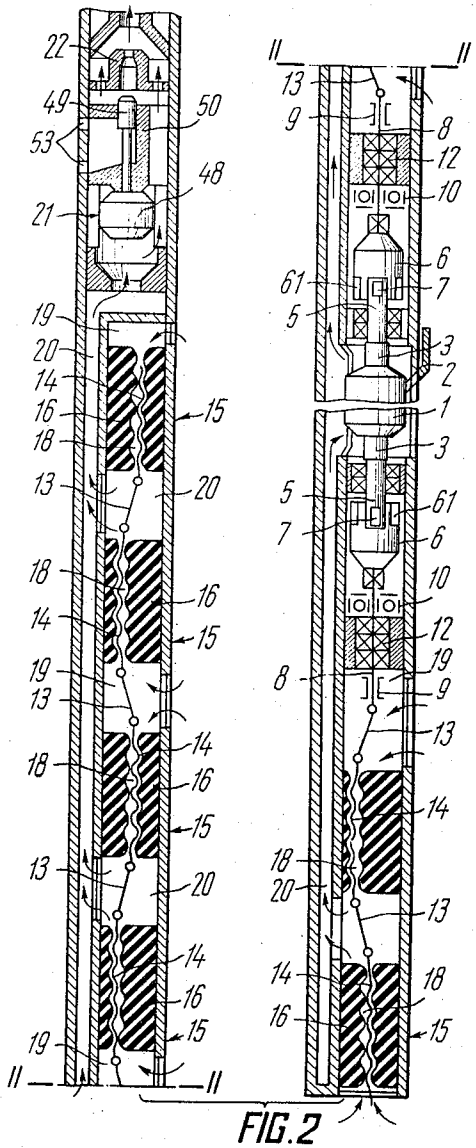

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FIG. 2

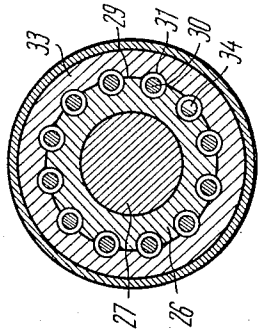


FIG. 8

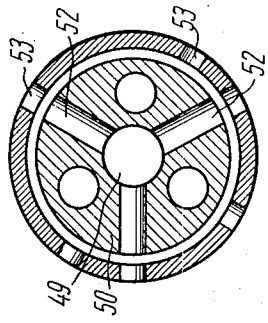


FIG. 14

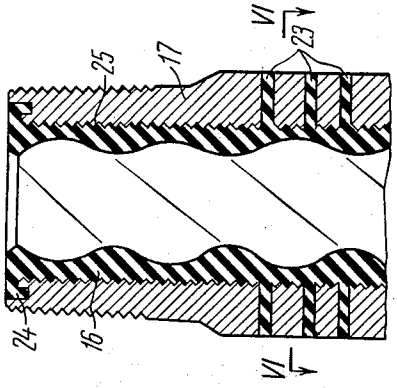


FIG. 5

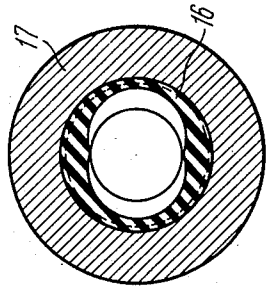


FIG. 6

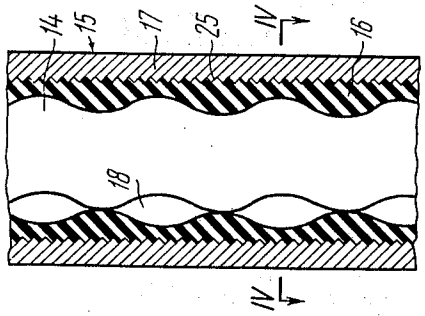


FIG. 3

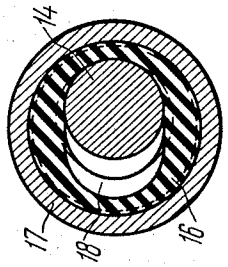


FIG. 4

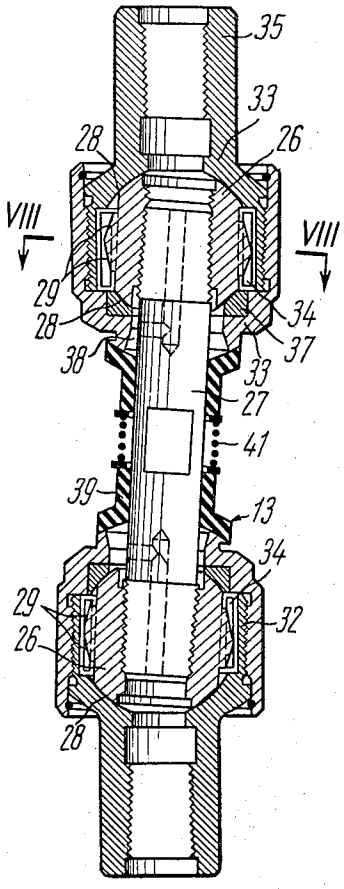


FIG. 7

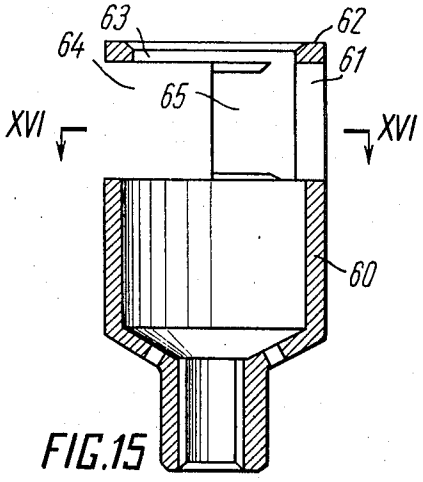


FIG. 15

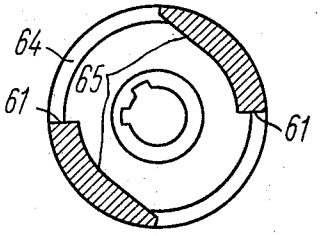


FIG. 16

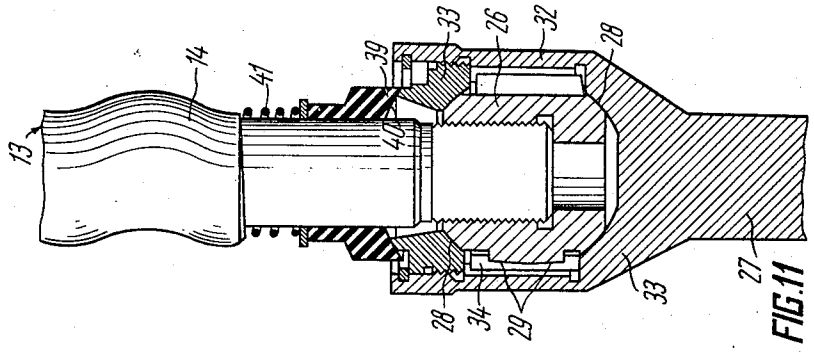


FIG.11

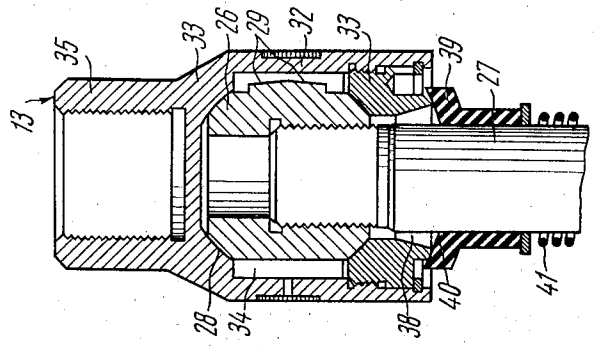


FIG.10

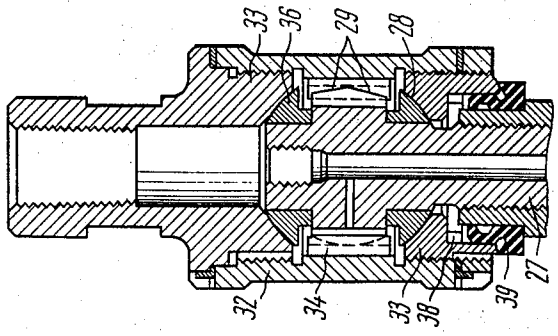


FIG.9

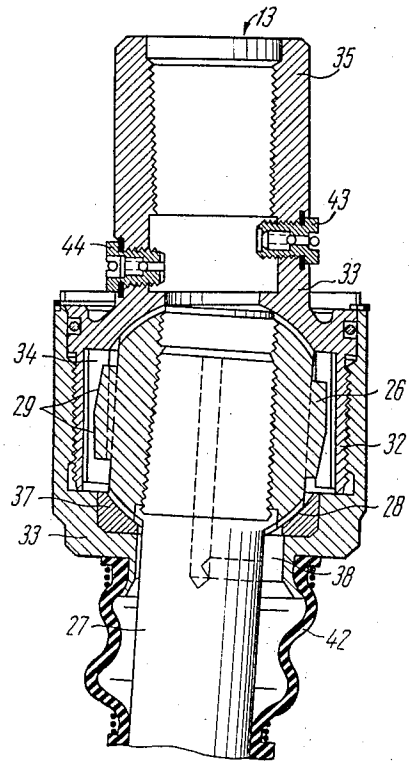

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FIG.12

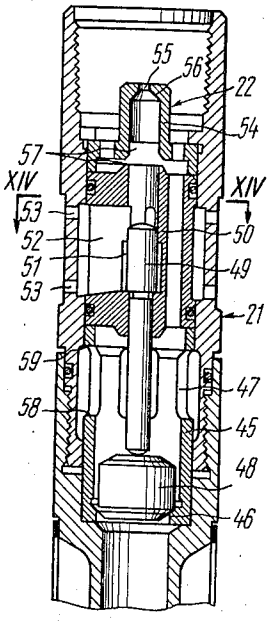

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FIG.13

SUBMERSIBLE SCREW PUMP

CROSS REFERENCES TO RELATED APPLICATIONS

This application is a continuation of application Ser. No. 188,729 filed Oct. 13, 1971, now abandoned.

BACKGROUND OF THE INVENTION

The present invention relates to the pump-making industry and, more particularly, to submersible screw pumps for transferring fluids, substantially viscous fluids containing gases at a high pressure, for example, in oil pumpage from deep oil wells or in transferring oil products from oil reservoirs.

PRIOR ART

Known in the art are submersible pumps of this type. A screw pump has been proposed heretofore, comprising a drive with a starting device, coaxially secured stators formed by rigid casings with elastic bushings fixed within the casings coaxially therewith, with the elastic having inner screw threads and being protected by a safety valve with a piston and a pilot valve, and twin screw rotors.

One of the rotors has a right-hand screw thread and the other rotor a left-hand screw thread, with each screw rotor being disposed inside the corresponding stator having a number of starts of the thread exceeding that of the rotor by one. In principle, both rotors can rotate in opposite directions. Provided between the screw threads of the rotor and stator are closed spaces disposed along the stator axis from the intake chamber of each rotor to the common delivery chamber.

The screw rotors are usually connected to the driving shaft and to each other through double universal joints in which both heads are interconnected by an elastic member.

In the stators of the known screw pumps, the length of the elastic bushing is only 5 to 10 times the diameter of the screw rotor cross section. Such pumps can operate during a long period at a pressure not exceeding a 200-meter column of the liquid pumped.

The double universal joints employed in such pumps for connecting the screw rotors, are provided with two heads interconnected through a flexible member, for example, a steel rope.

The coupling elements transmitting the torque are made defined as balls located within the slots. The pump is provided with a protective device made as a safety valve, in which the travel of the pilot valve in the direction of flow of the liquid being pumped is by virtue of the velocity pressure of the flow, and in the opposite direction due to the pressure of liquid in the delivery chamber.

The known screw pumps are disadvantageous in that they fail to develop a pressure exceeding a 1,000-m column of the liquid pumped and cannot operate under such conditions during several months continuously and without repair.

In the known pumps operating at a pressure of a 1,000-m column of liquid the elastic bushings of the stators are liable to break and fail, the flexible member, viz., the steel rope interconnecting the heads of the joints breaks down, the pilot valve of the safety valve and the driven collar of the starting clutch are apt to get damaged during the start of the rotors.

The known screw pumps are inefficient when employed for oil pumpage from deep wells and, sometimes, they are absolutely useless under such operating conditions; therefore, a new engineering solution of the problem is made necessary. This solution should be suitable not only for oil pumpage from a well but also for transferring viscous oil products from underwater reservoirs.

The known screw pumps are equipped with safety valves having guide sleeves of the gates which are slotted in their upper portion. The purpose of these slots is to reduce the effort of the velocity pressure applied to the gate at the end of its rise. This purpose, however, can be met only at a definite ratio between the length of these slots and the distance from the seating of the valve guide sleeve to the nearest step of the slots, and the length of the valve gate cylinder generatrix.

An object of the present invention is to eliminate the above-mentioned disadvantages.

SUMMARY OF THE INVENTION

The specific object of the invention is to provide a high-pressure and long-life submersible screw pump for oil pumpage from deep oil wells, in which there are provided definite ratios between the sizes determining the hydraulic and mechanical characteristics of the screw rotors and stators of the pump. This necessitates to combine the great length of the elastic bushings of the stator capable of creating high heads, with the strength of the balanced screw rotors, pivot joints, pilot valve, and the components of the starting clutch to provide a reliable start of at least two substantially long screw rotors. Such a pump intended for oil-production industry, must operate in a well continously during 6–12 months without repair.

The elastic bushings must be reliably fixed to the rigid casings of the stators and protected against dry friction and destruction by excess pressure.

The pilot valve with a gate should automatically travel in its bushings in one direction not only under the action of the velocity pressure of the liquid at the stator outlet but also due to the hydraulic resistance of the buffer at the end of the upward motion of the pilot valve, and in the opposite direction under the pressure of the liquid effective in the delivery chamber.

High pressure developed by the pump results in heavy axial loads and torques which are particularly increased during starting. Therefore, the double universal joints should be free of any weak links such as a flexible member interconnecting the heads of the joints and balls transmitting torque through a point contact of the surfaces.

In addition, it is necessary to strengthen the starting device by eliminating breakdowns of the driven collars of the starting clutch.

According to the invention, these and other objects are attained owing to the fact that in the submersible screw pump, the length of the elastic bushing of each stator is at least 20 times the cross-sectional diameter of the corresponding screw rotor, while the oblong elastic bushings are protected against dry friction and excess pressure by automatically connecting the delivery chamber to or disconnecting it from the intake chambers of the rotors by means of a gate and a pilot valve, located coaxially within the safety valve, depending on the output of the rotors and the pressure effective in the pump delivery chamber, in which case the

centers of the extreme closed spaces of each rotor adjacent to the delivery chamber are located on a common line passing through the axis of the stators and the axis of the rigid member interconnecting the two heads of the joints transmitting torque from one rotor to the other.

Each head of the double universal joints is preferably formed by two part-spherical surfaces partitioned by two symmetrically arranged truncated cones adjoining each other with their bases and having an angle of taper corresponding to the angle of deviation of the axis of the rigid member interconnecting the heads, from the axis of the stators, the surface of these cones being provided with longitudinal part-cylindrical slots which correspond to the slots of the head cap bushings encompassing the heads and accommodating loosely set rollers which are in a linear contact with the lines of intersection of the truncated cones of the heads with the part-cylindrical slots to form an engagement of the heads with the bushings of the head caps capable of transmitting torque at an angle to the axis of the universal joint rigid member and to that of the screw rotor.

One of the head caps of each universal joint is preferably made integral with the cap bushing, thus forming a single member.

The heads of the double universal joint are preferably secured to the ends of the screw rotors and the driving shaft and each head is slidably encompassed by two caps through part-spherical surfaces and with a gap, in which case the two adjacent caps of one joint are preferably interconnected by a bridge to form a single member.

It is desirable to make the part-spherical surfaces in the form of changeable inserts.

According to the specific object of the invention, one of the caps of the joint head is preferably provided with an opening which is sealed by an elastic cup which follows the shape of the head cap end face.

It is also advantageous to make the seal of the opening in the head cap as an undulated pipe and to provide inlet and outlet valves on the head caps of the universal joints.

Advantageously, the rigid casings of the stators of metal are preferably provided with at least two rows of equally spaced orifices while grooves are on the inner surfaces and end faces of the rigid casings of the stator.

Then, in the safety valve protecting the elastic bushings of the stators against dry friction and excess pressure, the valve being located in the delivery chamber and consisting of a guide sleeve provided with a seating and slots, a gate disposed in the sleeve, a pilot valve arranged in its bushing whose hollow space is communicated through channels with the intake chambers of the screw rotors, the pilot valve being adapted to cut off these channels from the delivery chamber, it is advisable to locate coaxially to the gate and the pilot valve, a hydraulic buffer having, in its sleeve, a calibrated hole with a gentle transition along an oblique surface from the hole to the pilot valve which, together with the gate, moves in one direction under the action of the differential pressure created by the flows of liquid within the guide and buffer sleeves and moves in the opposite direction under the pressure of the liquid effective in the delivery chamber. Advantageously, the safety valve is provided with a limiter of the pilot valve travel in the direction of flow of the liquid being pumped which limiter is constituted as a hydraulic buffer disposed on the side opposite to the gate. To prevent a pressure drop within the buffer sleeve as against the pressure within the delivery chamber, it is expedient to make symmetrical channels between the adjacent end faces of the pilot valve bushing and the buffer sleeve, with the cross-sectional area of channels being equal to at least a doubled cross-sectional area of the pilot valve.

The guide sleeve of the safety valve is preferably provided with at least two symmetrical slots whose size along the axis of the valve exceeds the length of the gate cylinder generatrix, the lower step of the slots being spaced from the gate seating for a distance at least twice as large as the length of the gate cylinder generatrix. Furthermore, advantageously the starting device is defined as a starting clutch disposed between the drive and the drive shaft and having a driven member constituted by a collar with two inner claws which during rotation of the driving member, located inside of the driven member, mesh with spring-interconnected pawls, in which case the end face of the collar of the clutch driven member on the drive side is preferably defined by annular bridges interconnecting the end faces of the claws.

The collar of the starting clutch driven member is preferably provided with apertures disposed symmetrically to each other on one side throughout length of each claw, with an oblique surface being near each aperture on the collar inner side opposite to the claw.

The proposed novel combination of the components of a submersible screw pump makes it possible to considerably increase the efficiency of such a pump when pumping oil from deep wells.

The submersible screw pump is of the volume type and combines the advantageous features of centrifugal and piston pumps.

The proposed pump provides for a continuous flow of liquid in a system oil bed - well - casing head with minimum emulsification of the liquid without pulsation due to the principal operating characteristics of the pump and due to the absence of suction and discharge valves and also provides for a high head and delivery of fluid at a low cross section and a small weight of the pump (the head may be as high as 1,500–2,000 m and even higher).

The pump has a high efficiency which only slightly changes within a wide range of pressures and reaching 70–80 percent when pumping a viscous liquid.

The simple construction of the pump facilitates its mounting and repair. The starting clutch facilitates the start of the pump with a standard submersible electric motor equipped with a hydraulic protection system, while the presence of a safety valve provides for filling the pumping tubes with liquid when lowering the pump into a well and discharge of the liquid therefrom when the pump is withdrawn from the well, and prevents dry friction of the rubber of the elastic bushings of the pump stators when the gate is closed or in the case of clogging, the discharge pipeline or absence of liquid in the intake chambers of the rotors. The provision of such a valve makes it possible to wash the pumping tubes with a hot liquid to remove paraffin deposits and to pump a thick liquid into a flowing well through the central string prior to raising the pump. The valve operates automatically depending on whether the liquid is present in the well.

An unexpected technical effect has been obtained during pumpage of oil wells with a yield below the rated

output of the pump. Due to the automatic operation of the safety valve which controls the cooling and lubrication of the oblong elastic bushings of the stators, the dynamic level of the oil reservoir in the well with inadequate oil yield is subjected to rhythmical oscillations and this increases the oil inflow from the reservoir into the well. The yield of the well, as well as the production of oil are increased.

The use of the present invention makes it possible to considerably widen the field of application of the screw pumps in the oil production industry. The pump can also be effectively employed as a reservoir pump, for example for transferring viscous oil products from underwater reservoirs into tankers.

The present invention will be better understood from the following detailed description of an embodiment thereof with reference to the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal section of the submersible screw pump according to the invention having a pair of screw rotors with right-hand and left-hand threads;

FIG. 2 shows an embodiment of the submersible screw pump comprising several pairs of twin screw rotors interconnected in pairs and coupled to both ends of the shaft;

FIG. 3 the view looking in the direction of the arrows is a longitudinal section of the screw rotor and stator, taken on an enlarged scale;

FIG. 4 is a sectional view taken along the line IV—IV in FIG. 3 the view looking in the direction of the arrows;

FIG. 5 is a fragmentary longitudinal section of the stator with a left-hand two-start screw thread, taken on an enlarged scale;

FIG. 6 is a sectional view taken along the line VI—VI in FIG. 5 the view looking in the direction of the arrows;

FIG. 7 is an enlarged-scale view of a longitudinal section of the double universal joint;

FIG. 8 is a sectional view taken along the line VIII—VIII in FIG. 7 the view looking in the direction of the arrows;

FIG. 9 is an enlarged-scale fragmentary longitudinal section of a modification of the universal joint head;

FIG. 10 is an enlarged-scale fragmentary sectional view of a modification of the universal joint head;

FIG. 11 is an enlarged-scale fragmentary longitudinal section of a modification of the universal joint head secured to the end of the screw rotor;

FIG. 12 is an enlarged-scale fragmentary longitudinal section view of still another modification of the universal joint head;

FIG. 13 is an enlarged-scale fragmentary longitudinal section view of a portion of the pump with a safety valve and a hydraulic buffer;

FIG. 14 is a cross section taken along the line XIV—XIV in FIG. 13 the view looking in the direction of the arrows;

FIG. 15 is a longitudinal section on an enlarged scale of the driven portion of the starting clutch, and

FIG. 16 is a sectional view taken along the line XVI—XVI in FIG. 15 the view looking in the direction of the arrows.

DETAILED DESCRIPTION OF THE INVENTION

The following terms will be used in the description of the present disclosure.

The term "submersible screw pump" is applied to a pump of an eccentric-screw or rotary type immersed into a well under the liquid level to pump this liquid to the daylight surface.

The "drive" is a motor, preferably a submersible electric motor.

The "starting clutch" is a clutch adapted for unidirectional working rotation of the unit, for example a claw or centrifugal clutch, which places the pump into operation after the motor rotor has been accelerated to a predetermined speed.

The "flexible coupling" is a device interconnecting two shafts of which at least one performs planetary motion and transmits torque and axial loads from one shaft to the other. The flexible coupling can be a double universal joint or a double axially movable coupling transmitting torque at an angularity of the driving and driven shafts and designed for loads corresponding to the pump parameters.

The "universal joint" is a device capable of transmitting torque, tensile and compressive loads, or power from one shaft to the other whose axis is coplanar with that of the former shaft and makes an angle therewith variable during operation.

The "double universal joint" is a device consisting of two universal joints.

The "screw rotor" is a rotor with a screw thread, for example one-start thread, which in its cross section is a circle, the centers of all the cross sections being regularly arranged on a screw line about the axis of the screw rotor.

The "twin screw rotors" are two screw rotors, one having a right-hand thread and another having a left-hand thread, interconnected, for example, through a double universal joint.

The "elastic bushings" of the stators are usually of oil-resistant rubber which is cured onto a rigid, preferably metal, stator sleeve. The inner screw surface of these bushings corresponds to the screw surface of the rotor and has one extra start of the thread as compared to the screw surface of the rotor.

The "closed spaces" are spaces between the screw rotor and the elastic bushing of the stator formed due to the provision of an "extra" start of the screw surface of the elastic bushing of the stator, with such spaces being filled with liquid. Due to the interference between the rotor and the stator bushing, these closed spaces have an indefinite spiral shape formed by two screw surfaces, i.e. by the surfaces of the stator and rotor and the contact line, that is the line of intersection of these surfaces.

The submersible screw pump (FIGS. 1 and 2) comprises a drive 1, for example an electric motor powered through a cable 2. Mounted on a shaft 3 of the drive 1 is a starting device constituted by a starting clutch 4 consisting of a driving member 5 and a driven member 6 interconnected through two pawls 7. The driven member 6 is mounted on a driving shaft 8 of the pump rotating in bearings 9 and 10 with shock absorbers 11 and having a seal 12. Mounted on the other end of the driving shaft 8 is a double universal joint 13 connecting the shaft 8 to one of two screw rotors 14 positioned into an elastic bushing 16 (FIGS. 1, 2, 3 and 4) of a stator

15 which, in turn, is coaxially fixed in a rigid casing 17.

The other end of the screw rotor 14 projecting from the elastic bushing 16, mounts another universal joint 13 connecting the second screw rotor 14 with the first rotor. One of the rotors 14 has a right-hand thread while the other has a left-hand thread.

Each screw rotor 14 is located within a corresponding elastic bushing 16 of the stator 15 having a thread with an extra start as compared to that of the rotor 14 so that between the screw threads of the rotor 14 and the elastic bushing 16 of the stator 15, there are formed closed spaces 18 disposed along the pump axis from an intake chamber 19 of each rotor 14 to a common delivery chamber 20. In this case, the centers of the closed spaces 18 of each rotor 14, which are the closest to the delivery chamber 20 are disposed on a common line passing through the axis of the stators 15 and the axis of the double universal joint 13 connecting the two rotors 14.

Located within the delivery chamber 20 is a safety valve 21 provided with a hydraulic buffer 22.

The rigid casings 17 of the stators 15 (FIGS. 5 and 6) are of metal and provided with three rows of uniformly disposed orifices 23, while V-shaped or rectangular grooves 25 are provided inside each rigid casing 17 and at its end faces 24.

Each double universal joint 13 (FIG. 7) consists of two heads 26 interconnected through a ridid member 27, for example a shaft or bridge (FIG. 11). Each of the heads 26 (FIG. 7) is formed by two part-spherical surfaces 28 divided by two symetrically located truncated cones 29 whose bases adjoin each other (FIGS. 7, 9–12). The angles of the truncated cones 29 are equal by magnitude to the angle of deviation of the axis of the rigid member 27 from the axes of the screw rotors 14 (FIGS. 1 and 2). The cones 29 (FIG.8) are intersected by longitudinal part-cylindrical slots 30 corresponding to longitudinal part-cylindrical slots 31 of busings 32 (FIGS. 7-12) of caps 33 encompassing the heads 26. Loosely set in the slots 30 and 31 (FIG. 8) are rollers 34 (FIG. 7) adapted for engagement of the heads 26 with the bushings 32 through the linear contact of the truncated cones 29 with the rollers 34.

Each head 26 is embraced by two caps 33 having part-spherical surfaces 28 corresponding to the heads 26. One of the caps 33 of the heads 26 of each universal joint 13 is integral with the bushing 32, thereby forming a single member 35 (FIGS. 10 and 12), in which case, the caps 33 may be located both on the inner side of the double universal joint 13 (FIGS. 7, 10 and 12) and on the outer side thereof FIG. 11. FIG. 9 shows an exemplary embodiment of one member of the double universal joint 13 made as two caps 33, inner and outer, thread-joined to the bushing 32.

FIGS. 7, 9 and 12 show exemplary embodiments of the part-spherical surfaces 28 of the head 26 and the caps 33 made as changeable inserts 36 or 37.

One of the caps 33 of the head 26 has an opening 38 (FIG. 10) which is sealed with an elastic cup 39 whose sealing flange 40 follows the shape of the face of the cap 33 of the head 26. The cup can be pressed to the cap 33 by a spring 41.

FIG. 12 shows an exemplary embodiment of the seal of the opening 38 of the cap 33 by means of an undulated pipe 42, in which case an inlet valve 43 and an outlet valve 44 are disposed on the caps 33 of the heads 26.

The safety valve 21 (FIG. 1) is located in the chamber 20 and comprises a guide sleeve 45 (FIG. 13) provided with a seating 46 and at least two slots 47 in which is slidably mounted a gate 48 of a pilot valve 49 arranged in a bushing 50 of the pilot valve 49. A cavity 51 of the bushing 50 through channels 52 and ports 53 communicates with the intake chambers 19 (FIG. 1) of the screw rotors 14 and are shut off from the delivery chamber 20 by the pilot valve 49.

Located coaxially with the gate 48 (FIG. 13) and the pilot valve 49 on the side opposite to the gate 48, is a hydraulic buffer 22 having in a sleeve 54 a calibrated hole 55 featuring a gentle transition along an oblique surface 56 from the hole 55 to that portion of the sleeve 54 in which moves the end of the pilot valve 49 after it has left the bushing 50.

Provided between the adjacent faces of the bushing 50 of the pilot valve and the sleeve 54 of the buffer 22 are symmetrical channels 57, whose cross-sectional area is at least equal to the double cross-sectional area of the pilot valve 49. The channels 57 may have either annular or rectangular shape as shown in FIG. 13.

The length of the symmetrical slots 47 of the guide sleeve 45 along the axis of the valve 21 from a lower step 58 of the slots 47 to a upper step 59 exceeds the length of the generatrix of the cylinder of the gate 48.

The lower step 58 is spaced from the seating 46 for a distance which is at least twice as large as the length of the generatrix of the gate 48.

The driven member 6 (FIGS. 1 and 2) of the starting clutch 4 of the pump is defined by a collar 60 (FIG. 15) provided with two internal claws 61. A face 62 of the collar 60 at the side of the drive 1 is constituted as annular bridges 63 connecting the faces of the claws 61. These bridges 63 form apertures 64 in the collar 60 (FIG. 16), with the apertures being arranged symmetrically relative to each other on one side of each claw 61 along the entire length thereof. Near each aperture 64, the inner side of the collar 60 opposite to the claw 61 has an oblique surface 65.

A sludge pipe 66 is located above the safety valve 21 (FIG. 1) forming a gap between its walls and pumping tubes 67 which serves as a settler for large solids of mechanical impurities.

OPERATION

The present submersible screw pump operates as follows.

The pumping unit is sunk into an oil well by means of tubes 67 (FIG. 1) and the electric motor 1 of the drive is powered from the daylight surface through the cable 2.

The torque of the electric drive 1 during its operation is transmitted to the driving member 5 of the starting clutch 4 mounted on the shaft 3. The driving member is located within the collar 60 (FIG. 15) of the driven member 6 (FIG.1) of the starting clutch 4. Under the effect of centrifugal force the pawls 7 located in the driving member 5 overcome the action of the return spring and mesh with the claw 61 (FIGS. 15 16) of the collar 60 (FIG. 1). Upon deactivating the electric motor of the drive 1, the pawls 7 under the action of the non-return spring are attracted to each other, and the driven member 6 is disengaged from the driving member 5. The driven member 6 is mounted on the drive shaft rotating in the bearings 9 and 10. The bearings 10 may be radial-thrust bearings provided with rubber-

metal shock absorbers 11 which increase the life of these bearings.

From the driving shaft 8, the torque through the two double universal joints 13 is transmitted first to one and then to the other screw rotor 14.

The screw rotors 14 rotate in the elastic bushings 16 of the stators 15 eccentrically and execute planetary motion. Since the rotors 14 are arranged in the bushings 16 with a negative allowance, they can be started with the standard submersible pump 1 only through an impact by means of the starting clutch 4.

In principle, the present screw rotors 14 can rotate in both directions. However, the direction of rotation must be strictly definite at a predetermined arrangement of the intake chambers 19 and the delivery chamber 20 with a safety valve 21. The opposite direction of rotation would be unsuitable since the pump cannot operate without being primed. The starting clutch 4 serves as a non-return coupling to avoid such a possibility.

During undesirable reverse rotation, the pawls 7 do not engage the claws 61 (FIG. 15, 16) of the collar 60 due to the presence of the apertures 64 and the oblique surface 65 on the inner side of the collar 60 (FIG. 1).

In the case of emergency axial displacement of the driven member 6 of the starting clutch 54 downwards or upwards, the pawls 7 disengage the claws 61 (FIGS. 15, 16) and the starting clutch 4 (FIG. 1) disconnects the electric drive 1 from the pump, thus serving as a safety device.

In order to balance the axial loads due to the pressure developed by the pump in the delivery chamber 20 and to unload the radial thrust bearings 10, the pump is equipped with double screw rotors 14, one having a right-hand thread and the other a left-hand thread. During the rotation of the rotors 14, the liquid from the intake chambers 19 fills the closed spaces 18 between the rotors 14 and the elastic bushings 16 of the stators 15, and the flows of liquid move in these spaces 18 along the axis of the stators 15 towards each other into the delivery chamber 20 housing the double universal joint 13.

The combined flow of liquid from the two rotors 14 washes the stator 15 which is more distant from the drive 1 and is fed first into the safety valve 21 and then through the sludge pipe 66 into the tubing 67 through which the liquid is pumped to the daylight surface.

The connection of the screw rotors 14 through the double universal joints 13 permits the rotors 14 to be self-aligning units occupying the optimum position within the elastic bushings 16 of the stators 15, thus compensating for the technological errors associated with the process of manufacture and assembly of the pump. Owing to the fact that the rotors 14 rotate eccentrically about the axis of the stators 15, to balance the centrifugal forces appearing during rotation, the rotors 14 are displaced from the axis of the stators 15 in diametrically opposite directions, while the centers of the extreme closed spaces 18 adjacent to the delivery chamber 20 are located on a common line passing through the axis of the rigid member 27 of the double universal joint 13.

The remaining unbalanced moment due to the pair of contrifugal forces acting in opposite directions, creates a certain vibration of the pump which improves filling of the intake chambers 19 and the adjacent closed spaces 18 of the screw rotors. This effect is particularly important during pumping of viscous liquids.

In this case, the suction capacity of the pump is of up to 8.5 m of the column of liquid.

In addition, the vibration of the pump, which does not reduces the strength of its components, effectively acts upon the oil flow passing from the oil-bearing bed into the well and this also is an advantage of the present submersible pump.

In order to provide for high heads of 1,000–2,000 meters of the liquid column which are necessary for pumping oil from deep wells and to meet the requirements of the strength of the pump components, the elastic bushings 16 of the stators 15 are oblong, in which case their length is at least 20 times the cross-sectional diameter of the corresponding rotor 14.

In order to provide for a reliable coupling of the elastic bushings 16 with the rigid, for example metal casings 17 of the stators 15, the inner surface of the casings 17 and the end faces 24 are provided with grooves such as thread or rectangular cuts. The three rows of uniformly disposed orifices 23 are used for the same purpose.

The torque from the driving shaft 8 to the screw rotors 14 is transmitted through the rollers 34 (FIGS. 7–12) of the double universal joints 13. The truncated cones 29 of the heads 26 have an angle of taper corresponding to the angle of deviation of the axis of the rigid member 27 of the joint 13 from the axes of the rotors 14. Such making of the cones 29 provides for a linear contact between the rollers 34 and the lines of intersection of the truncated cones 29 with the part-cylindrical slots 30 of the heads 26.

The gap between the part-cylindrical slots 30 and 31 and the rollers 34 placed between these slots limits the angle of deviation of the member 27 from the axis of the rotor 14 and is selected so that, with free rotation of the double universal joint 13 beyond its one cover, the other free end of the joint 13 will not touch housing 68 (FIG. 1) of the pump.

The axial load arising in the delivery chamber 20 is transmitted from the rotors 14 to the part-spherical surfaces 28 (FIGS. 7, 9–12) of the caps 33 and the heads 26 of the double universal joints 13 and is taken by the rigid member 27 which expands. If the intake chamber 19 is disposed between the rotors 14 (FIG. 1), then the rigid member 27 (FIGS. 7–12) will be compressed. To provide for making of the part-spherical surfaces 28 of the caps 33 and the heads 26 of hard wear-resistant material, they are in the form of interchangeable inserts 36 and 37.

The safety valve 21 (FIG. 13) protects the oblong elastic bushings 16 against dry friction, excess pressure and eliminates their untimely destruction. After sinking the pump into a well, the gate 48 and the pilot valve 49 of the valve 21 are in the bottom position. Through the open ports 53 and the channels 52, the liquid from the well is admitted into the bushing 50 of the pilot valve 49 and fills the empty tubing 67 which is used for sinking the pump into the well. The gate 48 (FIG. 13) resting on the seating 46 does not allow the liquid to enter the space between the rotors 14 (FIG. 1) and the shaped screw surface of the elastic bushings 16 of the stators 15. Therefore, if the liquid contains mechanical impurities, such cannot accumulate below the gate 48 and thus impede the start of the pump during the operation.

After the pump has been started, the flow of liquid lifts the gate **48** (FIG. **13**) in the guide bushing **45** up to the slots **47.** The pilot valve **49** raises, together with the gate **48,** and closes the passage of the liquid through the channels **52** to the delivery chamber **20** (FIGS. **1** and **2**) of the tube **67** and back. The total amount of liquid transferred by the pump is delivered to the daylight surface through the tubes **67.**

The pilot valve **49** (FIG. **13**) and the gate **48** continue to move by inertia and under the action of the liquid pumped. The pilot valve **49** enters the sleeve **54** of the hydraulic buffer **22** wherein it is smoothly decelerated due to the presence of the calibrated hole **55.**

The diameter of the hole **55** is determined depending on the characteristics of the pump and the liquid being pumped. A portion of the liquid is discharged from the space of the sleeve **54** of the buffer through the channels **57** and the gap between the pilot valve **49** and the sleeve **54.** The gentle transition along the oblique surface **56** to the hole **55** is necessary for eliminating the possibility of deposition of paraffin and sand in the inner space of the sleeve **54** in advance of the hole **55.**

Thus, the pilot valve **49** moves, together with the gate **48,** under the action of the difference of the forces of the flows of liquid in the guide bushing **45** and in the bushing **54** of the buffer **22.**

The hydraulic buffer **22** provides for a shock-free and reliable operation of the entire valve and the pump as a whole. The gate **48** is located within the guide bushing **45** with a gap, the magnitude of which depends on the output of the pump and the viscosity of the liquid.

The pilot valve **49** moves downwards under the action of the pressure of the liquid in the tubing **67** because the pilot valve **49** (FIG. **13**) is stepped and its cross section is different along its axis.

If the pump output exceeds the yield of the well or, for any reason, the yield decreases during prolonged (several months) operation of the pump, the dynamic level is reduced and the oil column, which always contains gas, particularly in its upper portion, descends to the delivery chambers **19** (FIG. **1**) of the screw rotors **14.** The output of the pump is reduced due to an entrapping of the gas by the rotors **14** together with the liquid and compression of this gas within the closed spaces **18.**

Simultaneously with the drop in the output capacity of the pump, the velocity of the liquid in the gap between the gate **48** (FIG. **13**) and the walls of the guide bushing **45** is decreased. The remaining constant force acting on the pilot valve **49** from above due to the liquid pressure becomes greater than the force acting on the pilot valve **48** from below. Thus, the pilot valve **49** moves downwards and allows the liquid to pass from the delivery chamber **20** (FIG. **1**) into the well to the intake chambers **19** of the screw rotors **14,** thereby preventing the appearance of dry friction of the threads of the elastic bushings **16** of the stators **15.**

The liquid circulates through the safety valve **21** until it is accumulated in the well and until the dynamic level is restored necessary for reestablishing the rated output of the ump. In pump. case the gate **48** (FIG. **13**) emerges and overcomes the force of pressure of the liquid acting on the pilot valve **49.**

The pilot valve breaks the flow of liquid from the chamber **20** through the channels **52** and the ports **53,** and the pump begins pumping the liquid through the tubes **67** (FIG. **1**) to the daylight surface again.

Similarly, the safety valve **21** drops the liquid into the well at a pressure exceeding the rated value and protects the elastic bushings **16** and the double universal joints **13** against damage. The pressure in the delivery chamber **20** can overpass the permissible limit in the case of clogging the tubing **67** with sand and paraffin or in the event of the gate on the surface being inadvertently left closed.

When the pressure in the delivery chamber **20** rises at a constant yield of the pump, the pilot valve **49** (FIG. **13**) drops down and opens the discharge ports **53** for a short period of time. After reducing the pressure in the delivery chamber **20,** the gate **48** raises the pilot valve **49** again, and the latter closes the discharge channels **52** and the ports **53.** When the tubing **67** is closed, the pressure rises over again and the "up-down" movement of the pilot valve **49** with the gate **48** is repeated.

When the pump stops to operate, the gate **48** (FIG. **1**) is no more affected by the effort on its bottom, and the pilot valve **49** lowers under the action of the pressure of the liquid in the tubing **67.** The gate **48** will rest on its seating **46,** thereby closing the way for the liquid into the pump and its clogging. The liquid will flow from the tubing **67** into the well, therefore the pump can be lifted up from the well with the empty tubes should the need for its repair arise.

The presence of the safety valve **21** provides for making a number of technological operations during exploitation of oil wells: washing of the tubing **67** for removing paraffin deposits if the pumped oil contains paraffin, suppressing the wild flowing by means of thick liquid prior to the raising of the pump if its output is insufficient.

We claim:

1. A submersible screw pump adapted for operation in an oil well for pumping oil therefrom comprising in combination: a drive provided with a shaft; a starting device mounted on the shaft of said drive; first and second stators mounted in a co-axial relationship with said drive, each of said stators having elastic bushings mounted within a rigid casing; a first screw rotor arranged eccentrically in said first stator, said first stator having an inner screw surface with an extra screw thread as compared to the screw surface of said first screw rotor; a first double universal joint for connecting said starting device with said first screw rotor and transmitting a turning moment; a second screw rotor arranged eccentrically in said second stator, said second stator having an inner screw surface with an extra screw thread compared to the screw surface of said second screw rotor; the direction of the screw surfaces of said second stator and screw rotor being opposite to the direction of the screw surfaces of said first stator and screw rotor; said screw surfaces of said first and second stators and screw rotors being in contact with each other and forming closed spaces disposed along the axes of the stators, the closed spaces to be filled with liquid being pumped; a second double universal joint for coupling said first and second screw rotors, transmitting a turning moment and carrying axial loads due to the pressure of the liquid; at least two intake chambers, one chamber being located at the extreme end of each of said first and second screw rotors and being connected with the space of said oil well; a delivery chamber located between the two other ends of said first and second screw rotors, said second double universal joint being located in this delivery chamber; a

safety valve communicating through a channel with said delivery chamber and with the space of the tubing serving for lowering said pump into said well where said pump operates and protecting said pairs of rotors and stators from dry friction and excessive pressure by admitting the oil from the space of said tubing into the space of said well; the length of the bushings of each of the first and second stators being at least 20 times the cross-sectional diameter of the corresponding screw rotor.

2. The submersible screw pump of claim **1**, wherein said first and second stators are formed by rigid sleeves and elastic bushings secured inside said rigid sleeves, the centers of the extreme closed spaces of each screw rotor adjoining said delivery chamber being located along a single line passing through the axis of said first and second stators and the axis of said double universal joint located in the delivery chamber.

3. The submersible screw pump of claim **1** wherein each of said first and second double universal joints comprises: two heads, a rigid member interconnecting said heads, an internal and external cover for accommodating said heads therebetween, at least two pairs of rollers for pivot connection of said heads with said covers, the pair of said external covers being secured to the adjacent ends of said screw rotors.

4. The submersible screw pump of claim **3** wherein each said head of said double universal joints comprises two part-spherical surfaces, two symmetrically located truncated cones with their bases adjoining one another and having a taper angle corresponding to the angle of deviation of the axis of said rigid member from the axes of said screw rotors; at least two longitudinal semicylindrical slots on said truncated cones, each said cover accommodating said heads having a part-spherical surface, and at least two longitudinal semicylindrical slots located correspondingly to said longitudinal semicylindrical slots on said heads, said rollers being arranged freely and with a gap in said longitudinal semicylindrical slots of said heads and said covers and having linear contact with the lines of intersection of the surfaces of said truncated cones with the surfaces of said semicylindrical slots of said heads to ensure transmission of the turning moment between said heads and said covers at an angle to the axes of said rigid member and said screw rotors.

5. The submersible screw pump of claim **3** wherein one pair of said internal or external covers is made integral with the casing of said cover, thus forming a single member.

6. The submersible screw pump of claim **3** wherein said heads are secured at the ends of said screw rotors and said driving shaft, two of said adjacent internal covers of said double universal joint being interconnected through a bridge to form an integral member.

7. The submersible screw pump of claim **4** wherein said part-spherical surfaces of said heads and covers are made in the form of changeable inserts.

8. The submersible screw pump of claim **3** wherein said covers have an opening for mounting said rigid member which is sealed by a sealing cup, the end face of the sealing cup conforming to the end face shape of said cover.

9. The submersible screw pump of claim **8** wherein the opening in the cover is sealed by an undulated pipe, and inlet and outlet valves are mounted on said cover.

10. The submersible screw pump of claim **2** wherein said rigid sleeves of the stators are made of metal and have at least two rows of uniformly arranged orifices and grooves on the inner surface and on the end faces of said flexible bushing.

11. The submersible screw pump of claim **1** wherein said safety valve includes a gate and a guide bushing, a pilot valve bushing having channels for connection with the space of said well, a pilot valve having two transverse diameters for disconnecting the space of said tubing from the space of said well, a hydraulic buffer having a bushing which is provided with a calibrated hole with a gentle transition along a tapered surface from said hole to said pilot valve, said pilot valve moving together with said gate in one direction under the effect of the difference of the forces of the accelerated flows of liquid in said guide bushing and in said buffer bushing and moving in the other direction under the effect of the liquid pressure in the space of said tubing.

12. The submersible screw pump of claim **11** wherein said hydraulic buffer is disposed on the side opposite to said gate and serves as a limiter restricting the movement of said pilot valve in the direction of motion of the flow of pumped fluid.

13. The submersible screw pump of claim **12** wherein, in order to prevent a pressure drop within said buffer bushing compared to the pressure in the space of said tubing, between the adjacent end faces of said pilot valve bushing and said buffer bushing there are located symmetrical channels having a cross sectional area equal to a double cross sectional area of said maximum diameter of the pilot valve.

14. The submersible screw pump of claim **11** wherein, for reducing the effect of the velocity flow of liquid on the velocity of motion of said gate and said pilot valve during the process of starting said pump, said guide bushing has at least wo symmetrical two the dimension of which along the axis of said valve exceeds the length of the generating line of the cylinder of said gate, while the lower step of said slots is spaced from said seat by a distance at least twice as great as said length of the generating line of the cylinder of said gate.

15. The submersible screw pump of claim **1** wherein said starting device is made in the form of a starting clutch disposed between said drive and said driving shaft and comprising a collar as a driven member, two inner teeth, the end face of said collar on the side of said drive having annular bridges interconnecting the faces of said teeth, and a driving member in the form of two claws and a spring for resetting said claws when said drive is stopped.

16. The submersible screw pump of claim **15** wherein said bushing of said driven member has apertures located symmetrically relative to each other along the entire length of said teeth and an oblique portion on the inner surface of said collar opposite to said tooth along said aperture.

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UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,802,803 Dated April 9, 1974

Inventor(s) Alexandr A. Bogdanov, et al

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

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Commissioner of Patents

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