Aug. 6, 1963 J. C. FISHER 3,099,998

FLUID RECTIFIER

Filed April 11, 1960 4 Sheets-Sheet 1

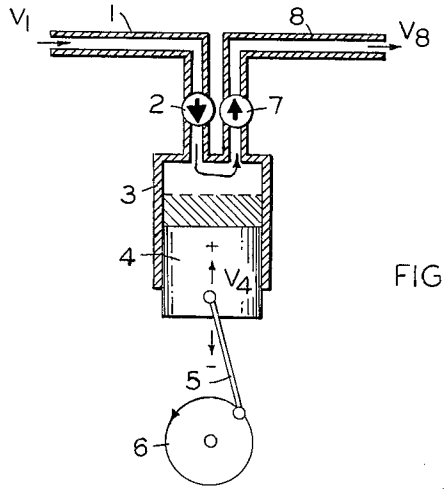


FIG. 1

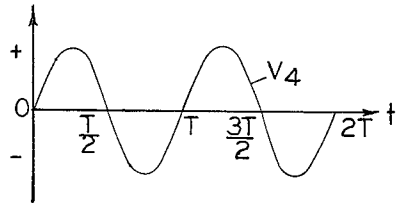


FIG. 2

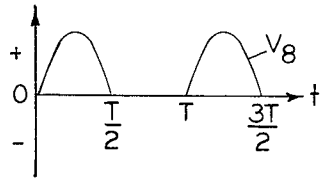


FIG. 3

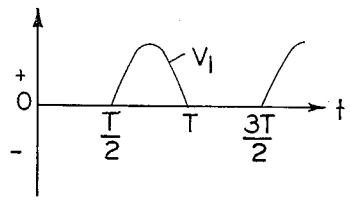


FIG. 4

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FLUID RECTIFIER

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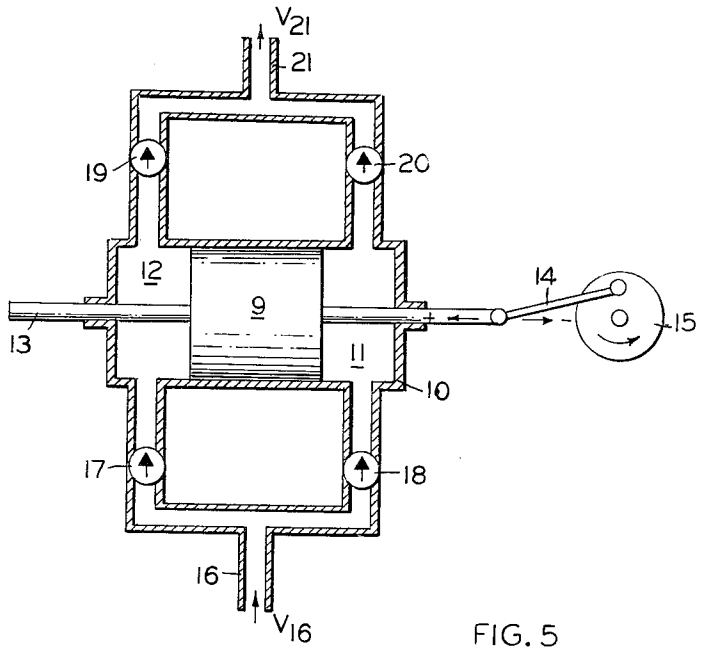


FIG. 5

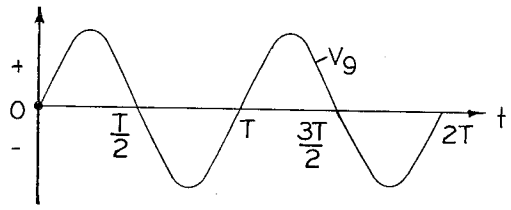


FIG. 6

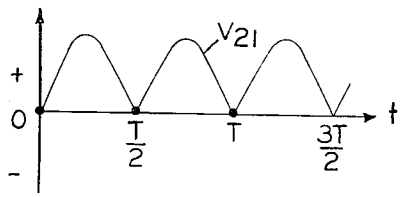


FIG. 7

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FLUID RECTIFIER

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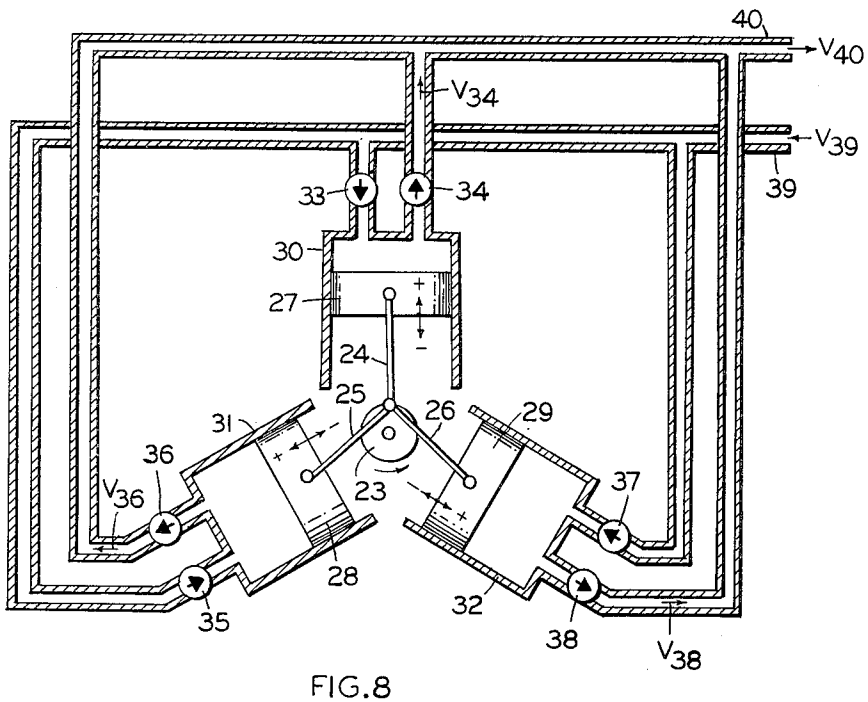


FIG. 8

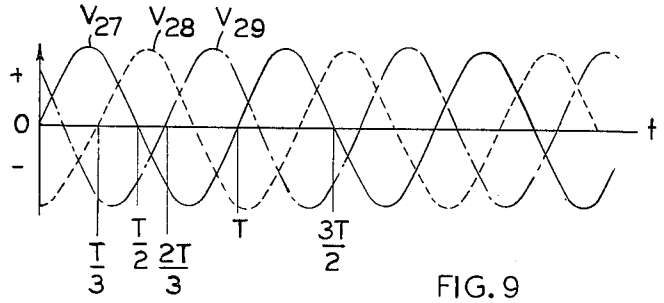


FIG. 9

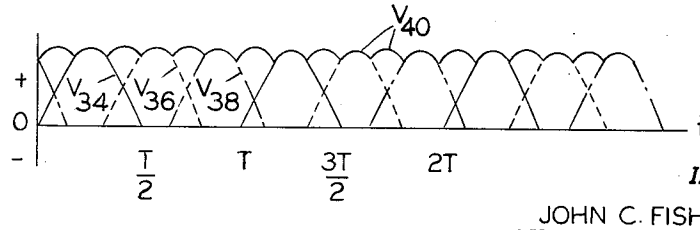


FIG. 10

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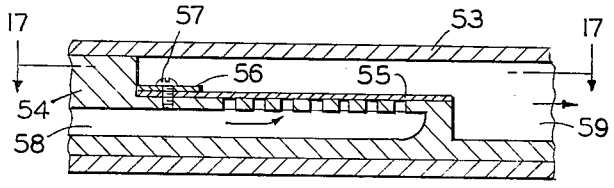


FIG. 11

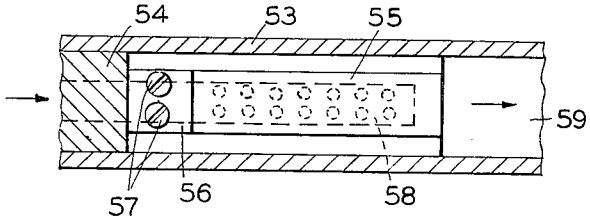


FIG. 12

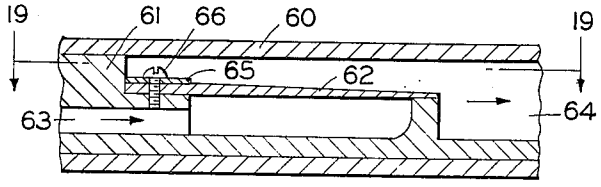


FIG. 13

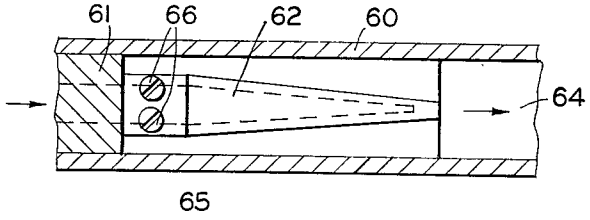


FIG. 14

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United States Patent Office 3,099,998 Patented Aug. 6, 1963

3,099,998

FLUID RECTIFIER

John C. Fisher, Cambridge, Mass., assignor to Am Dyne Trust, a trust of Massachusetts

Filed Apr. 11, 1960, Ser. No. 21,412
3 Claims. (Cl. 137—525.3)

This invention relates to a novel and improved fluid-rectifier valve for use in pumping devices of the type or types which produce a reciprocating or alternating flow of fluid internally and employ check valves or the like to convert this reciprocating flow to a unidirectional flow. This application is a continuation-in-part of my copending application Serial No. 636,597, filed January 28, 1957, now abandoned.

Examples of fluid pumps of the type with which the invention is concerned are: (1) single-cylinder piston pumps; (2) multi-cylinder piston pumps; (3) diaphragm pumps; (4) accelerator-tube pumps (sometimes called "inertia" or "liquid-piston" pumps) of the type described in my copending patent application Serial No. 553,015, dated December 14, 1955, now Patent No. 2,936,713; and (5) piston-tube pumps of the type described in my U.S. Patent No. 2,898,858, issued August 11, 1959. Such pumps when operated at low frequencies can satisfactorily employ rectifier valves of the long-familiar ball-check type. This type of valve has proved very reliable over a long period of use and evolution in low-frequency reciprocating pumps. However, when such pumps are operated in the frequency range from 20 cycles per second upward, the use of ball-check valves causes the volumetric efficiency of these pumps to decline steadily with increasing frequency. This is so because the check element in the ordinary ball-check valve is essentially a free mass coupled to the moving stream only by frictional forces. To cause such a mass to reciprocate through a fixed displacement requires a force which increases in proportion to the square of the frequency. This parabolically increasing force can be provided only by a relative velocity between the stream and the check element, such that the stream moves faster (absolute velocity) than the check element. The situation can be improved somewhat by the use of a hollow check element to reduce its mass, and by employing a shape which has a high drag coefficient, but inevitably, as frequency increases, the displacement of the check element from its seat lags further and further behind the velocity variation of the stream. The result is that considerable backflow occurs through the valve, and the volumetric efficiency of the pump declines progressively toward zero. Further disadvantages of the free-check-element type of rectifier valve are the noisy operation and the mechanical hammering of the check element against the seat and the forward stop as the operating frequency rises. The time-honored means by which this problem has been attacked has been the use of a spring element so arranged as to push the check element against the seat. The spring thus provides an additional force over and above the frictional drag acting to drive the check element back onto the seat when the forward stream reverses. By this means the time-lag between flow reversal and closing of the valve can be reduced.

However, the mere addition of a spring does not reduce the overall time lag between check valve displacement and the velocity variation of the fluid to the extent desired for optimum operation and will, of course, tend to increase the time lag on opening of the valve. This is partly due to an increase in effective mass of the system caused by the addition of the spring. Also when the check element is immersed in a fluid, and particularly a liquid, the movement of the check element through the

fluid results in a change in speed or direction of the fluid, resulting in an inertial force on the check element which affects system behavior in the same as if the mass of the check element had been increased.

To some extent, this mass-loading of the valve system can be reduced by streamlining the check element and the spring. It can also be partially compensated for by increasing the stiffness of the spring. However, simply increasing stiffness is not entirely beneficial, because it increases the forward pressure difference necessary to open the valve, and also the forward pressure drop with valve open at any given flow rate. This causes a loss of energy from the flowing stream, and it also causes the volumetric efficiency to suffer if such a valve is used on the intake side of a pump, because the forward pressure drop causes a fall in the density of a flowing gas or a possibility of cavitation in a flowing liquid.

Accordingly, it is the object of this invention to provide a novel and improved rectifier valve for use with fluid pumping systems of the type providing an alternating output fluid flow which will operate in substantial time phase relationship with the frequency of velocity variation of the output fluid flow thus materially reducing, if not substantially eliminating, back flow through the valve; which will operate quietly and with a minimum of wear; which will require a relatively small pressure drop to effect actuation thereof in an opening direction; and which is of a relatively simple construction and will provide an extended trouble-free service life.

In the drawings:

FIG. 1 is a diagrammatic representation of an exemplary single acting piston pump system incorporating rectifier valves;

FIG. 2 is a graphic representation of the velocity-time relationship of the movement of the piston of the pump of FIG. 1;

FIG. 3 is a graphic representation of the velocity-time relationship of the output flow of the pump of FIG. 1;

FIG. 4 is a graphic representation of the velocity-time relationship of the input flow of the pump of FIG. 1;

FIG. 5 is a diagrammatic representation of an exemplary double acting piston pump system incorporating rectifier valves;

FIG. 6 is a graphic representation of the velocity-time relationship of the movement of the piston of the pump of FIG. 5;

FIG. 7 is a graphic representation of the velocity-time relationship of the output fluid flow of the pump of FIG. 5;

FIG. 8 is a diagrammatic representation of an exemplary three-phase reciprocating fluid pumping system incorporating rectifier valves;

FIG. 9 is a graphic representation of the velocity-time relationship of the movement of the pistons of the pumps of FIG. 8;

FIG. 10 is a graphic representation of the relationship of the velocity variation in the output fluid flow in the system of FIG. 8 to the three piston velocities;

FIG. 11 is a cross sectional view of an alternative form of a rectifier valve;

FIG. 12 is a cross sectional view substantially along the line 17—17 of FIG. 11;

FIG. 13 is a cross sectional view of an embodiment of a rectifier valve incorporating the present invention; and

FIG. 14 is a cross sectional view substantially along the line 19—19 of FIG. 13.

In order to appreciate the many physical arrangements of pump systems with which this invention is concerned, reference will now be made to FIGS. 1–10 which relate to the more well-known configurations which can be used to pump liquids or gases. In each case, the pumping

device itself produces an internal fluid displacement and velocity which is essentially alternating or reciprocating, and rectifier valves are provided to operate upon this alternating flow to produce in the external fluid circuit an essentially unidirectional, but usually pulsating flow. For simplicity, the pumping elements have been shown as pistons, but it is to be understood that other structures such as diaphragms, accelerator tubes, etc., could also be used to motivate the fluid.

FIG. 1 shows the schematic arrangement of a single acting piston pump for fluids. For simplicity, a crank-and-connecting-rod drive is shown, but, as will be apparent, a cam mechanism, an electrodynamic vibration motor, etc., could be employed. An intake pipe **1** leads the fluid through an intake rectifier valve **2** into a cylinder **3**. Received in the cylinder is a piston **4** actuated by means of a connecting rod **5** which is in turn driven by a crankshaft **6**. From the cylinder **3**, on the discharge stroke of the piston, the fluid emerges via rectifier valve **7** and discharge pipe **8**. The time-variation of the piston velocity V_4 is substantially a sine wave as shown in FIG. 2, and if the rectifier valves perform their function effectively, the variation in the velocity V_1 of the fluid in the intake pipe and the variation of the velocity V_8 in the discharge pipe will, as shown in FIGS. 3 and 4, respectively, be substantially a sinusoid in which only the positive half-wave remains, the negative half-waves having been removed by the action of the rectifier valves. It is to be noted that the positive half-waves in pipe **1** represent a periodic function which is displaced positively on the time scale by one-half period, $T/2$, relative to the positive half-wave train in pipe **8**. This is so because the piston is single-acting, and the intake pipe can conduct fluid forward only when the discharge pipe is not conducting, and vice-versa.

In regard to FIG. 1, it should also be noted that within the volume swept through on each complete cycle by the top of the piston (shown shaded in FIG. 1) there is a purely alternating flow of the fluid as long as the liquid or gas remains in contact with the piston top (cavitation does not occur, with a liquid). If there were no valves **2** and **7**, this purely alternating flow would extend from the intake pipe through the pump to the discharge pipe, and no useful external fluid transfer would occur. The action of the valves converts the purely alternating fluid velocity at the piston top to a unidirectional, pulsating fluid velocity in each of the pipes **1** and **8**. Hence the term "rectifier valve" is fully justified in reference to valves **2** and **7**.

FIG. 5 shows a double-acting piston pump. This type of pump is often preferable to the single-acting variety, because it produces a greater discharge for the same piston displacement-rate (in terms of volume per unit time), and its external flow pulsations are smaller in relation to the average flow than those of the pump in FIG. 1. In FIG. 5, a piston **9** divides a closed cylinder **10** into two identical chambers **11** and **12**. Each chamber is traversed by an extension of a shaft **13** which carries piston **9**. The shaft **13** is driven by a connecting rod **14** and a crankshaft **15** in such a way as to produce a substantially sinusoidal piston velocity, V_9 as shown in FIG. 6. The chamber **11** is connected via an intake rectifier valve **18** to an intake pipe **16**, and via a discharge rectifier valve **20** to a discharge pipe **21**. Similarly, chamber **12** is connected via an intake rectifier valve **17** to a pipe **16**, and via a discharge rectifier valve **19** to a pipe **21**. Inflow of fluid to the pump is via a pipe **16** and outflow is via a pipe **21**. By analogy with the rectification of alternating electric current, it may be said that the rectifier valves **17**, **18**, **19**, and **20** are connected in a single-phase, full-wave bridge arrangement. This bridge arrangement of valves serves to convert the purely alternating fluid velocity at each end face of piston **9** into a unidirectional, pulsating fluid velocity V_{16} at the intake pipe and V_{21} at the discharge pipe. The time-variation of fluid velocity is here the same at both pipes **16** and **21**, and is essentially a rectified full-sinusoid. It should be noted that as shown in FIG. 7 there are two major pulsations of fluid velocity in every period T, where there was but one in the pump system of FIG. 1.

FIG. 8 shows a 3-phase, single-acting pump system. The chief advantage of this arrangement, and its logical extensions to higher numbers of phases, is the reduction in the magnitude of the external flow pulsations relative to the average flow. The system of FIG. 8 is essentially three pumps like the pump of FIG. 1, with their respective intake pipes in parallel and their respective discharge pipes in paralley. A crankshaft **23** drives the pistons **27**, **28**, and **29** through connecting rods **24**, **25**, and **26**, respectively. The three connecting rods are driven by the same crankpin, and the pistons they control are arranged to move back and forth along axes which are spaced 120° apart around the center of the crankshaft. As shown in FIG. 9, the time-phase difference between the velocity of any one piston and the next is thus ⅓-cycle. If a higher number of phases had been used, for example, six, then the phase shift between successive pumping elements would be ⅙-cycle or 60°. Cylinders **30**, **31**, and **32** are each associated with an intake rectifier valve **33**, **35**, and **37**, respectively; and a discharge valve, **34**, **36** and **38**, respectively. Flow enters the common intake pipe **39**, divides equally among the three pumping elements, and emerges from a discharge pipe **40**. It should be noted here that, as shown in FIG. 10, the major frequency of pulsation in the flow through pipes **39** and **40** is six times the piston frequency, there being six major pulsations in this flow in every period T.

It will be readily apparent that many other pumping arrangements of this general type are possible. The foregoing examples merely serve to show some of the more important pumping systems in which rectifier valves are employed.

Ideal behavior for a rectifier valve in a fluid pump of the type with which this invention is concerned is as follows: the valve should open with the minimum possible forward pressure difference at the instant when forward fluid velocity rises from zero; the opening of the valve, that is the displacement of the check element from its seat, should increase continuously as the forward fluid velocity increases; the opening of the valve should decrease continuously as the forward fluid velocity diminishes; and the valve should just be seated again at the time when the forward fluid velocity reaches zero and would, in the absence of the valve, reverse in direction. Stated more concisely, the rectifier valve should cause minimum forward pressure drop in the fluid stream, and its displacement off the seat should be exactly in time-phase relation with the forward fluid velocity. Secondary criteria of proper behavior are: the valve should have but one degree of freedom, that is it should be constrained against all motions other than the direct one on and off the seat parallel to the average direction of the fluid stream; and it should strike the seat on closing with the least possible momentum. This latter condition implies that the effective mass of the valve element must be as low as possible, and that its backward velocity at the instant of closing be as near zero as possible. Finally, the back-and-forth motion of the valve element should occur with a minimum of rubbing or sliding contact between solid surfaces. This last consideration is important because it affects the rate of mechanical wear of the valve parts, and it partially determines the forward pressure drop across the valve since friction in the valve must be overcome by an externally applied force in order to keep the valve element moving.

I have found that the above criteria for a rectifier valve which will satisfactorily operate with "reciprocating" pumps operating in the frequency range from 20 cycles per second up to as high as 250 cycles per second, is met in a valve having as its principal operative element a sub-

stantially flat, cantilever reed, of solid highly resilient material not subject to permanent set in normal operation of the valve, with one end of the reed being rigidly secured and the other end covering an opening or port through which fluid may flow in such a direction as to push the free end of the reed away from the port, but not in the opposite direction, and with the reed having a lowest undamped natural frequency, when immersed in the fluid, which is greater than the frequency of the velocity variation of the fluid passing through the valve. The effective static stiffness of the reed for forward deflections (off the seat) must be as low as possible consistent with the requirement that the immersed undamped natural frequency of the reed be at least equal to but preferably greater than the frequency of the variation in fluid velocity. All other factors being equal, I have found that those rectifier valves which have the highest ratio of immersed undamped natural frequency to effective forward static stiffness, yield the highest volumetric efficiency when used to rectify the internal fluid velocity produced by reciprocating pumps like those shown in FIGS. 1, 5 and 8. Volumetric efficiency of a reciprocating pump is defined as the ratio of the average fluid volume per unit time delivered out the discharge pipe, to the total volume per unit time swept through inside the pump by all the motivating elements, whether they be pistons, diaphragms, etc.

Although it is possible to compute analytically the lowest undamped natural frequency of a cantilever reed when immersed in air or any common gas, it is extremely difficult to compute by purely theoretical means the lowest undamped natural frequency of such a reed when immersed in a liquid. The lowest natural frequency of the system is proportional to the square-root of the ratio of effective systems stiffness (spring rate), measured statically, to effective system mass. With the reed immersed in a fluid the effective mass of the reed is increased; this is not serious with gases, but with liquids it considerably raises the effective mass of the reed. This is so because the check element cannot move through the liquid without causing some change in the absolute velocity thereof, either a change in speed (rate) or a change in direction. This change of velocity is responsible for an inertial force on the reed, and this inertial force affects the system behavior in exactly the same way as if the mass of the reed had been increased. The best method, from a practical point of view, is to design the reed on the basis of gas-immersion, and to modify the design by use of empirical factors derived from tests on liquid-immersed reeds of similar construction. In most cases, I have found that those reeds which exhibit the highest ratio of lowest undamped natural frequency to effective static stiffness in air, also exhibit the highest ratio in liquids like water, oil, kerosene, etc. which are commonly encountered in industrial pumping systems.

The cantilever valve reed may be made with uniform width and thickness along its entire length, or the width and/or the thickness may be varied in some definite manner along the length. The end of the reed which covers the valve port may be made flat, or it may be furnished with a protuberance of some sort which serves to cover the valve port. The fixed end of the reed may be clamped by welding, bolting, riveting, or any of the commercially accepted means of fastening two solid objects together. The material for the reed may be any of several homogeneous materials such as steel, copper, brass, aluminum, titanium, Monel, and other suitable metals; or unreinforced plastics like poly-tetrafluoroethylene (Teflon) or poly-methylmethacrylate (Plexiglas); or reinforced plastics like melamine-bonded woven fiberglass fabric and a whole family of similar composite materials.

In each complete rectifier valve there may be but one cantilever reed and associated valve port, or there may be several such reeds and ports arranged in parallel so as to divide the fluid flow to obtain a greater effective port area and a reduced forward fluid velocity. The valve reed may be clamped in such a way that there is a definite preloading in the closed position, i.e. the reed is under some bending stress when seated. This helps to reduce back leakage through the valve when the reed is in closed position, but it also increases the forward pressure drop through the valve. The flow through the valve port may be in a direction which is substantially perpendicular to the general plane of the valve reed when the valve is closed. However, I have found by experiment that it is often beneficial to construct the valve so that the reed when in seated position is obliquely inclined to the average direction of the forward-flowing fluid. It is clearly apparent that any design of rectifier valve which works well in a single-phase pumping device like that of FIG. 1 will also work well in any polyphase pumping device like that of FIG. 8. Hence, what has already been said, and the description which follows, are equally relevant for the operation of rectifier valves in both single-phase and polyphase reciprocating pumps.

Although exact analytical relationships cannot be derived for the important practical case of liquid-immersed rectifier valves, a study of the case of an air-immersed cantilever reed having uniform width and thickness along its length will reveal the important parameters of the problem. Assume that a reed is mounted so that one end is rigidly clamped to a stationary support and the opposite end is free to move in a direction perpendicular to the general plane of the reed. As long as the deflection of the free end of the reed does not exceed 20% of the free length of the reed, which it will seldom do in practical cases, the following analytical relationships are valid. The effective static stiffness of the reed is defined as the static force applied perpendicular to the plane of the reed at its free end, divided by the deflection which results there and is expressed as:

$$K=\frac{1}{4}ED\left(\frac{d}{L}\right)^3 \tag{1}$$

where

K=the effective (free-end) static stiffness of the reed, in units of (Force)/(Length);

E=Young's modulus of elasticity for the reed material, in units of (Force)/(Area);

D=the width of the reed, in units of (Length);

d=the thickness of the reed, in units of (Length);

L=the free length of the reed, in units of (Length).

The undamped natural frequency of the reed in air, in its lowest natural mode of transverse vibration, i.e. that mode in which every point along the free length of the reed at any instant is deflected to the same side of the rest position, is given by

$$\omega_n=1.0162\frac{d}{L^2}\sqrt{\frac{Eg}{w}} \tag{2}$$

where

ω_n=the lowest undamped natural frequency in air, in units of (Radians)/(Time);

g=the earth's gravitational acceleration, in units of (Length)/(Time)^2;

w=the specific weight of the reed material, in units of (Force)/(Volume).

From Equations 1 and 2, we may derive the ratio of undamped lowest natural frequency in air to effective static stiffness of the reed:

$$\frac{\omega_n}{K}=\frac{4.065\sqrt{gL}}{Dd^2}\sqrt{\frac{1}{Ew}} \tag{3}$$

On the basis of Equation 3 we may select either an optimum available material if the dimensions are given, or an optimum set of dimensions if the material is specified.

One further criterion of a good valve may be introduced at this point. Every cantilever reed, when deflected to the full-open position, must be subjected to an internal bending stress. For every material of which such a reed may be made, there is an upper limit to the value which

this stress may reach if the reed is to survive a predetermined number of cycles of opening and closing. This is generally referred to as an "endurance limit." If we denote the endurance limit by S_e, then we may arbitrarily write a new criterion for a rectifier valve of the cantilever-reed type, namely the product of the ratio given by Equation 3 and this endurance limit:

$$\frac{\omega_n S_e}{K}=\frac{4.065\sqrt{gL}}{Dd^2}S_e\sqrt{\frac{1}{Ew}} \quad (4)$$

On the basis of Equation 4 a preferred choice of material is a laminated fiberglass sheet such as N.E.M.A. Grade G–10, because it has a low modulus E by comparison with metals, low specific weight w by comparison with metals, but a high endurance limit S_e comparable to that for low-carbon steel. Of course, such a material cannot be used except where it has satisfactory resistance to chemical attack by the fluid to be pumped.

If a cantilever reed like that discussed above be subjected to a perpendicular force at its free end which varies according to the relation

$$f=\hat{f}\sin \omega t, \quad (5)$$

where

f=the instantaneous value of the force, in units of (Force);

$\hat{f}$=the crest or peak value of the force, (Force);

ω=the frequency of the force variation, in units of (Radians)/(Time);

t=time measured from the instant at which the force first increases positively from zero, (Time);

then the free end of the reed will experience a displacement from rest position, a velocity, and an acceleration. Each of these kinematic variables will have a sinusoidal time-variation, but they will be displaced, respectively, by phase-shifts of one-quarter cycle from one to the next, the acceleration being first in time-phase, the velocity one-quarter cycle behind acceleration, and the displacement directly opposite in phase to the acceleration. In the absence of frictional forces acting merely to halt the motion of the reed, the reed will behave either like a pure-mass element or like a pure-stiffness element, depending upon whether its lowest natural frequency ω_n is lower or higher, respectively, than the frequency ω of the disturbing force. If ω_n be lower than ω, then the acceleration of the reed will be in time-phase with the disturbing force, and in this regard it will behave essentially like the free mass of a ball-check valve, except that its apparent dynamic mass will be less than its true mass. Because of the reduction of apparent dynamic mass below true mass, the acceleration of the reed at the free end will be greater than it would if the force $\hat{f}$ were statically applied, but the displacement of the free end from rest position is just one-half cycle out of time-phase with the disturbing force. Since one of the criteria of good performance for a rectifier valve is that the displacement of the check element from rest position must be in time-phase with the forward velocity of the fluid stream (and the drag force due to the stream is exactly in time-phase with the fluid velocity), then clearly a reed valve whose lowest undamped natural frequency ω_n is lower than the frequency ω of the fluid-velocity variation cannot perform satisfactorily. This has been verified by me by experiments with reed valves of many shapes and sizes.

On the other hand, if the lowest undamped natural frequency of the reed is greater than the frequency of the force variation ($\omega_n>\omega$), the displacement of the reed will be in time-phase with the disturbing force. Because the apparent dynamic stiffness at the free end of the reed will always be less than the static effective stiffness, the displacement of the free end will be greater than it would if the force $\hat{f}$ were statically applied. However, because of the time-phase coincidence of tip displacement and disturbing force, a cantilever reed whose lowest undamped natural frequency is greater than the frequency of the fluid-velocity variation is suitable as an element for a rectifier valve. If the apparent dynamic stiffness at the free end of the reed be denoted by K_a, then the relationship of this parameter to static effective (free-end) stiffness is

$$K_a=K\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right] \quad (6)$$

From Equation 6 it can be seen that when $\omega=\omega_n$, the apparent dynamic stiffness is zero, and the only restraint upon the displacement, velocity, and acceleration of the free end of the reed is the internal (molecular) friction, which is never zero in any known material, although it is small enough in most spring materials to be ignored for many practical purposes. This is a condition of resonance, and the tip of the reed would undergo a very large alternating displacement. In most materials this would cause an early fatigue failure because of the very large internal bending stresses. However, at the exact condition of resonance, the velocity of the reed-tip would be in phase with the disturbing force, and again the reed would be unsatisfactory for use as an element in a rectifier valve. It should be noted that when such a reed is immersed in a fluid, the frictional force on the reed due to the velocity of the fluid relative to the reed is the disturbing force, and not a damping force.

The preceding analysis serves to substantiate the statement that any spring-loaded rectifier valve having only one degree of freedom will yield high volumetric efficiency when used with a reciprocating pump above 20 cycles per second only if the lowest undamped natural frequency of the moving system in the valve is greater than the frequency of velocity variation of the fluid stream. The foregoing analysis was directed toward valve elements of the cantilever-reed type, but it is equally valid for the case of a spring-loaded ball-check valve, provided that the correct equations are used for effective static stiffness and natural frequency.

I have found that maximization of the ratio given by Equation 3, namely ω_n/K, is best accomplished by the use of a cantilever reed whose width D and thickness d each decrease in some regular fashion from maximum values at the clamped end of the reed to minimum values at the free end. In every case such a tapering reed will have a higher ratio ω_n/K than a reed of the same material having the same maximum width and thickness but of constant width and thickness along the free length. The analytical proof of this is very difficult, because it requires the solution of the differential equation of the elastic curve of a cantilever beam with variable cross section. This differential equation is

$$EI\frac{d^2y}{dx^2}=M \quad (7)$$

where

I=the moment of inertia of the cross section of the beam, in units of (Length)^4;

y=deflection from rest position of any point on the free length of the beam in a direction normal to the rest plane of the beam, (Length);

x=distance along the free length of the beam, measured parallel to the rest plane from any convenient point on the beam (Length);

M=bending moment acting on any cross section of the beam in a plane normal to the rest plane of the beam, in units of (Force)×(Length).

For a uniform beam (constant width and thickness), Equation 7 can readily be solved, and it leads to the derivation of Equations 1 and 2. However, for a tapered beam, Equation 7 becomes a second-order differential equation with a variable coefficient, which is difficult to solve without the aid of a computing machine. The variable factor is I, which for a rectangular cross section is given by

$$I=\frac{Dd^3}{12} \quad (8)$$

In spite of the analytical difficulties in the solution of Equation 7 for tapered beams, it can readily be demonstrated by experiment that the ratio ω_n/K is higher for tapered beams than those of the same material, same maximum width and thickness, and constant width and thickness.

In the embodiment of FIGS. 11 and 12, a valve body in the form of a rigid duct 53 having a rectangular or square cross section is divided by a tight-fitting rigid insert 54 into an upstream passage 58 and a downstream passage 59. The valve also includes a flat cantilever reed 55, of constant thickness and width, whose fixed end is securely clamped by screws 57 between clamping plate 56 and a flat mating shelf in insert 54. Fluid flow can occur only in the direction shown by the arrows, namely from passage 58 to passage 59. In certain applications the rectifier valve may be required to withstand high pressure differences in the reverse or back flow direction of flow through the valve. Accordingly, it may be necessary to support the reed 55 on its upstream side. In the embodiment of FIGS. 11 and 12 this support of the reed is provided by the top wall of the passage 58. As can be seen from the drawings, the reed 55 is in overlying engagement with the top wall of the passage 58, and the top wall of the passage 58 is provided with a plurality of closely spaced openings for the flow of fluid therethrough.

In the embodiment of FIGS. 13 and 14 the valve comprises a body forming duct 60 having rigid walls and rectangular cross section. The duct 60 is divided by a tight-fitting rigid insert 61 into an upstream passage 63 and a downstream passage 64. Passage 63 is of rectangular section, and at its downstream end, its upper wall is cut away to form a triangular aperture. This aperture is covered on its downstream side by a cantilever reed 62, which is securely clamped at its upstream end between a clamping plate 65 and a mating flat shelf formed in the insert 61. Screws 66 are set into tapped holes in the insert 61 to provide the clamping force. The reed 62 has constant width and thickness under the clamping plate 65, but from the downstream edge of the plate 65 outward, the thickness of the reed tapers linearly to a minimum at the free end. Similarly, the width of the reed 62 tapers linearly from the clamped edge outward to the free end. For reasons of simplicity in forming the reed and the seat, it is desirable to put all the taper on the downstream (upper) face of the reed. However, this is not an absolute necessity, and the taper could be divided in any reasonable manner between the two faces. Similarly, the taper in width could all be put on one edge, or could be divided in any way between both edges. In any event it is necessary only that the bottom face of the reed engage the rim of the orifice at all points around its periphery, and that the edges of the reed project just beyond the mating edges of the orifice, in order to prevent backflow when the reed is seated.

The advantage of the construction shown in FIGS. 13 and 14 over that shown in FIGS. 11 and 12 is that the ratio ω_n/K is highest for the tapered reed (other things being equal), and yet the escape area for the fluid around the edge of the reed is large so that the forward pressure drop is reduced.

It should be apparent from the above that the specific differences in structure between the embodiments described above are merely illustrative of the many variations in construction of the rectifier valve which are possible within the scope of this invention. Further it should be apparent that each of the valves described above could be modified in accordance with the teachings of another of the embodiments described and that additional modifications of and variations in cantilever supported reed rectifier valves of the type described could be made without departing from this invention, the critical limitation being that the lowest undamped natural frequency of the reed when immersed in a fluid passing through the valve must be greater than the frequency of the velocity variation of the fluid flowing through the valve.

Inasmuch as many changes could be made in the above construction and many apparently widely different embodiments of this invention could be made without departing from the scope thereof, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

It is also to be understood that the language in the following claims is intended to cover all of the generic and specific features of the invention herein described and all statements of the scope of the invention which, as a matter of language, might be said to fall therebetween.

Having thus described my invention, I claim:

1. In a pumping system of a type providing an alternating fluid flow having a frequency of velocity variation of at least 20 cycles per second, a rectifier valve comprising means providing an opening for fluid flow through the valve, and means for controlling fluid flow through said opening including an elongated cantilever supported reed of resilient material, and means on the reed seated on the downstream side of said opening, the reed having a major and a minor transverse dimension and being tapered in the plane of its minor transverse dimension longitudinally of the reed over its entire free length, the reed having a lowest undamped natural frequency while immersed in the fluid flowing through said valve which is greater than the frequency of velocity variation of said alternating fluid flow.

2. In a pumping system of a type providing an alternating fluid flow having a frequency of velocity variation of at least 20 cycles per second, a rectifier valve comprising means providing an opening for fluid flow through the valve, and means for controlling fluid flow through said opening including an elongated cantilever supported reed of resilient material, and means on the reed seated on the downstream side of said opening, the reed having a generally rectangular cross section to provide the reed with major and minor transverse dimensions and being tapered in the planes of both the major and minor transverse dimension over the entire free length of the reed having a lowest undamped natural frequency while immersed in the fluid flowing through said valve which is greater than the frequency of velocity variation of said alternating fluid flow.

3. In a pumping system of a type providing an alternating fluid flow having a frequency of velocity variation of at least 20 cycles per second, a rectifier valve comprising an elongated hollow housing having a wall extending longitudinally up the housing to divide the housing into upstream and downstream passages, that wall being provided with aperture means for the passage of fluid therethrough, and means for controlling the flow of fluid through the valve including an elongated cantilever supported reed of resilient material overlying said wall and aperture means on the downstream side thereof, the reed having major and minor transverse dimensions and being tapered in the planes of both said dimensions over its entire free length, the reed further having a lowest undamped natural frequency when immersed in the fluid flowing through the valve which is greater than the frequency of velocity variation of said alternating flow.

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