June 7, 1955 | P. H. THOMAS | 2,710,165

ROTARY GAS OR VAPOR DEVICE HAVING SERIES ARRANGED ROWS OF BUCKETS

Filed Sept. 23, 1947

FIG. 2.

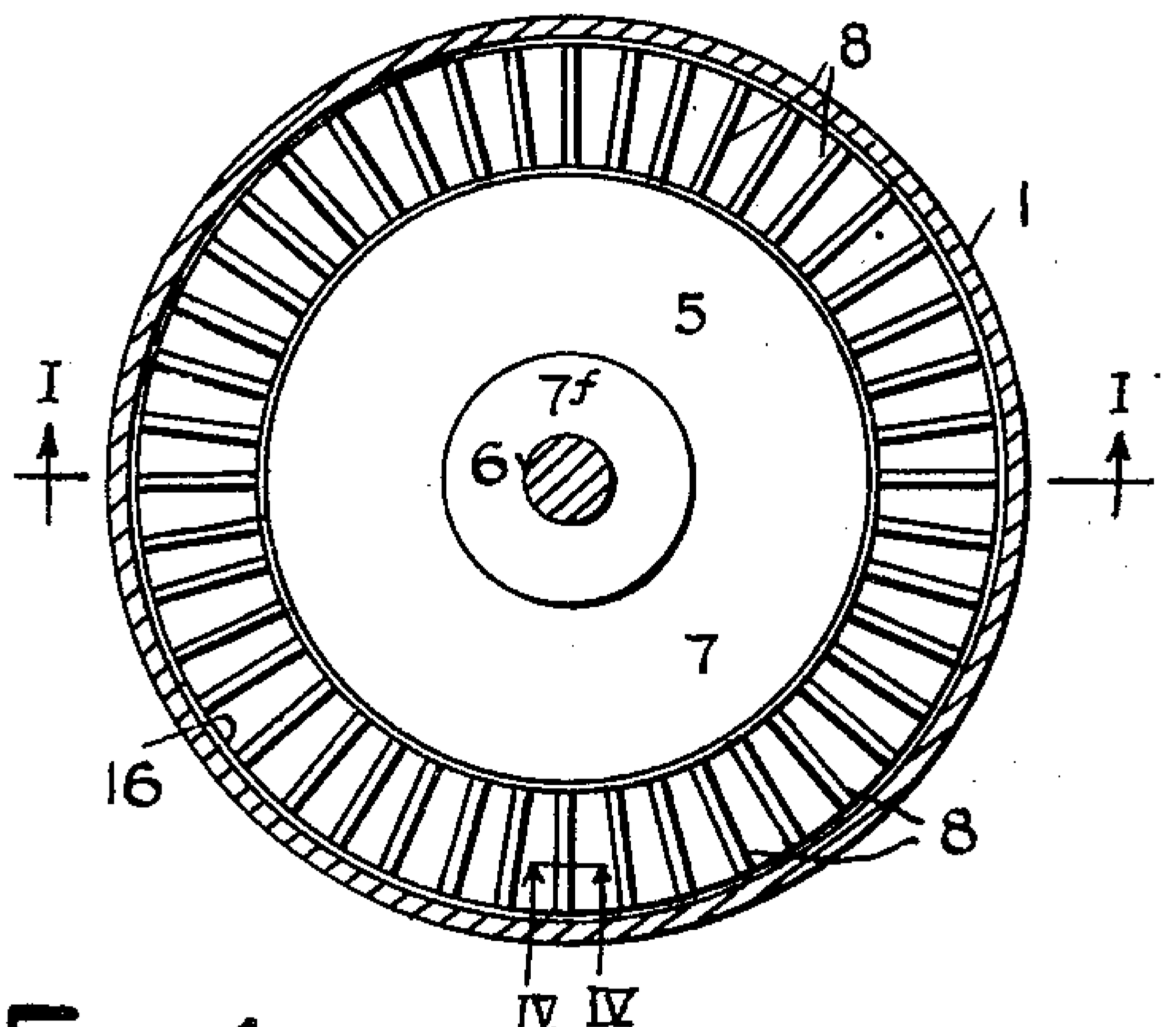


FIG. 4.

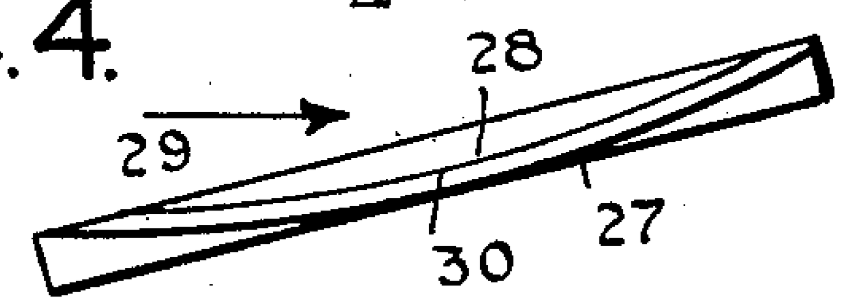


FIG. 4a.

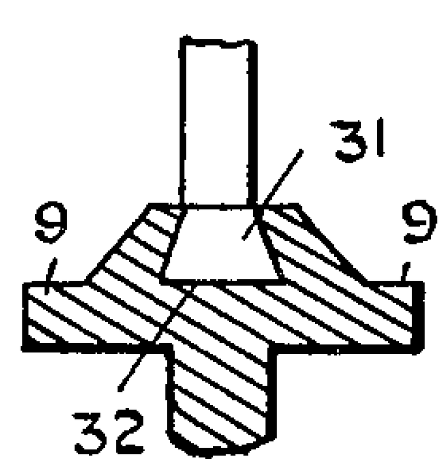


FIG. 3.

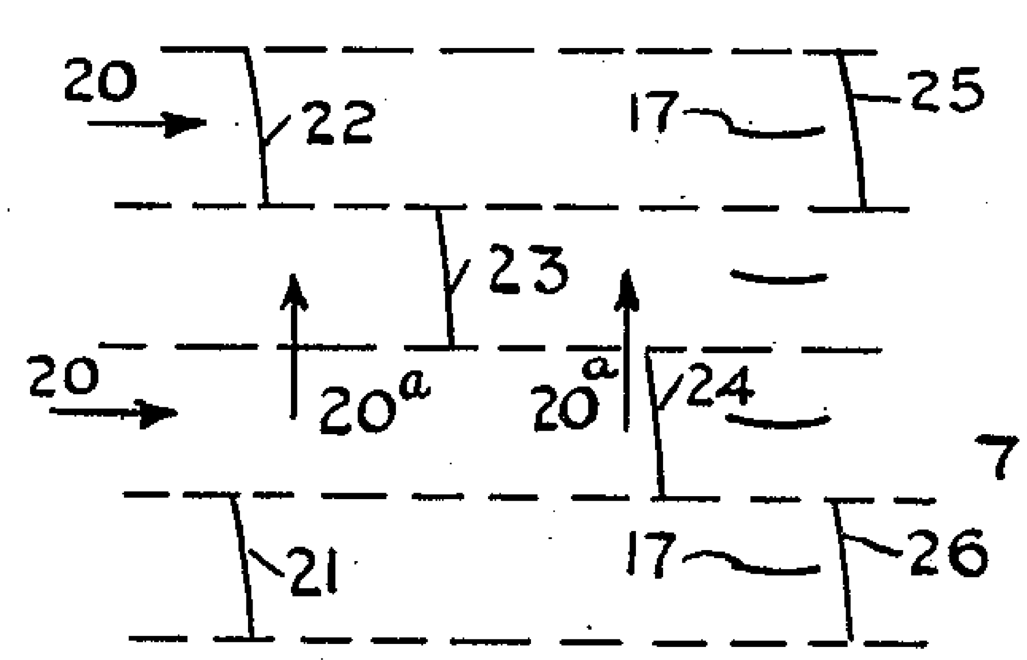


FIG. 5.

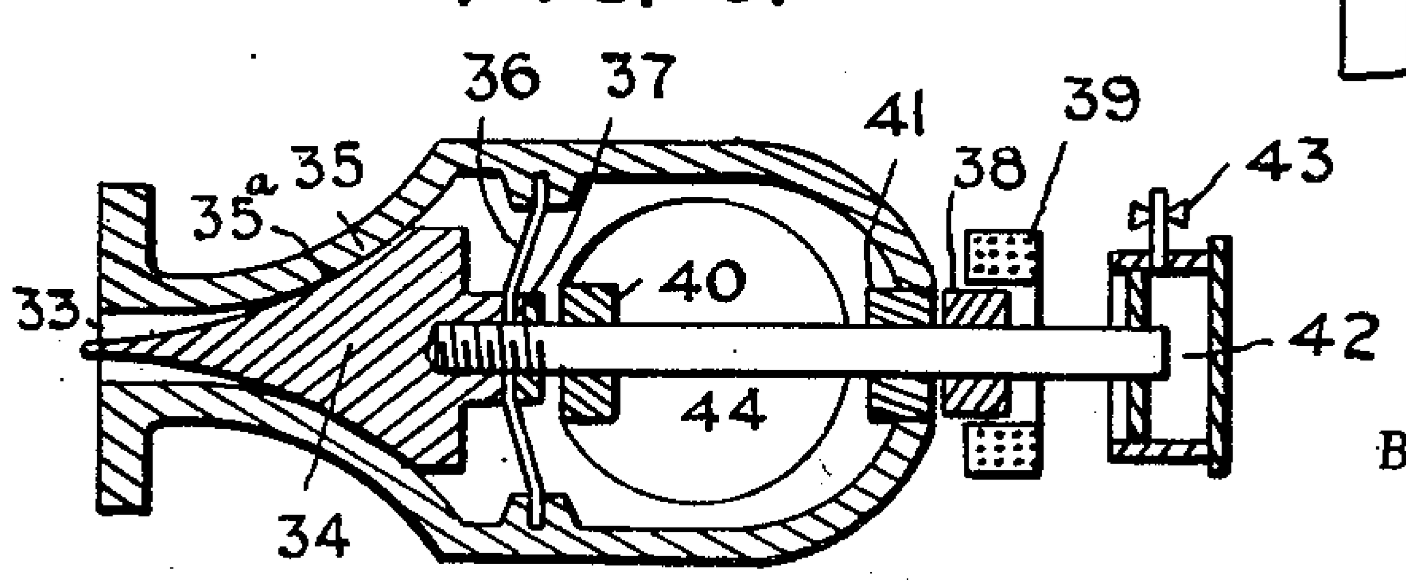
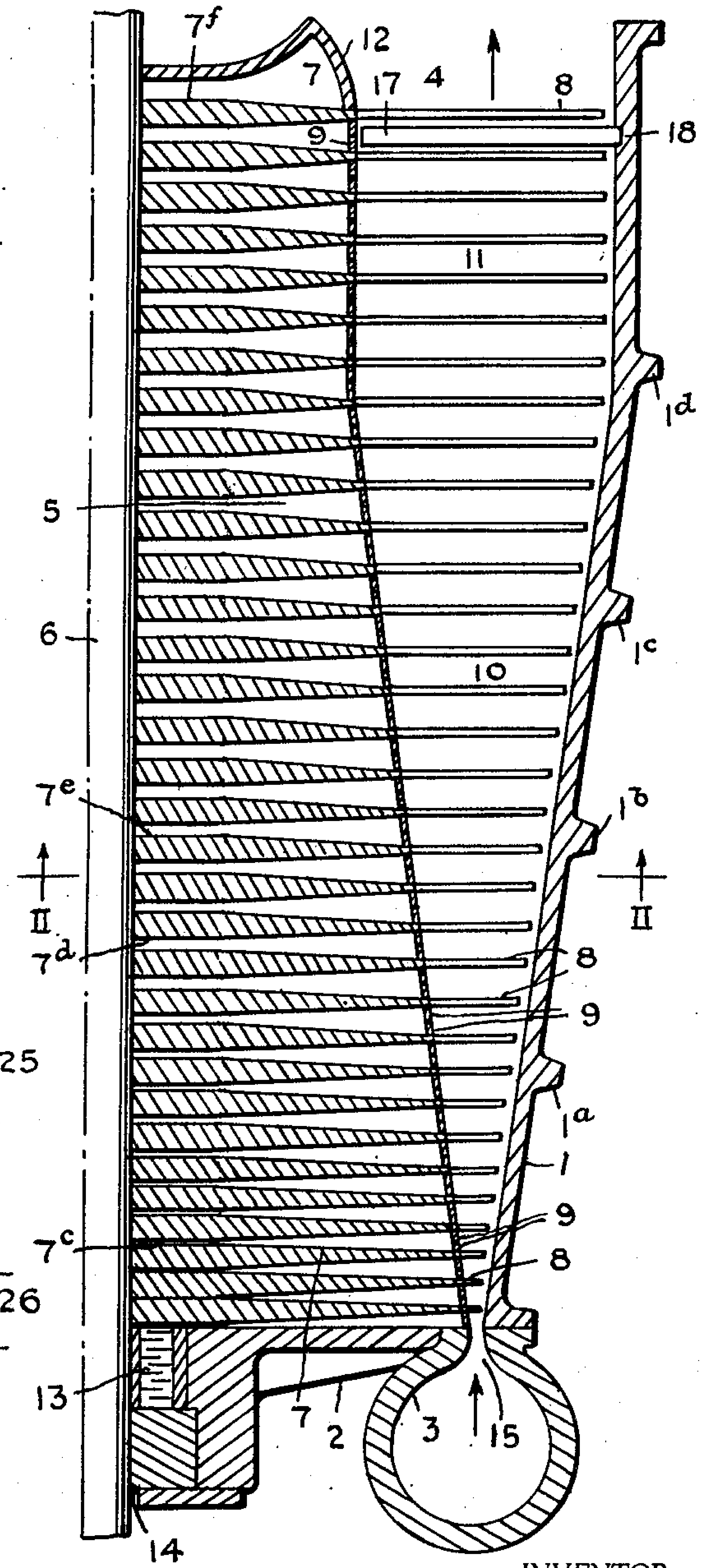


FIG. 1.



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ROTARY GAS OR VAPOR DEVICE HAVING SERIES ARRANGED ROWS OF BUCKETS

Percy H. Thomas, Washington, D. C.

Application September 23, 1947, Serial No. 775,631

4 Claims. (Cl. 253—69)

Applicant's invention relates to devices serving to extract power from or deliver power into a rapidly moving stream of steam or other compressible gas or vapor, and particularly from a stream of confined steam under pressure. This invention secures unusual simplicity of structure and high efficiency, as well as providing for a wide range of operable conditions.

Basically this transfer of power is accomplished directly through the impressed static pressure of the steam or other compressed gas, acting as a force, and through a very high bucket speed, acting as a speed, power being measured as the product of force by speed. With these characteristics is combined a novel construction for maintaining a substantially straight through flow of the gas, the new feature involving the introduction of definite, circumferential spaces, or gaps, between the rotor buckets. By "straight through" is meant steam flow without those transverse deflections (or "jets") which are characteristic of operation of the conventional steam turbine, and which are introduced to produce the pressure on the buckets which causes them to rotate. Reliance is placed upon the inertia of the stream to control the pulsations of flow resulting.

By "static pressure" as the actuating force in my turbine is meant the force exerted against the face of the moving blade by the expansive pressure in the steam or other fluid, due to its being a confined gas or vapor, such as is exerted on the piston of an engine by the motive fluid within. It excludes pressure produced by the deceleration of a jet through the conversion of its kinetic energy into mechanical energy. The critical character of this distinction on the effectiveness of steam turbines, for example, may be seen from the fact that the maximum energy derivable from a decelerating jet impinging on a slower moving surface is limited by the jet velocity (energy$=\frac{1}{2}\,M\,V^2$), and that no bucket moving faster than this limit can receive any energy therefrom. Obviously if the stationary jet or bucket is moved at increasing speed, the pressure on the surface from the jet becomes progressively less and less down to the zero point. This condition definitely limits the maximum speed of a turbine bucket receiving energy from a jet to a speed lower than the jet velocity, and in the conventional steam turbine, nozzles are provided to produce high velocity steam jets in each stage, predominantly in the tangential direction of flow with respect to the axis of rotation, in which nozzles the steam pressure causes the acceleration of the particles of steam until the pressure available in the stage has been exhausted in converting the available energy of the steam into kinetic energy. At the exit of the nozzles, the jet enters into an open free space and is no longer accelerating because it is no longer subject to accelerating pressure, since the static pressure has been dissipated. When the jet meets obstruction at the bucket which is moving at a speed less than the jet a pressure is exerted whereby energy is developed, but this force is not the original "static pressure," and is not a "component force" thereof as I use that term.

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On the other hand, in my axial flow turbine where the static pressure is exerted directly on the blade surface, positioned at a comparatively small pitch angle to the plane of rotation, the speed of the blade is determined entirely by the pitch, increasing and decreasing with the pitch without limit and without any change in the average velocity of the stream in the channel, which is moving in the axial direction at right angles to the plane of rotation of the blades. In normal operation of my turbine, the torque force developed in the direction of motion at the high rotating speed at which it is designed to operate is largely a component of the static pressure and no significant pressure on the blades is due to the velocity of the stream. The pressure derived from the velocity of the stream is insignificant in my turbine because it is flowing at a very much lower speed than that of the blades than in the conventional turbine, the ratio being of the magnitude of one to three or more. Furthermore in my turbine there is no substantial change in either the velocity or direction of the stream in passing through the wheel because much the greater part of the torque or thrust developed is derived as a geometrical component (in the direction of their rotation) of the static pressure normal to the blades due to the hydraulic head on the wheel. In other words, in my turbine the operating force is direct axial pressure, not pressure derived from kinetic energy in a high speed tangential jet. Thus the kinetic energy of the conventional turbine jet is not a significant component of the energy developed by my type of turbine when operating under normal conditions.

A high rotational blade speed is a vital feature of the design of my improved type of turbine, and has many advantages for the higher the bucket velocity in any type of turbine the less the torque force required for a given output, and the lighter the construction.

Very broadly, applicant's invention, as applied to steam power, embodies a long annular channel through which the steam flows parallel to the axis, an annular channel being prescribed (as in conventional turbines) to exclude the buckets or blades from the cylindrical space lying close around the shaft where the energy converting parts of the rotor would have too low a velocity to be effective. If this device is to be a steam turbine, the pressure is high at the entrance end and as low as may be feasible at the discharge end. Since steam will expand greatly in passing from high to low pressure states, while giving up energy, the section of the channel is made to expand over a wide range from the high pressure end to the low pressure end. As the steam passes through the channel, the static pressure impresses itself on a large number of radial buckets which are carried on the rotor, arranged in many stages, these buckets being set with a certain pitch with reference to their plane of rotation. The buckets are thus arranged circumferentially in stages, groups, or rings, along the length of the rotor.

The effect of introducing this pitch is for the axial pressure on the buckets to exert a component of force on the buckets in the direction of their rotation, which makes the necessary elements for the transfer of power from the steam to the buckets. The amount of power developed upon any one bucket is exceedingly small, but the aggregate from the very large number of buckets is a close approximation to the amount of all the recoverable power in the steam, with a well designed rotor.

By extracting power from the pressure of the steam rather than from its velocity, as is usual with steam turbines, much waste of energy is avoided in the elimination of the tortuous passages through the channel. This results from the elimination of the high velocity jets.

Since the pressure on the steam is not allowed to develop the natural spouting velocity, it is necessary that the buckets be strong enough to withstand the static pressure difference from stage to stage. Since the buckets in a ring or a stage are spaced apart for the steam to pass between in an axial direction, the steam pressure and resulting velocity in these gaps must also be restrained. "Axial" is in the sense of lying in a plane containing the axis, and "axial," like "straight through," precludes transverse deflections of the stream by stationary guide vanes, such as are characteristic of conventional steam turbines. This is accomplished by making use of many and narrow buckets and high bucket speeds, with motion directed directly across the path of the steam, so that during the resulting very brief period during which any space between adjacent buckets in the same stage is open for axial flow, the acceleration of the slug of steam moving in this space will be too small to cause any deleterious action. During the brief period while any such slug of steam is accelerating in any one space, its momentum is accumulating, even if the added velocity attained appears very small. As soon as the next bucket appears in its rotation, this slug of steam gives up its accumulated momentum to that bucket, which rotates with a definite pitch and at constant speed.

As a result, the buckets, even tho they occupy only a minor fraction of the circumference, have impressed upon them the full static head, as tho they covered the whole circumference. As explained, the momentum acquired by the accelerating steam while freely moving carries the static pressure from the open spaces to the bucket.

To avoid an infinite pressure when a flat bucket cuts across the slug of steam, the rising high pressure will cause slipping of enough steam around the trailing edge of the bucket from the upstream to the downstream side or a sufficient amount of local compression to relieve the peak of the pressure, with the result that the transfer of the momentum of the slug to the blade is progressive, and is distributed over the whole period during which the bucket is restraining the slug. By curving the buckets, so that the angle of attack at the nose is zero and so that the steam delivered at the trailing edge is in a prescribed direction, eddies are avoided. The overall result is that these buckets in any stage, in effect, carry the full difference in pressure between this stage and the next even though they span only a minor fraction of the channel cross section. The pressure per square inch in the buckets will be as much greater than the average unit steam pressure between stages as the total channel cross section is greater than the total area of the buckets in that stage.

If the rotor is to develop a given amount of power, the necessary total weight of steam required must pass through the channel, and the channel is so laid out that this passing of the chosen weight of steam will occur at a suitable velocity at each point according to the purposes of the designer. At the same time the varying cross section is adjusted to the volume of the steam from stage to stage, as it grows with the reduced pressure and the extraction of power.

The control of the average velocity of the steam at any stage is secured by the adjustment of the pitch of the buckets in relation to their speed, and so on, stage by stage. Since the ratio of the velocity of the steam axially to the speed of the bucket determines the pitch and also determines the magnitude of the component of the steam pressure on the bucket acting in the direction of bucket movement, the efficiency would be 100% were there no losses.

Since there is a certain difference in velocity of the axial steam flow at the leading edge of the bucket from that at the trailing edge, due to the transfer to the intercepting buckets of the momentum acquired momentarily by the various slugs of steam, these buckets are slightly curved, as already stated, so that the steam may be left at the trailing edge with the actual velocity desired by the designer. The buckets are preferably as thin as mechanical considerations permit, so as to reduce steam losses.

On account of the pitch of the blades, there may be in some cases a certain component of the steam pressure on the buckets parallel to the bucket surface tending to cause rotational movement of the steam in the channel, but as the duration of this tendency is very brief, the actual accomplished rotational movement will be negligible.

After the passage of the blade at any point there is an axial flow of steam through the open space which suppresses any previously acquired rotational velocity. In case it should be desirable, stationary curved guides or baffles may be inserted between bucket stages (as seen between the last two stages in Figure 1) to straighten out any such rotational velocity. This is accomplished substantially without loss of energy.

As stationary guide vanes or nozzles are an essential part of all types of turbines heretofore known, and as my invention requires no such guide vanes, it may be explained that these guide vanes as heretofore used have the essential function of accelerating the steam from the stage ahead to form predominantly tangential jets. While I may provide baffles or guide vanes, these are structurally and functionally different from those heretofore used. As seen in Fig. 3, they are not nozzles, but are slightly curved vanes, almost parallel to the axial flow; they cannot serve for the formation of tangential jets, and their purpose, as already explained, is to bring back into the axial direction any small divergence from that direction that may have accumulated over a number of turbine stages from minor imperfections in the operation. No change in steam velocity or production of kinetic energy occurs by reason of the presence of these vanes.

To use this rotor as a pump or compressor for steam or other vapor or for gas, it is necessary only to reverse the direction of rotation by supplying the necessary power to the rotor, and to introduce the steam at the low pressure end. The pitch of the buckets will maintain an average velocity of flow corresponding to the bucket speed. All the way through the channel, the pitch at each stage and the length of buckets will be adjusted to the velocity and volume of the gas or vapor appropriate to that stage. The received steam or gas will be delivered at a higher pressure than that at the start, according to the design.

There will be a certain pulsation in such axial flow, due to the losing of momentum by the steam against the head pressure in the free space between buckets and its reestablishment by the next bucket. The buckets are slightly curved so as to discharge the steam upstream at a slightly higher velocity than the velocity at which it was received.

Fixed curved guide vanes may be used between stages to straighten out any lateral components of velocity that may develop; especially in the low pressure stages. All this is merely the reversal of the turbine actions and requires no further explanation. The low skin friction, the avoidance of spouting velocity in the steam and the fact that the steam passes almost directly through the channel are great advantages. The flow of steam around the trailing edge of the bucket from the pressure side to the suction side to cushion, when necessary, the shock of the locally accumulated momentum on the blade involves only a very small portion of the steam.

High bucket speed, and proper width of buckets for the fall or rise of pressure between stages and the axial steady flow, coordinated with the space between buckets are important. The spacing of the stages, one from another, controlling the axial rate of fall of pressure, is also of importance. The number of stages offers a wide range of choice. The fact that the steam passes straight through means lower velocities for the same length blades and less losses than for the conventional type.

The saving potentially possible over conventional turbines is shown by the very great discrepancy between the actual and the ideal "Carnot cycle" efficiencies.

It is unimportant that the clearance between the steam channel and the end of the buckets be close, because the intermittent, impact type operation causes the checking action to control a wedge shaped body of steam both upstream and downstream of the buckets, extending to the outer wall of the channel.

As a matter of fact the local special type of flow at the buckets, with its intermittency, extends only a short distance upstream and downstream from the blades, leaving the flow practically steady elsewhere between stages.

The changes of axial velocity are very abrupt, but are of small numerical amplitude. The forces involved may be high, giving good power output with a minimum of eddy losses. As the areas of the buckets are small, relatively heavy unit stresses do not require thick buckets, the thin bucket being advantageous.

This new rotor is available over a very wide range of steam pressures and also over a wide range of density in the fluid.

In cases of heavy starting torque, it may be necessary to assist the turbine in starting, as the steam flow is not normal when the rotor is stationary or moving slowly. This may be done in the conventional manner through the generator driven by the rotor, it being used as a motor.

To permit a clear and definitive descriptive statement of certain elements of applicant's claims, the following definitions are set up for certain terms used therein.

"Compressible fluid."—This term is to be taken to include any compressible gas or vapor.

"Unobstructed."—By this term applied to a gas or vapor channel or passage, is meant the condition of the channel such that the gas or vapor is free to flow from the high pressure to the low pressure, limited only by the spouting velocity (velocity due to the free action of the pressure) affected by any friction in the passage. Means for checking or deflecting the flow are excluded, except of course when introduced as elements in a claim, separately specified.

"Axial," "in line with axis."—By these terms as applied to flow of gas or vapor is meant flow confined to a fixed plane passing through the common axis of the shell and the rotor; it permits small, undesired and unavoidable deviations, due to exigencies of design or construction, such as the momentary eddies in the gas or vapor caused by the blade torque reaction, but excludes purposeful deviations or deflections, such as are caused by the nozzles of the conventional steam turbine.

"Crossing and blocking."—Applied to the flow of gas or vapor through buckets in a channel, this term requires that the bucket cross and momentarily block the path later followed by the gas or vapor, as distinguished from the action of nozzles and buckets in conventional turbines, which have the function of deflecting the flow as it would naturally flow through the channel. The distinction is more analogous to the difference between a red light temporarily stopping cars in their path, and the detour sign, directing traffic over a different path continuously, without interruption.

The nature and some of the advantages of this apparatus having been shown, it will now be described more in detail in connection with the accompanying drawings in which Fig. 1 is a longitudinal cross section of half of the cylinder and rotor on the line I—I of Fig. 2; Fig. 2, a cross section normal to the axis on the line II—II of Fig. 1; Fig. 3, a diagram showing the arrangement of blades or buckets; Fig. 4, a single bucket in plan; Fig. 4(*a*), a partial elevation thereof and cross-section on the line IV—IV in Fig. 2; and Fig. 5, a needle type steam regulating valve for governing purposes, shown in section on its longitudinal axis.

In Fig. 1, which is an illustration of a steam turbine adapted to capacities up to 50,000 kw. or higher, and for steam pressure of upwards of 500 pounds and temperatures upwards of 800° F., the buckets range from ¼ inch wide or less, and 1 inch long or less, at the entrance, to buckets several inches wide and several feet long at the discharge, these being mounted in parallel circular rows for rotation within and spread along the axis of the cylinder **1**, which is circular in section and locally strengthened by ribs, **1***a*, **1***b*, **1***c*, **1***d*. This cylinder is closed at the high pressure end by the head **2**, and the steam distributor **3**, forming a part thereof. The other end of the cylinder at **4** is open to the condenser, which condenser is not shown, as the particular form of condenser is not a part of this invention.

The rotor **5**, is carried on the shaft **6**, and consists of a series of bucket-carrying discs or rings **7**, **7**, fast on the shaft. These discs have hubs **7***c*, **7***d*, **7***e*, **7***f*. As this rotor may have a speed as high as 3,600 R. P. M., I prefer to form these discs as shown, thickened near the hub on account of the very large centrifugal forces. This particular feature of disc is not part of this invention. The buckets carried on the rim of the discs are shown at **8**, **8**, **8**. Only sample buckets and discs are illustrated as all are of the same type. They extend, located at intervals, the full length of the channel. The discs are provided with collars **9**, **9**, at their circumferences, meeting collars on the adjacent discs, so that these collars all taken together form a smooth cone inside the main cylinder, this being the inner wall of the steam channel **10**, the cylinder **1**, being the outside wall. This construction results in a tapered annulus, with a very small width at the high pressure end, a gradually increasing width as the pressure drops, and the maximum width at the condenser end of the channel. I have shown a certain length at the low pressure end of the steam channel as of constant width at **11**. This is here shown as an expedient, in case the strength of material or other limitation would not permit longer blades for these last stages.

At **12**, I show a fairing at the end of the series of discs to ease the flow of steam to the condenser. I indicate a customary labyrinth seal for high pressure steam at **13**, and a thrust bearing at **14**. The ring type or annulus port or nozzle for the supply of steam to the first stage is shown in section at **15**, as a part of the steam distributor **3**. At this point the steam is given the desired entrance velocity and a proper axial direction of flow for its passage through the steam cylinder. This is accomplished by properly proportioning the diameter and width of the opening, and curvature of the nozzle walls. The nozzle is itself in the form of a ring, coaxial with the channel.

It is desirable to use a shrouding or "tire" **16**, Fig. 2, securing the ends of the larger buckets to prevent vibration and to maintain the proper angular position. As the construction of this shrouding is not part of this invention, it is not further illustrated.

Located between the discs **7***f* and **7***f*, Fig. 1, are shown fixed guide vanes or deflectors **17**, arranged around a circle, just as are the buckets **8**, **8**, except that the guide vanes are fast in a groove in the cylinder **1**, as shown at **18**. The physical construction, as far as mounting is concerned, may be similar to that of the fixed buckets in a normal steam turbine. Their purpose is to straighten out any rotational flow introduced by the pitch of the buckets or otherwise, as explained.

In Fig. 2 is shown a cross section of the cylinder taken at the line II—II, where the parts shown are marked as in Fig. 1. Only one ring of blades is shown and all are alike and equally spaced on each ring.

In Fig. 3 is shown the arrangement of a group of buckets located in four consecutive stages.

At **20**, **20**, are arrows showing the direction of the axial flow of steam, and **20**–*a*, **20**–*a*, the direction of the movement of the buckets. The preferred relative setting of the buckets in the successive stages as they rotate is shown, the buckets being **21**, **22** in the 1st stage; **23** in the 2nd; 24 in the 3rd; and **25** and **26** in the 4th stage; other buckets not shown complete the ring at any stage. The space between buckets in the same stage is twice the width of the buckets and the buckets in adjacent stages are staggered so that no open path is free more than three stages long. However, this exact arrangement of buckets is by no means necessary.

The action may be explained as follows. The buckets are assumed to be moving to the right (with respect to the direction of steam flow) at a speed about ten times, more or less, the average axial stage steam velocity, Fig. 3. Consider the instant after the bucket **22** has passed with the arrow from the position shown. The pressure difference impressed on the steam between the first and second stages may be taken, for purposes of illustration, as 5 pounds per square inch for this stage, and the average steam velocity at this stage along the axis as 95 F. P. S. If the space between stages be taken as 4 inches long, a column of steam 4 inches long just released by the bucket **22**, will accelerate under 5 pounds pressure until the next bucket **21**, reaches the column and cuts across its path. This bucket then absorbs such acceleration velocity as may have been attained between stages by the column, as has been explained. In this case, the time required for the bucket **21**, to cross the open space will be $\frac{1}{8,550}$ sec. if this space be 2 in. wide. During that interval this column of steam will attain an acceleration velocity of about 112 F. P. S. This, taken with the above average axial velocity, will give a momentary velocity on the arrival of the bucket **21**, of approximately 95+112/2 or 151 F. P. S. The entrance edge of the buckets in these stages is given a pitch equal to the angle of the relative motion between the steam and the bucket, that there may be no eddy. The curvature of the bucket, see Fig. 4, is such that the steam is decelerated during the passage of the bucket to the point where it has such a velocity as to secure the specified average velocity for the stage. In this case the leaving steam velocity would be about 95−112/2=39 F. P. S. Similarly with the action of the buckets of the other stages.

The useful torque force on the buckets in the direction of rotation will be approximately that of the difference of steam pressure between the two adjacent stages, times the tangent of the effective pitch angle of the bucket taken to the plane of rotation, that is the tangent of the relative motion angle. These figures will be slightly modified by any rotational flow and by friction and eddy losses. The very small loss from these causes as well as the great reduction in bucket friction surface, compared to normal turbines, will cause a great reduction in the large discrepancy below the Carnot cycle efficiency suffered by those turbines, and this saving will be more important because any percentage increase in output is equivalent to the saving of the same percentage in the total cost of plant; boilers, buildings, coal storage, water handling, equipment, etc.

In Fig. 4 is shown in plan, enlarged, one of my buckets. The suction side of the bucket is shown at **27**, and the pressure side at **28**. I prefer to have an approximately circular surface in the suction side, so that the steam may be kept in contact with the bucket surface without too much reduction in pressure. The bucket is given sharp edges and just enough thickness in the central portions **30**, to give mechanical strength and stiffness. The leading edge may be slightly rounded, where varying values of entering steam velocities are expected. The arrow **29**, shows the direction of rotation. The amount of the curvature and the effective pitch will vary from stage to stage as the relation of the velocity of the steam to the bucket speed varies. Fig. 4–*a* shows the end of the bucket of Fig. 4 upset to form a mortise. The mortise is shown at **31** and a locking "Dutchman" at **32** to secure the bucket.

Fig. 5 shows a governing throttle in the form of a needle valve to control the speed of the turbine by limiting the amount of steam supplied as the output varies. This is intended to act, either to "wire-draw" the steam by gradually and partially closing the opening between the needle and the seat, or intermittently, so that the valve will rest either open or closed, but not at an intermediate point. This choice is controlled by a dash pot. The advantage of the intermittent control is that, when operating, the rotor is always fully loaded. There is some loss of efficiency with the simple wire-drawing to give less flow, as the distribution of velocities and stage pressures in the rotor will be somewhat disturbed at certain loads. The average velocity will still be established by the pitch of the buckets as long as the speed of rotation is kept constant. Lower pressures will reduce the output.

In this figure, **33** is the nozzle opening where the steam is delivered, **34** is the movable needle valve, **35** is the seat on the valve wall and **35**–*a* is the coacting surface on the needle. A leaf spring **36** is held fast on the valve stem by the nut **37**, and at its ends rests in a slot in the surrounding wall. This spring is so adjusted as to substantially balance the steam pressure when the valve is closed, but will still retain enough excess displacement to cause the needle **34**, to close the valve quickly when the magnet core **38**, is released by its magnetizing coil **39**. The valve spindle moves in bearings **40** and **41**, and is controlled by the dash pot **42**. The dash pot is adjusted by the vent **43**. When the valve is to have the intermittent action, the dash pot vent **43**, is open. To get gradual action for wire-drawing, it is given a certain limited vent to adjust the speed of action to the desired regulation. As the load on the rotor varies, up and down, the valve lagging behind slightly will then operate in intermediate positions. The magnetizing coil **39**, is actuated from a centrifugal governor of the usual type, not shown, through an electric circuit and electric contacts, closed by the governor, also not shown. The needle and moving parts are made very light to give quick action and to avoid serious hammer. At **44** is seen the opening through which steam is fed to the nozzle proper. By these means this steam turbine may be given any desired type of speed control.

While this invention has been illustrated more particularly for a high pressure steam turbine, it is just as applicable to other types of turbines, to the use of gases in place of steam and for use as a compressor for any of these fluids. I have pointed out the effectiveness of the intermittency of the bucket action; the relatively low steam velocity; the high bucket speed with relation thereto; the absence of nozzle type, or reaction type, steam jets between stages; the direct, substantially axial flow of steam; the restricted or absent stationary guide vanes; the staggering of pitched buckets in consecutive stages; the true, smooth surface, expansion channel; the generous end clearance for the buckets; the importance of proportioning of the average and acceleration steam velocities from stage to stage; and other novel features. It is desired to make the following claims as broad as the prior art will permit.

I claim as my invention:

1. Apparatus for the interchange of mechanical power to and from a stream of compressible fluid flowing under a difference of pressure comprising a hollow tapered shell of circular cross-section open at both ends, a rotatable member concentrically located therein and spaced therefrom to form therebetween an annular tapered passage extending from end to end thereof without obstruction except for radial buckets on said rotatable member, and circular end closure members for said shell coaxial with said rotatable member each with an unobstructed annular opening registering with said tapered passage, in combination with a plurality of rings of impact buckets secured to said rotatable member and spaced apart axially along said tapered passage constituting means for interchanging energy with said stream

while maintaining axial direction of flow thereof, each bucket ring consisting of narrow flattened buckets separated by channel providing spaces between adjacent buckets of greater width than the width of the buckets in said ring, the open spaces between said buckets permitting passage of the greater part of said stream and each immediately following ring of buckets intersecting said stream in said open spaces, said buckets being set at a pitch angle to their plane of rotation insufficient to deflect said stream from its axial path when rotating at a speed substantially greater than that of said stream, the resulting impact between said buckets and stream serving to change the velocity of said stream in part causing axial pulsations therein, and producing direct pressure on said buckets having a circumferential component, whereby rotational energy is interchanged between said buckets and stream in accordance with changes in the axial pressure and movement of said stream.

2. Apparatus as set forth in claim 1 wherein the buckets of each row have a curvature concave to the higher pressure end and their entering edges have a pitch angle whose tangent is the ratio of the axial velocity of said stream at said row to the peripheral speed of the buckets of said row.

3. Apparatus as set forth in claim 1 wherein the buckets in successive rows are disposed in axial alinement with the spaces in the preceding rows, respectively, said buckets in two successive rows predominantly blocking said annular passage.

4. Apparatus as set forth in claim 1 wherein the spaces in each row are of the order of twice the width of the buckets in the same row and three rows of buckets subtend the entire cross-sectional area of said annular passage.

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