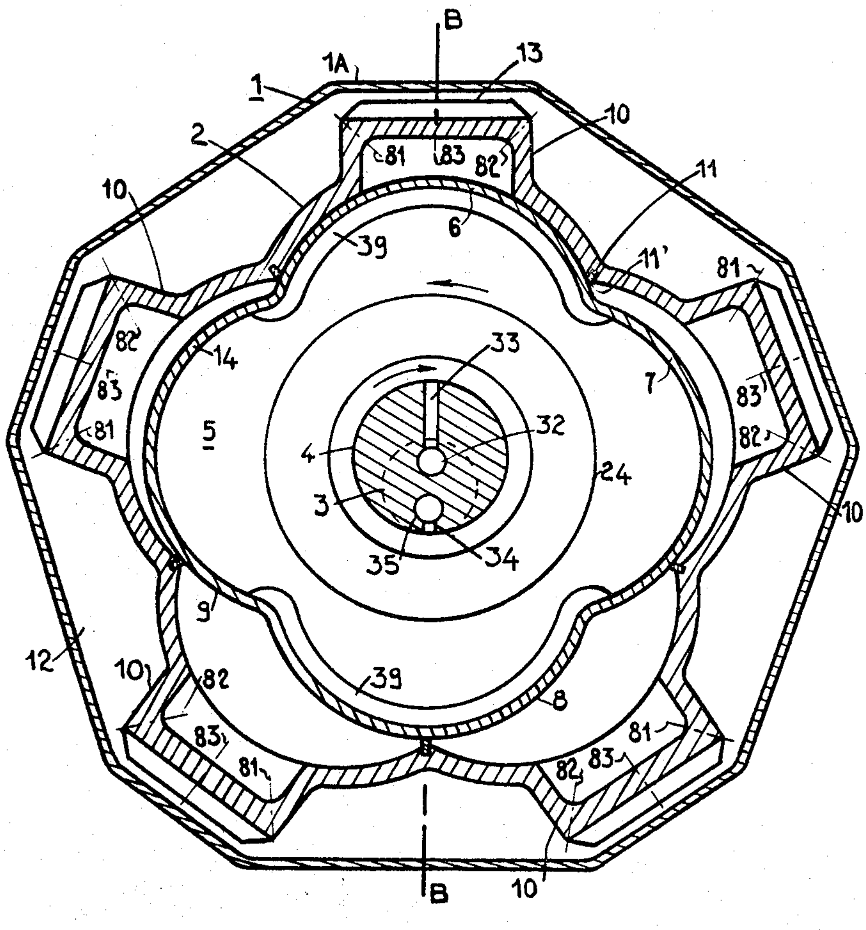
TWO-STAGE COOLING SYSTEM FOR HEAT MACHINE COMPONENTS

Filed Aug. 23, 1966 4 Sheets-Sheet 1

FIG 1



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TWO-STAGE COOLING SYSTEM FOR HEAT MACHINE COMPONENTS

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Fig 2

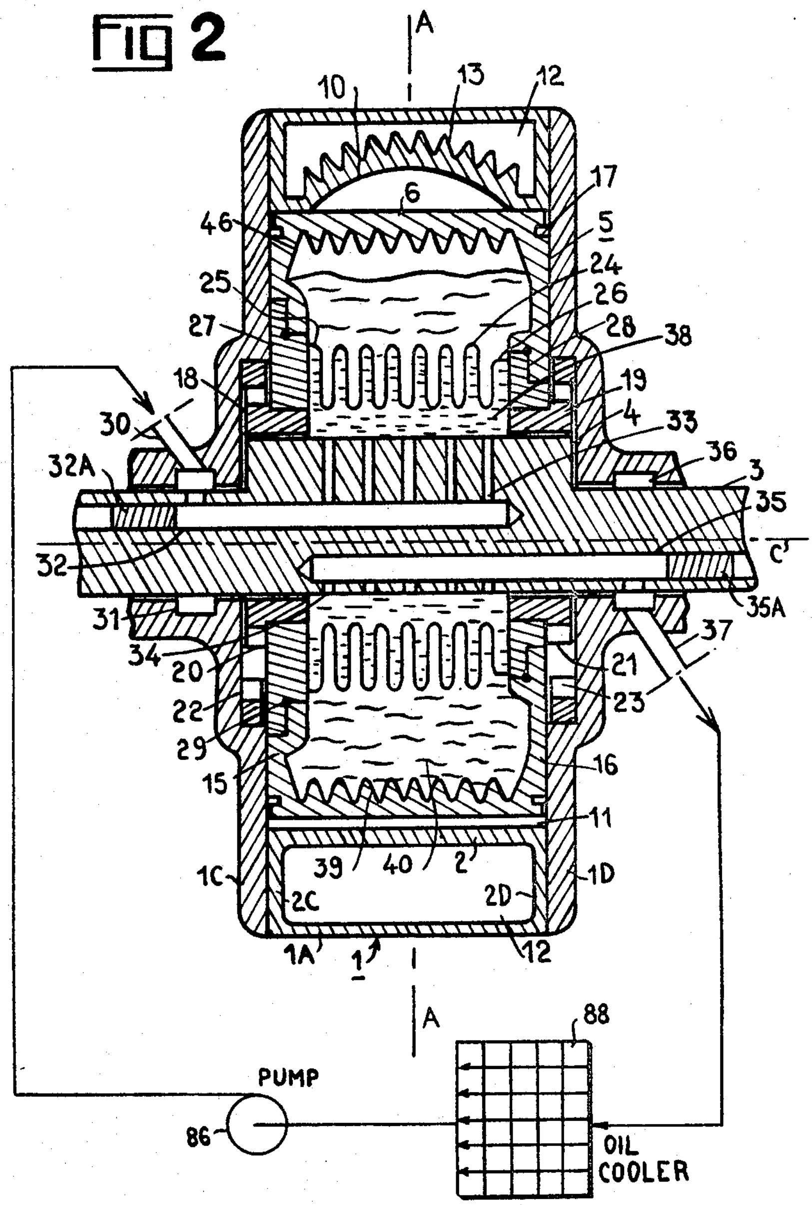
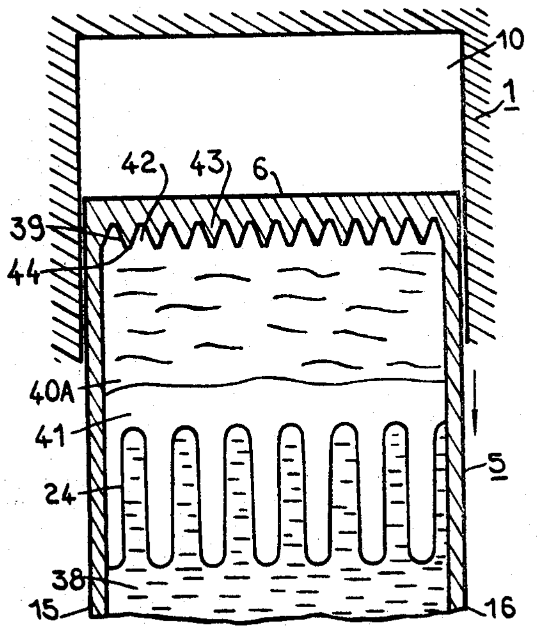


Fig 3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6
39
44
40A
41
24
5
38
15
16

Fig 4

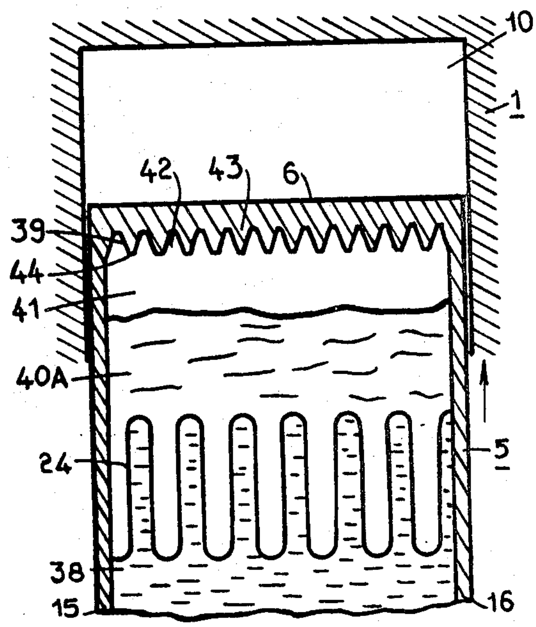
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FIG 5

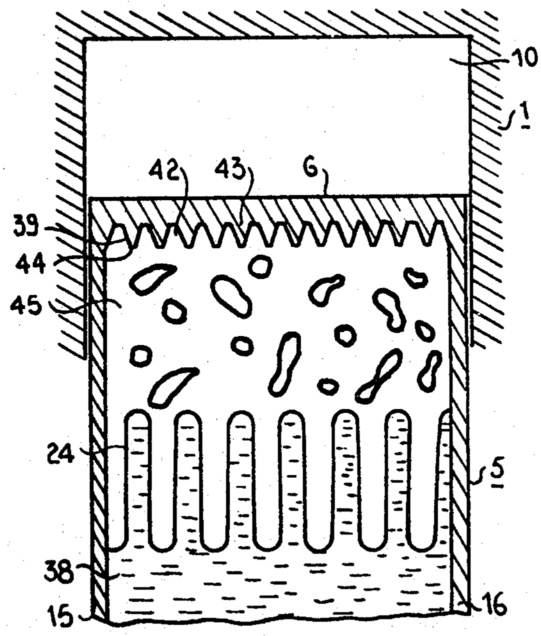
10
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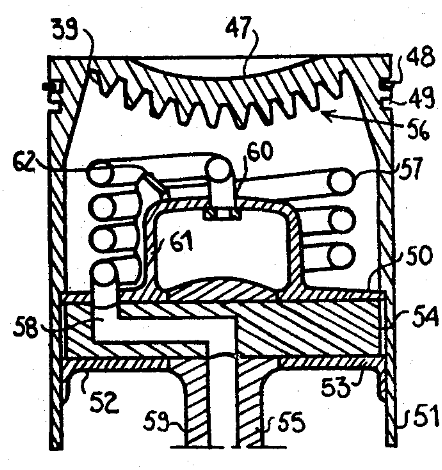
FIG 6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United States Patent Office

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TWO-STAGE COOLING SYSTEM FOR HEAT MACHINE COMPONENTS

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Claims priority, application France, Sept. 7, 1965, 30,619
17 Claims. (Cl. 123—8)

ABSTRACT OF THE DISCLOSURE

A sealed primary chamber containing a vaporizable liquid such as water is arranged adjacent the wall of the piston exposed to heat. This liquid is submitted to stabilized vaporization and condensation, transferring its heat through a separating wall to a secondary coolant fluid, specifically lubricant oil, circulated through a secondary, inner chamber of the piston.

This invention relates to cooling systems for the heated components of heat machines, including both engines and compressors.

The invention has especial utility in connection with the cooling of pistons of combustion engines, both of the reciprocatory-piston type and the rotary-piston type. It will, therefore, be described in terms of such use, but this should not be construed as a limitation on the scope of the invention.

The pistons of combustion engines serve to define variable-volume spaces or compression chambers containing hot gases and/or combustion mixtures under high pressures, and the piston must provide a tight seal against such pressures with the help of piston rings or equivalent sealing elements. If the pistons and their seals are to operate correctly, it is essential that the large amounts of heat generated in the compression chambers to which piston surfaces are exposed, shall be dissipated at a sufficiently high rate to the exterior.

The problem of providing adequate rates of heat dissipation from the pistons of large-sized engines has been found especially difficult to solve in the case of the multi-lobed rotary pistons, or rotors, of the so-called rotary-piston engines, in which the rotor and stator are formed with interengaging lobes differing in number by unity, so that the rotor revolves in a complex path within the stator to define the variable-volume chambers therein. The path of the rotor within the stator may be epicycloidal (sometimes called "epitrochoidal") or hypocycloidal (also called "hypotrochoidal") depending on the geometry of the machine. Such paths will generically be referred to as "trochoidal" herein.

The problem of providing an adequate rotor cooling system has been one of the main causes that has so far held up the practical development of rotary-piston machines, which otherwise are very attractive and hold great promise for the powering of vehicles and other applications.

Since the moving parts of a heat machine must be lubricated, it would appear natural to use the lubricant as a coolant to dissipate the generated heat. This concept has been widely utilized in rotary-piston engines, and many different types of rotary-piston machine constructions have been disclosed, wherein the lubricant is circulated through suitable recesses and compartments formed in the rotor in order to carry away with it the heat generated adjacent the rotor surfaces exposed to high temperatures.

However, the results have been only moderately successful. Lubricant oils are poor heat transfer agents and

do not allow a rate of heat dissipation as high as would be desirable. Oils are chemically unstable and break down rapidly at the high temperatures that would be advantageous in the operation of such machines, requiring frequent lubricant replacement. Further, the high inertial forces generated by the reciprocatory components of the trochoidal motion of the rotor in a rotary-piston machine cause foaming and frothing of the oil and separation of the liquid from the heated surfaces, thereby further reducing heat-transfer efficiency.

It has also been proposed to seal a body of a liquid that is a good conductor of heat, such as potassium or sodium (which liquefy at the temperatures involved) in one or more sealed cavities formed in the rotor of a rotary-piston engine near the rotor surfaces exposed to high temperatures, in order to abstract the heat therefrom and transfer it to a secondary coolant such as lubricating oil circulated through the rotor. This expedient has made it possible somewhat to improve the cooling of rotor surfaces remote from the axis of rotation, but the improvement in over-all heat dissipating rate achieved in this way has been slight and has hardly justified the increase in complication and expense.

The applicant has been engaged for many years in the study of evaporation cooling processes. Evaporation cooling, in which heat is abstracted from a heated surface by a boiling liquid in contact with the surface, is very attractive in theory because it should enable attainment of much higher heat dissipation rates than would be possible with a single-phase liquid coolant, due to the high latent heats of vaporization of certain common liquids including water. Early attempts at evaporation cooling encountered frustrating difficulties because of so-called spheroidal-state ebullition and resulting burn-out. That is, it was found that wherever the temperature of the heat-dissipating surface locally exceeded a rather moderate critical temperature (about 125° C. for water at ordinary pressure), spheroidal-state ebullition would set in and the temperature would tend to jump almost instantly to an enormously higher value (of the order of 1000° C.), causing destructive burnout at a point of the metal surface.

In a number of earlier U.S. patents and other publications, the applicant has disclosed means for positively avoiding this runaway temperature condition and consequent burn-out. The applicant's evaporation cooling systems, known commercially as "Vapotron" broadly involve the provision of heat-dissipating formations in the form of protuberances (bosses and/or ribs) separated by channels or grooves, projecting from the heat-dissipating surface into the boiling liquid, and so shaped and dimensioned that stable temperature gradients are established along their sides. These stable temperature gradients which encompass the so-called "critical" temperature referred to above, are found to be a positive safeguard against runaway temperature and consequent burn-out. The applicant's "Vapotron" technique have thus made it feasible to reach rates of heat dissipation and cooling efficiencies very greatly superior to those achievable with single-phase liquid cooling. In the past years, Vapotron technique have been successfully applied on a worldwide industrial scale to the cooling of high-power electron discharge tubes, and have made it possible to construct such tubes having power ratings very many times greater than was earlier possible. Vapotron coolers have also been applied to the cooling of fuel elements in nuclear power plants, and in other applications.

It has been the applicant's main object in the present invention to apply the so-called Vapotron, or non-isotherm evaporation-cooling, techniques, to the moving parts of heat machines, and particularly to the pistons of rotary-piston and reciprocatory-piston combustion engines.

As indicated earlier herein, lubricant oils are unsuitable for use not only as single-phase liquid cooling agents but also as two-phase evaporation-cooling agents because, inter alia, of their chemical instability at high temperatures. The evaporating liquid must necessarily be a liquid having a boiling point in the range of temperatures permissible for the part to be cooled, and one that will not break down in that temperature range. Water satisfies these requirements. However, the provision of a system for circulating water or another evaporable liquid, in addition to lubricant, through a reciprocatory piston or a rotor, raises very serious design and constructional problems owing to the sealing requirements, and the resulting system would be unsatisfactory from an engineering standpoint. It has been an object of this invention to eliminate these difficulties entirely, through the provision of a two-stage cooling system in which a body of evaporable liquid (such as water) is permanently enclosed in a sealed chamber of the rotor or other moving part to be cooled, adjacent the heated surfaces thereof, and in which the sealed evaporable liquid transfers its heat to a secondary coolant fluid, specifically lubricant oil, circulated through another chamber of the moving part in heat transfer relation with the evaporable liquid.

In applying the above concepts to heat machines in which the moving parts have components of motion subject to periodically varying accelerations, as is the case with the pistons and rotors of both reciprocating-piston and rotary-piston engines, a further problem is encountered. Due to the inertial forces generated in the sealed body of primary evaporable liquid by the varying accelerations, the heated wall surfaces are intermittently stripped of liquid during certain periods of the cyclic motion, and are exposed merely to the vapor so that during such periods heat transfer becomes very poor, and there is introduced a definite risk of subsequent spheroidal-state ebullition and burn-out. It is an object of this invention to eliminate this danger entirely, and it is in fact a further object hereof to utilize the cyclic inertial forces created by the reversing accelerations of the reciprocatory motion of a piston (whether in a reciprocatory-piston machine or in a rotary-piston machine) actually to improve the efficiency of the heat transfer process.

The above and further objects of the invention will be made clear from the ensuing description of exemplary embodiments with reference to the accompanying drawings, wherein:

FIG. 1 is a simplified cross sectional view of a rotary-piston engine provided with cooling means according to the invention, the section being on the line A—A of FIG. 2;

FIG. 2 is a sectional view on line B—B of FIG. 1;

FIG. 3 is a fragmentary view showing the general conditions of the fluid in the primary chamber during the inward stroke of the rotary piston;

FIG. 4 similarly shows the conditions during the outward stroke of the piston;

FIG. 5 shows the conditions at an intermediate stage during movement reversal; and

FIG. 6 is a simplified sectional view of a reciprocating-engine piston provided with cooling means according to the invention in a different embodiment.

A rotary-piston engine of generally conventional layout is illustrated in FIG. 1 as comprising a five-lobed stator **1** and an inner four-lobed rotor **5**.

The stator **1** includes an outer casing wall **1A** of generally pentagonal shape and an inner wall **2** having a contour resembling a five-pointed star. The inner stator wall **2** internally defines five concaves lobes **2**. The meeting of each pair of adjacent lobes defines a cusp at which an inwardly-opening slot **11** is formed, with a seal strip **11'** positioned in the slot. In the central region of each lobe the inner stator wall **2** projects outwards to form a suitably shaped compression chamber **10** sealed at its outer end which is spaced inward from the outer casing wall **1A** so as to define a peripheral space **12** between the inner wall **2** and outer wall **1A** of the stator. As can be seen from FIG. 2, the outer wall 1A and the inner wall 2 of the stator **1** constitute an integral casting having parallel spaced transverse end walls **2C** and **2D**, so that the space **12** constitutes a sealed enclosure. The structure thus described is firmly, and preferably dismountably, assembled as by bolts (not shown) between the transverse end flanges **1C** and **1D** of the stator.

The rotor **5** is in the form of a four-lobed recessed shell having transverse walls **15** and **16** rotatably slidable in close mating engagement with the stator sidewalls or end flanges **1C** and **1D**. Annular sealing strips **17** inserted in slots formed in the outer surfaces of the rotor sidewalls **15** and **16** near the outer edges thereof engage the inner surfaces of stator sidewalls **1C** and **1D**. The rotor shell extending across the rotor end walls **15** and **16** is four-lobed as is clearly apparent from FIG. 1, and each of the four outwardly convex lobes **6, 7, 8** and **9** of the rotor is adapted for close mating engagement with each of the five concave lobes of the stator shell **2**. It will be understood that in the operation of the engine only one of the rotor lobes can at any particular time be fully seated in a concave stator lobe, and in FIG. 1 this condition is shown for the uppermost rotor lobe **6**.

The rotor end walls **15** and **16** are provided in their central regions with hubs **18** and **19** respectively, which are secured to said end walls by way of annular insert members **27** and **28**, secured in said end walls by way of interfitting shoulder joints and annular seals **29** as shown, for convenience of assembly.

The hubs **18** and **19** are formed with cylindrical inner bearing surfaces which rotatably surround an eccenter member **4**. The smooth bearings described may be replaced with antifriction bearings if desired. The eccenter **4** is part of the engine output shaft **3**, which is rotatably received in cylindrical bearing surfaces formed internally of end hubs formed integrally with the stator sidewall flanges **1C** and **1D** as shown. The smooth stator bearings of the rotor may likewise be replaced with antifriction bearings.

Secured around the rotor hubs **18** and **19** are externally toothed gear annuli **20** and **21**, which mesh with internal annuli **22** and **23** secured in shoulders defined within the stator sidewalls **1C** and **1D**.

Each of the five stator combustion chambers **10** is equipped with conventional fuel injection, ignition, and exhaust means, here shown only schematically as including an intake valve and an exhaust valve **81** and **82** projecting into each compression chamber **10** of the stator, and an ignition plug **83** (or a fuel injector, as the case may be) projecting centrally into the chamber.

The operation of this type of rotary-piston engine is well known in the art and will only be briefly described to the extent that it is pertinent to the invention.

The rotary piston or rotor **5** is constrained to describe a complex motion, epitrochoidal in character, within the stator, during which each of the four rotor lobes **6–9** is successively seated in a stator lobe, thereby sealing the corresponding stator compression chamber **10**. It can be shown that as the engine shaft **3** rotates, the rotor **5** has a component of rotation about the center axis C of the shaft, in a direction reverse from that of the shaft at an angular rate four times slower than that of the shaft as indicated by the arrows. At the same time, due to the eccenter **4** the rotor **5** has another component of motion which is a circular translation described bodily with the circular translation of the center of said eccenter **4** about the shaft axis C, and this last component of motion imparts to each of the rotor lobes **6–9** a radial reciprocation toward and away from the center axis C. The result of this composite movement of the rotor is to cause a cyclic variation of the effective volume of the sealed chambers defined between the rotor and stator. The timing of the intake and exhaust valving and the ignition means is so predetermined as to ignite the combustible fuel-air mixture in each of the five stator chambers **10** in succession, as said chamber is in the minimum-volume sealed condition shown for the uppermost chamber **10** in FIG. 1. The ignited mixture then expands as the sealed volume is progressively increased, and is exhausted through the exhaust valve **82** of the chamber, after which a fresh amount of fuel mixture is admitted into the chamber by the operation of the intake valve **81**, and is thereafter compressed as the sealed volume diminishes. In the embodiment described the engine thus operates in a four-stroke cycle. It is important to note that in this type of operation, the coaction of the rotor and stator is such that it is always the same pair of rotor lobes, herein lobes **7** and **9** respectively, which are exposed to the gases at the intake and exhaust strokes of the cycle, and always the other pair of rotor lobes, as shown the lobes **6** and **8**, which are exposed to the gases during compression and expansion. As a consequence, the rotor lobes are unequally loaded thermally, the lobes **6** and **8** being exposed to considerably more heat than the lobes **7** and **9**.

It will be understood that the illustrated embodiment is but one example of the wide variety of operative rotary-piston engines that can be constructed, and to all of which the invention is applicable.

The lubricating and cooling system of the improved engine according to the invention will now be described.

The inner space of the recessed rotor **5** is divided into a radially outer, or primary, chamber **40** and a radially inner or secondary **38** by an annular wall **24** which, in the disclosed embodiment, is formed from any suitable fluid-tight sheet material resistant to corrosion and to the operating temperatures involved and having good heat-conductivity, such as suitable copper alloy or steel sheet. Wall **24** is preferably formed with accordion pleats as shown or with other suitable convolutions, in order to maximize the total area of heat transfer between its two sides. The axial ends of the tubular wall **24** are suitably attached to the annular inserts **27** and **28** of the rotor end walls by welding or brazing. Thus the primary and secondary chambers are completely sealed from each other. The primary or outer chamber **40**, moreover, is itself completely sealed in operation. Chamber **40**, after evacuation of air therefrom, is partly filled with a body of primary heat transfer liquid, which may be water or other suitable liquid vaporizable and chemically stable under the pressure and temperature conditions of operation as will be later described. The inner, secondary chamber **38** contains in operation a body of liquid which constitutes both a secondary heat-transfer fluid and a lubricant for the moving parts of the engine, specifically for lubricating the stator and rotor bearings and gearings, which liquid may be any suitable and usual grade of oil. The oil in the secondary chamber **38** inwardly of annular separating wall **24** is connected with an external flow circuit, schematically shown, through passages as follows:

An oil inlet tube **30** is externally connected to a source of pressure lubricant such as an oil pump **86**, extends through one of the stator hubs, the left one as shown, and connects internally with an annular groove **31** formed in the inner bearing surface of the stator, thereby to lubricate said bearing. Groove **31** is connected by a short radial duct drilled into the engine shaft **3** with a longitudinal bore **32** formed in said shaft, which bore is shown plugged at **32A** at its axially outer end, and as extending in the other or inward axial direction over a major part of the axial length of eccenter **4**. A number of axially spaced radial holes **33** drilled through eccenter **4** discharge at their outer ends into the secondary chamber **38**. A second longitudinal bore **35** is drilled into shaft **3** and eccenter **4** from the opposite direction and is plugged at **35A**, the bores **33** and **35** being transversely spaced from each other in a plane coinciding with the common diametric plane of shaft **3** and eccenter **4** as is visible in FIG. 1. Bore **35** is connected with the secondary chamber **38** at the side of eccenter **4** diametrically opposite the side containing the radial drill holes **33**, by way of a similar series of axially spaced radial drill holes **34**. Bore **35** is connected by a short radial passage with an annular groove **36** formed in the corresponding stator hub similar to inlet groove **31**, and connecting with groove **36** is an oil outlet tube **37** connected to the external lubricant system, specifically an air radiator or other oil cooler **88** as here shown, from which the cool oil is returned to pump **86**. The intake and discharge grooves or chambers **31** and **36** are connected by way of narrow channels or passages, as schematically indicated, to deliver oil to suitable points of the bearing surfaces of the assembly.

It will be observed that in view of the relative arrangement of the longitudinal oil conduits **32** and **35** in the eccentric **4**, the radial ducts **33** through which oil is discharged from conduit **32** into the secondary chamber **38** are longer than the radial ducts **34** through which oil is returned from said secondary chamber into the conduit **35**. There is thus created in operation a net differential centrifugal oil pressure which aids the circulation of the oil around the secondary cooling circuit.

The primary chamber **40** defined between the outer surface of the accordion-pleated wall **24** and the inner surface of rotor shell **14** is partly filled with a vaporizable liquid which may be permanently sealed within the cavity of chamber **40**. The primary liquid may be water with a suitable proportion of antifreeze agent mixed with it, or it may be any other liquid having a suitable boiling temperature, preferably within the range from 70 to 140° C. at ordinary pressure. Thus, ethyl alcohol, trichloroethylene and tetrachloroethylene are examples of liquids usuals as the primary cooling fluid of the invention instead of or in addition to water. The primary chamber **40** is preferably filled with liquid to more than half its total capacity, so that most of the inner surface of separator wall **24** is at all times contacted by said liquid even at rest and at idling rates of the motor. Suitable deflecting baffle means, not shown, may be provided in the chamber **40** in a generally conventional manner to ensure that a substantial contact area of the liquid with the heated surfaces of the rotor shell is present even at low engine rates.

In the illustrated embodiment, the four-stroke operating cycle of the engine shown is such that in each of the five stator chambers **10**, the intake and exhaust strokes occur when the rotor lobes **7** and **9**, respectively, are seated with the stator lobe under consideration, whereas the compression and expansion strokes take place as the rotor lobes **6** and **8**, respectively, are thus seated. Hence, as earlier noted, the lobes **6** and **8** are exposed to maximum heating, and the inner surfaces of these rotor lobes are, accordingly, shown as being provided with "Vapotron"-type heat dissipating projections **39** in the form of parallel ribs and grooves extending in a circumferential direction over the inner surface of each lobe. The ribs and grooves are shown with a triangular cross sectional contour similar to that disclosed in the applicant's co-pending U.S. patent application Ser. No. 512,090, filed Dec. 7, 1965. Alternatively they may be constructed as disclosed in applicant's U.S. Patent 3,235,004. As disclosed in said patent and said patent application, the heat-dissipating projections **39** have as their general function to establish and maintain stable temperature gradients over the surfaces thereof in contact with the vapour in the primary chamber **40**, whereby higher surface temperatures can be safely reached without any danger of local destruction or "burnout" of the metal.

Before considering the detailed mechanism of the evaporation heat transfer process occurring in the primary chamber **40** from the ridged inner surface of rotor **5** to the boiling fluid in the chamber, it is important to consider the forces that act on the body of fluid during the operation of the engine, since those forces govern the distribution of the liquid and vapour phases in relation to the heat transfer surfaces at any time. There are different types of force in action, the first being gravity which is quite negligible in regard to the others and will be disregarded, and the others being inertial forces created by the movement of the rotor. As earlier indicated, the rotor describes a complex trochoidal type of motion with respect to the stator, and its movement can be broken down into a component of rotation due to the rotation of shaft 3 about its center axis C, and a component of translation due to the circular displacement of the eccentric 4 about the shaft axis C. The first, rotational, component is considerably the lower of the two, and is comparatively unimportant; its effects will be briefly referred to later. By far the principal forces acting on the primary liquid are the inertial forces created by the second or translatory component of rotor motion which produces in effect, a radial reciprocation of the rotor relative to the axis of the stator. Due to this radial reciprocation, the primary liquid sealed in chamber 40 is subjected to high radial accelerations towards and away from the stator axis, as a result of which said liquid is alternately forced outwards against the ridged rotor surface 17, and inwards against the accordion-pleated surface of the annular separator wall 24. Between these two conditions, there occurs an intermediate phase during which there is a mixture of liquid and vapour substantially completely filling the annular primary chamber and contacting both its outer (rotor) surface and its inner (separator 24) surface. While it is readily understood that the three main phases or conditions just indicated merge continuously with one another during operation of the engine, it is convenient to consider the three typical phases separately and they will thus be described with reference to FIGS. 3–5.

FIG. 3 illustrates the condition that obtains as the rotor lobe 6 is travelling radially inwards towards the stator axis as indicated by the arrow, so that the liquid phase 40A is forced into contact with the ridged inner surface of said lobe, while vapour is contacting the outer surface of the pleated separator 24. At this time there is intense local boiling and vaporization of the liquid along the sides of the ridges 39 and particularly in the bottoms of the grooves 42 between them. The vapour that is thus formed is discharged radially inwards by the high centripetal pressure generated by the radial inward rotor movement at this time, and is forced through the annular body of liquid 40A to collect in the inner vapour space indicated at 41, where the vapour partially condenses in contact with the relatively cool surface of separator 24. During this phase, therefore, intense dissipation of heat from the hot rotor lobe 6 to the primary liquid 40A takes place.

As the rotor lobe 6 reverses its movement (see FIG. 5) and starts moving radially outward, the liquid 40A is forced through the vapour 41 that has collected in the radially inward region during the preceding phase by the strong inertial forces and an intensely active mixing action occurs, producing what is effectively a liquid-vapour emulsion 45 filling the chamber, as illustrated in FIG. 5. The vapour condenses in the midst of the liquid considerably cooler than it, and the average temperature of the body of liquid 40A rises.

The third phase, shown in FIG. 4, sets in as the rotor lobe 6 moves radially outward, so that the body of liquid 40A is now pressed against the pleated separating wall 24. It is chiefly during this phase that the heat accumulated by the primary liquid during the first two phases is transferred through separator 24 to the secondary liquid, the oil, in the secondary chamber 38 forming part of the lubricating circuit earlier described.

In this third phase of the heat-transfer cycle, it will be noted that the heat-dissipating formations 39 of the rotor lobe 6 are exposed to vapour, so that their temperature rises due to the heat flux applied thereto from the outer surface of the rotor lobe. This temperature rise occurs mainly at the grooves 39 or roots of the protuberances, whereas the apices 44 are heated at a substantially slower rate, a stable temperature gradient being maintained continuously along the sides of the protuberances 39. Hence, at the termination of this phase when the rotor will start moving inward again to recommence the first phase shown in FIG. 3, the liquid forced against the ridged wall is able to resume immediate contact with the relatively cool apices of the protuberances and the normal evaporation heat transfer operation earlier described can at once be resumed, with the continuous stable temperature gradients being present from the cool tips to the hot roots of the protuberances, which may be carried to temperatures well above the so-called "critical" boiling temperature (about 125° C. for water at ordinary pressure), which it would not be permissible to exceed in the absence of the protuberance owing to the danger of burn-out, as explained in the applicant's earlier patents and publications referred to.

In order to ensure that the protuberances 39, while exposed to the vapour and out of contact with liquid during the third phase of the cycle, will not rise to an excessively high temperature before they recontact the liquid as just described, it is preferred according to the invention that said protuberances are made relatively massive so as to have substantial thermal inertia.

It will be seen from the above detailed description of the cyclic evaporation heat transfer process occurring in the rotary-piston engine described, that the over-all heat transfer from the outer surface of the hot rotor lobe 6 by way of the primary boiling fluid 40A to the secondary lubricating liquid, is extremely efficient. The efficiency is due in part to the relatively high temperatures attainable at the hottest points, i.e. in the grooves, of the ridged rotor surface without danger of burnout, and is also due largely to the intensely active mixing and turbulence of the vapour and liquid during the intermediate (emulsion) phase described with reference to FIG. 5 of the described heat transfer mechanism. As a result there is only a comparatively small temperature drop between the primary heat transfer wall (the ridged wall of rotor lobe 6) and the secondary heat transfer wall (separator 24) and hence the secondary coolant-lubricant liquid, regardless of the distance between the two walls and hence the size of the engine, the heat being so to speak transported by the body of primary liquid from the primary wall to the secondary wall by the inertial forces inherent to the operation of the engine. The over-all heat resistance of the two cascaded heat exchange means remains very low even when the flux density of the heat to be dissipated is high, so that it becomes possible to use heat flux densities several times higher than the highest values heretofore usable in engines of the type described.

As earlier mentioned, the circular component of rotor movement also has an action on the heat transfer cycle. This component acts to impart to the annular body of liquid in the primary chamber a turbulent bodily rotation with respect to the rotor. The resulting centrifugal forces only have a minor modifying effect on the cyclic process described above. However, the bodily rotation of the liquid annulus is beneficial in that it serves to equalize the temperatures between the four lobes of the rotor which, as earlier indicated, are unequally loaded thermally in the 4-stroke embodiment described. The effect is especially marked during periods of acceleration and deceleration due to the increased turbulence at such times. This improves the uniformity of the heat transfer and, additionally, tends to equalize the thermal expansion effects to which the rotor is exposed.

In the embodiment of the invention just described, additional cooling is used for the stator, and for this purpose the peripheral chamber 12 earlier referred to, is filled with a vaporizable liquid for instance water. The outer surfaces of each of the five combustion chambers 10 is partly formed with heat dissipating protuberances 13, such as ridges and grooves, which may be formed similar to the protuberances 39. The chamber 12 is preferably connected with an external circulatory system by way of an inlet and an outlet not shown. Alternatively it may be sealed and cooled by the surrounding air, or a secondary cooling liquid.

Returning to the rotor cooling system and the cyclic heat transfer process earlier described with reference to FIGS. 3–5, it will be observed that the pressure within the sealed primary chamber **40** is subjected to large and rapid variations during each cycle, the pressure rising during the phase shown in FIG. 3 and dropping in the phase of FIG. 4, and the pressure being especially low in the emulsion phase of FIG. 5. The average pressure, however, will clearly increase with increasing heat dissipation in the engine, due to the increase in pressure of the vapour produced. Now, any increase in operating pressure will improve the efficiency of each of the heat transfer processes involved, including the efficiency of the vaporization heat transfer, that of the condensation heat transfer and that of the conduction heat transfer through separator **24**. In other words, the efficiency of the heat transfer increases as the operating rate of the engine is increased. This constitutes in effect a natural feedback action which is very desirable.

Some numerical values for the temperatures and pressures present in various parts of the rotor cooling system of a rotary-piston engine constructed in accordance with the invention, will now be given by way of example.

The oil in the secondary cooling system is pumped with a flow rate such that its temperature rise on contacting the separator **24** does not exceed 20° C. at the maximum engine operating rates, and is passed through an external cooling system such as a radiator **88** whereby the average temperature of the oil is held at about 70° C. The separating or secondary heat transfer wall **24** is arranged to transfer a heat flux of about 10 watts per sq. centimeter with a temperature drop across said wall not exceeding 30° C. It is noted that this conduction rate is readily achieved in view of the great tubulence of the primary liquid contacting the wall. The primary liquid, therefore, has an average temperature of about 70+30=100° C. The highest pressure reached by the vapour in the limited vapour space provided in the primary chamber **40**, which vapour is generated in the grooves of the protuberances **39** as earlier described, is found to be in the range of from 2 to 4 atmospheres, which corresponds to a boiling temperature of about 120° C. It is here noted that said maximum vapour pressure is limited only by the condensation simultaneously occurring in the liquid. The condensation produced by mixtures of the vapour with liquid in a turbulent emulsion of the kind present during the transitional phase of the heat transfer cycle previously described herein (FIG. 5), is very efficient and its efficiency rises rapidly as the temperature difference between the vapour and liquid phases increases. Experience has shown that a temperature difference of only 20° C., as present between the 100° C. primary liquid and its vapour at 120° C., is amply sufficient to ensure more than the required condensation capacity.

It should be observed that the 2 to 4 atmospheres pressure range in the primary chamber **40** is substantially of the same order of magnitude as the oil pressures normally used in engine lubricating systems. The pressures on both sides of the separating wall **24** are therefore about the same, and this simplifies the construction of this wall e.g. from corrugated metal sheet having good heat transfer characteristics, since said wall will not be subjected in operation to high mechanical stresses.

Turning now to the primary heat transfer wall constituted by the rotor shell **51**, it is important that the heat dissipating extensions **39** be proportioned in accordance with the teachings of the prior patent or application identified above, in order to provide full assurance against the danger of burnout of the rotor. As described in application Ser. No. 512,090, filed Dec. 7, 1965, the dimensions of the extensions **39** can be determined from the heat conductivity of the metal constituting said wall, the nominal value of the heat flux density present through the wall at the maximum rate of engine operation, and certain other physical properties of the primary liquid used. The parameters available make it possible to predetermine said dimensions for a good economy and maximum performance in each particular instance of use. The rotor wall **5** when thus constructed will be capable of transferring heat flux densities of several hundred watts per sq. cm., while having an average temperature in the groove **43** between the extensions **39** only 20 or 30° C. higher than the temperature of the vapour generated in said grooves.

From the temperature values given above for both the secondary and the primary heat transfer assemblies described, it can be seen that the maximum temperature at the primary heat transfer surface **6** is substantially less than 200° C. Thus the outer surface of said rotor wall which is exposed to the heated gases does not greatly exceed about 200° C. even in the presence of an intense heat flux. Tests have confirmed these results, and it is evident that greatly reduced operating temperatures obtainable with the cooling system of the invention will correspondingly improve the operating conditions of the engine, including the efficiency ratio, the lubrication of the relatively moving surfaces and generally the performance and service life of the engine.

An additional advantageous feature of the construction described is that the piston rings or seals such as **17** are exposed to very active cooling owing to their proximity to the heat-dissipating formations **39**. At higher operating loads; the evaporation of the primary coolant liquid is not restricted to said heat dissipating formations but extends also to the adjacent surface areas **46** of the rotor and flanges, thereby assuring energetic cooling of said piston rings.

In FIG. 6 the invention is shown as embodied in a reciprocatory piston, e.g. of a marine engine or the like. The piston comprises a skirt **51** and an endwall **47**, shown outwardly concave, formed as an integral casting, and the inner surface of the endwall **47** being formed with heat dissipating extensions of the type disclosed e.g. in the applicant's last-identified patent application. Conventional piston rings **48** are seated in grooves **49** formed in the upper peripheral surface region of the skirt. A plugging disk **50** is sealingly secured, as by welding, across the inner bore of the skirt **51** not far from its open and, so as to define between the disk **50** and the ridged surface of the endwall **47** a sealed space or chamber. This constitutes the primary chamber of the two-stage cooling system of the invention and is partly filled with a suitable vaporizable liquid, which may be water or any of the other liquids earlier mentioned as suitable.

The sealing disk **50** is formed with aligned bearings **52** and **53** in which the cross-pin **54** of the connecting rod **55** is journalled. The secondary cooling system in this embodiment comprises a coil pipe **57** mounted in the primary chamber on the inner surface of disk **50** and having an inlet end connected to a passage **58** in the crosspin **54** and thence a passage **59** in the connecting rod **55**. The free end of the coil **57** is connected by way of a calibrated orifice **60** with the interior of a sealed chamber defined by a dome **61** upstanding from the disk **50**. The coil **57** is mounted coaxially around this dome and is supported in place by spacers such as **62**. The secondary circuit may comprise the usual accessories not shown including an oil pump for circulating the oil through passage means in the crankshaft (not shown) and by way of the crank bearing, which it may serve to lubricate in passing, up the passages **59** and **58** through the coil **57** and calibrated orifice **60**, which serves to determine the flow rate of the oil, into outlet chamber defined in dome **61**. From this chamber the oil may flow out by suitable outlet means not shown, directly into the crankcase, preferably lubricating the crosspin bearing **54** on its way out.

The plugging disk **50** may have a sealable filling plug, not shown, extending through it for introducing the primary liquid (e.g. water) into the primary chamber within the piston.

The operation of this embodiment will be easily understood from the explanations given in connection with the first embodiment of the invention, and will not be described in detail. As in the case of the rotary-piston engine, the heat transfer process proceeds in a cycle determined largely by the movements of the body of primary liquid within the piston chamber, under the inertial forces created by the reciprocation of the piston. The heat generated in the compression chamber of the engine (not shown) and applied to endwall **47** of the piston, is thus carried away from the ridged inner wall **56** provided with the "Vapotron" heat dissipating extensions **39**, and is transferred through the wall of the coil **57** to the lubricant, to be dissipated thereby in the external lubricant system.

It will be understood that the cylinder heads, not shown, of the reciprocatory engine including the piston just described, may if desired be provided with evaporation cooling means generally similar to the means described and shown for the corresponding parts of the rotary-piston engine of FIGS. 1–2, including if desired the "Vapotron" like heat dissipating extensions shown at **13** in FIG. 2.

The heat dissipating formations referred to in the last paragraph and at other points of the disclosure, in particular the formations **39** provided on the inner surfaces of the rotor lobes in FIGS. 1 and 2 and the inner surface of the piston and wall in FIG. 6, may assume a wide variety of forms. Preferably they are of either of the types disclosed in the applicant's prior U.S. Patent 3,235,004 or in the applicant's copending patent application Ser. No. 512,090, filed Dec. 7, 1965.

According to the said patent, the extensions or protuberances and intervening channels are so dimensioned as to satisfy substantially the relations

$$d<b/3$$

and

$$b=m\sqrt{ac}$$

wherein d represents the average transverse width of an interprotuberance channel, a the average transverse width of a protuberance between adjacent channels, c the heat conductivity factor of the wall material, and m a numerical coefficient within the range from about 0.7 to about 1.8, when a and b are expressed in centimeters and c in watts transmitted heat per centimeter and per degree centigrade.

According to the said copending application, the extensions or protuberances and intervening channels are so dimensioned as to satisfy substantially the relations

$$b_2=kc\theta/\Phi$$

and

$$s_1/s_a=k\Phi/pq$$

wherein b_2 represents the height of a protuberance, s_a and s_1 the base area and total side surface area of protuberance respectively, c the heat conductivity coefficient of the wall material, q the critical value of heat flux density of said liquid at the operating pressure, θ a specified temperature drop from the base to apex of a proturberance, Φ the maximum specified value of heat flux density per unit area of the heat input surface, k a numerical safety factor selectable over the range from 1 to 2, and p a numerical efficiency factor selectable over the range from 0.8 to 1.6.

What I claim is:

1. In a machine including a moving member and wherein heat is generated adjacent a wall surface of said member, a cooling system for said member comprising:

means defining a sealed primary enclosure in said member adjacent to said wall surface thereof;

a body of a primary coolant liquid that is vaporizable in the range of temperature permissible for said member, sealed in the primary enclosure;

means defining a secondary enclosure in said member in a region spaced from said wall surface;

means defining a flow system for a secondary coolant fluid connected with the secondary enclosure to circulate said fluid therethrough; and

a heat exchanging and separating wall extending between said primary and secondary enclosures, said heat exchanging and separating wall having good heat conducting characteristics and being shaped as a heat exchanger between two moving liquids;

whereby said primary liquid will abstract heat from said wall surface, and transfer the abstracted heat through the separating wall to the secondary coolant fluid.

2. The combination defined in claim **1**, wherein the surface of said heat exchanging wall directed into the enclosure is provided with heat-dissipating formations, said heat-dissipating formations comprising protuberances separated by channels and so dimensioned as to establish stable temperature gradients over the width of the wall directed towards the primary enclosure.

3. The combination defined in claim **2**, wherein said extensions are dimensioned in substantial accordance with the relations

$$d<b/3$$

and

$$b=m\sqrt{a.c}$$

wherein d represents the average transverse width of an interprotuberance channel, a the average transverse width of a protuberance between adjacent channels, c the heat conductivity factor of the wall material, and m a numerical coefficient within the range from about 0.7 to about 1.8, when a and b are expressed in centimeters and c in watts transmitted heat per centimeter and per degree centigrade.

4. The combination defined in claim **2**, wherein said etxensions are dimensioned in accordance with the relations

$$b_2=kc\theta/\Phi$$

and

$$s_1/s_a=k\Phi/pq$$

wherein b_2 represents the height of a protuberance, s_a and s_1 the base area and total side surface area of a protuberance respectively, c the heat conductivity coefficient of the wall material, q the critical value of heat flux density of said liquid at the operating pressure, θ a specified temperature drop from the base to the apex of a protuberance, Φ the maximum specified value of heat flux density per unit area of the heat input surface, k a numerical safety factor selectable over the range from 1 to 2, and p a numerical efficiency factor selectable over the range from 0.8 to 1.6.

5. The combination defined in claim **1**, wherein said heat exchanging and separating wall is formed with convolutions for increasing the surface area of its contact with said primary liquid and secondary fluid.

6. The combination defined in claim **1**, wherein the primary liquid has a boiling temperature in the range of from 70 to 140° C. at ordinary pressure and is chemically stable in said range.

7. In a machine including a moving member whose motion has a component subject to a periodically varying acceleration and wherein heat is generated adjacent a wall surface of said member, a cooling system for said member comprising:

means defining a sealed primary enclosure in said member adjacent to said wall surface thereof;

a baby of a primary coolant liquid that is vaporizable in the range of temperatures permissible for said member, sealed in the primary enclosure;

means defining a secondary enclosure in said member in a region spaced from said wall surface;

means defining a flow system for a secondary coolant fluid connected with the secondary enclosure to circulate said fluid therethrough; and

a heat exchanging and separating wall extending between said primary and secondary enclosures in a general direction transverse to said component of motion of said member, said heat exchanging and separating wall having good heat conducting characteristics and being shaped as a heat exchanger between two moving liquids whereby said sealed body of primary liquid will be intermittently thrown against said separating wall by the inertial forces created by said periodically varying acceleration to promote heat transfer from said primary cooling liquid to said secondary coolant fluid.

8. The combination defined in claim **7**, wherein the surface of said heat exchanging and separating wall directed into the primary enclosure is provided with heat-dissipating formations.

9. The combination defined in claim **7**, wherein said secondary coolant fluid comprises a lubricant liquid for said machine.

10. The combination defined in claim **7**, wherein said machine is a rotary-piston machine of the type including a multi-lobed rotor described a trochoidal motion relative to a stator so as to define variable-volume working chambers therein, and wherein said rotor comprises said moving member.

11. The combination defined in claim **10**, wherein said machine includes a shaft mounted for rotation in the stator, an eccenter secured on the shaft, and means mounting said rotor for slidable motion on the eccenter, and wherein said separating wall comprises an annular wall surrounding the eccenter and defining an annular outer primary enclosure outwardly limited by the inner wall surfaces of the rotor lobes, and an annular inner secondary enclosure inwardly limited by the eccenter.

12. The combination defined in claim **11**, wherein said separating wall is in the form of an accordion pleated sleeve.

13. The combination defined in claim **11**, including passage means formed in said shaft and said eccenter for connecting the secondary enclosure with said flow system.

14. The combination defined in claim **11**, wherein an inner surface of at least one of said rotor lobes is formed with heat dissipating formations in the form of protuberances projecting into said primary enclosure and channels separating said protuberances, and so dimensioned as to establish stable temperature gradients over the side of said wall directed towards the primary enclosure.

15. The combination defined in claim **7**, wherein said machine is a reciprocatory-piston machine, and the piston constitutes said moving member.

16. The combination defined in claim **15**, including means defining a wall sealingly extending across said piston and spaced from the piston end wall to define said primary enclosure having said body of primary coolant liquid sealed therein.

17. The combination defined in claim **16**, including a tubular coil positioned within said primary enclosure and defining said secondary enclosure connected with the secondary fluid flow system.

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