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(54) VOLUME-EXPANSION STEAM ROTOR ENGINE.

(57) A volume-expansion steam rotor engine with a spherical rotor consisting of a disk-shaped diaphragm (2) hingedly connected to mutually perpendicular blades (3, 4) located on the opposite sides of the diaphragm (2) and forming, together with the latter, inside the casing, working chambers of variable volume. Two volume dosers (6, 7) are provided for dosed feeding of the working medium each of them being connected by its input to a main (32) for feeding the working medium, and by its output to a corresponding section of the working machine in a zone where the working chambers have a minimum volume.

ROTARY STEAM EXPANSION ENGINE

Technical Field

The invention relates to the mechanical engineering, and in particular, to rotary steam expansion engines.

The invention may be most advantageously used in vehicle engines ensuring reverse movement, movement by inertia and engine braking.

The invention may be used in engines in various power producing plants.

Background of the Invention

It is generally known to use both steam piston engines and turbines as steam power producing plants and engines for vehicles. Advantages of steam engines, especially in urban use, in comparison with internal combustion engines, is the possibility of using any fuel, absence of no-load operation and negative effects on ecological parameters of the environment.

The main disadvantages of steam pistoon engines are:

- heavy-weight reciprocating parts limiting engine speed;
- dead volumes in cylinders of steam engines lowering their volumetric efficiency;
- unwieldy reverse device because of its kinematic connection with the power transmission;
- absence of possibility of increasing adiabatic steam expansion in a single cylinder without an increase in its size while retaining piston force which materially unfluences power-to-weight ratio.

Steam turbines do not have reciprocating parts, but their employment on vehicles with a low power-to-weight ratio is rather ineffective because of a low efficiency under varying load.

The overall efficiency of an engine includes the thermal efficiency (the ratio of maximum possible work to consumed thermal energy), the mechanical efficiency (the ratio of maximum work to a work without taking into account friction losses) and the volumentric efficiency (the ratio of volume of utilized fluid to maximum volume of the working chamber).

Known in the art is a reversible air vane motor (SU, A, 435358) comprising a casing accommodating a stator and a rotor having radial slots accommodating radially movable vanes, and a metering device for supplying fluid. The stator walls have primary exhaust ports located on either side of its plane of symmetry and equally spaced therefrom and having devices for their automatic shutting on the inlet side. Energy losses are high in this motor because of the action of centrifugal forces upon the vanes with an increase in the rotor speed since the vanes move with friction with the inner surface of the stator. Misalignment of the vanes results in an additional friction in the slots. These were the reasons why the mechanical efficiency of the air motor decreased. Throttling of fluid also results in energy losses in the form of thermal losses to reduce the thermal efficiency. In view of the above consideration, the overall efficiency of the air motor is rather low. In addition, the reverse device in this air motor is very complicated as it has a large number of valves operating only in the presence of fluid. These disadvantages also impair weight and size characteristics of the motor. In case such a motor is used in a vehicle, fluid is consumed under braking conditions, hence energy is consumed. Engine braking without fluid is impossible. The design does not provide for a rotor speed control with forward and reversed rotation thus limiting application of the air motor as a steam engine for a vehicle.

Known in the art is a reversible rotary engine (SU, A, 1298407) comprising a cylindrical stator having two manifolds (with inlet and exhaust ports), each directly communicating with the nearest minimum volume chamber and individually connecting, via controlled valves, to a fluid supply line and, via a spool valve, to an exhaust line, a rotor having radial slots mounted eccentrically with respect to the stator, and radially movable vanes installed in the slots and dividing the interior space of the stator into working (expansion and displacement) chambers communicating with the manifolds via exhaust valves. There is also provided a fluid metering device and power takeoff shafts connected to the rotor.

This prior art engine is deficient in a comparatively low efficiency because of a low mechanical efficiency owing to energy losses through friction of the vanes with an increase in engine speed and their misalignment under the action of fluid pressure, a decrease in thermal efficiency because of thermal energy losses in operation of forward and reverse throttling valves provided in the metering device, and a decrease in the volumetric efficiency as it is not possible to ensure a reliable sealing at the end faces of the vanes in the slots. The decrease in the mechanical, thermal and volumetric efficiency determines a rather low overall efficiency. In addition, in view of a comparatively low ratio of the stator diameter to the rotor diameter, working chambers of the engine have small capacities so as to impair the power-to-weight ratio of the engine.

Summary of the Invention

The invention is based on the problem of providing a steam expansion engine, design of a rotary machine, and a system for dluid supply so as to enhance efficiency and power-to-weight ratio for an effective application as an engine in a vehicle.

The invention essentially consists in that in a rotary steam expansion engine having a casing accommodating a rotary machine connecting to a line for a metered supply of fluid and a line for control and exhaust of fluid, according to the invention, the rotary machine comprises a known per se rotary machine having a spherical rotor formed by a diaphragm in the form of a disc-shaped partition mounted for rotation about the center of a spherical interior space of the casing and defining a pair of mutually isolated compartments, and a pair of vanes pivotally connected to the diaphragm on either side thereof in two mutually perpendicular diametrical planes and defining with the diaphragm and inner surface of the casing sealed variable-capacity working chambers, each vane being rigidly secured to a respective power takeoff shaft, the axes of the shafts extending at an angle with respect to each other and intersecting each other at the center of the spherical interior space, and in that there are provided two volumetric metering devices, each having an inlet communicating with the fluid supply line and an outlet communicating with a respective compartment of the rotary machine in a zone where working chambers of this compartment are of minimum capacity.

Prior art reversible rotary engines with a spherical rotor have unbalanced members - vanes. During rotation of the rotor, the vanes are pressed by centrifugal forces against the inner surface of the casing, and energy losses to overcome friction occur along lines of contact. The same losses occur along lines of contact between the vanes and the rotor. In operation of the engine, the vanes are subjected to a pressure differential, and the resultant forces are transmitted from the vane to the rotor through their contact, but the vane reciprocates with respect to the rotor in a misaligned position. Unlike the prior art, the engine according to the invention is free from the abovementioned disadvantages. Its advantages reside in an increase in the mechanical efficiency. Fluid supply and engine speed control in the prior art were carried out by means of a throttling valve. This control method is accompanied by inevitable losses during throttling of a vaporous fluid because of a drop of temperature and pressure. The engine according to the invention is free from this disadvantage. Control in this case is effected by the volumetric metering of fluid, and a batch metered by the metering devices has parameters fully identical with the initial parameters of fluid. Therefore, the fluid carriers substantially all initial energy without losses. As work is performed upon the adiabatic expansion of fluid only on the account of its internal energy, the thermal efficiency of this engine is increased. The design of the engine according to the invention does not involve linear mating portions between working chambers which mate along a spherical surface so as to allow efficient sealing members to be used which are characterized by droplet leakage only. Accordingly, the engine according to the invention has an increased volumetric efficiency.

The combination of structural features according to the invention makes it possible to have a larger capacity of working chambers with the same volume and weight of the engine as in the prior art so as to allow a higher output at the power takeoff shafts to be achieved, hence, to enhance power-to-weight ratio.

It is preferred that each volumetric metering device have a body with a cylindrical bore of a diameter substantially equal to the diameter of a power takeoff shaft extending therethrough, the shaft body having an interior space accommodating a piston which divides this space into two parts, radial ports being made in the shaft wall which are spaced axially along the shaft and diametrically opposed to each other, the body of the metering device having two pairs of radial ports open into the interior space of the cylindrical hole, the ports of each pair being axially spaced along the shaft at the same distance as the radial ports of the shaft, the pairs of ports being diametrically opposed to each other and communicating with fluid supply and control lines.

The piston is preferably made of two parts defining a space therebetween and a passage is preferably provided in the shaft wall and in the body of the metering device to establish permanent communication of this space with the interior space of a control cylinder.

These structural features make it possible to simplify the engine control system and to provide working chambers using the intire volume of the casing of the rotary machine so as to improve power-to-weight ratio of the engine.

Brief Description of the Drawings

The invention will now be described with reference to its specific embodiment illustrated in the accompanying drawings, in which:

Figure 1 shows a sectional view of a rotary steam expansion engine according to the invention with a schematically shown control system;

Figure 2 is a diagrammatic view of a rotor of a rotary machine.

Best Mode for Carrying Out the Invention

A rotary steam expansion engine according to the invention comprises a casing 1 accommodating a known per se rotary machine. The rotary machine is a machine having a so called "spherical" rotor. The rotor is formed by a diaphragm 2 in the form of a disc-shaped partition mounted for rotation about the center of a spherical interior space of the casing 1. The diaphragm 2 defines in the interior space of the casing a pair of mutually isolated compartments. Vanes 3, 4 are provided on either side of the diaphragm 2 and are pivotally connected thereto to extend in two mutually perpendicular diametrical planes. The rotor is diagrammatically shown in Figure 2.

Each vane 3, 4 is a part of a sphere defined by a pair of planes intersecting at an acute angle, the line of intersection of the planes being drawn through the sphere diameter. The inner surface of the casing 1 mating with the outer surfaces of the diaphragm 2 and vanes 3, 4 is also spherical.

The vanes 3, 4 (Figure 1) are rigidly secured to power takeoff shafts 5, 5' having their axes a, b (Figure 2) extending at an angle α with respect to each other and intersecting each other at the center of the spherical interior space.

There are also provided volumetric fluid metering devices 6, 7, which in the embodiment shown in Figure 1 are accommodated in the power takeoff shafts 5, 5'. It is obvious that the metering devices 6, 7 may also be provided outside the shafts 5, 5' , and their preferred design will be described below.

The pivotal connection of the diaphragm 2 to the vanes 3, 4 may be as follows. For example, cylindrical projections 8 (shown conventionally) are provided on the peripheral surfaces of the diaphragm 2 in such a manner that their longitudinal axes of symmetry intersect each other at the center of this plane at right angles to each other. Concave cylindrical surfaces mating with the surfaces of the respective cylindrical projections 8 are provided on the end faces 9 of the vanes 3, 4. Therefore, the vanes 3, 4 are provided on either side of the diaphragm 2, and the latter is capable of moving with respect thereto. The power takeoff shafts 5, 5' are mounted to extend along axes of symmetry of the vanes 3, 4 and pass through holes of the casing 1.

The casing 1 has a pair of inlet ports 10, 11 connecting to passages 12 for fluid supply and four exhaust ports connecting to an exhaust manifold. The inlet ports 10, 11 are provided on either side of the diaphragm 2 at points corresponding to minimum capacities of the working chambers. The exhaust ports are provided pairwise on either side of the diaphragm 2. The first port of the pair is located at the leadingend of a maximum-capacity working chamber and the other port is located at the trailing end thereof. If exhaust ports 13, 14 located at the trailing end of the maximum-capacity working chambers they are forward movement exhaust ports, the other two ports (not shown in the drawing) located at the leading end of the maximum-capacity working chambers (on either side of the diaphragm) are reverse exhaust ports. The exhaust ports for forward movement and reverse are part of fluid exhaust devices 15 which, in addition, comprise a twin forward movement valve 16 and a twin reverse valve (not shown in the drawing).

The twin forward movement valve 16 has an electromagnetic actuator and a hydraulic actuator in the form of an industion coil 17 and a hydraulic power cylinder 18 having its piston mechanically coupled to the valve 16. The induction coil 17 has the same coupling. The electromagnetic actuator ensures rapid opening and shutting of the exhaust ports and the hydraulic actuator performs the same functions slowly and smoothly so as to vary their throughput capacity.

The location of a coil 19 and a hydraulic cylinder 20 for actuation of the twin reverse valve is conventionally shown in Figure 1. In addition, reverse ports 21, 22 are provided in the casing 1 of the engine on one (any) side of the diaphragm 2 and on either side of the vane 3 or 4.

The engageable surfaces of the parts such as the outer surface of the diaphragm 2, outer surfaces of the vanes 3, 4 inner surface of the casing 1, surfaces of the cylindrical projections 8 and concave cylindrical surfaces of the end faces 9 mate with one another over the whole area so as to allow effective sealing to be provided (not whosn) to enhance sealing of working chambers 23, 24, 25, 26 (Figure 2).

The working chambers 23, 24, 25, 26 are defined pairwise by the vanes 3, 4 , diaphragm 2 and inner surface of the casing 1 in respective compartments of the casing defined by the diaphragm 2 and have variable capacity which varies during rotation of the vanes 3, 4.

In a zone where the working chambers 23, 26 or 24, 25 (Figure 2) of a respective compartment have the minimum capacity, this compartment communicates with the outlet of the respective metering device 6 or 7.

In the description of the metering devices 6, 7 that follows below identical parts of the metering devices will be shown at identical reference numerals for the sake of simplicity as the metering devices 6, 7 are of identical design.

Each metering device 6, 7 has a body 27 rigidly secured to a bearing assembly of the shaft 5 or 5' in the engine casing 1. The power takeoff shaft 5 or 5' extands through an axial bore of the body 27 of the metering device for rotation with respect to the body of the metering device. An interior space 28 extending axially along the shaft is provided in the body of the shaft 5 or 5'. A pair of free pistons 29 are provided in the interior space of the shaft.

A pair of ports 30 for fluid supply are provided at the ends of the interior space 28, adjacent to the end walls thereof. They are spaced at 180° C with respect to each other. A control port 31 is provided in the intermediate part of the peripheral wall of the interior space 28 of the shaft.

The ports 30 of the body 27 of the shaft connect to lines 32 via fluid supply and discharge ports 12, 12' and to a control line. The control port 31 connects to the line 33 through an annular groove of the body 27 of the metering device (not shown). Communication between the ports 30 and the line 32 and passages 12 occurs when the shaft is in predetermined positions with respect to the body 27. The control port 31 permanently communicates with the control line 33 owing to the provision of the annular groove. The line 33 communicates with a hydraulic movement control cylinder 34 having a piston with a spring mechanically coupled to a movement pedal 35. In order that the piston of the cylinder 34 be not subjected to the influence of pressure differentials in the line 32, its piston chamber is connected to the latter.

A braking system has a brake pedal 36, a hydraulic brake cylinder 37 having a piston mechanically coupled to the brake cylinder pedal 36. The cylinder 37 is hydraulically coupled to a spool valve 38 which, in turn, is connected to the actuator cylinders 18, 20 of the twin forward movement valve 16 and twin reverse valve.

A reverse system has a reverse lever 39 mechanically coupled to a piston of the spool valve 38 and to a piston of a reverse spool valve 40. The spool valve 40 connects, via a valve 41, to the fluid supply line and, on the other side, to the reverse ports 21, 22. An opening device of the valve 41 is mechanically coupled to the movement pedal 35 and to a device for its automatic shutting (not shown). The induction coils 17, 19 are connected to an electric power supply via a switch 42 having its contact plate mechanically coupled to the reverse lever 39, and switches 43 having their contact plates mechanically coupled to the movement and brake pedals 35 and 36, respectively.

It is preferred that in an embodiment of the engine designed for use in a vehicle the piston 29 be made up of two parts defining a space therebetween. This space preferably communicates through the control line 33 with the interior space of the control cylinder 34. This interior space communicates through the passage 31 in the wall of the shaft 5 or 5' and body 27 of the metering device 6 or 7 with the interior space of the control cylinder. This construction facilitates the engine control and enhances its reliability.

In the description of operation of the rotary steam expansion engine shown in Figure 1, reference will also be made to Figure 2 illustrating the formation and operation of the varying-capacity working chambers 23, 24, 25, 26. The engine functions in the following manner. The initial position of the engine is when it is stopped: the movement pedal 35 and the brake pedal 36 are free, both switches 43 are opened, the reverse lever 39 is in the "Forward" position in which the coil 19 of the electromagnetic actuator of the reverse exhaust valve can be energized. The spool valve 38 connects through a hydraulic line the brake cylinder 37 to the actuator cylinder 18 of the forward movement valve 16 and disconnects a hydraulic line between the brake cylinder 37 and the reverse hydraulic cylinder 20. The brake cylinder 37 is filled, and the cylinders 20 and 18 are empty. There is no fluid in the movement cylinder 34, and the fluid is available in the space between the two parts of the piston 29 which are in the interior space of the metering devices 6, 7. The parts of the piston 29 are in their limit position in which they cover the ports 30. The forward movement ports 13, 14 and reverse ports of the fluid exhaust device 15 are open, and the interior spaces of the working chambers 23, 24, 25, 26 communicate with one another through the exhaust manifold. For moving forward, the movement pedal 35 is depressed to close one of the switches 43; the electric current flows through the closed contacts of the switch 42 to the coil 19 of the twin reverse valve 19, and the exhaust reverse ports are shut off, the valve 41 is opened, and fluid is admitted through the reverse spool valve 40 to the reverse port 21. The diaphragm 2 starts moving to cause rotation of the vanes 3, 4 and power takeoff shafts 5, 5' in the predetermined direction.

The outer surfaces of the diaphragm 2 and vanes 3, 4 slide over the inner spherical surface of the casing 1, and the surfaces of the cylindrical projections 8 slide over the cylindrical surfaces of the end faces 9.

After the beginning of rotation the valve 41 is automatically shut off. Further depression of the movement pedal 35 will cause liquid to move from the interpiston space of the metering devices 6, 7 through the control line 33 to the movement control cylinder 34 so as to bring the parts of the piston 29 closer to each other. The fluid supplied through the line 32 fills the space behind the pistons in the metering devices 6, 7. Rotation of the power take-off shafts 5, 5' ensures a cut-off of a batch of fluid which depends on the capacity of the space behind the pistons, and the volume of this space depends, in turn, on the position of the parts of the pistons 29. When one of the ports 30 is brought in registry with the fluid supply port 12 in the body 27 of the metering device and the other port is brought in registry with the fluid supply line 32, the free parts of the piston 29 start displacing the fluid batch from the space behind the pistons into the port 12 and further through the inlet port 10 or 11 into the minimum-capacity working chamber. Therefore, the position of the movement pedal 35 determines the amount of the fluid batch for admission to the working chamber. When the pedal 35 is released, the piston of the movement control cylinder 34 forces liquid into the interpiston space under the action of the spring, the pistons cover the ports 30, and the fluid is not admitted to the working chambers.

The reference is now made to the vehicle movement by inertia with the engine members in the initial state. For stopping the engine the brake pedal 36 is depressed to close tie respective pair of contacts of the switch 43 and to energize the induction coil 19 of the electromagnetic actuator of the reverse movement, the twin reverse valve (not shown) being held shut. Liquid from the brake cylinder 37 is admitted through the cylinder of the spool valve 38 to the actuator cylinder 18, and the hydraulic actuator will gradually shut off, the twin forward movement valve 16. Rotation of the vanes 3, 4 is decelerated because of the back-pressure build-up in the working chambers, and the engine is stopped. The pedal 36 is released, and all elements of the engine return to the initial position.

For reverse movement, the reverse lever 39 is set to the "Reverse" position. Movement of the piston of the spool valve 38 will now connect the brake cylinder 37 to the actuator cylinder 20 of the reverse valve, and the other pair of contacts of the switch 42 will be closed. The piston of the reverse spool valve 40 establishes communication of the valve 41 with the reverse port 22. When the movement pedal 35 is depressed, electric power supply is connected to the induction coil 17 to shut off the twin forward movement valve 16, the twin reverse valve being open, and the valve 41 is opened. The diaphragm 2 and the vanes 3, 4 start moving in the opposite direction. Owing to the fluid supply through the reverse port 22, the valve 41 is shut off, and the movement pedal 35 controls the reverse movement through the metering devices 6, 7.

Fluid in this engine is in the form of a liquid supplied to the working chamber of the engine. The liquid boils in the working chamber during adiabatic expansion and performs work owing to vapour expansion.

Industrial Applicability

The invention may be most advantageously used in vehicle engines whose structural features ensure reducing of weight and dimension characteristics of the vehicle engine as a whole while maintaining high efficiency thereof.

Claims

1. A rotary steam expansion engine having a casing (1) accommodating a rotary machine connecting to a line (32) for a metered fluid supply and to a control and fluid exhaust line (33) **characterized** in that the rotary machine comprises a known per se rotary machine having a spherical rotor formed by diaphragm (2) in the form of a disco-shaped partition mounted for rotation about the center of a spherical interior space of the casing (1) and defining a pair of mutually isolated compartments, and by a pair of vanes (3, 4) pivotally connected to the diaphragm (2) to extend on either side thereof in two mutually perpendicular diametrical planes to define with the diaphragm (2) and with the inner surface of the casing (1) sealed varying-capacity working chambers (23, 24, 25, 26), each vane (3, 4) being rigidly secured to a respective power takeoff shaft (5, 5'), the axes (a, b) of the shafts extending at an angle (α) with respect to each other and intersecting each other at the center of the spherical interior space, and in that there are provided a pair of volumetric metering devices (6, 7), each having an inlet communicating with the fluid supply line (32) and an outlet communicating with the respective compartment of the rotary machine in a zone where the working chambers (23, 26 or 24, 25) of this compartment have the minimum capacity.

2. An engine according to claim, 1 **characterized** in that each volumetric metering device (6, 7) has a body (27) having a cylindrical bore of a diameter substantially equal to the diameter of the power takeoff shaft (5 or 5') extending therethrough, an interior space (28) being provided in the body of the shaft which accommodates a piston (29) dividing this interior space (28) into two parts, and in that radial ports (30) are made in the walls of the shaft (5, 5') which are axially spaced along the shaft and are diametrically opposed to each other, the body (27) of the metering device (6, 7) having two pairs of radial ports (12, 12') opening into the interior space of the cylindrical hole, the ports (12, 12') of each pair being axially spaced along the shaft (5, 5') at the same distance as the radial ports (30) of the

shaft, the pairs of ports (12, 12') being diametrically opposed to each other and communicating with the fluid supply and control lines (32, 33), respectively.

3. An engine according to claim 2, **characterized** in that the piston (29) of each metering device (6, 7) is made up of two parts defining a space therebetween, and in that a passage (31) is made in the wall of the shaft (5, 5') and in the body (27) of the metering device (6, 7) permanently connecting this space to the interior space of a control cylinder (34).

INTERNATIONAL SEARCH REPORT

International Application No PCT/SU 89/00134

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [6]

According to International Patent Classification (IPC) or to both National Classification and IPC

IPC[5]: F 01 C 3/00, G 01 F 11/00

II. FIELDS SEARCHED

Minimum Documentation Searched [7]

Classification System	Classification Symbols
IPC[4]:	F 01 C 3/00, 3/06, G 01 F 3/00, 11/00 - 11/46

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT [9]

Category *	Citation of Document, [11] with indication, where appropriate, of the relevant passages [12]	Relevant to Claim No. [13]
A	DE, A1, 2619474, (Vaguero Manuel Biedma, Jsla Cristina, Huelva), 3 February 1977 (03.02.77), figure 1	1
A	US, A, 4631011 (Roger R. Whitfield), 23 December 1986 (23.12.86), column 5, lines 28-35; column 6, lines 1-34; figure 19	1
A	US, A1, 1298407 (L.P. Proglyada), 23 March 1987 (23.03.87), column 1, lines 18-48; figure 1	1-2
A	SU, A1, 322626 (Kuznetsky nauchno-issledovatelsky ugolny institut), 11 November 1972 (11.11.72), column 1, lines 26-30; column 2, lines 1-26; the drawing	2-3

* Special categories of cited documents: [10]

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
19 December 1989 (19.12.89)	09 February 1990 (09.02.90)
International Searching Authority	**Signature of Authorized Officer**
ISA/SU	