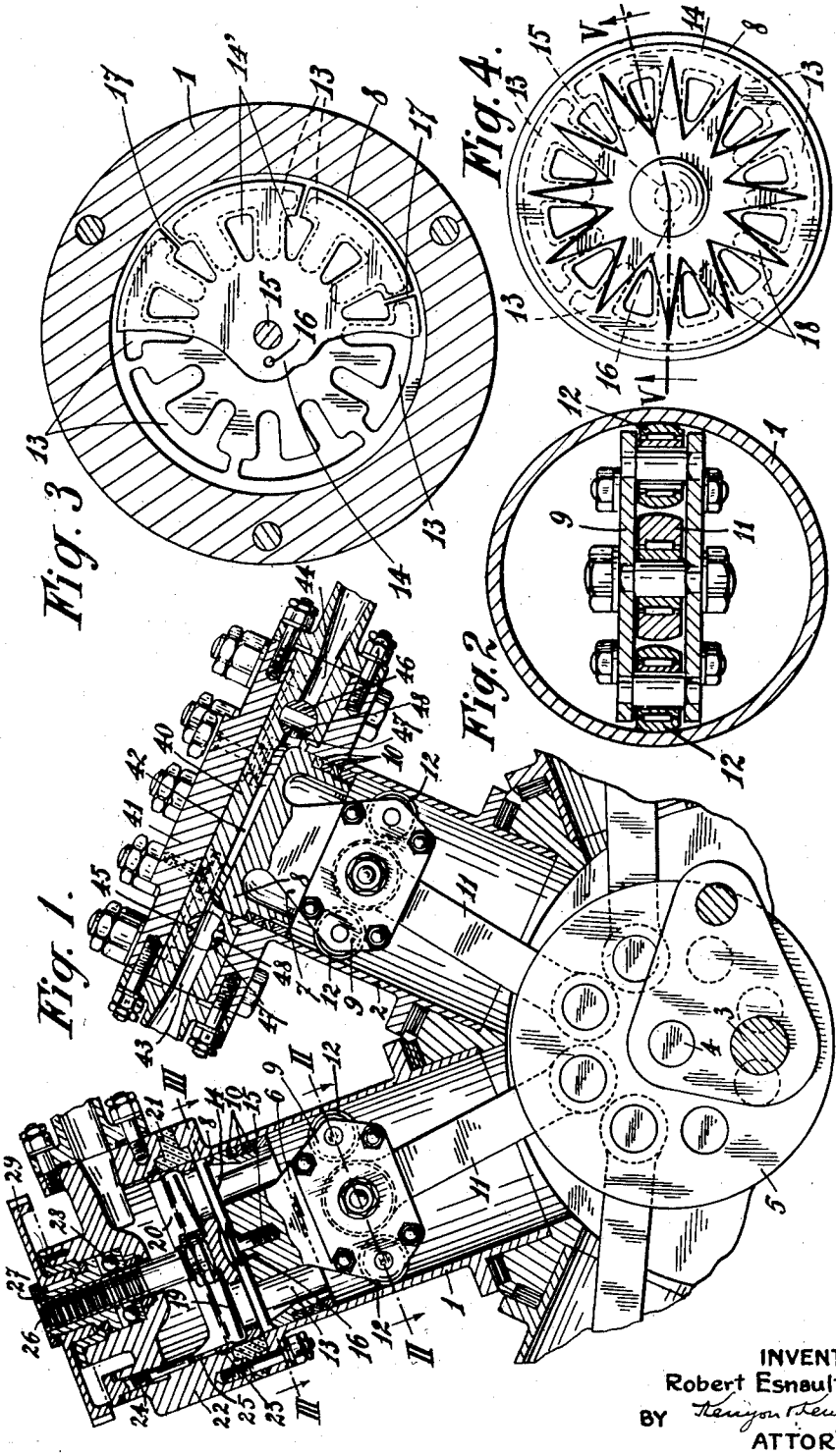
Aug. 31, 1943. R. ESNAULT-PELTERIE 2,328,439

HEAT TRANSFER APPARATUS

Filed Dec. 7, 1940 5 Sheets-Sheet 1



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HEAT TRANSFER APPARATUS

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Fig. 5

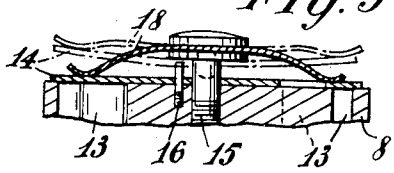


Fig. 6.

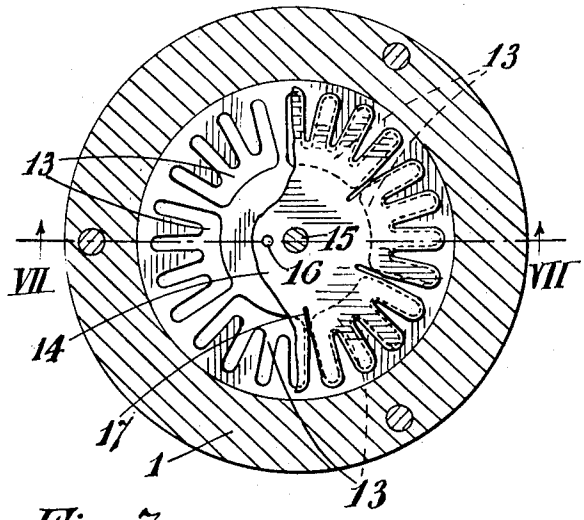


Fig. 7.

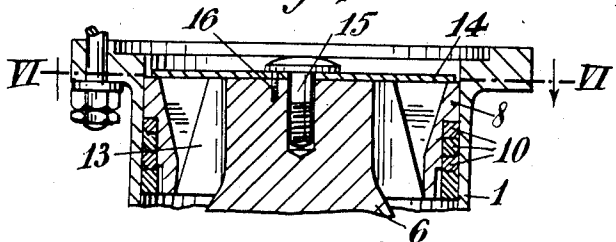
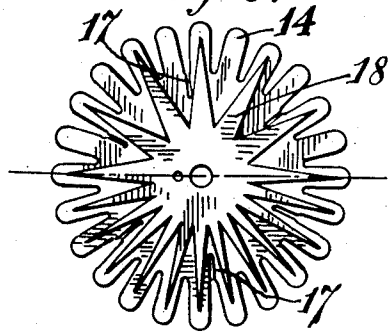


Fig. 8.



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Fig. 9.

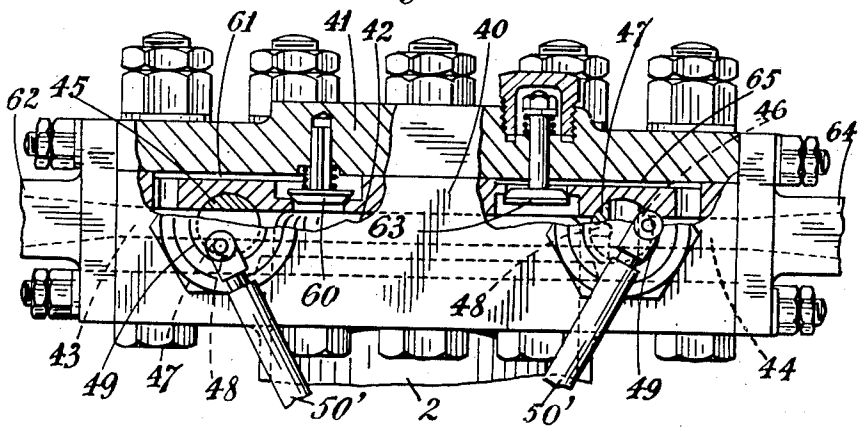
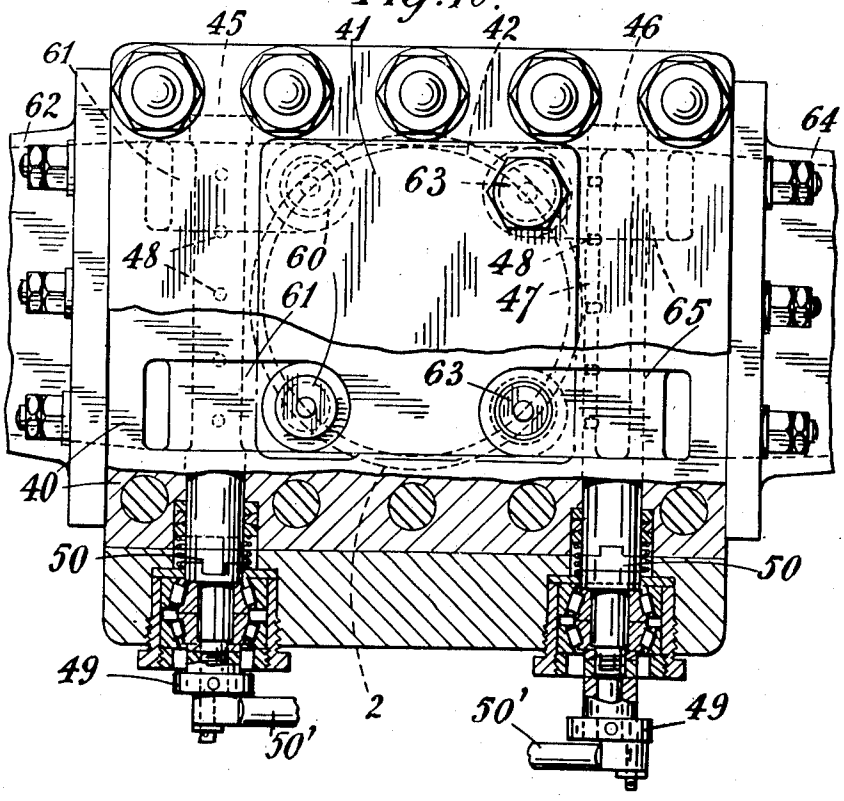


Fig. 10.



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Fig. 12.

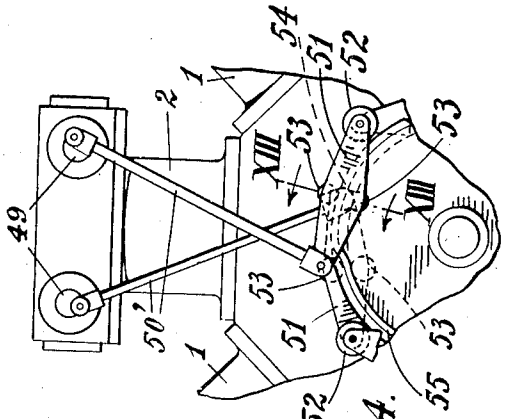


Fig. 14.

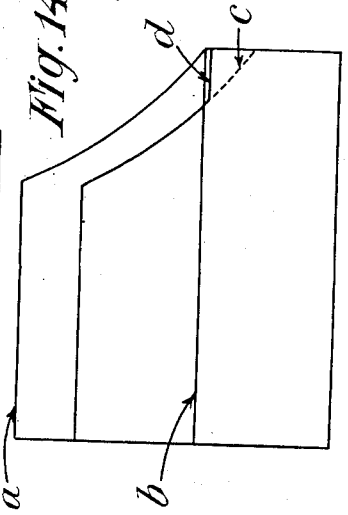


Fig. 11

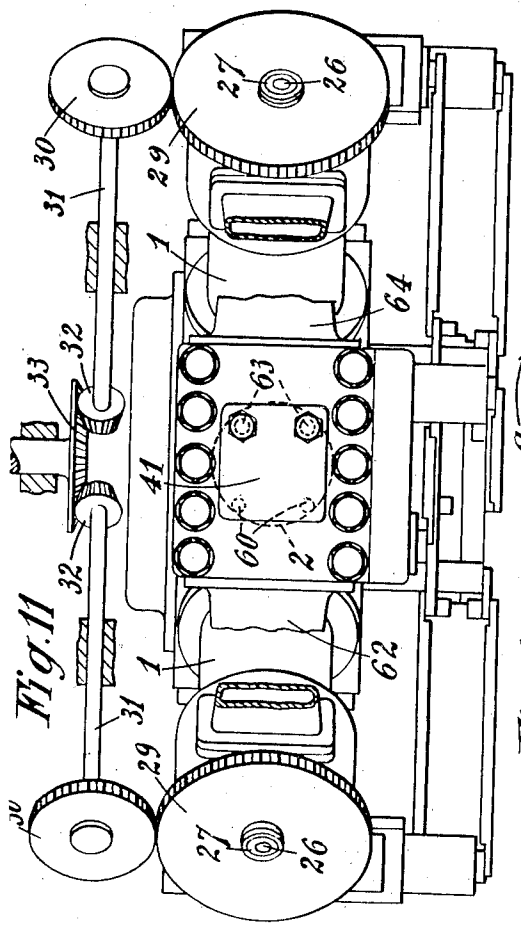
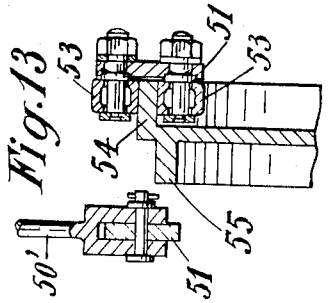


Fig. 13



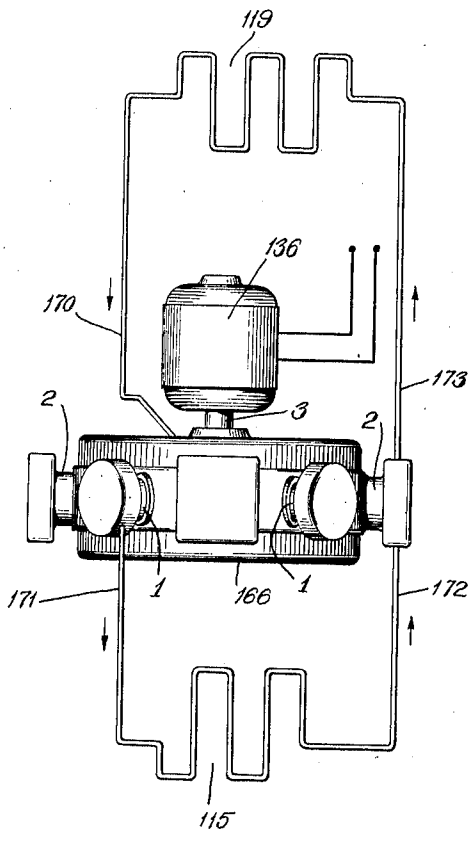
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Fig. 15.



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Patented Aug. 31, 1943

2,328,439

UNITED STATES PATENT OFFICE

2,328,439

HEAT TRANSFER APPARATUS

Robert Esnault-Pelterie, Boulogne-Billancourt, France; vested in the Alien Property Custodian

Application December 7, 1940, Serial No. 369,033
In Great Britain December 21, 1939

9 Claims. (Cl. 62—136)

This invention relates to improvements in or modifications of the heating and cooling process and apparatus claimed in my prior application, now Patent No. 2,310,520, and the present application is in the nature of a continuation in part of my said prior application (hereinafter referred to as the "parent specification") to which reference is hereby made for certain descriptions which it is unnecessary to repeat at length herein.

In the said parent specification there is described and claimed an apparatus operating according to the principle of the thermo-pump and using a non-condensing gas as the working fluid, the low pressure of the cycle being essentially at least equal to 20 kg./cm.^{2} The apparatus comprises pipes forming a closed circuit (closed except for make-up and bleed) for a non-condensing working fluid or gas such as air, this circuit including the several heat interchangers there described, and the cylinders of a combined motor-pump unit having six cylinders arranged star-fashion around a common crankcase, where the pistons of all the cylinders are connected to a common crank. Pump cylinders and motor cylinders alternate, there being three of each. An electric motor is connected to the common crank, and furnishes power needed to make up for friction and heat losses. Air at the low pressure prevailing in the circuit enters the crankcase, and thence passes through valves located in the heads of the pistons working in the pump cylinders and is discharged by the pump cylinders into a high pressure part of the system. After passage through a heat interchanger the highly compressed air passes to the motor cylinders where its entrance and discharge is governed by cam-actuated spring-loaded valves. One of the features of the present invention is a change in the form and mode of operation of these valves in order greatly to reduce the frictional resistance thereof and in that and other ways to increase the efficiency of the apparatus. From the motor cylinders the expanded air passes through a second heat interchanger and thence to the common crankcase, being now at the low pressure prevailing in the system. Means are provided for supplying make-up air and for bleeding air from the circuit, as needed, and control of output is under thermostatic control. The apparatus, by manual operation of valves and a switch, is adapted to be reversed in the effect which it produces, i. e., either to heat or to cool the atmosphere of an enclosed space. In an apparatus of this kind operating according to the Joule cycle it can be shown that the greater the ratio of the volumes for a given ratio of pressures, the higher the efficiency. When using a non-condensing gas, for example ordinary atmospheric air, as the working medium the ratio of the volumes is small compared with that obtainable in apparatus using a condensible vapour, and it is necessary in order to obtain a sufficiently high thermal efficiency (theoretical co-efficient of performance multiplied by the mechanical efficiency) for commercial utility to attain a mechanical efficiency substantially in excess of 90%, as compared with the efficiency of 80 to 85% common in vaporising refrigerating machines.

Various constructional features and dispositions have been described in the parent specification with a view to obtaining this high efficiency, and the present invention relates to an improved motor-pump unit with the same object in view.

It has been found that the most important losses in reciprocating pumps and motors occur at the valves, especially where these are of the automatic type, and to reduce this loss the suction and delivery valves of the main pump each comprise a crenellated opening in a partition member and a flexible lamina overlying said opening in the manner of a flap valve. Although it is proposed to use pressure differences between the two sides of the valves of the order of 25 kg./cm.^{2} it is found that valve laminae ½ mm. thick composed of fibrous material bonded with a synthetic resin can successfully be used with crenellated openings the slots of which are 6 mm. wide. The crenellated openings can extend over a substantial area and the lightness and flexibility of the laminae ensure the rapid opening of the valves. Metallic laminae of less thickness can be used but are inferior to the bonded fibrous material from the point of view of flexibility and heat-insulating quality.

At the valves of the motor losses occur due to fluid friction, the non-instantaneous opening and closing of the valves, pneumatic losses due to sudden enlargements, contraction and bends, and solid friction losses due to the inertia of the parts and the use of continuously acting closing springs. These losses may be substantially reduced according to a feature of the invention by providing valves of the kind having ports in relatively sliding surfaces and by opening and closing these valves by the sole agency of posi-

tively acting means. In particular, it is preferred to use separate admission and exhaust valves each in the form of a ported flat plate slidably mounted between two members having flat ported surfaces, or separate rotary valves similar to the Corliss valves used for many years in large steam engines. In either case the valves are preferably operated by an annular cam having inner and outer cam surfaces giving positive operation of the valve both in the opening and in the closing direction.

The use of annular cams rotating at the same speed as the crankshaft makes it practicable to obtain quicker opening and closing of the valves than with cams rotating at half the crankshaft speed as described in the parent specification, but the disadvantage is encountered that it is then impracticable to regulate the heating and cooling output by varying the valve timing by means of a planetary gear. According to a feature of the present invention, the regulation of the amount of heating or cooling produced is effected by varying the volume of the dead space of the pump cylinders. In effect, therefore, the regulation is produced by varying the difference between the high and the low pressure of the cycle, while the point of cut-off in the motor cylinders remains fixed. Due to this fact it will be appreciated that it is not possible to ensure the expansion of the working fluid down to exactly the predetermined low pressure of the cycle at all loadings. In other words, prior to and at the moment of opening of an exhaust valve there may be a difference of pressure between its two sides, which will give rise to losses due to unbalanced back pressure on the pistons, increased friction in the valve mechanism and pneumatic disturbances in the gas flow. To avoid this disadvantage, and similar disadvantages which may occur at the period of opening of the inlet valves, according to a feature of the invention the motor cylinders are provided with a valve allowing fluid to pass into the high pressure conduit in the case of excess pressure in the cylinder, and a valve allowing fluid to pass into the cylinder from the low pressure conduit in the event of the cylinder pressure falling below the low pressure of the cycle.

The sliding friction of the pistons accounts for a further substantial loss, and this may be reduced by the use of special pistons comprising a crown portion provided with suitable packing means and a crosshead portion carrying the gudgeon pin and provided with rollers arranged to run on the cylinder walls and to transmit the lateral forces thereto. All points of articulation both in the piston-connecting-rod crankshaft system and in the valve mechanism are furthermore provided with needle, roller or ball bearings with the same end in view.

As pointed out in the parent specification, the radial arrangement of the pump and motor cylinders is particularly advantageous in an apparatus of this kind, and it was proposed to use either a six-cylinder or a ten-cylinder unit. Further investigation has shown that with an eight cylinder unit the resultant loading of the common connecting rod bearing, while being substantially lower than that of a six-cylinder unit, is about equal to that of a ten-cylinder unit, so that having regard to the greatly reduced number of moving parts, an eight-cylinder unit presents substantial advantages over all other arrangements.

The invention will be further described with reference to the constructional example shown in the accompanying drawings, in which:

Figure 1 is a section through two cylinders, of an eight-cylinder motor-pump unit;

Figure 2 is a section along the line II—II in Figure 1;

Figure 3 is a section along the line III—III in Figure 1, it being noted that this line is below the valve lamina on the left-hand side of the cylinder axis and above it on the right-hand side;

Figures 4 and 5 show a modification of the valve arrangement shown in Figure 3;

Figure 6 is a view similar to Figure 3 showing a further modification of the valve arrangement;

Figure 7 is a section along the line VII—VII in Figure 6;

Figure 8 shows in its upper and lower halves two modifications of the valve arrangement shown in Figure 6;

Figure 9 is an elevation of a motor cylinder head, with parts broken away to show the arrangement of the pressure-equalising valves;

Figure 10 is a plan corresponding to Figure 9;

Figure 11 is a plan view of the motor-pump unit showing diagrammatically the means for varying the dead space of the pump cylinders;

Figure 12 is a fragmentary view showing the valve operating mechanism;

Figure 13 is a section along the line XIII—XIII in Figure 12;

Figure 14 is a pressure-volume diagram for the motor cylinder showing the effect of the pressure-equalizing valves; and

Fig. 15 is a schematic diagram of the system of the invention herein.

Referring to the drawings, the heat transfer apparatus in which my invention is useful may be of the type shown in the parent specification. For convenience I show here in Figure 15 a schematic diagram of such an apparatus (omitting valve actuating mechanism and pipe connections to all but two of the cylinders) which in the present instance incorporates a motor-pump unit of eight cylinders. In a system of this kind, compressed gas from the pump cylinders 1 passes through pipe 171 to the high pressure part of the system containing one or more heat interchangers 115, where it gives off heat, and then returns through pipe 172 to the motor cylinders 2 where it is expanded. Expanded gas passes from the pump cylinders 2 through pipe 173 to the low pressure part of the system containing one or more heat interchangers 119, where it absorbs heat, and then returns through pipe 170 to the enclosed crankcase 166 of the motor-pump unit whence it is drawn through inlet-valve openings in the heads of the pump pistons, thus completing the circuit. An electric motor 136 connected to shaft 3 supplies power needed to make up for heat and friction losses. The pipes 171, 172 and 173 are shown for convenience in Fig. 15 as being connected to but one each of the pump and motor cylinders. As shown in the parent specification similar connection may be made with all of these cylinders by use of suitable headers. Also, by means of reversing valves, and the like, as shown in the parent specification, the system may be utilized either to heat or to cool the space in which a particular heat interchanger is located.

The eight-cylinder motor-pump unit, of which one pump and one motor cylinder are shown in Figure 1 at 1 and 2 respectively, comprises a crankshaft 3 having a crank-pin 4 on which is

rotatably mounted a ring 5 to which all the connecting rods are articulated. By means of suitable gearing the ring 5 is caused to rotate about the crank-pin 4 at the same rate as the crankshaft rotates, but in the opposite direction, so that any diameter remains always parallel to its starting direction. This arrangement has the advantage relatively to the more usual master connecting-rod system that the strokes of all the pistons are equal. In Figure 1 the gearwheels have been omitted to clarify the drawings.

The pistons of the pump and motor cylinders are shown at 6 and 7 respectively and are similar in design in that they both comprise a crown portion 8 and a cross-head portion 9. The crown portion 8 carries piston rings 10, while the cross-head 9, to which the connecting-rod 11 is articulated by means of a needle bearing, carries two rollers 12 also by means of needle bearings. The rollers 12 run on the cylinder walls and transmit thereto the lateral thrusts resulting from the obliquity of the connecting rod. As shown in Figure 2, the generatrix of the rolling surface of the rollers is an arc of smaller radius than that of the cylinder.

The crown portions 8 of the pump pistons 6 are provided with a number of inlet-valve openings 13 (Figure 3) arranged regularly at a distance from the axis of the piston and having their inner margins crenellated so as to resemble the outspread fingers of a hand. A flexible lamina 14 overlies the openings 13 and is fixed against rotation by a central screw 15 and an eccentrically arranged pin 16. The lamina 14 has a plurality of openings 14' formed therein so that the crenellated margins of the openings 13 are overlapped by the lamina to a substantially constant extent. The lamina 14 is preferably composed of fibrous material bonded with a synthetic resin, but thin sheet metal or other suitable materials can also be used. In order to increase the flexibility of the marginal portions of the laminae, and thereby obtan a wider opening of the inlet valves, a plurality of slits 17 may be provided in the portions overlying the piston surface between adjacent openings 13. Said slits would form a radial extension of certain of said openings 14' (Figure 3).

Figures 4 and 5 show a modification in which the valve laminae 14 are not rigidly clamped to the crown portions of the pistons 6. The centre of each lamina is capable of sliding upon screw 15 but is seated upon the openings 13 by the resilient lamina 18 acting as a spring. By this means great flexibility of the marginal portions of the laminae is ensured without the necessity of providing slits therein.

Two modifications of the above valve arrangements are shown in Figures 6 and 7 and Figure 8.

In Figures 6 and 7 the inlet-valve openings 13 have their outer margins crenellated so as to resemble the outspread fingers of a hand. The flexible lamina 14 which overlies the openings 13 and is held in position by screw 15 and pin 16 has its periphery crenellated in a manner corresponding to the openings 13 so that the crenellated margins of these latter are overlapped by the lamina to a substantially constant extent. Slits 17 are provided to increase the flexibility of the marginal portions of the laminae.

Figure 8 shows an advantageous modification of Figures 6 and 7 in which the valve lamina 14 is made a little more flexible. The lamina 14 is partly covered by a resilient lamina 18 acting as a spring and assisting the seating of the valve proper. The upper part of Figure 8 shows a star-shaped resilient lamina with points engaging alternate fingers of the valve lamina, while the lower part of the figure shows a construction in which the star has the same number of points as the valve lamina 14 has fingers.

Referring to Fig. 1, a discharge valve 19 of similar construction is carried by a shallow piston-like head 20 which is axially slidable in a block 21 composed of a synthetic resin material incorporating friction-reducing substances. The block 21 constitutes a heat barrier between the pump cylinder 1 and the cylinder head 22 and is shaped at 23 to form a flexible lip engaging the sliding head 20 in the manner of a cup-leather packing. Rotation of the sliding head 20 is prevented by a screw 24 engaging a slot 25, while axial movement is provided for by means of a screw-threaded shaft 26 fixed to the sliding head and passing through a corresponding nut 27 supported by a thrust bearing 28 and keyed to a gearwheel 29. By rotating the gearwheel 29 it will be understood that the dead space of the pump cylinder is varied by the axial movement of the sliding head 20. As shown in Figure 11, the gearwheel 29 of each pump cylinder is engaged by a gearwheel 30 mounted on a radial shaft 31. At their inner ends the radial shafts carry bevel pinions 32 meshing with a central bevel pinion 33, so that the turning of this latter varies equally the dead spaces of all the pump cylinders and therefore also the difference between the high and the low pressure of the cycle. The heating or cooling output of the apparatus naturally varies in a corresponding manner. As in the parent specification, the heating and cooling may be controlled by thermostatic means acting on a motor for turning the shaft of the bevel wheel 33.

Referring to Figs. 1, 9 and 10, motor cylinders 2 are provided with heads comprising a block 40 and a top plate 41 bolted to a flange at the upper end of the cylinder. The block 40 is also composed of a synthetic resin material incorporating friction-reducing substances and on each side of the cylinder head space 42 formed in it, it is provided with a passage 43, 44 extending laterally over substantially the whole width of the cylinder. In the passages 43, 44 respectively are transversely arranged the inlet valve 45 and the exhaust valve 46, both being rotary valves having a transverse passage in the manner of an ordinary plug cock. In order to avoid the possibility that the segmental portions 47 of the valves, which are located on the high pressure side when the valve is closed, might be bent by this pressure, a number of holes 48 are bored through these segmental portions, so that the pressure becomes equalised on the two sides. The other segmental portions of the valves are at this time supported by the seating formed in the block 40 and cannot bend. Each rotary valve is driven from a small crank 49 supported by roller bearings through an Oldham coupling 50 which ensures that no side thrusts are transmitted to the valve proper. As shown in Figures 9 and 12, the small cranks 49 are operated by rods 59' the other ends of which are articulated to the ends of levers 51 which are pivoted at 52 and each carry a pair of rollers 53 engaging the inner and outer cam surfaces of one of the annular cams 54, 55. The cam 54 controls all the exhaust valves, while the cam 55 controls the admission valves. Since the dead spaces of the motor cylinders cannot be totally suppressed,

owing to the necessity for providing adequate admission and exhaust passages, the exhaust valves are preferably arranged to close shortly before the end of the exhaust stroke so that a certain quantity of gas is enclosed and compressed to a pressure not exceeding the high pressure of the cycle.

In order to ensure that the pressure in the motor cylinders cannot rise above the predetermined high-pressure of the cycle, pressure-equalising valves 63 are provided in the cylinder heads (see Figs. 9 and 10). The valves 60 open outwardly under excess pressure in the cylinder space and allow working fluid to pass along passages 61 leading to the high pressure supply conduits 62. Similar pressure-equalising valves 63 are provided opening inwardly so that when the ratio of the high and low pressures (*a* and *b* respectively of Figure 14) is reduced by increasing the dead space of the pump cylinders whereby the final pressure in the motor cylinders falls towards a value *c* which is lower than the low pressure *b* of the cycle, working fluid can enter the cylinders from the low pressure conduits 64 along the passages 65 before the exhaust valves 46 are operated. This maintains the final pressure *d* in the motor cylinders as near the low pressure *b* as is possible. Frictional forces resisting the operation of the valves are thus substantially reduced and unbalanced back pressures on the pistons avoided. The pressure-equalising valves may advantageously be constructed in the form of crenellated flap valves of the kind illustrated in Figures 3 to 8.

Instead of using rotary valves of the kind shown in Figures 1, 9 and 10, other types of valve giving a large port area and not having a positive abutment at either end of their travel may be used.

What I claim is:

1. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a low pressure part into which the motor discharges and from which the pump is supplied, the combination of a pump cylinder having a displaceable head member adapted to vary the volume of the dead space in the cylinder, a cam-actuated valve for controlling the admission of gas from a high-pressure line to a motor cylinder, a second cam-actuated valve for controlling the escape of expanded gas from said last-named cylinder to a low-pressure line, and spring-head pressure-equalizing valves cooperating with each of said cam-actuated valves respectively for permitting escape of gas from said motor cylinder to the high-pressure line and from the low-pressure line to the said cylinder when the pressures within said cylinder respectively exceed or fall below the desired working range.

2. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a lower pressure part into which the motor discharges and from which the pump is supplied, the combination of means to vary the ratio of high to low pressure in the gas circuit by varying the dead space of a pump cylinder and a yieldably-held pressure-relief valve permitting gas to return from a low pressure part of the circuit to a motor cylinder when the pressure in the latter falls below said low pressure.

3. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a low pressure part into which the motor discharges and from which the pump is supplied, the combination of means to vary the ratio of high to low pressure in the gas circuit by varying the dead space of a pump cylinder, a cam-actuated exhaust valve for a motor cylinder, and a spring-held valve arranged to permit gas to return from a low-pressure part of said circuit to said motor cylinder while the exhaust valve thereof is closed when the pressure in said cylinder falls below the pressure in said part of said gas circuit.

4. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a low pressure part into which the motor discharges and from which the pump is supplied, the combination of means to vary the ratio of high to low pressure in the gas circuit by varying the dead space of a pump cylinder, cam-actuated inlet and exhaust valves for a motor cylinder, and spring-held valves adjacent each of said cam-actuated valves arranged respectively to permit return of gas from a low-pressure part of said circuit to said motor cylinder and from said motor cylinder to a high-pressure part of said circuit when negative pressure develops at either of said values while the corresponding cam-actuated valve remains closed.

5. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a low pressure part into which the motor discharges and from which the pump is supplied, the combination of means to vary the ratio of high to low pressure in the gas circuit by varying the dead space of a pump cylinder, a movable valve for controlling the flow of gas between a motor cylinder and an external part of said gas circuit, said valve being capable of closing without meeting a positive abutment, and cam surfaces and paired cam-followers operating together to actuate and control the movement of said valve at every position in its path of travel.

6. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a low pressure part into which the motor discharges and from which the pump is supplied, the combination of means to vary the ratio of high to low pressure in the gas circuit by varying the dead space of a pump cylinder, a movable valve for controlling the flow of gas between a motor cylinder and a part of said gas circuit, cam surfaces and paired cam-followers operating together to actuate and control the movement of said valve at every position in its path of travel, and a spring-held valve arranged in shunt with said cam-actuated valve to permit backward flow of gas when negative pressure develops at the cam-actuated valve while the latter is closed.

7. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a low pressure part into which the motor discharges and from which the pump is supplied, the combination of a movable valve for controlling the flow of gas between a motor cylinder and a part of said gas circuit, said valve being capable of closing without meeting a positive abutment, cam surfaces and paired cam-followers operating together to actuate and control the movement of said valve at every position in its path of travel, and a spring-held valve arranged in shunt with said cam-actuated valve to permit backward flow of gas when negative pressure develops at the cam-actuated valve while the latter is closed.

8. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a low pressure part into which the motor discharges and from which the pump is supplied, the combination of a valve for controlling the flow of gas between a motor cylinder and a part of said gas circuit, said valve being capable at that end of its path of travel corresponding to closed position of further movement beyond the position at which the flow of gas is shut off, and means to actuate and control the movement of said valve at every position in its path of travel, said means consisting of inner and outer cam surfaces, paired cam-followers coacting therewith and connections thereof with the valve, whereby the valve is moved to permit and to shut off gas flow free of yieldable restraint opposing said movement in either direction.

9. In heat transfer apparatus which includes a motor-pump unit, a source of added power therefor, and a non-condensing gas circuit having a high pressure part into which the pump discharges and from which the motor is fed and a low pressure part into which the motor discharges and from which the pump is supplied, the combination of a rotary valve for controlling flow of gas between a motor cylinder and a part of a closed gas circuit, said valve being capable at that end of its path of travel corresponding to closed position of rotating further beyond the position at which the flow of gas is shut off, and means to actuate and control the movement of said valve at every position in its arc of rotation, said means consisting of inner and outer cam surfaces, paired cam-followers coacting therewith and connections thereof with the valve, whereby the valve is rotated to permit and to shut off gas flow free of yieldable restraint opposing said rotation in either direction.

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