

[54] **REFRIGERATION SYSTEM, HEAT RECOVERY SYSTEM, REFRIGERATED GAS COMPRESSION SYSTEM AND BRAYTON CYCLE SYSTEM**

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[57] **ABSTRACT**

A refrigeration system in which cold refrigerant liquid is produced by permitting refrigerant vapor to flash from a batch

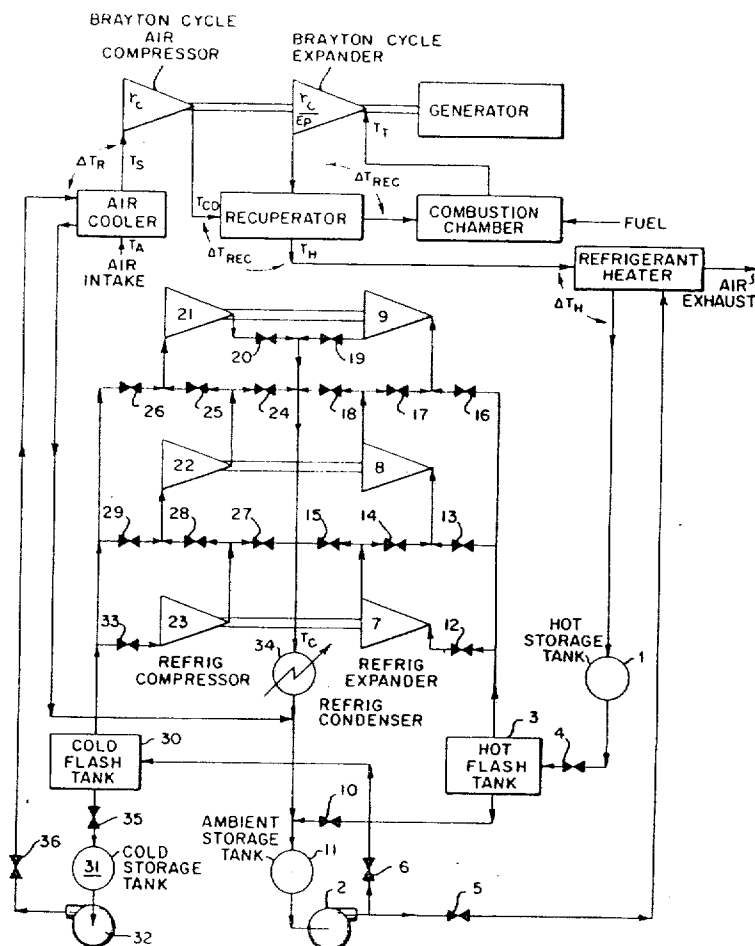
of ambient temperature refrigerant liquid at progressively decreasing temperatures and pressures, which cools down the unvaporized liquid. Vapors are compressed by refrigerant compressors, cooled and condensed at ambient temperature, and used for succeeding refrigerant liquid batches.

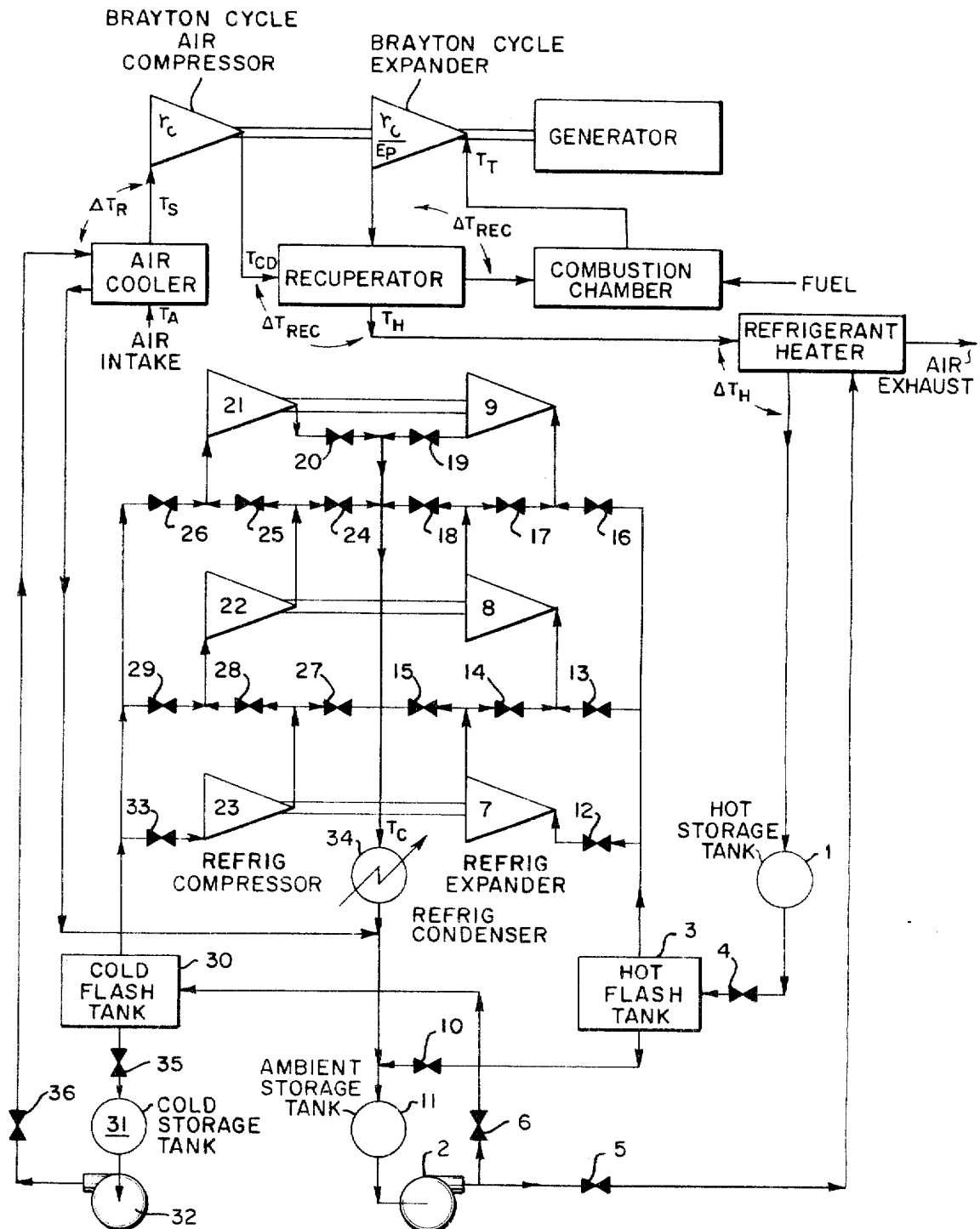
The above refrigerating system is preferably driven by a hot refrigerant liquid at progressively decreasing temperatures and pressures, thus cooling the remaining liquid to about ambient temperature. The vapors flashed drive refrigerant turbines, each of which drives a refrigerant compressor, and after expansion the vapor is cooled and condensed at ambient temperature.

At the start of each cycle, as the batch of hot refrigerant liquid starts to flash the pressure available to the refrigerant expanders is at a maximum. At the same time, as the batch of ambient temperature refrigerant liquid starts to flash, the compression ratio required of the refrigerant compressors is at a minimum. Conversely, at the end of each cycle the pressure available to the expanders is at a minimum and the compression ratio required of the compressors is at a maximum. The refrigerant expanders and compressors therefore are valved sequentially so that at the start of the cycle the expanders are in series and the compressors are in parallel and at the end of the cycle the expanders are in parallel and the compressors are in series.

In another form of the invention, in an open recuperated Brayton cycle engine, air is compressed with refrigeration at compressor inlet at least 50° F. below ambient and to from 0.9 to 1.1 of a calculated optimum compression ratio raised to the 0.286 power.

3 Claims, 1 Drawing Figure





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REFRIGERATION SYSTEM, HEAT RECOVERY SYSTEM, REFRIGERATED GAS COMPRESSION SYSTEM AND BRAYTON CYCLE SYSTEM

BACKGROUND OF THE INVENTION

The efficiency of a refrigeration system is determined by the work which is required to remove the necessary quantity of heat from a process stream which it is desired to cool to some chosen temperature. When this heat is removed in stages of progressively lower temperatures, the work which is required is reduced in accordance with the number of stages employed; the greater the number of stages, the less is the work required and the greater is the efficiency of the refrigeration system.

The efficiency of a heat recovery system is determined by the work which it produces from the heat which it removes from a process stream when it cools the process stream through a chosen temperature range. When this heat is recovered and is used in stages of progressively lower temperatures, the work which is produced is increased in accordance with the number of stages employed; the greater the number of stages, the greater is the work produced, and the greater is the efficiency of the heat recovery system.

If the gas which is about to enter a gas compressor is cooled by refrigeration, less work is required to compress the gas through a desired compression ratio, and if the inlet gas is cooled to a suitable temperature the gas compression work which is saved is more than the refrigeration work which is needed to cool the gas. The total work, which is the sum of the cooled-suction gas compression work and the refrigeration work, is less than the uncooled-suction gas compression work alone. The amount of work which is saved depends on the efficiency (the number of stages) of the refrigeration system and on the gas compression ratio; the more efficient the system and the greater the ratio, the greater the saving. With a given compression ratio, and when cooling to a given temperature, a two stage refrigeration system provides a greater saving than a single stage, and a three stage system provides a still greater saving.

In a Brayton cycle engine, air enters the engine at atmospheric pressure, is compressed, is heated and then is expanded back to atmospheric pressure. The net work output of the engine is the relatively small difference between two quite large numbers, i.e., it is the difference in the total work produced by its air expander and the work consumed by its air compressor. The work produced by the air expander of a 5.4 ratio simple Brayton cycle engine is about 2.77 times the net work output of the engine, and when the compressor takes suction at ambient temperature (for example 100° F.) the work consumed by the air compressor is about 1.77 times the net work output. If the ambient temperature air is refrigerated before it enters the compressor, the work output of this Brayton cycle engine increases because the compression ratio increases, and the expansion ratio increases accordingly; because the work produced by the air expander therefore increases; and also because the mass flow air through the engine increases, due to the greater density of the cold air. The work which is required to refrigerate the inlet air must, of course, be deducted from the work which is produced by the Brayton cycle engine, but even when an inefficient single stage refrigeration system is used the refrigerated suction engine delivers more usable shaft work than does the same engine if it takes suction at 100° F.

SUMMARY OF THE INVENTION

One aspect of the present invention concerns an improved refrigeration system.

In the refrigeration system, cold liquid phase refrigerant is used countercurrently to cool a process stream from about ambient temperature to some chosen lower temperature. The cold refrigerant liquid thereby is heated to about ambient temperature. The liquid refrigerant is maintained under sufficient pressure so that it does not boil at the top temperature which it reaches as it countercurrently cools the process stream. A

batch of cold liquid refrigerant is produced by permitting a batch of ambient temperature refrigerant liquid to flash in a flash tank. During its flashing, the refrigerant liquid boils at progressively lower pressures and temperatures. The vapor which evolves is removed continuously by refrigerant compressors and then is compressed, is cooled, and is condensed at about ambient temperature. The flashing results in cooling down that portion of the refrigerant liquid which does not flash, and when the liquid thus has been cooled down sufficiently it is stored temporarily in a cold storage tank and subsequently it is used countercurrently to cool the process stream. Although the cold liquid is produced in batches, the cold storage tanks permits using it at a continuous rate to cool the process stream. Since the batch of ambient temperature refrigerant liquid is flashing at progressively lower temperatures, heat is being removed from it continuously at progressively lower temperatures. The process thus is equivalent to one in which heat is removed and rejected in a very large (in theory, an infinite) number of stages of refrigeration. Consequently, in theory this refrigeration system is reversible thermodynamically and is potentially the most efficient one which can be employed.

The heat recovery system ambient temperature refrigerant liquid is heated and is maintained under sufficient pressure so that it does not boil while it is being heated. The hot liquid is stored temporarily in a hot storage tank. A batch of the heated liquid is transferred from the hot storage tank to a hot flash tank, where it is permitted to flash at decreasing temperatures and pressures until finally it reaches a predetermined lower temperature, usually about ambient temperature. The flash vapor passes through refrigerant expanders, each one driving a corresponding refrigerant compressor, as has been referred to above. The refrigerant expander may drive its refrigerant compressor directly or it may drive the compressor through a constant or through a variable speed changing device. The flash vapor which leaves the last expander is condensed at about ambient temperature. Since the batch of hot refrigerant liquid is flashing at progressively lower temperatures, it is giving up heat continuously at progressively lower temperatures. The process thus is equivalent to one in which heat is recovered and used in a very large (in theory, an infinite) number of stages of heat recovery. Consequently, in theory this heat recovery system is reversible thermodynamically and is potentially the most efficient one which can be employed.

At the beginning of the cycle the vapor coming off the hot flash tank is at its maximum temperature and pressure and can produce the maximum amount of expansion work. At the beginning of the cycle the vapor coming off the cold flash tank is also at its maximum temperature and pressure and it requires the minimum amount of compression work. Therefore, at the beginning of the cycle the available overall expansion ratio is used in three expanders which are valved so that they operate in series, and the required compression ratio is provided by the paired three compressors which are valved so that they operate in parallel. (Three pairs of expanders and compressors form a very satisfactory compromise between efficiency and equipment costs, and so the general summary of the invention will be described in connection with such an arrangement, it being realized that there may be only two pairs of expanders and compressors or more than three.) As the cycle progresses the pressure of the flashing refrigerant in the hot flash tank decreases and the pressure of the flashing refrigerant in the cold flash tank also decreases, so the work which is produced by each refrigerant expander decreases and the work which is required by each refrigerant compressor increases. When the cycle has progressed to the point where each expander is unable to drive its paired compressor, the valving is switched so that the hot vapor passes through two expanders in series and the cold vapor passes through two compressors in parallel. (One expander and its paired compressor are cut off temporarily by valves.) The smaller overall expansion pressure ratio available at this time is used in only two, rather than three, expanders and each expander produces

more work (sufficient to drive its paired compressor). After a further drop in the pressure of the vapor from the hot flash tank, valves are again switched so that two expanders in parallel drive two compressors in parallel. After a still further drop in the pressure of the hot flash tank, two expanders in parallel drive two compressors in series, and finally, at the end of the cycle three expanders in parallel drive three compressors in series.

By switching the compressor valves it is possible to operate with first one, then two, and then three stages of compression. When the overall compression ratio which is needed is produced in three stages rather than in one, less work is required by each compressor, but there is a concomitant decrease in the total rate at which vapor flows from the cold flash tank, and there is a corresponding decrease in the rate at which the flash tank cools down.

By switching the expander valves it is possible to operate with first three stages, then two stages, and then one stage of expansion. When the overall expansion ratio which is available is used in one stage of expansion rather than in three, more work is produced by each expander, but there is a concomitant increase in the total rate at which vapor flows from the hot flash tank, and there is a corresponding increase in the rate at which the flash tank cools down.

At any given time during the course of the flash cycle the rate at which vapor flows from the hot flash tank through the expanders is usually quite different from the rate at which vapor flows from the cold flash tank through the compressors, but at each and every instant of the flash cycle the work which is produced by each refrigerant expander is exactly the same as the work which is required by its paired compressor.

During the whole of the cycle the vapor leaving the last refrigerant expander is cooled to substantially ambient temperature and is condensed to a liquid. The refrigerant compressor discharge vapor likewise is cooled and condensed. The ambient temperature condensed refrigerant liquid may be the feed both for the cold flash tank and for the waste heat recovery system. (It is, of course, possible, and in some cases it may be desirable, to use different fluids for the refrigeration cycle and for the work producing cycle. In this case two separate condensers are used.) When a cycle is completed the ambient temperature liquid which is left in the hot flash tank is transferred to the ambient storage tank; the cold liquid which is left in the cold flash tank is transferred to the cold storage tank. Then a fresh batch of hot refrigerant liquid is transferred from the hot storage tank to the hot flash tank, a fresh batch of ambient temperature refrigerant liquid is transferred from the ambient storage tank to the cold flash tank, and the cycle is repeated.

In the refrigeration system of the present invention the heat value of the refrigeration work (W_R) which is required to cool a process stream (for example, a lb. mol of gas, typically air) from ambient temperature (T_A) to some chosen lower temperature (T_S) is

$$W_R = \frac{C_p}{E_R} \left[T_C \ln \frac{(T_A - \Delta T_R)}{(T_S - \Delta T_R)} - (T_A - T_S) \right]$$

where C_p = the molal specific heat of the gas (for air about 7.0); T_C = the chosen condensing temperature; ΔT_R = the chosen temperature difference between the gas which is leaving the cooler and the refrigerant liquid which is entering the cooler; and E_R = the chosen efficiency of the refrigerant compressor.

As an example, assume that $T_A = 560^\circ \text{R.}$, $T_C = 580^\circ \text{R.}$, $\Delta T_R = 20^\circ$, $E_R = 0.8$, and that it is desired to cool the gas to 400°R.

$$W_R = \frac{7.0}{0.8} \left[580 \ln \frac{540}{380} - (560 - 400) \right] = 381 \text{ BTU/lb. mol.}$$

For any chosen refrigeration system a coefficient of performance C_R (at a particular T_S) can be calculated, where C_R is the ratio of W_R (in heat units) to the heat removed from a lb. mol of gas when the gas is cooled from ambient temperature T_A to a chosen lower temperature T_S . Mathematically

$$C_R = \frac{W_R}{C_p(T_A - T_S)},$$

and for the assumed set of conditions,

$$C_R = \frac{381}{7.0(560 - 400)} = 0.34 \text{ BTU}$$

of work required for each BTU which is removed in cooling the gas. Although for the chosen system C_R is dependent to some extent on the values of C_p , T_C , ΔT_R , T_A , and E_R , it is strongly dependent on T_S . As an illustration, when the values of C_p , T_C , ΔT_R , T_A , and E_R are the same as in the previous example, but T_S is 450°R. rather than 400°R. , $C_R = 0.254$ rather than 0.34.

In the heat recovery system of the present invention, the heat value of the recovery work (W_H) which is produced from the heat released by a process stream (for example a pound-mol of gas, typically air) as the stream cools from a super-ambient temperature (T_H) to ambient temperature (T_A) is

$$W_H = C_p E_H \left[(T_H - \Delta T_H) - T_C \left(\ln \frac{(T_H - \Delta T_H)}{T_C} + 1 \right) \right]$$

where E_H is the chosen efficiency of the refrigerant expander, and ΔT_H is the chosen temperature difference between the gas and the refrigerant.

As an example, assume that $E_H = 0.8$, $\Delta T_H = 50^\circ$, $T_C = 580^\circ \text{R.}$, and that it is necessary that the system produce 381 BTU of work per lb. mol,

$$W_H = 7.0 \times 0.8 \left[(T_H - 50) - 580 \left(\ln \frac{(T_H - 50)}{580} + 1 \right) \right] = 381 \text{ BTU/lb. mol.,}$$

from which $T_H = 950^\circ \text{R.}$

For any chosen heat recovery system a coefficient of performance C_H (at a particular T_H) can be calculated, where C_H is the ratio of W_H (in heat units) to the heat which becomes available when a lb. mol of gas cools from an initial temperature T_H to ambient temperature T_A . Mathematically,

$$C_H = \frac{W_H}{C_p(T_H - T_A)},$$

and for the assumed set of conditions,

$$C_H = \frac{381}{7.0(950 - 560)} = 0.14 \text{ BTU}$$

of work which is produced from each BTU of available heat. For the chosen system, C_H is somewhat dependent on the values of C_p , E_H , ΔT_H , T_A and T_C , but it is strongly dependent on T_H . As an illustration, when the values of C_p , E_H , ΔT_H , T_A and T_C are the same as in the previous example, but T_H is 900° rather than 950°R. , $C_H = 0.114$ rather than 0.14.

In another aspect of the present invention the ambient temperature power producing refrigerant liquid is used counter-currently to cool the gas which is discharged from a gas compressor, and the ambient temperature power producing refrigerant liquid is heated thereby. When the gas suction temperature and the gas compression ratio are suitably matched, the heat of compression of the gas heats the refrigerant power liquid to a temperature which is high enough so that the power liquid provides all of the work which is needed to refrigerate the gas which is about to enter the gas compressor, and no external work is needed to operate the refrigeration cycle. (Throughout the remainder of the specification this self-driven, sequentially valved refrigerant expander-compressor system will be referred to as the Treadwell System. This can be considered as the most economical or the preferred way of operating this form of the present invention, although as an alternate the refrigeration cycle portion of the system may be powered in whole or in part by an independent motor or steam turbine driver.) When the heat of gas compression provides the work of refrigeration, T_H is equal to the gas compressor discharge temperature, and $T_H = (T_S/E_C) (r_c^n - 1) + T_S$, wherein E_C is the efficiency of the gas compressor; r_c is the gas compression ratio; and n is the numerical value of adiabatic exponent ($k-1/k$) (for air, $k = 1.4$, and $n = 0.286$). When for a desired compression ratio r_c it is desired to determine the

matching T_s , a trial T_s is selected and a corresponding T_H is calculated from the preceding formula. The refrigeration work W_R which is required for a chosen refrigeration system to cool the air to the trial T_s is calculated by the method previously explained. This W_R is compared to the calculated heat recovery work W_H which is produced by a chosen heat recovery system (using the calculated T_H which corresponds to the trial T_s). A series of values of T_s is tried until the refrigeration work which is required for the trial T_s is equal to the heat recovery work which is produced when the corresponding calculated T_H is used.

As an example, assume that a compression ratio of 15.0 is desired and that the self-driven Treadwell system is to be used. Several values of T_s are tried, which finally converge on 400° R., and as a check, this value for T_s , together with the desired value of 15.0 for r_c , is substituted in the previously given equation

$$T_H = (T_s/E_c)(r_c^n - 1) + T_s.$$

In substituting,

$$T_H = (400/185)(15.0^{0.286} - 1) + 400,$$

from which $T_H = 950°$ R. It was previously shown that with the Treadwell System, when $T_H = 950°$ R., the work which is produced by the heat recovery system supplies the work which is required by the refrigeration system when $T_s = 400°$ R.

With this suction temperature the gas compression requires only 71.5 percent of the single stage adiabatic work which is required when suction is taken at 560° R. In the prior art a compression ratio of 15.0 cannot be achieved in a single stage compressor with ambient temperature (560° R.) suction because the discharge temperature of 1,330° R. (870° F.) is much too high, and because far too much compression work is consumed, so a compression ratio of this magnitude usually requires two expensive, intercooled stages of compression. However, when the Treadwell System is used to cool the suction gas to 400° R., the same compression ratio of 15.0 is readily conducted in a single stage compressor which produces a discharge temperature of only 950° R. (490° F.), and at the same time the net work is less than the work which is required by the more expensive two stage compressor. If other less efficient refrigeration and heat recovery systems are used in place of the Treadwell System, more refrigeration work is needed to cool the suction to 400° R., and less heat recovery work is produced from the T_H of 950° R. Therefore, the heat recovered cannot provide refrigeration to a temperature as low as 400° R. and the net gas compression work is greater.

When the Treadwell System is used to cool the gas which is about to enter a gas compressor, the subsequent work of adiabatic compression closely approximates the work of isothermal compression when the isothermal compression process is conducted at ambient temperature. In fact, if ΔT_R and ΔT_H are made infinitely small, and T_c is made the same as T_A , when the Treadwell System is used to cool the suction gas the work of adiabatic compression exactly equals that of ambient temperature isothermal compression.

Isothermal compression requires the least amount of work because in theory the process is reversible thermodynamically. With adiabatic compression, the gas which is discharged from the compressor is at a higher temperature than the gas which enters the compressor, and the heat energy which is required to produce this increase in temperature is provided at the expense of additional work energy which has been delivered to the compressor. The compressed gas is discharged from the compressor at a relatively low temperature level and its heat normally is wasted by being rejected to cooling water in an inter or an after cooler. The direct rejection of this heat to cooling water is a completely irreversible process thermodynamically. By contrast, in the heat recovery portion of the Treadwell System heat is also rejected to cooling water, but only after it has produced work in the refrigerant expander. As a result, in the Treadwell System, in theory the heat rejection is completely reversible thermodynamically. Similarly, in the refrigeration portion of the

Treadwell System, in theory the heat rejection is completely reversible. When in theory the Treadwell System is used with an adiabatic gas compressor, the gas initially is at ambient temperature and after compression and heat recovery is also at ambient temperature; the refrigeration process is reversible; the heat recovery process is reversible; and the adiabatic compression process is reversible. Since the final temperature of the compressed gas is the same as its initial temperature, and since in theory all of the processes involved are reversible, in theory adiabatic compression using the Treadwell System is equivalent to isothermal compression.

It will be noted that when the compressed gas supplies the heat which furnishes the work which is required by the refrigeration system, the system is self-regulating. If the gas compressor discharge temperature rises, more heat is available, more work is developed and more refrigeration work is available to lower the temperature of the gas which is about to enter the compressor. When this temperature is lowered, the temperature of the gas which is discharged from the compressor is in turn lowered. If the gas compressor discharge temperature falls, less heat is available, less work is developed and less refrigeration work is available, so there is an increase in the temperature of the gas which is about to enter the compressor, and this increase in turn raises the temperature of the gas which is discharged from the compressor. This automatic self-regulation is an important operating advantage of this aspect of the present invention.

The refrigeration system also can be used to cool substances other than gas. In such a case heat from another source may be used to raise the temperature of the refrigerant power liquid to a level high enough so that it will provide all the work which is needed by the refrigerant compressors. However, work is saved to the extent that waste heat is furnishing at least some of the work for the refrigeration system, even though it may not be all of the work.

When the Treadwell System is used to cool the gas entering a gas compressor, the heat of gas compression need not be the only source of heat for the power producing refrigerant liquid. There may be other sources, which further can increase the amount of self-driven refrigeration that can be produced, and this can permit a still lower gas compressor inlet temperature, with a still further saving in compressor work.

The combination of the Treadwell System with a Brayton cycle engine constitutes another preferred form of the present invention wherein the air which is about to enter the compressor of a recuperated Brayton cycle engine is refrigerated and all the work of refrigeration is provided by the heat which is recovered from the exhaust air which is leaving the recuperator of the same engine. According to the present invention, it has been discovered that the maximum Brayton cycle work is produced when r_c^n has an optimum value defined by optimum $r_c^n = [(E_c E_T E_P T_T / T_s)^{1/2}]^{1/2}$, (Eq. 1), where $E_P \times (r_c / r_e)^n$ (the expansion ratio r_e is smaller than the compression ratio r_c because of parasitic pressure losses in the system); E_c and n are as previously defined; E_T is the air expander efficiency; T_T is the air expander inlet temperature; and T_s is a chosen suction temperature, which is usually selected for practical reasons, such as the cost and the performance of available refrigeration equipment.

It has been discovered according to the present invention that if the improvement in performance is to be of practical significance, T_s should be at least about 50° F. below the ambient air temperature that is ordinarily encountered. It has further been discovered that satisfactory results can be achieved over a range of from 10 percent greater to 10 percent smaller than the r_c^n actually calculated.

For any chosen T_s there is a unique value of r_c^n at which the work produced in a Brayton cycle is a maximum. The net work output, i.e. the Brayton cycle work less the refrigeration work, depends, of course, on the efficiency of the refrigeration system which is chosen, but once the refrigeration system is chosen, at the chosen T_s the net work output is a maximum at the same unique value of r_c^n at which (for the same T_s) the Brayton cycle work output is a maximum.

It was shown previously that when the Treadwell System is used at the assumed conditions, the exhaust air must enter the heat recovery system at 950° R. in order for the heat recovery system to provide the refrigeration work which is needed when the refrigeration system cools from 560° R. to 400° R. the air which is about to enter the air compressor. It was also shown previously that for maximum Brayton cycle work output the optimum $r_c^n = (E_C E_T E_P T_T / T_S)^{1/2}$, (Eq. 1).

As an example, assume that $E_C = 0.85$; $E_T = 0.87$; $E_P = (1.05)^n = 1.014$; $T_T = 1,960^\circ \text{R.}$ (1,500° F.); and $T_S = 400^\circ \text{R.}$, $(0.85 \times 0.87 \times 1.014 \times 1,960/400)^{1/2} = 1.915 = r_c^n$.

With this r_c^n , E_C and T_S , the air compressor discharge temperature (T_{CD}) is about 830° R. The recuperator temperature approach (ΔT_{REC}) is the difference between the temperature of the compressed air which is entering the recuperator and the temperature of the exhaust air which is leaving the recuperator and is, therefore, 120° F. (950 - 830). This is approximately the difference between the temperature of the compressed air which is leaving the recuperator and the temperature of the exhaust air which is leaving the expander and is entering the recuperator. For the assumed r_c^n , the assumed E_T and the assumed E_P , the exhaust air leaves the expander at a temperature of about 1,160° R., so the compressed air is heated recuperatively to about 1,040° R. (1,160 - 120) and the cycle operates at a thermal efficiency of about 40.2 percent.

It is to be noted that when the Brayton cycle operating conditions (T_S and its related r_c^n , T_T , E_P , E_T) and the Treadwell System operating conditions (T_A , T_C , ΔT_R , ΔT_H , E_R , E_H) are known, the recuperator temperature approach (ΔT_{REC}) is uniquely fixed. (This is true only when the work of refrigeration is provided by the heat which is available in the recuperator exhaust.) Because the air expander inlet temperature (T_T); the system pressure losses which fix E_P ; the component efficiencies (E_C , E_T , E_R , E_H); the heat exchanger temperature approaches (ΔT_R , ΔT_H); the condensing temperature (T_C); and the ambient temperature (T_A) are all constants for any selected system, the recuperator temperature approach (ΔT_{REC}) is a function only of T_S and r_c^n . Since T_S and r_c^n are related by the previously given (Eq. 1) for optimum r_c^n for maximum Brayton cycle work output, r_c^n is a function only of ΔT_{REC} . Therefore, when for economic or other reasons a specific recuperator temperature approach is chosen, this choice also determines the optimum r_c^n which is required for maximum work output.

This optimum r_c^n is given by the equation, (Eq. 2), $r_c^n =$

$$E_T E_P E_T \pm \left[E_T E_P T_T^2 - 4 E_T E_P T_T \left(T_A \left[1 + \frac{C_R}{C_H} \right] - \Delta T_{REC} \right) \left(1 - E_C \left[1 + \frac{C_R}{C_H} \right] \right) \right]^{1/2} \\ \frac{2 \left(T_A \left[1 + \frac{C_R}{C_H} \right] - \Delta T_{REC} \right)}{2 \left(T_A \left[1 + \frac{C_R}{C_H} \right] - \Delta T_{REC} \right)}$$

wherein E_T , E_P , T_T , T_A , and E_C are as previously defined; ΔT_{REC} is the chosen recuperator temperature approach; C_R is the coefficient of performance of the chosen refrigeration system at the T_S at which it operates; and C_H is the coefficient of performance of the chosen heat recovery system at the T_H at which it operates. It is to be noted that although T_S does not appear explicitly in the equation, it is inherent in the calculation of C_R and C_H . The preceding equation holds true only when the work of refrigeration is supplied by the heat which is available in the recuperator exhaust.

In using this equation, a trial T_S is selected, and from $r_c^n = (E_C E_T E_P T_T / T_S)^{1/2}$, relating r_c^n to T_S , a trial optimum r_c^n is calculated. The corresponding air compressor discharge temperature T_{CD} then is calculated

$$(T_{CD} = T_S + (r_c^n - 1) \frac{T_S}{E_C}),$$

and to it is added the desired recuperator temperature approach (ΔT_{REC}) to give T_H . The coefficient of performance C_H of the chosen heat recovery system is calculated for this T_H , and the coefficient of performance C_R of the chosen refrigeration system is calculated for the same trial T_S . The values for

T_T , E_T , ΔT_{REC} , C_R , C_H , T_A and E_C are substituted in Equation 2, and the resulting r_c^n is compared with the trial r_c^n , Eq. 1. If this resulting r_c^n is not the same as the trial r_c^n , a new trial r_c^n is calculated from a new trial T_S , a new C_H and a new C_R are calculated, and a new resulting r_c^n is calculated. This procedure is repeated until the calculated resulting r_c^n is the same as the calculated trial r_c^n .

With a recuperator temperature approach of 10° F., (requiring an extremely large and very expensive recuperator), by calculation the suction temperature is 442° R, the optimum r_c^n is 1.825, the cycle thermal efficiency is 42.8 percent and the power production is 7.7 BTU of work for each cu. ft. of air displaced by the compressor, i.e. for each cu. ft. of compressor volumetric capacity. Other things being equal, the cost of a compressor is related to its volumetric capacity, and the cost of the compressor is a substantial part of the cost of a Brayton cycle engine. The work produced per unit of compressor capacity is, therefore, a measure of the cost of the equipment used to produce power in a Brayton cycle engine.

With a recuperator temperature approach of 100° F., the suction temperature is 407° R., the optimum r_c^n is 1.898, the cycle thermal efficiency is 40.7 percent and the power production is 8.6 BTU of work per cu. ft. of compressor capacity.

With a recuperator temperature approach of 120° F., the suction temperature is 400° F., the optimum r_c^n is 1.915, the cycle thermal efficiency is 40.2 percent, and the power production is 8.96 BTU of work per cu. ft. of compressor capacity.

With a recuperator temperature approach of 150° F., the suction temperature is 390° R., the optimum r_c^n is 1.94, the cycle thermal efficiency is 39.7 percent, and the power production is 9.42 BTU of work per cu. ft. of compressor capacity.

When no recuperator is used, and the heat of the air which is leaving the air expander is used only to power the refrigeration system, the suction temperature is 343° R., the optimum r_c^n is 2.07, the cycle thermal efficiency is 37 percent, and the power production is 12.3 BTU of work per cu. ft. of compressor capacity.

When taking suction at 560° R., with an r_c^n of 1.62 and a recuperator temperature approach of 150°, in the prior art a recuperated uncooled Brayton cycle engine produces 3.94 BTU of work per cu. ft. of compressor capacity, at a thermal efficiency of about 29.2 percent. The recuperator of this engine exhausts at about 1,120° R., and when this exhaust heat is used to make 50 psig steam in a waste heat boiler, the steam produces in an expensive separate steam turbine about 1.38 BTU of additional work, for a total of 5.32 BTU for each cu. ft. of capacity of the air compressor of the Brayton cycle engine. The combined cycle thermal efficiency is about 39 percent.

With the same recuperator temperature approach of 150°, an engine designed in accordance with the present invention has a suction temperature of 390° R., operates with a cycle thermal efficiency of about 39.7 percent, and produces a net work output of about 9.42 BTU per cu. ft. of compressor capacity, which is about 2.39 times that of the uncooled standard recuperated cycle engine and about 1.77 times that of the uncooled combined recuperated cycle engine. It is to be noted that this 9.42 BTU work output is produced by the Brayton cycle engine alone, and that there is no need for an expensive separate power producing steam turbine.

All the preceding Brayton cycle examples have been based on using the Treadwell System, but with a Brayton cycle it is possible to use other, less efficient combinations of self-driven refrigeration-heat-recovery systems. For example, a single or a multi-stage refrigeration system can be powered by a heat recovery system employing a single or a multi-stage boiler. For a chosen heat recovery system and for a chosen refrigeration system it is necessary to give due consideration to all the pertinent factors, such as the heat exchanger temperature approaches, the component efficiencies, the number of stages, and the like, and to calculate for each specific T_H its coefficient

cient of performance C_H and for each specific T_S its coefficient of performance C_R . The previously described procedure using a trail T_S is then employed to determine the optimum r_c^n for a chosen ΔT_{REC} .

The preceding Eq. 2 for r_c^n is, therefore, applicable regardless of the type of refrigeration system or of the type of heat recovery system which is used.

The thermal efficiency of a prior art Brayton cycle engine suffers considerably if it becomes necessary to operate the engine at a reduced capacity. For example, the previously described prior art engine, with a recuperator temperature approach of 150° , an r_c^n of 1.62, and a suction temperature of 560° R. (ambient temperature = 560° R.), operates at a design point thermal efficiency of 29.2 percent. When called on to operate at 42 percent of its design point capacity, the same engine operates at a thermal efficiency of 17.8 percent, which is only 61 percent of its design point efficiency. This is because the most practical way to reduce the capacity of a prior art engine is to lower its firing temperature, which simultaneously lowers its Carnot cycle efficiency as well. In this example, at the design point the firing temperature is $1,500^\circ$ F., but at 42 percent capacity the firing temperature is only $1,090^\circ$ F.

As engine built in accordance with the present invention, with a recuperator temperature approach of 150° , an r_c^n of 1.94, and a suction temperature of 390° R. (ambient temperature = 560° R.), operates at a design point thermal efficiency of 39.7 percent. When this engine is called on to operate at 42 percent of its design point capacity, it operates at a thermal efficiency of 29.5 percent, which is almost 75 percent of its design point efficiency. In this case, the capacity is reduced by allowing the level of suction refrigeration to rise to 560° R. The engine capacity is easily varied between 42 percent and 100 percent of its design capacity by adjusting the temperature level to which the suction air is cooled. (This adjustment can readily be made in various ways, for example, by throttling the flow of cooling water to the refrigeration condenser.) When the temperature of the suction air is raised, the engine capacity is reduced. The firing temperature remains at $1,500^\circ$ F. for this entire range of engine capacities. If the firing temperature is lowered, the engine capacity can be reduced to even less than 42 percent of design capacity, at the penalty of somewhat lowered thermal efficiency.

BRIEF DESCRIPTION OF THE DRAWINGS

The drawing shows, in purely diagrammatic form, the combination of the refrigerant system for cooling the air compressor for a recuperated open Brayton cycle.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

On the drawing, air at ambient temperature T_A enters the air cooler at the point marked Air Intake, and is cooled to a temperature T_S which is at least 50° F. below the ambient temperature at which it is intended to operate the engine. The cooled air enters the air compressor, in which it is compressed through the calculated optimum compression ratio r_c . The compressed air enters the recuperator at a discharge temperature which is determined by this optimum r_c and suction temperature T_S . In the recuperator the air is heated by heat exchange with the exhaust from the Brayton cycle expander and passes into a conventional Brayton cycle combustion chamber. In this chamber fuel is burned and the temperature of the compressed air is raised further to T_r , which is the maximum temperature that the materials of the expander can withstand. The maximum permissible level of T_r is in no sense changed by the present invention.

The compressor is driven by the expander. The difference in the work which is produced by the expander and the work which is required by the compressor constitutes the net work output of the Brayton cycle. This is symbolized on the drawing by the power output shaft being connected to and driving the generator.

The expander exhaust gases go to a recuperator, which they leave at a temperature T_H . T_H is determined by the discharge temperature of the compressed air and by the temperature differential ΔT_{REC} indicated. The exhaust gases then pass through a refrigerant heater in which pump (2) keeps the refrigerant liquid at a sufficient pressure so that it does not boil. The amount of liquid which goes to the heater is determined by the adjustment of valves (5) and (6). In the heater the liquid refrigerant is heated up to temperature T_H minus the small temperature differential ΔT_H which is required for heat exchange. The exhaust gases then are exhausted as indicated, ordinarily at ambient temperature plus the same small temperature differential ΔT_H .

The hot refrigerant liquid flows from the refrigerant heater into a suitably insulated hot storage tank (1).

From time to time valve (4) is opened, and a batch of hot liquid is transferred from hot storage tank (1) to hot flash tank (3). The liquid holding capacity of hot storage tank (1) is sufficiently greater than that of hot flash tank (3) to permit substantially continuous operation. The drawing is diagrammatic, so only a single hot flash tank is shown, but multiple tanks can be used, if desired.

In hot flash tank (3) the heated refrigerant liquid, initially under such pressure as may be needed to prevent boiling in the refrigerant heater, flashes at decreasing temperatures and pressures until it reaches a minimum temperature and pressure, normally about ambient temperature. Valve (10) then is opened, and the remaining unvaporized liquid is permitted to flow into ambient storage tank (11).

Three refrigerant expanders (7), (8) and (9) constitute the power generating portion of the refrigeration system. The pattern of flow through the expanders is controlled by valves (12), (13), (14), (15), (16), (17), (18), and (19). At first, when the vapor in the hot flash tank is at maximum temperature and pressure, valves (12), (14), (17) and (19) are opened, and valves (13), (15), (16) and (18) are closed. As a result, refrigerant vapor passes in series through (7), (8) and (9). These expanders drive corresponding refrigeration compressors (21), (22) and (23). This is symbolized on the drawing as a common shaft connecting expander (7) and compressor (23), a common shaft connecting expander (8) and compressor (22), and a common shaft connecting expander (9) and compressor (21).

At the start, the temperature and pressure in hot flash tank (3) is at a maximum and the flash vapor passes through expanders (7), (8) and (9) in series. At the same time, the pressure and temperature in cold flash tank (30) is at a maximum, and the load on refrigeration compressors (21), (22) and (23) is at a minimum. The pattern of flow through these compressors is controlled by valves (20), (24), (25), (26), (27), (28), (29) and (33). At the start the three compressors operate in parallel, valves (20), (24), (26), (27), (29) and (33) being open, and valves (25) and (28) being closed. The load on compressors (21), (22) and (23) increases as the temperature and pressure of the refrigerant in cold flash tank (30) drops. When expanders (7), (8) and (9) can no longer produce sufficient power to drive the compressors, valves (17), (19), (20) and (26) are closed. This has the effect of cutting off expander (9) and compressor (21); and now expanders (7) and (8) in series drive compressors (22) and (23) in parallel.

After a further lapse of time, the pressure and temperature of the refrigerant in hot flash tank (3) and of the refrigerant in cold flash tank (30) drops. When the load on compressors (22) and (23) increases and the power output of expanders (7) and (9) decreases to the point where the expanders can no longer drive the compressors, valves (13) and (15) are opened and valve (14) is closed. Now expanders (7) and (8) in parallel drive compressors (22) and (23) in parallel.

After a further drop in the temperature and pressure of the refrigerant in tanks (3) and (30), valves (27) and (29) are closed and valve (28) is opened. This results in two expanders, (7) and (8), in parallel driving two compressors, (22) and (23), in series.

When the temperatures and pressures in tanks (3) and (30) have dropped still further, valves (16), (19), (20) and (25) are opened and valve (24) is closed. Now the three expanders (7), (8) and (9) operate in parallel to drive the three compressors (21), (22) and (23) in series. It will be noted that during the whole operation exhaust vapors from the expanders and compressed vapors from the compressors flow into a conventional water cooled refrigeration condenser (34), where the vapors are condensed at practically ambient temperature. The condensate is discharged into ambient storage tank (11). When expanders (7), (8) and (9) no longer have sufficient power to drive compressors (21), (22) and (23), valve (10) is opened, and the unvaporized liquid in hot flash tank (3), now at substantially ambient temperature, also is discharged into ambient storage tank (11). The unvaporized cold liquid in cold flash tank (30) is discharged into cold storage tank (31) through valve (35). As in the case of hot storage tank (1), cold storage tank (31) should have sufficient capacity so that continuous operation is possible.

In the meantime pump (32) continuously has been pumping cold refrigerant liquid from cold storage tank (31) through the air cooler, which has been mentioned above. Flow of the cold liquid is controlled by valve (36). It will be seen from the drawing that the refrigerant liquid leaves the air cooler at substantially ambient temperature and flows into ambient storage tank (11), in which it is joined by the condensate which is formed in refrigeration condenser (34). Valves (10) and (35) now are closed and valves (4) and (6) are opened. A new batch of refrigerant liquid from hot storage tank (1) thus is introduced into hot flash tank (3) and a new batch of ambient temperature liquid thus is introduced into cold flash tank (30). The refrigeration cycle then is repeated. The system is self-regulating. If temperature T_s at the inlet of the air compressor tends to increase, the temperature of the compressed air entering the recuperator also increases and so, likewise, does T_H . This, in turn, heats the refrigerant liquid to a higher temperature. The flashing of this hotter liquid in hot flash tank (3) produces more power which in turn reduces the temperature of the refrigerant in cold storage tank (31) and lowers T_s . If T_s tends to decrease, the process is reversed. This self-regulation is an advantage when the Treadwell System of refrigeration is combined with the recuperated Brayton cycle.

The preferred embodiment shown by the drawing utilizes all of the advantages of a full Treadwell System and represents a preferred modification, but the invention is not limited to using all of the advantages, and may use only part of them.

It will be noted that combining the Treadwell System of refrigeration with a Brayton cycle gives optimum results. However, the air which is about to enter the compressor of the Brayton cycle may be cooled by any other refrigerating means, though with some loss in overall work output or efficiency.

Looking at the refrigeration system alone apart from a Brayton cycle, the parallel-series flow pattern of the refrigeration compressors may be used with any type of driver, and is not restricted to waste heat powered expanders. However, where waste heat is available, it is desirable to use it to the maximum extent possible. As has been pointed out before, in the Treadwell System it is not essential that the same refrigerant which is used in the power producing cycle be used in the refrigeration cycle, but when the Treadwell System is combined with a Brayton cycle, it is ordinarily more convenient and economical to use the same refrigerant liquid both for power production and refrigeration.

I claim:

1. A refrigerating system in which a volatile refrigerant is chilled by volatilizing a portion thereof to produce cold liquid refrigerant, the volatilization being at decreasing pressures, comprising in combination a flash tank for liquid refrigerant, a plurality of compressors, power sources for driving each compressor, each compressor having a suction inlet for a volatilized refrigerant and a compressed vapor outlet, valved conduits extending from the refrigerant flash tank at a level above liquid refrigerant therein to the suction inlets of the compressors, and valved conduits connecting the compressed vapor outlet from at least one of the compressors to the suction inlet of another compressor, whereby upon actuation of the valves vapors from the refrigerant flash tank can be directed to the suctions of all of the compressors in parallel or to two or more compressors in series, a cooled compressed vapor condenser, a source of coolant to the condenser at a temperature sufficiently low to condense compressed refrigerant vapors in the condenser, valved conduits connecting the outlets of the compressors to the cooled compressed vapor condenser, a condensed refrigerant vapor storage container and conduit means connecting the condenser thereto, whereby condensed refrigerant vapors flow into the storage tank, means for controlling the valves in the valved conduits to operate the compressors in parallel until the pressure in the refrigerant flash tank has reached a predetermined value and then successively connecting compressors in series until finally all of the compressors are in series and the pressure in the refrigerant flash tank reaches a minimum, means for then discharging the unvaporized and cooled refrigerant liquid from the flash tank to a cold refrigerant liquid storage tank, means for transferring a fresh charge of condensed compressed refrigerant vapors from the condensed vapor storage tank and means for repeating the above cycle by connecting the flash tank to the inlets of the compressors in parallel.

2. A refrigerating system according to claim 1 in which each compressor is connected to its own power turbine to drive it, a source of hot liquid refrigerant, a hot liquid refrigerant flash tank, means for transferring a charge of hot refrigerant liquid to the flash tank, each power turbine having a turbine vapor inlet and outlet, valved conduits connecting the turbine inlets to the hot flash tank at a point above the hot refrigerant liquid level therein, valved conduits connecting power turbine outlets to power turbine inlets of another power turbine, means for actuating the valves in the conduits so that at first the power turbines are in series while the refrigerant compressors are in parallel, whereby the hot refrigerant liquid flashes to predetermined temperature and pressure, then connecting power turbines in parallel as compressors are connected in series, until finally all of the power turbines are in parallel and all of the compressors are in series until the hot refrigerant flash tank has flashed to a minimum temperature and pressure, valved conduits connecting power turbine exhausts to the compressed refrigerant vapor condenser, whereby these turbine exhausts are condensed to a liquid and added to the condensed refrigerant liquid storage tank, and finally, when the hot and cold refrigerant flash tanks have flashed to predetermined minimum temperatures and pressures, means to transfer the unflashed refrigerant from the hot refrigerant flash tank to the compressed refrigerant vapor storage tank and to introduce from the hot refrigerant storage tank to the hot refrigerant flash tank a fresh charge, and repeat the above cycle.

3. Refrigerating system according to claim 2 in which the number of turbines and compressors is three each.

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