

Dec. 15, 1942.

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2,305,162

METHOD OF REFRIGERATION

Filed March 16, 1939

3 Sheets-Sheet 1

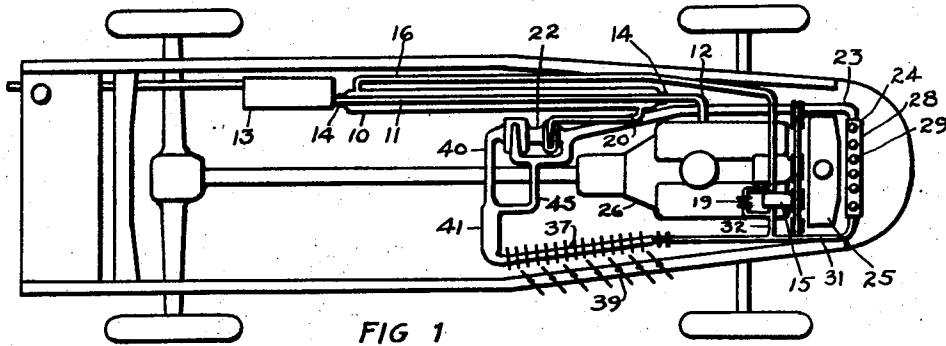


FIG 1

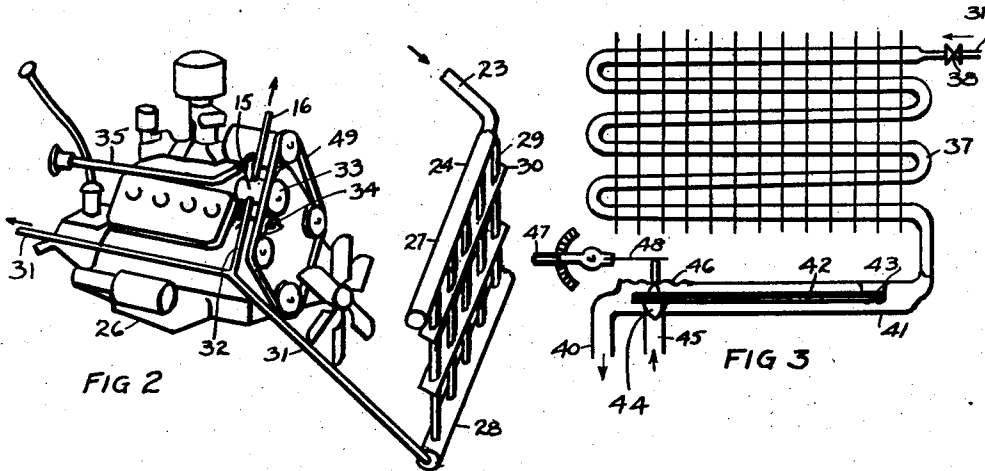


FIG 2

FIG 3

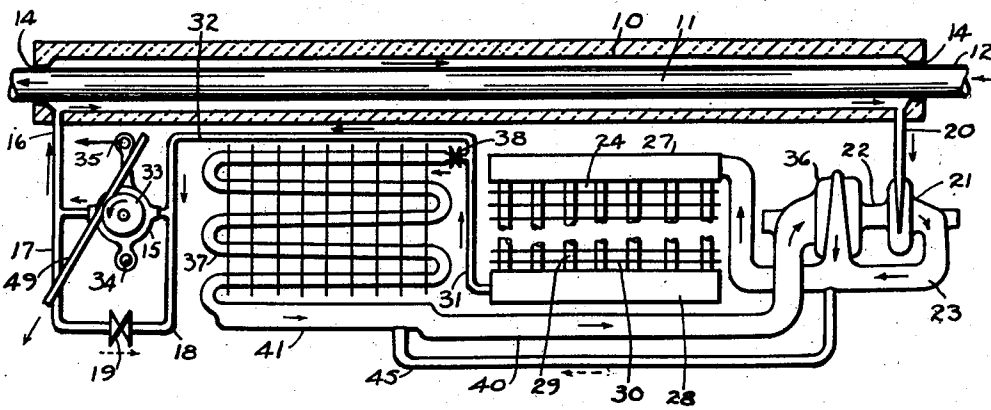


FIG 4

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METHOD OF REFRIGERATION

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3 Sheets-Sheet 2

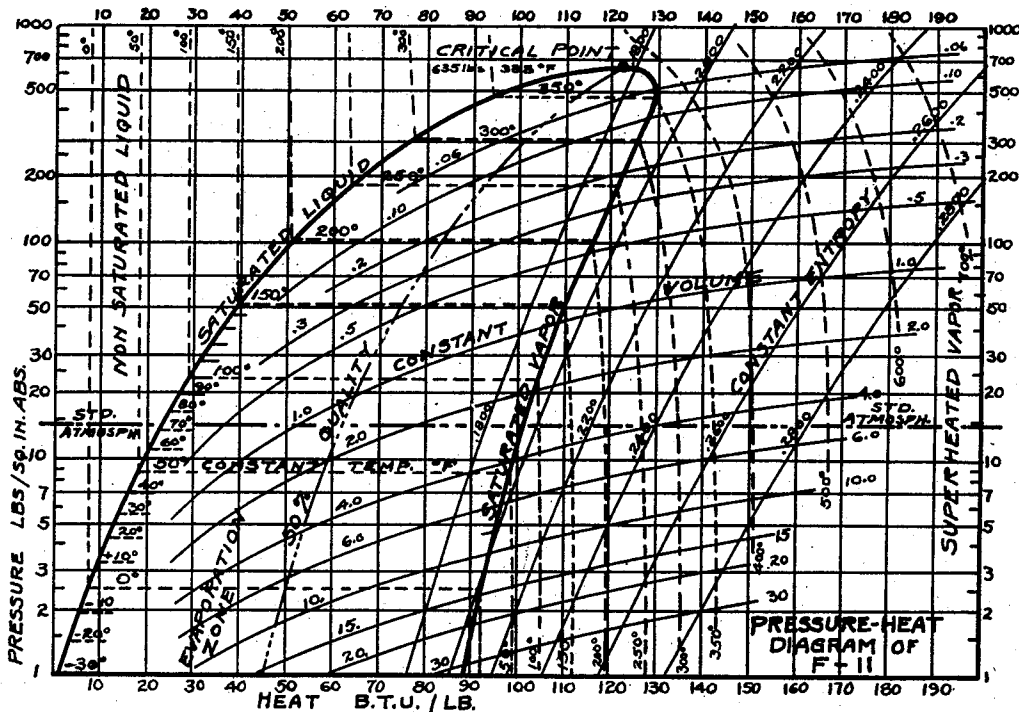


FIG 5

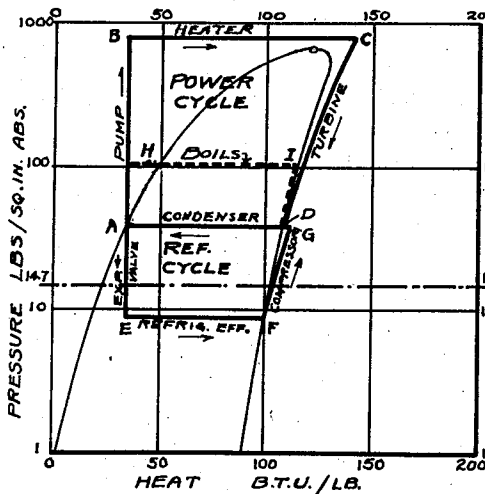


FIG 6

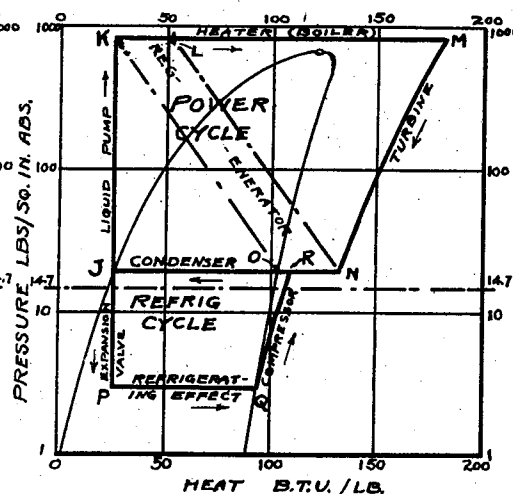


FIG 8

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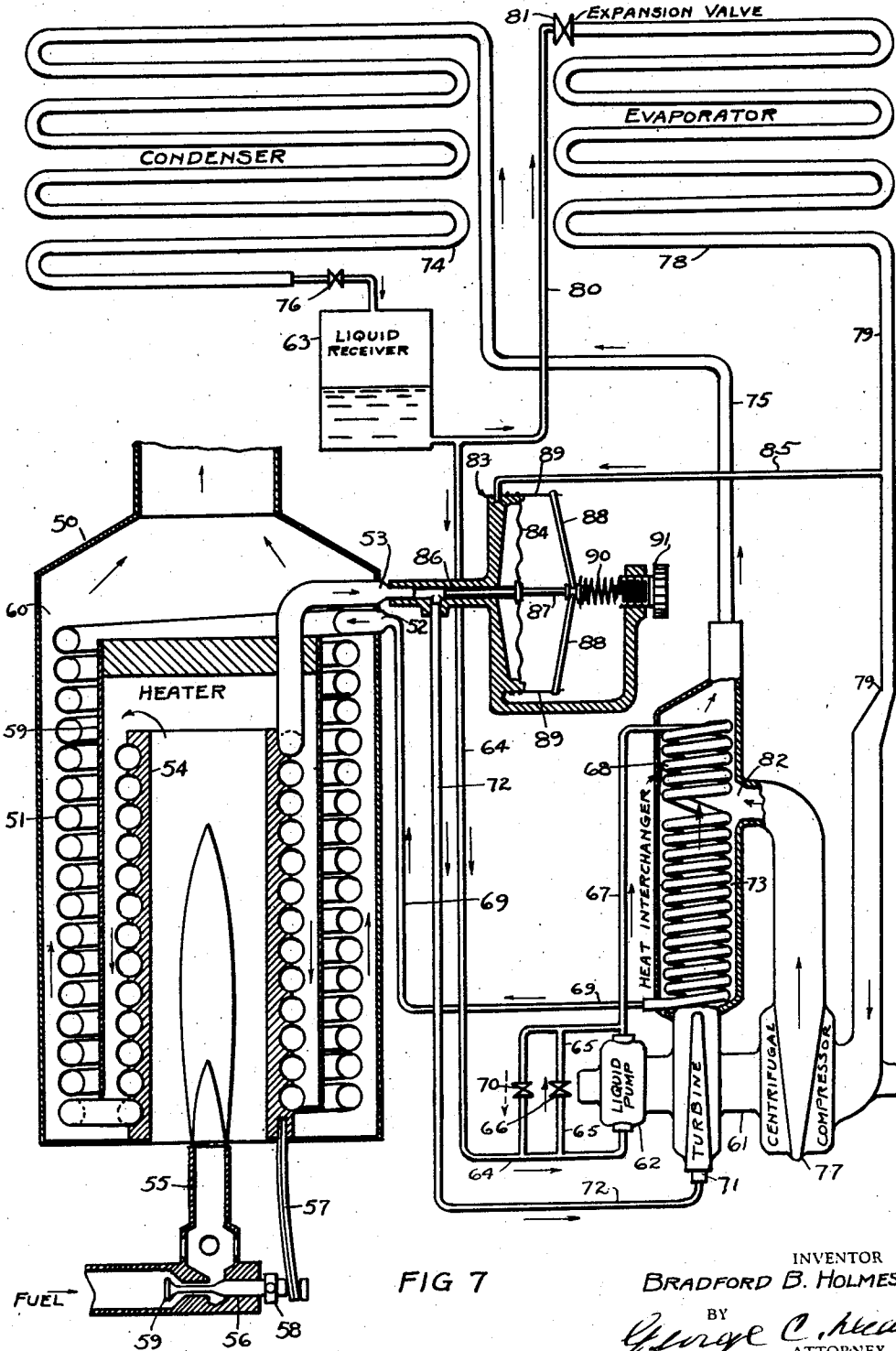
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METHOD OF REFRIGERATION

Filed March 16, 1939

3 Sheets-Sheet 3



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2,305,162

METHOD OF REFRIGERATION

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Application March 16, 1939, Serial No. 262,165

13 Claims. (Cl. 62-178)

My present invention relates to refrigeration and more particularly to a novel method of refrigeration wherein heat is transferred by a process of evaporating a liquid to produce compressed vapor; then expanding the vapor and utilizing the effects of expansion; and then re-liquefying the vapor, in continuously repeating cycle. The expansion of the vapor is commonly utilized to produce power, as in steam engines or turbines, where water is the conventional liquid; or to utilize the temperature drop, as in mechanical refrigerators, where volatile, low-boiling-point liquids are commonly used.

A specific novel feature of the method of my invention depends upon the discovery of certain of these volatile low-boiling-point refrigerator liquids, which are phenomenally well adapted for use as motive power in engines and turbines. One very novel feature of such use is combining motive power production, with refrigeration; that is to say, by my invention liquid from a single source, expanded in an evaporator to produce refrigerating effect may be also employed as the motive power for driving the compressor whereby the expanded vapor is compressed in the compressor and passed to a condenser for re-use in the heater and evaporator.

An even more basic feature of the method of the invention is my discovery that certain of these refrigerator liquids when used for motive power purposes, can be made to operate according to a new cycle; that is to say, certain of them have critical temperatures and pressures low enough so that it is possible to vaporize the liquid without boiling it. To this end the outlet is throttled enough so that the pump can raise the pressure above critical pressure, and keep it there while the liquid is being progressively heated by countercurrent flow through the heater, up to critical temperature, and thereafter superheated to such extent as may be desired. This makes for smooth continuous flow of the liquid through the heater and into the throttled outlet into the turbine, all without any of the bubbling or turbulence which is usually localized in the boiler, necessitating ample spaces for separation of the bubbles of vapor from the body of the boiling liquid.

Furthermore, the working fluid can be heated in such a manner that it gains temperature and heat substantially uniformly as it progresses through the heater, and the furnace gases lose heat and temperature substantially uniformly as they pass from the fire to the stack. This countercurrent type of heat transfer is most efficient.

Another important fact in my discovery is that owing to the high molecular weight and density of these fluids as compared with water, the turbine or heat engine, can be much smaller than one employing steam at the same power.

Furthermore, the volume of vapor per horsepower to be condensed is far smaller than that

of steam at the same condensing temperature.

Both of the above features are of the greatest importance, particularly where a mobile, as distinguished from a stationary power plant, is desired.

For power purposes, either alone or combined with the novel method of refrigeration disclosed herein, I have discovered one which combines many uniquely important characteristics. It is trichloromonofluoromethane, CCl_3F , commonly known as "Freon F-11." I have investigated its characteristic properties particularly in the zones of high temperature and pressure operation for power purposes; and have discovered:

(1) That its vapor is particularly suitable as a means for transforming heat energy into power either in a turbine or piston engine;

(2) That in accomplishing this, a new heat cycle is followed;

(3) That in certain power applications it has decided advantages over steam for the above purpose;

(4) That it has a particular application as a heat driven refrigerator, one of the reasons being that it can be used both as a refrigerant and in a prime mover in a completely enclosed turbine driven centrifugal compressor;

(5) That in a heat driven refrigerator it has an additional special application to motor vehicles such as busses, automobiles, refrigeration trucks and airplanes as a refrigerator or air cooler, where the waste heat from the vehicle's internal combustion engine can be used as a source of energy. This is a case where a fluctuating source of heat is used to supply uniform refrigeration.

There are others of these carbon-chlorine-fluorine refrigeration fluids, that will be referred to later.

The above and other objects of this invention will be brought out more fully in the specification which follows.

The invention is illustrated in the following drawings, in which

Fig. 1 is a more or less schematic plan view of the chassis of an automobile equipped with an exhaust heat cooling system;

Fig. 2 is a schematic perspective view of an automobile engine with liquid pump, liquid pump clutch and condenser;

Fig. 3 is a detail vertical view of the evaporator or cooler with the thermostatic temperature control;

Fig. 4 is a diagram of the complete system removed from the chassis for greater clarity;

Fig. 5 shows a pressure-heat Mollier diagram of the preferred fluid, "F-11";

Fig. 6 shows the waste heat-refrigeration cycle as superimposed on the Mollier diagram;

Fig. 7 shows a diagram of a more refined and efficient heat-refrigeration system in which gas

is the fuel. A boiler and regenerator are shown in vertical cross-section, as well as details of the automatic control; and

Fig. 8 shows the cycle of Fig. 7 superimposed on the Mollier diagram.

The waste-heated refrigerator will be discussed first as it is the simplest.

Motor vehicles and airplanes derive their motive power almost exclusively from internal combustion engines. Automobile engines have a thermal efficiency of about 20% to 25% (Marks' Mechanical Engineers' Handbook), which means that 75% to 80% of the heat of the fuel burned is wasted. About 40% of the total heat escapes from the exhaust. The exhaust temperature of the gases after leaving the exhaust valve range from 575° to 1000° F. (Marks). The temperature will, of course, be less when the engine is idling or throttled down.

If an engine is delivering 20 brake horsepower, about 40 horsepower, or $40 \times 42.4 = 1696$ B. t. u. per minute, escapes through the exhaust pipe and the temperature of the gases will range from 300 to 600° or 800° F. This is far more energy than is needed to provide adequate air cooling or refrigeration for the vehicle even though all of it is not available for use. The refrigerator should be designed to operate as near the idling condition of the engine as possible and provision should be made to prevent excessive refrigeration when the engine is under heavy load as when the vehicle is speeding or climbing. Provision must also be made to render the refrigerator nonoperative when refrigeration is not wanted.

One form of a refrigerating system for carrying out the novel method of the invention, as applied to a motor vehicle, is shown in Figs. 1, 2, 3, and 4 in which 10 is a tube surrounding the exhaust pipe 11 between the motor exhaust outlet 12 and muffler 13. The tube is sealed to the exhaust pipe at its ends 14, 14, forming a pressure tight chamber. The exhaust pipe must have sufficient area and heat transfer surface to transmit the required heat units to the fluid "F-11" in the space between the pipe 11 and tube 10. The fluid "F-11" is forced into the boiler or heater by means of a pump 15, to be described later, through the inlet pipe 16.

The inlet and outlet of the pump are connected by pipes 17 and 18 and a by-pass pressure relief valve 19. This valve may be of any standard form and is adjustable so that when a predetermined pressure is built up in the boiler, the pump will by-pass liquid and the boiler pressure is thus prevented from exceeding the desired pressure. Thus the liquid enters the boiler at a point near where the exhaust pipe enters the muffler, and travels countercurrent to the exhaust gases and leaves the boiler through the outlet pipe 20 which is located near the engine where the exhaust gases are hottest and enters the turbine 21. Thus the fluid gains heat and temperature substantially uniformly (provided it does not boil or evaporate) in its passage through the heater, and the hot exhaust gases will be correspondingly cooled in passing from the engine to the muffler.

Fig. 5 shows the physical characteristics of "F-11" in terms of temperature in degrees F., pressure in pounds per square inch absolute, entropy, heat content in B. t. u. per pound and volume in cubic feet per pound.

Pressure heat coordinates are chosen rather than the more usual heat-entropy coordinates

because when pressure is plotted in a logarithmic scale and heat on a decimal scale, the diagram gives a clear picture of the behavior of the fluid. Also because constant specific volumes can be readily shown. The heat-entropy coordinates are not well adapted to show volumes.

The pressures, volumes and temperatures were calculated partly from known formulae and partly from formulae devised by me. In the regions where the "F-11" is commonly employed for refrigeration, only, the diagram is very accurate; and in the regions of higher temperature and pressure where I employ it for power purposes, the diagram is doubtless less accurate, but it is certainly accurate enough so that the examples do not contain serious error—that is, the discussion is accurate in principle and reasonably accurate in degree.

The behavior of the lines of constant temperature outside of the saturated fluid line in the vicinity of the critical point are not definitely known, but they have no significance when the vapor is superheated.

Assume that the atmospheric temperature is such that the condensing temperature will be 130° F., that the by-pass relief valve is set for 800 lbs. and that there is sufficient heat in the exhaust to heat the fluid escaping from the boiler to about 475° at 800 lbs. pressure.

The condition of one pound of liquid "F-11" leaving the condenser is shown at point A, Fig. 6, $P=39$ lbs., $T=130^\circ$ F., Heat= 34.7 B. t. u. After leaving the pump, its condition is shown at B. $P=800$, $T=130$, $H=34.7$ (neglecting heat added by the work done on it by the pump).

As it passes through the boiler, it increases substantially uniformly in temperature until it reaches the critical temperature, 388°, when it ceases to be a liquid and becomes a vapor. It now continues along the line of constant pressure and is superheated to 475°, which is the assumed temperature at which it leaves the boiler.

Its condition on leaving the boiler is shown at C. $P=800$ lbs., $T=475^\circ$, $H=140$. It now leaves the boiler via pipe 20, Figs. 1 and 4.

Note that the liquid has not boiled during this process of heating because the liquid was under too high a pressure. It has passed from a liquid to a superheated vapor through its critical point. Thus the heater or boiler can be extremely simple and counter-current in character. In its passage through the boiler it has absorbed 140 B. t. u.— 34.7 B. t. u.— 105.3 B. t. u.

The fluid now enters the turbine portion 21 of the turbo-compressor 22. The turbo-compressor will not be illustrated in detail for such devices are well known in the art. A description of one can be found in Marks' Handbook. Suffice it to say that the turbo-compressor consists essentially of a shaft supported by bearings at the ends. The shaft carries a turbine rotor and a centrifugal compressor. The shaft, rotors and bearings are enclosed in a housing and that part of the housing surrounding the turbine is separated from that part surrounding the compressor by a packing usually of the labyrinth type. The turbine housing is provided with one or more nozzles in which the vapor expands approximately adiabatically. The high velocity vapor jet then impinges upon the rotor blades causing the rotor to revolve. All steam or vapor turbines produce mechanical work by virtue of the kinetic energy of the vapor which results from decrease in total heat energy during adiabatic expansion.

Referring again to Fig. 6, the vapor enters

the nozzle in the state shown at point C. It expands in the nozzle to the condenser pressure point D, which is assumed to be 39 lbs. absolute. The expansion takes place along the line CD which is a line of constant entropy since adiabatic expansion occurs without change in entropy. On the saturated vapor line D the condition of a pound of vapor is:

Pressure=39 lbs. per sq. in. absolute;
Temperature=130° F.;
Heat=107.8 B. t. u.

Thus the energy available for mechanical or shaft work is 140 B. t. u.—107.8 B. t. u.=32.2 B. t. u. The theoretical thermal efficiency is

$$\frac{32.2}{105.3} = 30.6\%$$

It will be shown later on how this efficiency can be increased in a more refined cycle.

After passing through the turbine, the expanded gases enter the outlet pipe 23 and flow to the condenser 24. The condenser is located in front of the engine radiator 25 of the motor vehicle engine 26. The form of condenser here shown comprises two headers 27 and 28 connected by tubes 29. Heat radiating surfaces or fins 30 are attached to the tubes to increase the radiating area.

In the condenser the fluid condenses until when it is all liquefied it again reaches point A and the cycle is completed. This is shown by line DA (Fig. 6) and the heat radiated is 73.1 B. t. u./lb.

The condensed liquid collects in the header or liquid receiver 28 and is returned to the pump 15 via pipes 31 and 32 for recirculation. This completes the power cycle.

The pump 15 can be a piston or rotary pump, and such pumps are well known and described in Marks' Handbook.

The pump has a V pulley 33. The pump is mounted on a hinge 34. A Bowden wire 35, Fig. 2, lever or other manual device is provided so that the operator can pull the pump pulley into clutching relation with the V belt 49 on the engine. This forms a convenient means without valves or other devices which might be subject to leakage, for rendering the machine operative or non-operative. When the pump is clutched in, liquid is pumped to the boiler and the system becomes operative if there is sufficient heat in the exhaust.

When the pump is unclutched no liquid enters the boiler and that in the boiler soon evaporates and collects in the condenser, and the pressure in the whole system drops to that of the condenser.

The power portion of the system having been described, the refrigerating portion will now be explained.

The compressor portions 36 of the turbo-compressor 22 becomes effective when the turbine rotates. It applies suction to the evaporator or cooler 37. This causes liquid from the receiver 28 to enter the evaporator via pipe 31 and expansion valve 38. The boiling of the "F-11" under the reduced pressure of the compressor suction causes a drop in temperature of the evaporator which cools air entering the vehicle through the louvres 39. This provides cooling for the vehicle.

The action of the refrigerating system is that of a standard vapo-compression system such as described in Marks' Handbook. To continue the

example, Fig. 6: Assume that the temperature of the evaporator is 50° F. "F-11" in entering the expansion valve from the receiver has the characteristics shown at A.

5 Pressure=39 lbs. per sq. in. absolute;
Temperature=130° F.;
Heat=34.7 B. t. u. per lb.

10 Passage through the throttling expansion valve takes place between A and E. The change is at constant enthalpy, that is, with no change in heat content, the result at E being:

15 Pressure=8.8 lbs. absolute;
Temperature=50° F.;
Heat=34.7 B. t. u. per lb.

The liquid evaporates in the cooler from E to F which represents the refrigerating effect. The condition at F is:

20 Pressure=8.8 lbs. per sq. in. absolute;
Temperature=50° F.;
Heat=98.3 B. t. u. per lb.

The refrigerating effect is:

25 98.3 B. t. u.—34.7 B. t. u.=63.6 B. t. u. per lb.

The vapor now leaves the evaporator via pipe 40 and enters the centrifugal compressor 36 where it is compressed and discharged into pipe 23.

30 Compression is shown from F to G, Fig. 6, and is adiabatic (neglecting friction) so follows a line of constant entropy. At G the condition of a pound of "F-11" vapor is:

35 Pressure=39 lbs. per sq. in. absolute;
Temperature=150° F. approximately;
Heat=111 B. t. u. per lb.

The theoretical shaft work is 111 B. t. u.—98.3 B. t. u.=12.7 B. t. u. per lb.

40 The theoretical coefficient of performance is

$$\frac{\text{B. t. u. } 63.6}{\text{B. t. u. } 12.7} = 5.01$$

which is about the same as in an ammonia system.

45 It has been shown that one pound of "F-11" in the power cycle theoretically provides 32.2 B. t. u. of shaft work with an input of 105.3 B. t. u. Thus the heat input is, theoretically.

50 $\frac{12.7}{32.2} \times 105.3 = 41.5$ B. t. u. per lb. of refrigerant

To allow for mechanical and heat losses, twice this, or say 83.0 B. t. u. may be actually required,

55 $\frac{\text{Ref. effect } 63.6}{\text{Heat input } 83.0} = .766$ $\frac{\text{Heat input } 83.0}{\text{Ref. effect } 63.6} = 1.3$

Assuming that a half ton of refrigerating capacity is required to cool the vehicle this would require absorbing 100 B. t. u. per minute. For this the heat input required would be 130 B. t. u. per minute, which is less than $\frac{1}{10}$ of the heat escaping in the exhaust in the example cited.

60 After compression the vapor enters pipe 23. Figs. 1 and 4, mixes with the exhaust from the turbine and enters the condenser where it is condensed and the refrigerating cycle is completed. This is shown in Fig. 6 as GA.

If the engine is idling or running slowly, the exhaust will not be hot enough to heat the boiler fluid to the condition shown at C in the previous example. Suppose that it can only heat the fluid to 200°, P=104 lbs. at point H, Fig. 6. In this case the temperature is below the critical and the liquid will be pumped to 104 lbs. and gain in heat and temperature until it reaches the saturated

liquid line. It then boils and leaves the boiler at I as a saturated vapor. It then expands in the turbine from I to D. It should be noted that the lines of constant entropy are nearly parallel to the saturated vapor line in this region, so substantially no condensation will occur during expansion.

At I, heat=115 B. t. u. per lb.; so 115-34.7-80.3 B. t. u. is the heat input; and 115-107.8-7.2 B. t. u. is the theoretical shaft work available.

The heat input per pound of refrigerant as calculated above, is

$$\frac{12.7}{7.2} \times 80.3 = 142 \text{ B. t. u.}$$

The turbine is not operating as efficiently as in the first example. This is because of the lower pressure. Assuming that $2\frac{1}{2}$ times the theoretical power is needed to supply the actual refrigeration, the heat input required would be $2.5 \times 142 = 355 \text{ B. t. u.}$

$$\frac{\text{Heat input}}{\text{Refrigerating effect}} = \frac{355}{63.6} = 5.59$$

On this basis, $\frac{1}{2}$ ton of refrigeration will require $100 \times 5.59 = 559 \text{ B. t. u.}$, per minute.

As the available heat from the engine is assumed to be less, due to reduced power, it is probable that 200° boiler temperature represents about the lower limit where the required refrigeration can be obtained.

It is thus seen that as the engine speeds up the exhaust pipe will become hotter and more heat becomes available; so if the cooling system is designed to operate at 200° boiler temperature, the refrigerating effect will be excessive. It is, therefore, desirable to provide means by which the operator can set a maximum cooling effect which will not be exceeded when the engine provides more than the necessary heat.

Referring to Fig. 3, the outlet of the evaporator 37 has an enlarged section 41 adjacent to the pipe 40. A bimetal thermostatic strip 42 is secured at one end to 41 at 43. The other end of the strip has a valve 44 which normally closes off the pipe 45 leading to pipe 23, Fig. 4. If the temperature in 41 drops sufficiently, the strip 42 will bend in a direction to open valve 44 permitting vapor from pipe 23, which is at a higher pressure than the vapor in the evaporator, to enter 41 via pipe 45. This prevents the pressure in the evaporator from decreasing further. Valve 44 could also be used to throttle the gas leaving the evaporator at 40, if valve and thermostat are arranged accordingly, and in such case by-pass 45 could be omitted. However, I prefer the by-pass arrangement because it can be operated without affecting the action of the compressor. As the temperature in the evaporator is a function of the pressure, the temperature will not drop below that necessary to open the valve even when the turbo-compressor speeds up. Chamber 38 is further provided with a diaphragm 46. A settable hand lever 47 changes the tension of a flat spring 48 which presses against the bimetal through the flexible diaphragm, and permits the thermostat to be adjusted to open at any desired temperature.

Thus, when the pump is clutched in, cooling will commence as soon as sufficient heat is available in the exhaust. The cooling will increase with the increase of engine power output until a settable maximum refrigerating effect is reached which maximum will be unaffected by further increases in engine output.

The above description applies to a refrigerating system utilizing waste heat where efficiency is not important.

Where heat is utilized which has to be paid for, a more efficient system requiring different controls is desirable.

Fig. 7 illustrates partly in diagram, partly in detail, an "F-11" gas-operated compression refrigerator suitable for cooling rooms or food refrigerators.

The numeral 50 denotes the boiler or heater. Heater is the better name for it as boiling does not normally take place—that is, the liquid is heated at a pressure above the critical pressure so that it changes, at the critical temperature, to a vapor without boiling. At this point the latent heat of vaporization is zero and the transition from liquid to vapor involves little apparent physical or thermal changes. Moreover, the specific volume of the vapor is identical with that of the liquid. This means that a boiler or heater to produce vapor at this condition not only would have to add less heat per pound but also would have a much smaller volume than if vapor were formed by boiling at say between 300° and 350° (see Fig. 5).

The heater consists, essentially, of a coil or coils, 51, into one end 52 of which liquid "F-11" is pumped and from the other end 53 superheated vapor is discharged. No provision for vapor separating space or liquid circulation is needed. The flow of fluid is countercurrent to the flow of heat. The heater is thus of the simplest and most efficient form.

That part of the tube which is nearest to the flame is wound on a high thermal conductivity metal core 54 such as copper. This is to protect the thin walled tube from becoming so highly heated in spots as to endanger the chemical composition of the fluid.

It is desirable to regulate the source of heat to maintain relatively constant temperature and pressure conditions. While a pressure regulator can be used for this purpose, a temperature regulator is simpler and provides the additional function of shutting off the fuel in event of fuel failure.

The gas burner 55 is provided with a combined regulating and shut-off valve 56. A bimetallic thermostatic strip 57 is embedded in core 54. When the core attains a desired temperature, the strip bends, causing the valve 56 to seat, cutting off the gas supply. An adjustable collar 58 prevents the valve from entirely closing so a small flame is left to serve as a pilot flame and to prevent cooling off of the boiler when the demand is intermittent.

If the gas fails the boiler will cool rapidly because of the evaporation of the fluid. This will cause the end 59 of valve 58 to seat, thus shutting off the gas inlet, and no gas will flow unless the valve is manually operated or the bimetal heated externally.

Heat from the flame passes up through the core, then down outside the core and coils as shown by arrows; then up outside the baffle 59 so that the gas temperature drops uniformly from the flame to the outlet 60 and the "F-11" temperature rises uniformly from the inlet 52 to the outlet 53.

The turbo-compressor 61 is similar to that already described except that it is provided with a rotary or piston liquid pump 62. Condensed liquid from the receiver 63 flows via pipe 64 through the by-pass 65, check valve 66, pipe 67,

heat interchanger 68, and pipe 69, into the heater by gravity when the machine is not in use so that the heater coil 51 is full. On lighting the flame, some of the liquid will boil away before enough pressure is developed to start the turbine, but as soon as the turbine starts, the pump becomes operative and pumps liquid from the receiver into the heater against the pressure head. A relief valve 70, set to open at the desired boiler pressure, prevents excessive boiler pressure, as in Fig. 4.

When the pressure head of the liquid in the container 63 exceeds the pressure in heater (boiler) 51, as when there is no heat in the boiler, valve 66 is open so that the pump 62 is by-passed and the boiler is filled with liquid from the reservoir 63 by gravity. On the other hand, when the boiler is in operation and the boiler pressure exceeds the pressure head of the liquid then valve 66 closes. The valve 70 is an ordinary spring relief valve which opens to by-pass fluid when the boiler pressure exceeds a predetermined amount.

Superheated vapor leaves the heater at 53 and enters the turbine nozzle 71 or nozzles via pipe 72. After expanding and rotating the turbine, it enters the exhaust pipe 73, still in a superheated condition. It then encounters the countercurrent heat interchanger coil 68. This coil is located between the liquid pump 62 and heater inlet 52. The exhaust vapor transfers its superheat to the liquid because of the temperature differential.

The exhaust vapor after leaving the heat interchanger, flows to the condenser 74 via pipe 75, where it is condensed and the liquid then enters the receiver 63 after passing the check valve 76. The purpose of the check valve will be explained later. The power cycle is now completed.

Referring to Fig. 8, the following example will be assumed for illustrative purposes. The standard ton conditions of 86° F. condenser temperature and 5° F. evaporator will be used. The heater (boiler) temperature set for 700° and the relief valve 70 for 800 lbs. Assume further that the machine has been in operation long enough so the pressure of 800 lbs. and temperature of 700° has been reached by the vapor leaving the heater.

Starting at J, the liquid entering the pump has the condition $T=86$, $P=18.3$, $H=25.3$, $V=.01094$ cu. ft. per lb. The work in B. t. u. required to raise its pressure from 18.3 to 800 lbs. =

$$\frac{(P_1 - P_2) (\text{lbs. sq.} \times \text{ft.}) V (\text{cu. ft.})}{\text{Joule}}$$

$$\frac{781.7 \times 144 \times .01094}{778} = 1.6 \text{ B.t.u.}$$

On leaving the pump at K its condition is $T=86$, $P=800$, $H=25.3+1.6=26.9$, the work done by the pump being absorbed by the liquid.

It now enters the heat interchanger, or regenerator 68, Fig. 7, and picks up 27.4 B. t. u. as will be explained later, so that at L its condition is $P=800$, $T=220$ approx., $H=26.9+27.4=54.3$ B. t. u.

It then enters the heater, changes from a liquid to a vapor without boiling, gains superheat until when it leaves the heater it has the condition M, $P=800$ lbs., $T=700^\circ$, $H=181$ B. t. u. The heat absorbed from the flame is $181-54.3=126.7$ B. t. u. per lb.

The superheated vapor now enters the turbine and expands to the condenser pressure, point N, $P=18.3$, $T=275^\circ$ approx., $H=130$, $181-130=51$

(pump work) = 49.4 B. t. u. available for shaft work.

The theoretical thermal efficiency is

$$\frac{49.4}{126.7} = 39\%$$

If steam at 800 lbs., 700° F. is completely expanded to the condenser pressure corresponding to 86°, the theoretical Rankin cycle thermal efficiency is 39.4%. While the theoretical efficiency of "F-11" in this example is slightly less than that of steam, the actual efficiency will probably be higher because of the greater density of the vapor and small expansion ratio, as will be shown later.

At R the vapor, still superheated, enters the refrigerator and cools substantially to condensing temperature by imparting its remaining superheat to the liquid. Its temperature at N is about 275°, whereas the temperature at L is about 220° so there is sufficient temperature differential for the heat transfer. It leaves the refrigerator at O, $P=18.3$, $T=86$, $H=102.6$ so that theoretically $130-102.6=27.4$ B. t. u. of superheat are returned to the system by heating the liquid from K to L.

At O the vapor enters the condenser and radiates its latent heat of 77.3 B. t. u. Another way of expressing the efficiency is

$$\frac{H \text{ in} - H \text{ out}}{H \text{ in}} = \frac{126.7 - 77.3}{126.7} = \frac{49.4}{126.7} = 39.0\%$$

As before, the centrifugal compressor 77 applies suction to evaporator 78 via pipe 79, lowering its pressure and causing it to cool because of the low boiling point of the refrigerant. This decreased pressure causes liquid to flow from the receiver 63 via pipe 80 to the expansion valve 81 and into the evaporator.

Compressed vapor leaves the compressor with some superheat and enters the heat interchanger pipe 73, at 82, where its temperature is about the same as the turbine exhaust at that point. Thus a small amount of additional heat is available for heating the boiler liquid. From this point on the compressor vapor combines with the turbine exhaust.

The refrigerating cycle is shown in Fig. 8. Liquid at J passes the expansion valve to P, evaporates to Q, compresses to R, and condenses back to J.

The refrigerating effect is $92.9-25.3=67.6$ B. t. u. per lb. Shaft work = $105-93=12$ B. t. u. Theoretical coefficient of performance

$$\frac{67.6}{12} = 5.64$$

about the same as ammonia.

Superheat = $105-103=2$ B.t.u.

Heat input per lb. of refrigerant =

$$\left(\frac{12}{49.4} \times 126.7 \right) - 2 = 28.8$$

$$\frac{\text{Refrigerating effect}}{\text{Heat input}} = \frac{67.6}{28.8} = 2.35 \text{ B.t.u.}$$

Assuming that thermal and frictional losses are double the theoretical heat input, the actual refrigerating effect per heat input is approximately 1.17 B. t. u.

In order to maintain automatic control of the temperature a regulator 83 is arranged to shut off the flow of vapor to one or more of the turbine nozzles when the evaporator pressure falls below a desired amount. The regulator comprises a diaphragm 84 one side of which is exposed to

the atmosphere and the other to the evaporator pressure via pipe 85. A valve 86 is attached to the diaphragm. A snap action mechanism comprising rod 87, toggles 88 and spring 89 provides a means whereby the valve shuts tight or opens wide to prevent losses by throttling. A spring 90 and screw 91 provide a settable adjustment. When the pressure in the evaporator falls to the desired amount, the spring plus atmosphere on the diaphragm exceed the evaporator pressure on the inner surface of the diaphragm plus the heater pressure on the valve stem and cause the valve to snap shut thus rendering the turbo-compressor non-operative or slowed down depending upon whether all or only some nozzles are cut off. As already explained, the thermostat 57 keeps the heater pressure and temperature reasonably constant.

When the evaporator pressure rises through to overcome the toggle, the valve 86 snaps open, permitting the turbo-compressor to start or speed up if only some nozzles are affected.

Check valve 76 prevents warm liquid in the receiver from evaporating through the condenser-compressor into the cold evaporator when the compressor is not running.

From the foregoing description, examples, Fig. 5, and from present commercial practice, the following principles become clear:

(1) "F-11" vapor is particularly suitable for a heat-driven compression refrigerating machine because

(a) It is non-toxic, non-combustible, non-corrosive to the kinds of material in ordinary use in refrigeration, and is chemically stable at suitable pressure and temperatures.

(b) Its molecular weight (7.63 times that of water) makes it particularly suitable for use in a centrifugal compressor and this same quality also makes it especially suitable for use in a turbine.

(c) When used as both the refrigerant and vapor for a heat engine in a closed system, the refrigerating and power portions do not have to be hermetically sealed from each other, and certain portions such as condenser and receiver, are common to both.

(d) Its vapor pressure at 75° is approximately atmospheric (14.7 lbs.) so when the system is idle there will be little or no differential pressure such as would tend to cause leakage, either inward or outward.

(e) It permits a hermetically closed system to be used with a single fluid serving as both the power vapor and refrigerant.

Water vapor is now used in jet-compressor air-cooling machines wherein steam is the source of power, but the vacuum needed to obtain the cooling effect is so high that it cannot be successfully maintained indefinitely in a hermetically sealed enclosure of this character. In such system, vacuum pumps are necessary to maintain such vacuum by discharging air leakage into the atmosphere; so the system cannot be said to be hermetically sealed.

With "F-11," however, a vacuum pump is not needed because the vacuum is never high and the system can be effectively sealed against in-leak of air.

(2) "F-11" vapor is particularly suited for a heat engine vapor, entirely irrespective of its use as a refrigerant, under certain conditions. This can best be brought out by a comparison

with steam which is substantially the only vapor in use for that purpose.

(a) Owing to its favorable critical point, pressure, approximately 635 lbs. per square inch, the liquid can be changed to a superheated vapor directly, without boiling, by heating it while at a pressure above the critical so that vapor and liquid circulation and separation space are not necessary. The heater can be countercurrent in character and need be no bigger than necessary to provide the needed heat absorbing surface. As contrasted with this, the critical pressure of steam is 3226 lbs. per sq. in.; and the temperature will be over 1000° F., to prevent excessive condensation during expansion. Such pressures and temperatures are too high for present usage, so steam vaporization must be by boiling the water. Consequently, the steam boiler must be larger per boiler horse power. Moreover, the boiling keeps the boiler at substantially uniform temperature except for the superheater. Consequently, it is unsuitable for countercurrent heating.

While the advantages that could be derived from passing a fluid through the critical point have long been known, it is believed that this combination of "F-11," a constant pressure liquid pump and a thermostatic temperature control forms the first practical application of this idea, and provides a non-toxic, non-combustible fluid at safe and convenient pressures and temperatures.

(b) "F-11" vapor when slightly superheated remains superheated during adiabatic expansion to normal condenser pressures, because it does not cross the saturated vapor line. Thus the "F-11" heat engine runs dry. Steam, however always crosses the saturated vapor line in expanding to a high vacuum even when highly superheated, which means the evil of condensation in cylinders and turbines. I have discovered, however, that with "F-11," a heat interchanger must be used to return the superheat remaining after expansion to the system, if serious heat losses are to be avoided. With steam, no heat interchanger for this purpose is needed because no superheat remains after expansion. With the "Regenerative" steam cycle, steam is bled between the throttle and exhaust and used for heating the feed water, but this occurs before the steam has been completely expanded so is quite a different thing.

(c) Perhaps the greatest advantage of "F-11" as a heat engine vapor is its small volume after expansion as compared with steam. The volume of completely expanded steam is enormous and makes the steam turbine undesirable where small compact power plants are needed, such as in portable plants, motor vehicles, aeroplanes, etc., particularly with air cooled condensers.

In the example shown in Fig. 8, 1 lb. of "F-11" expanded from M to N has 49.4 B. t. u. available for shaft work. Its volume at 0 where it enters the condenser is

$$2.24 \text{ cu. ft. B. t. u. per cu. ft.} = \frac{49.4}{2.24} = 22.0 \text{ B. t. u.}$$

Steam expanded from 800 lbs., 700° to the same condenser temperature would occupy approximately 440 cu. ft. per lb. and has 507 B. t. u. available for shaft work.

$$\text{B. t. u. per cu. ft.} = \frac{507}{440} = 1.15$$

and the above "F-11" ratio 22 to 1÷steam

ratio 1.15 to 1=19. Therefore, "F-11" vapor has 19 times as much heat energy available per cu. ft. of vapor to be condensed as steam has, in the example cited; so for "F-11," the condenser volume can be much smaller than that of a steam plant. It has been noted that the thermal efficiency for "F-11" and steam for this example were about equal.

(d) With steam a very high vacuum, about 1" Hg is needed in the condenser. This is difficult to maintain in a closed sealed circuit, so vacuum pumps are required. With "F-11," however, the condenser pressure is 18.3 lbs. abs. or slightly above atmosphere so vacuum pumps are not required.

(e) Owing to the high density and small volume per B. t. u., pistons, cylinders and turbines for an "F-11" engine will be much smaller than with steam for a given power.

All the above factors make an "F-11" heat engine highly desirable for many power applications, particularly mobile plants where light weight, compactness, low first cost, and low-grade fuel are important and where pressures and temperatures cannot be excessive. For the cases mentioned, the critical pressure of the fluid should not exceed 1,000 lbs.; and the temperatures should be substantially less than 1000° F., as low as 800° F. being much better.

As I have discovered and explained above, "F-11" is phenomenally suitable for use for both refrigeration and motive power in a heat refrigerating machine of the kind herein described, but so far as concerns a heat engine without the refrigerating machine, there are other fluids, particularly those that resemble "Freon-11" in that they are vaporizable working fluids belonging to the same class of compounds containing carbon, chlorine and fluorine as essential elements in stable chemical combination and which have critical pressures substantially below 1000 pounds per square inch absolute; also they are non-combustible, non-corrosive, non-toxic and are stable throughout a wide range of superheat above critical pressure.

Although two embodiments of apparatus have been disclosed for carrying out the novel method of refrigeration in accordance with the invention, various changes in the construction and arrangement of the apparatus, which may now become apparent to those skilled in the art, may be made without departing from the scope of the invention. Reference is therefore to be had to the appended claims for a definition of the limits of the invention.

I claim:

1. That method of producing refrigeration which comprises raising the pressure of a portion of a body of fluid to near or above its critical pressure, vaporizing and superheating said portion of fluid by external heat to produce a superheated vapor, expanding said superheated vapor to produce motive power, utilizing said motive power to compress another portion of said body of fluid, condensing and evaporating said other portion of fluid to produce a refrigerating effect, condensing the vapor of said first portion of fluid, and repeating the cycle.

2. A method of producing refrigeration according to claim 1, characterized by the fact that the body of fluid comprises a halo-fluoro derivative of an aliphatic hydrocarbon.

3. A method of producing refrigeration according to claim 1, characterized by the fact that the

body of fluid comprises carbon, chlorine and fluorine in stable chemical combination.

4. A method of producing refrigeration according to claim 1, characterized by the fact that the body of fluid comprises trichloro-monofluoromethane (CCl_3F).

5. That method of producing refrigeration which comprises raising the pressure of a portion of a body of fluid to near or above its critical pressure, vaporizing and superheating said portion of fluid to near or above its critical temperature by external heat to produce superheated vapor, expanding said superheated vapor to produce motive power, utilizing said motive power to compress another portion of said body of fluid, condensing and evaporating said other portion of fluid to produce a refrigerating effect, condensing the vapor of said first portion of fluid, and repeating the cycle.

6. A method of producing refrigeration according to claim 5, characterized by the fact that the body of fluid comprises a halo-fluoro derivative of an aliphatic hydrocarbon.

7. A method of producing refrigeration according to claim 5, characterized by the fact that the body of fluid comprises carbon, chlorine and fluorine in stable chemical combination.

8. A method of producing refrigeration according to claim 5, characterized by the fact that the body of fluid comprises trichloro-monofluoromethane (CCl_3F).

9. That method of refrigeration which comprises raising the pressure of a portion of a body of fluid to near or above its critical pressure, vaporizing and superheating said portion of fluid by external heat to produce a superheated vapor, expanding said superheated vapor to produce motive power, utilizing said motive power to raise the pressure of said first portion of fluid and to simultaneously compress another portion of said body of fluid, condensing and evaporating said other portion of fluid to produce a refrigerating effect, condensing the vapor of said first portion of fluid, and repeating the cycle.

10. A method of producing refrigeration according to claim 9, characterized by the fact that the body of fluid comprises a halo-fluoro derivative of an aliphatic hydrocarbon.

11. A method of producing refrigeration according to claim 9, characterized by the fact that the body of fluid comprises carbon, chlorine and fluorine in stable chemical combination.

12. A method of producing refrigeration according to claim 9, characterized by the fact that the body of fluid comprises trichloro-monofluoromethane (CCl_3F).

13. That method of producing refrigeration which comprises raising the pressure of a portion of a body of fluid consisting of trichloro-monofluoro-methane (CCl_3F) to approximately 800 pounds per square inch, vaporizing and superheating said portion of fluid by external heat to a temperature of approximately 700° F. to produce a superheated vapor, expanding said superheated vapor to produce motive power, utilizing said motive power to compress another portion of said fluid to a pressure of approximately 39 pounds per square inch, condensing and evaporating said other portion of fluid to produce a refrigerating effect, condensing the vapor of said first portion of fluid, and repeating the cycle.

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