

[54] **STEAM JET REFRIGERATION APPARATUS**

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 [51] Int. Cl. ....**F25b 41/04**  
 [58] Field of Search.....**62/100, 116, 226, 268, 500, 62/501, 469; 417/167, 168, 169**

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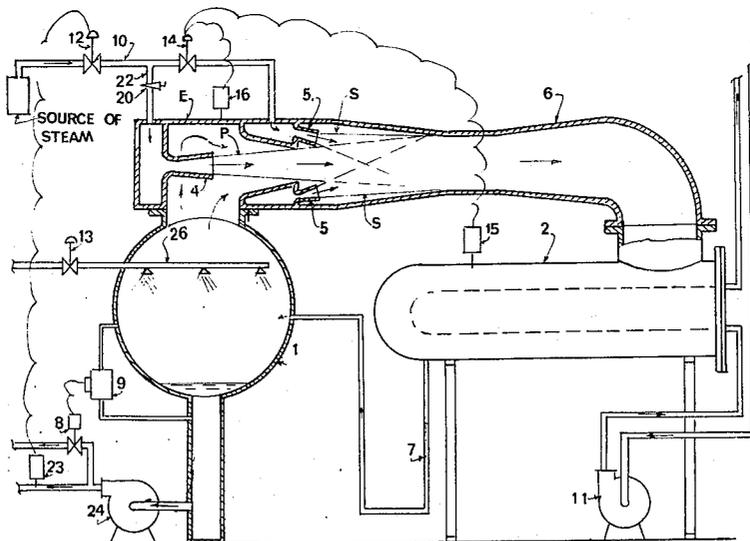
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[57] **ABSTRACT**

A steam jet refrigeration apparatus provided with a compound ejector for high pressure steam and which can be adapted to be utilized also with low pressure steam systems. The use of a compound ejector and a superheater lowers the rate of steam consumption of existing steam jet refrigeration systems, thereby making the apparatus economical in operation. The apparatus can also be provided with an improved automatic control, which approaches the quality of a modulating control.

**13 Claims, 10 Drawing Figures**





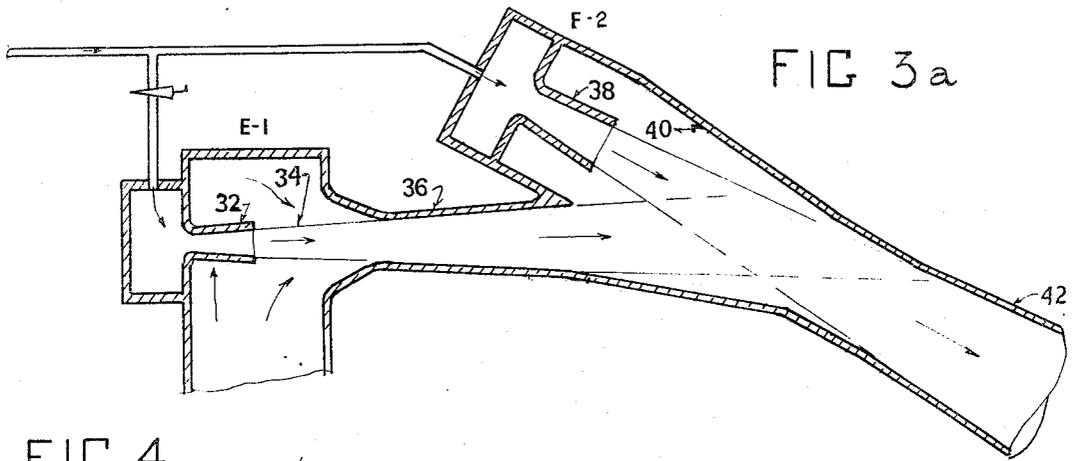
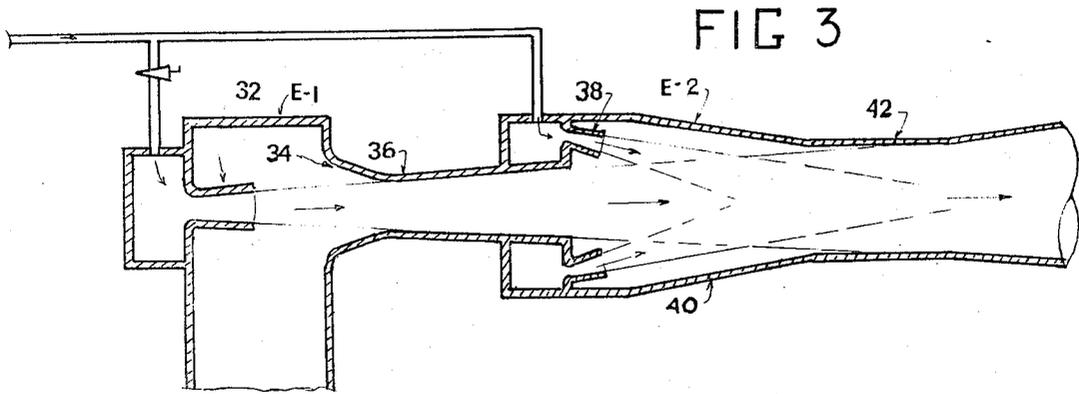


FIG. 4

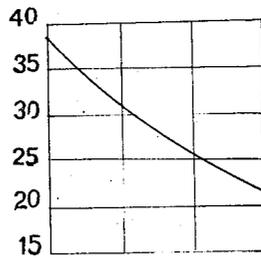
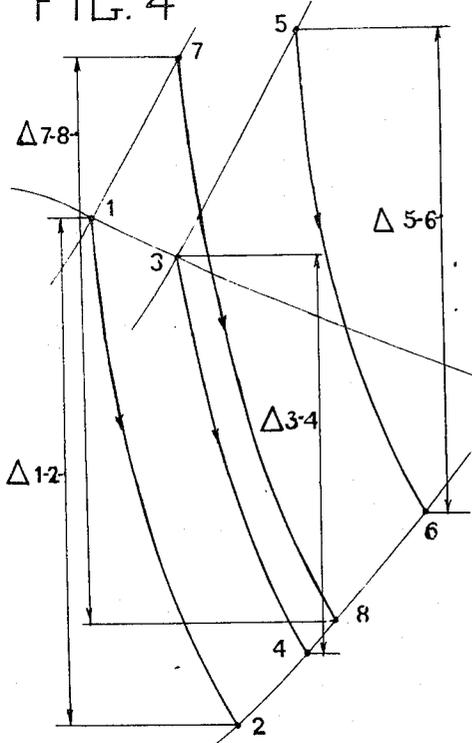
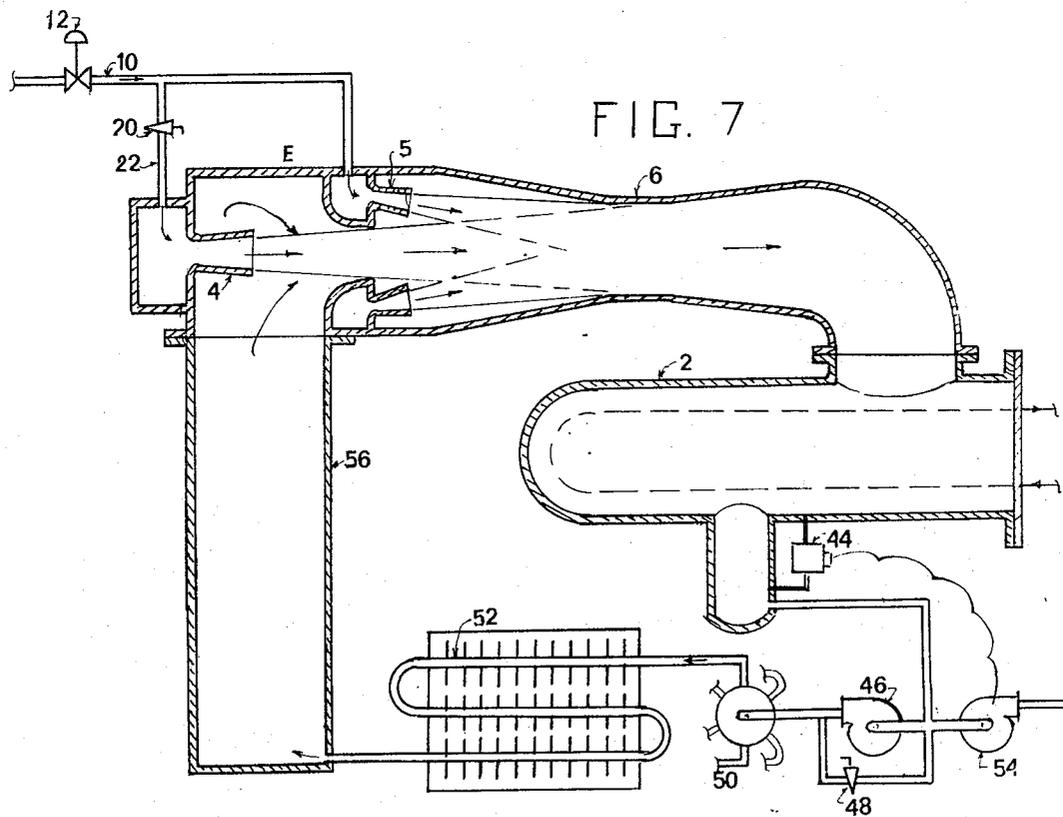
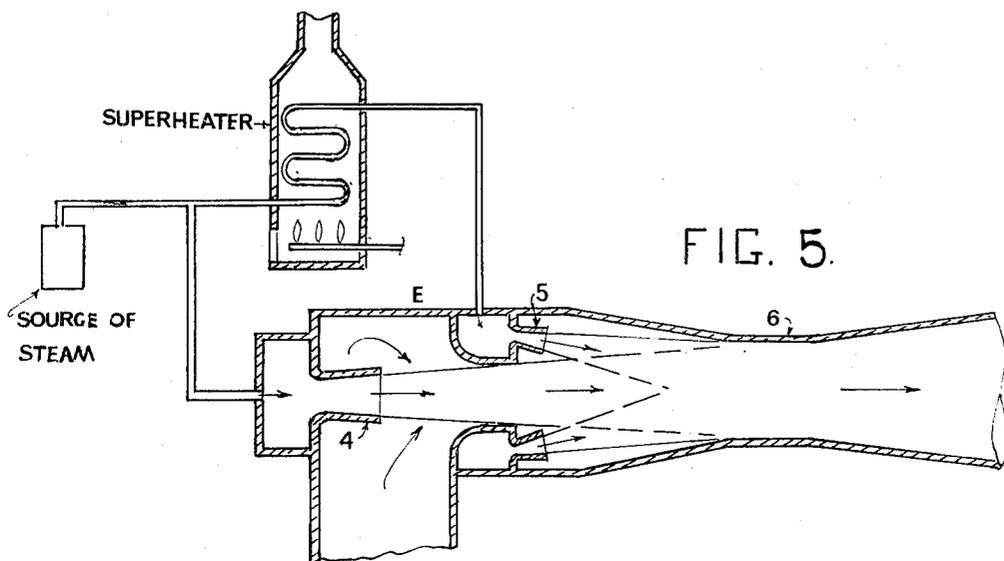


FIG. 6

35	40	45	50	1
40	45	50	55	2
45	50	55	60	3
0'1	0'12	0'14	0'18	4
1:1	1:1	1:1	1:1	5
1/11	1/92	1/79	1/61	6

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FIG. 8

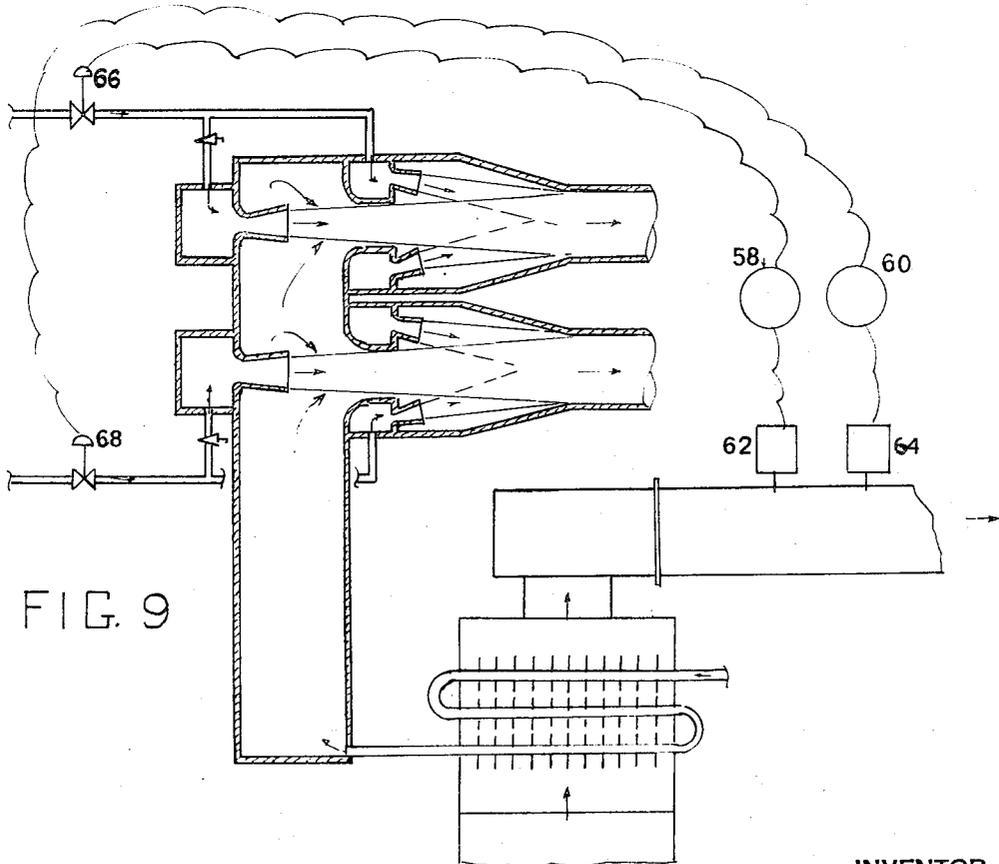
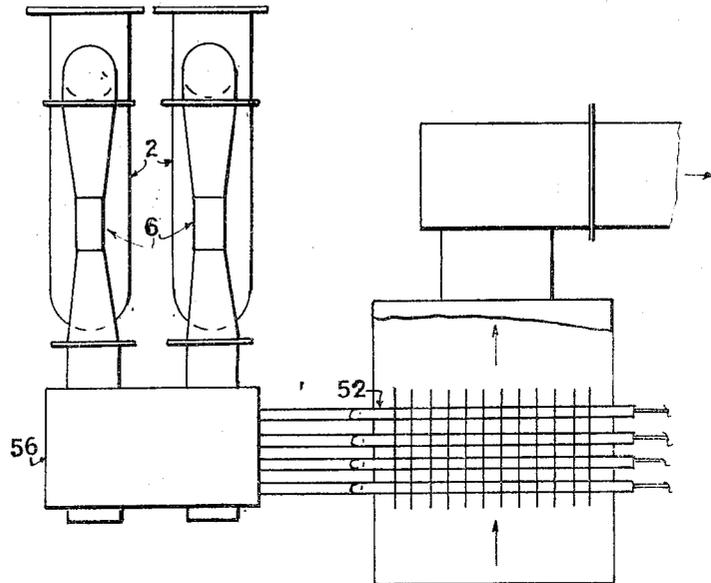


FIG. 9

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## STEAM JET REFRIGERATION APPARATUS

## BACKGROUND OF THE INVENTION

Steam jet refrigeration has not been widely used in the past because the steam consumption was so high as to make any system of this type impractical, the cause of the high steam consumption being the low efficiency of the known ejectors. In order to overcome the drawback of the known systems, a compound ejector was devised.

An important object of the present invention is the utilization of the compound ejector for steam jet refrigeration machines as liquid coolers and as air and gas coolers in different combinations.

The invention will now be further described with reference to preferred embodiments as shown in the drawings appended hereto, in which:

FIG. 1 is a diagrammatic view of the steam jet refrigeration apparatus as a water chiller, constructed in accordance with the teachings of the present invention;

FIG. 2 is a diagrammatic view of an alternative embodiment of the compound ejector of the present invention;

FIG. 3 is a diagrammatic view of a two-stage compound ejector for alternative use with the apparatus;

FIG. 3a is a diagrammatic view of a variation of the two-stage compound ejector shown in FIG. 3;

FIG. 4 is a graph showing a skeleton Mollier steam chart;

FIG. 5 is a diagrammatic view of a steam jet refrigeration machine with a superheater in the branch steam line to the secondary nozzle of the ejector;

FIG. 6 is a chart showing the steam consumption of existing steam jet refrigeration machines having conventional ejectors;

FIG. 7 is a diagrammatic view of the steam jet refrigeration apparatus used for air conditioning and constructed in accordance with the teachings of the present invention;

FIG. 8 is a plan view of the apparatus shown in FIG. 7; and

FIG. 9 is a diagrammatic view of an automatic control system for a steam jet refrigeration apparatus having multiple compound ejectors.

Referring to FIG. 1 of the drawings, a steam jet refrigeration apparatus is shown having a high pressure steam line 10 connected to a source of high pressure steam. The steam is admitted to the machine by opening valve 12 in the steam supply line 10. A primary nozzle 4 of the compound ejector, referred to generally by the letter E, is fed principally with steam of reduced pressure. For this purpose, a pressure reducing device 20 is located in branch line 22. Secondary nozzle 5 of the compound ejector E receives the full steam pressure. Thus, the steam jet leaving primary nozzle 4 of the compound ejector induces a flow of vapor from the evaporator 1 toward and into the jet of nozzle 4. These two streams form the primary mixture P. Around the stream of the primary mixture P there is a ring-shaped secondary nozzle 5, which forms a secondary jet stream S that mixes with the primary mixture P forming the combined mixture which is carried into the diffuser 6. Since the diffuser 6 is a conically shaped pipe, the combined mixture loses velocity and gains pressure thereby overcoming the higher pressure in condenser 2 into

which it enters. Accordingly, it should be apparent that the refrigeration effect of the system is achieved in the evaporator 1 by evaporation of part of the spray water, the heat of evaporation being supplied by the rest of the water, causing the water to be cooled. In the alternative, steam having the required pressures may be supplied to the first and secondary nozzles of the ejector from two different sources.

A conventional ejector with a single nozzle accomplishes two functions: the capturing of the vapor and its compression. While the first function is accomplished most economically by lower pressure steam as motive fluid, the compression work for higher compression ratios requires high pressure motive steam. These contradictory conditions dictate compromise solutions and are the cause of low efficiency.

In the compound ejector E the dual functions are divided. The primary jet accomplishes the capturing of the vapor, its acceleration, and little or no compression, while the secondary jet accomplishes the mixing of the primary mixture P with secondary jet S, and the compression work. Thus, for each nozzle the most advantageous steam pressure can be selected and each nozzle can be designed for its particular function. Optimum conditions can therefore be achieved. This has the most desirable advantage of ensuring a much higher efficiency of the compound ejector and consequently a lower steam rate of the refrigeration apparatus. Moreover, the compound ejector E has the desired result of reduced losses by shocks produced by the impact of the high velocity jet particles against the slow moving vapor particles penetrating into the jet. Furthermore, the system has a more homogeneous mixture of motive steam and vapor due to deeper penetration of the vapor in the lower velocity primary jet of the compound ejector and due to second mixing of the stream of the secondary jet with the primary mixture.

The condensate formed in the condenser 2 moves towards the evaporator 1 due to the slight difference of pressures between the condenser 2 and the evaporator 1. The condensate will be returned to the evaporator 1 by means of U-shaped tube 7. The amount of condensate corresponding to the weight of motive steam fed to the nozzles 4 and 5 is returned to the boiler feed water system by means of float switch 9 in the evaporator 1 and solenoid valve 8, the latter under the control of float switch 9. Connected to the bottom of the evaporator 1 is a chilled water circulating pump 24. Valve 12 in the steam supply line 10 may be manually operated or may be an automatic two-position control valve under the control of a switching thermostat 23 sensing the temperature of the chilled water. For larger capacities, two or more ejectors E may be used for the same evaporator. These ejectors can be activated and deactivated in sequence by a step thermostat (not shown).

Valve 13 is utilized for stopping the flow of chilled water into the spray header 26 of the evaporator when the chilled water pump 24 stops operating.

It should be apparent that compressor type refrigeration machines, as is well known, require fairly constant condenser water temperatures and condenser pressures to maintain the full capacity of the thermal expansion valve and to prevent freeze-ups. The evaporator pressure of a given steam jet refrigeration machine is determined by the design of the ejector and also by the pres-

sure of the motive steam. A reduction of the condenser water temperature will reduce the condenser pressure. However, the evaporator pressure will change only by a small amount. The compression ratio will be lower and the steam requirements for the compression work will be reduced. Therefore, it should be apparent that the steam supply to the secondary nozzle of the ejector possesses excess energy which can be saved. This is accomplished by throttling modulating control valve 14 in the steam branch to the secondary nozzle, as seen in FIG. 1, by the action of pressure controller 15 sensing the condenser pressure. Therefore, a decrease in condenser pressure will cause partial closing of control valve 14. This action has to stop when the condenser water temperature approaches the evaporator temperature or the freezing point.

In order to avoid excessive throttling by the valve 14 causing malfunction of the ejector E, preventative action must be taken. Thus, a second pressure controller 16 is utilized, the function of which is to maintain constant pressure in the mixing chamber of the ejector or in the evaporator. Controller 16, as seen in FIG. 1, takes corrective action by resetting controller 15 and by overriding controller 15 as the pressure in the mixing chamber or in the evaporator rises above the predetermined design pressure.

Since the maximum outdoor temperature and the maximum humidity for which a cooling tower is selected occur only during a limited number of days during the cooling season, in most instances, the cooling tower is able to supply colder water than at the maximum outdoor temperature or humidity. Therefore, the above-described controls by taking advantage of the lower condenser temperature achieve substantial savings of motive steam.

The principles of the above mentioned control are applicable also to air cooled condensers.

Although FIG. 1 shows a single primary nozzle 4 for larger size units, a cluster of small nozzles can be used instead of one large nozzle. Moreover, the ring-shaped secondary nozzle can be replaced by a number of single nozzles arranged in a circle around the stream P of the primary mixture. It should be noted that instead of water cooled condensers, air cooled condensers can be utilized; however, this would increase the steam consumption at high outdoor temperatures due to the higher condenser pressure and temperature. In FIG. 2, the functioning of the compound ejector is the same as that shown in FIG. 1, in which the jets of the primary nozzles 28 accomplish the capturing of the vapor and the jets of the secondary nozzles 30 accomplish the compression work. However, in the embodiment shown in FIG. 2 there is only one mixing chamber.

FIGS. 3 and 3a show two forms of a two-stage ejector, designed on basis of principles developed for compound ejectors. In FIG. 3, in this embodiment, the first stage ejector E-1 comprises a nozzle 32, a mixing chamber 34 and diffuser 36 connected in series with another ejector E-2 comprising nozzle 38, mixing chamber 40 and diffuser 42. As in the compound ejectors described hereinbefore, reduced steam pressure is used for the first stage and full steam pressure for the second stage ejector. The function of the first-stage ejector is to capture the vapor and to compress it slightly. The function of the second-stage ejector is to

capture the mixture exiting from the first-stage diffuser and to compress it to the final pressure.

The structure illustrated in FIG. 3a is the same as that shown in FIG. 3. However, the jet of the second-stage ejector is shown positioned at an acute angle to the first-stage ejector.

FIG. 4 shows a skeleton Mollier steam chart. On this chart lines connecting points 1-7, 3-5 and 2-6 are constant pressure lines for 100, 30 and 0.2 psia respectively. The line connecting points 1-3 is the saturation line. Line 1-2 on the chart represents a process of expansion of steam of 100 psia pressure in a nozzle from condition 1 to condition 2.  $\Delta 1-2 = h_1 - h_2$  is the difference of the enthalpies, equivalent to the energy available for the compression work. Similarly, if instead of 100 psia steam, 30 psia steam is expanded from condition 3 to condition 4,  $\Delta 3-4$  is substantially smaller than  $\Delta 1-2$ .  $\Delta 3-4$  is sufficient to capture the vapor, but it is not appropriate for the compression work as a disproportionately large amount of steam would be needed due to the small value of  $\Delta 3-4$ . It appears from the Mollier chart that the slope of the constant pressure lines is steep in the region of superheated steam above the saturation line and less steep in the region of moist steam below the saturation line. Consequently, if low pressure steam of the condition 3 is superheated at a constant pressure to condition 5 and then expanded in a nozzle to condition 6,  $\Delta 5-6$  will be greater than  $\Delta 3-4$ . Therefore, if heat is spent to raise the enthalpies of the motive low pressure steam, this will increase the difference of the enthalpies at the beginning and at the end of expansion. Accordingly, greater energy becomes available for the compression. Thus, by the measure of superheating the low pressure steam, which is fed to the secondary nozzle, low pressure steam becomes suitable for ejectors and particularly for steam jet refrigeration machines. Thus, the present invention becomes attractive for the installation of air conditioning or refrigeration in buildings provided with low pressure steam heating systems. This cannot be achieved by steam jet refrigeration machines having conventional ejectors in an economic manner.

FIG. 5 shows the incorporation of the superheater in the branch steam line to the secondary nozzle. Lines 7-8 in FIG. 4 represent the expansion of high-pressure superheated steam in a nozzle. Superheating of the steam fed to the secondary nozzle will decrease the steam consumption as  $\Delta 7-8$  is greater than  $\Delta 1-2$  and the available energy per pound of steam increases. While superheating of the high-pressure steam for a conventional single ejector is not beneficial due to the decreased capacity of capturing the pumped fluid at higher jet velocity, superheating is highly beneficial for compound ejectors, since only the steam fed to the secondary nozzle is superheated, and the functioning of the primary nozzle fed by steam of lower pressure is therefore not affected by the superheating.

FIG. 6 is a chart illustrating the steam consumption of existing steam jet refrigeration machines with conventional type of ejectors operating with 100 psia pressure steam when the condenser temperature is 105° F. The chart gives the steam consumption in pounds/hour/ton of refrigeration for different evaporator temperatures in degrees F., as shown on line 1. Line 2 indicates the corresponding chilled water tempera-

ture leaving the chiller in degrees F. Line 3 shows the air temperature leaving the air conditioner corresponding to the evaporator temperatures of line 1. Line 4 shows the value of the steam pressure in psia corresponding to the evaporator temperature of line 1. Line 5 shows the condensing pressure in psia corresponding to 105° F. condensing temperature. Line 6 shows the compression ratio (line 4:line 5). For example, for a chilled water temperature of 45° F. requiring an evaporator temperature of 40° F., the steam consumption is 30.6 pounds/hour/ton of refrigeration. This value is approximately double the steam consumption of turbine driven centrifugal type compressor units. Obviously, the chilled water temperature must be higher to improve the economy of this machine but this would render it impractical for air conditioning. Instead of cooling water to 45° F., air can be cooled to 60° F., which is the usual temperature of the supply air in most air conditioning systems. Therefore, calculating with a 10° approach, a 50° F. evaporator temperature is required, for which temperature the steam rate is 21.8 pounds/hour/ton. Thus, this value approaches the steam rate of steam absorption machines, and considering the reduction of the steam rate by use of compound ejectors of improved efficiency, and further considering the substantial savings that can be achieved by the control system permitting the use of colder condenser water when available it appears that air conditioning by steam jet refrigeration becomes very attractive from the point of view of economy. Moreover, the present system has additional advantages of a very simple design, no moving parts, low cost of the machine and the maintenance thereof, an absence of chemicals, such as freon or lithium bromide. Obviously, the steam rate can be reduced even further by using a higher air temperature, for example, 65° F. The reason for the reduced energy consumption at higher evaporator temperatures, is the reduced compression ratio and the reduced compression work.

FIG. 7 discloses in diagrammatic form a steam jet refrigeration apparatus for use in air conditioning or product cooling. Thus, by opening valve 12, high pressure steam is admitted to the line 10. A pressure reducing device 20 is located in branch line 22 to primary nozzle 4, while the high pressure steam is conducted directly into secondary nozzle 5. Thus, by opening valve 12 the cycle is started. There is a temperature difference between the air or other gas which has an average temperature of approximately 80° F. and the refrigerant water in the direct expansion cooling coil 52, which refrigerant is evaporating at 50° F. Consequently, there will be a flow of heat from the air or other gas to the refrigerant water in the coil 52. By giving off heat, the air or gas is cooled and by absorbing heat the liquid refrigerant evaporates. The vapor leaving the coil 52 enters a suction chamber 56 of the ejector, from where it flows toward and into the primary jet of the ejector E. The primary mixture captured by the secondary jet is carried into the diffuser 6 where it is compressed, so that it can be discharged into condenser 2. Thereafter, the condensate flows, either by gravity, or is pumped by a small pump 46 to distributor 50. From the distributor 50 the condensate, which is the refrigerant liquid, flows to several circuits of the cooling coil 52. By adjusting recirculating valve 48,

pump 46 delivers the exact quantity of liquid to the coil, as needed for the desired cooling capacity. This quantity remains constant as long as pump 46 continues to run. Condensate pump 54 is under the control of float switch 44 and this pump delivers the excess condensate to the boiler feed system. FIG. 8 shows the air conditioning apparatus of FIG. 7 in plan view.

FIG. 9 shows an automatic control system for a steam jet refrigeration machine with multiple compound ejectors. The system is an improved sequence control. For example, if in FIG. 9, an air conditioning apparatus is used having a steam jet refrigeration machine with two compound ejectors and high pressure steam as motive fluid, it becomes extremely desirable to control the operation of the machine, for example by maintaining a constant discharge air temperature. In this arrangement time clock 58 opens and closes the two-position steam control valve 66 a predetermined number of times, for example, 12 times per hour. The length of time the valve stays open, which is called the "on" period, is varied by the thermostat 62. At full load, the "on" period is maximum, that is 5 minutes. At partial load, when the discharge air temperature falls slightly, the thermostat 62 acts in such a way that it shortens the "on" period. Thus, the smaller the load, the shorter will be the "on" periods. Accordingly, at 50% load, the "on" period is zero, and the first ejector is deactivated. Time clock 60 operates in conjunction with the control valve 68, and thermostat 64 functions in the same way as thermostat 62. However, the set point of the thermostat 64 is lower than that of the thermostat 62. When the load is greater than 50 percent, the "on" period for valve 68 stays at a maximum. When the load falls below 50 percent thermostat 64 shortens the "on" period of valve 68. In no load, the "on" period of valve 68 is zero and the second ejector is deactivated. Of course, the reverse occurs when the load increases. The above-described control operates in very small sequential steps, ensuring very little variation of the controlled temperature. Thus, the quality of the control approaches the quality of a modulating control. It ensures economical operation since only that much steam is consumed as is required to match the load.

What is claimed is:

1. A steam jet refrigeration apparatus for cooling fluids comprising a source of relatively high pressure steam, a compound ejector having a primary nozzle, a secondary nozzle, a mixing chamber and a diffuser, a device operatively connected to said primary nozzle for reducing the steam pressure thereto, said secondary nozzle receiving the full steam pressure, an evaporator connected to the condenser, means connecting the evaporator to said mixing chamber of said ejector, the refrigeration being accomplished by evaporation of water in the evaporator at a lower pressure wherein the heat of evaporation is extracted from the fluid to be cooled, said low evaporator pressure being maintained by removing the vapor from the evaporator through said connecting means whereby said vapor is captured by the jet of the primary nozzle of said ejector, the flow of steam leaving said primary nozzle inducing the flow of said vapor from the evaporator through said connecting means to said chamber, wherein said secondary nozzle is ring-shaped and surrounds the stream of

steam issuing from said primary nozzle and the vapor captured thereby.

2. A steam jet refrigeration apparatus as claimed in claim 1, further comprising a superheater in the line between said steam source and said secondary nozzle.

3. A steam jet refrigeration apparatus as claimed in claim 1, wherein said secondary nozzle is supplied with high pressure superheated steam.

4. A steam jet refrigeration apparatus as claimed in claim 1, further comprising means for automatically reducing or increasing the amount of steam admitted to the secondary nozzle of said ejector to match the steam supply with the varying steam requirement, said changes in steam requirement being due to variations of the condenser pressure, while said means maintains a constant pressure in the evaporator of said ejector.

5. A steam jet refrigeration apparatus as claimed in claim 1, further comprising means for automatically reducing or increasing the steam supply to the secondary nozzle of said ejector including a throttling modulating control valve, a first pressure controller for sensing condenser pressure, a second pressure controller for maintaining a constant predetermined pressure in the evaporator, said first pressure controller sensing the condenser pressure acting upon a decrease in condenser pressure to close partially said control valve, said second pressure controller acting to reset said first pressure controller and/or to override the latter if the pressure in the mixing chamber or in the evaporator rises above said constant predetermined pressure therein.

6. A steam jet refrigeration apparatus as claimed in claim 1, wherein the primary nozzles of said compound ejector are located around the secondary nozzles of said compound ejector with one common mixing chamber for the primary and secondary nozzles.

7. A steam jet refrigeration apparatus for cooling liquids comprising a source of relatively high pressure steam, an ejector, a condenser, the ejector delivering to said condenser, an evaporator, said evaporator receiving the condensate from said condenser and connected to deliver vapor to said ejector, said ejector comprising a first nozzle means connected through a pressure reducing means to said steam source, the connection between the evaporator and the ejector opening to the ejector adjacent said first nozzle whereby vapor from the evaporator is captured by relatively low pressure steam issuing from said first nozzle, and a second nozzle receiving full steam pressure, said first and second nozzles opening to a common mixing chamber whereby steam from said second nozzle means is directed to mix with the steam of the first nozzle and the vapor captured thereby, the uncompressed mixture issuing to a single common diffuser wherein the velocity of said mixture is reduced and the pressure increased, the refrigeration being accomplished by evaporation of water in the evaporator at a low pressure wherein the heat of evaporation is extracted from the fluid to be cooled, said low evaporator pressure being maintained by removing the vapor through said ejector.

8. A steam jet refrigeration apparatus as claimed in claim 7, wherein the fluid to be cooled is water.

9. A steam jet refrigeration apparatus as claimed in claim 7, wherein the fluid to be cooled is air.

10. A steam jet refrigeration apparatus for cooling fluids comprising a source of relatively high pressure steam, a compound ejector having a primary nozzle, a secondary nozzle, a mixing chamber and a diffuser, a device operatively connected to said primary nozzle for reducing the steam pressure thereto, said secondary nozzle receiving the full steam pressure, an evaporator connected to the condenser, means connecting the evaporator to said mixing chamber of said ejector, the refrigeration being accomplished by evaporation of water in the evaporator at a low pressure wherein the heat of evaporation is extracted from the fluid to be cooled, said low evaporator pressure being maintained by removing the vapor from the evaporator through said connecting means whereby said vapor is captured by the jet of the primary nozzle of said ejector, the flow of steam leaving said primary nozzle inducing the flow of said vapor from the evaporator through said connecting means to said chamber, wherein a number of smaller secondary nozzles are arranged in a circle around the stream of steam issuing from said primary nozzle and the vapor captured thereby.

11. A steam jet refrigeration apparatus for cooling fluids, comprising a source of relatively high pressure steam, a compound ejector having a primary nozzle, a primary mixing chamber, a secondary nozzle, a secondary mixing chamber and a diffuser, a superheater in the line between said steam source and said secondary nozzle, a device operatively connected to said primary nozzle for reducing the steam pressure thereto, said secondary nozzle receiving full steam pressure, an evaporator connected to the condenser, means connecting the evaporator to said primary mixing chamber of said ejector, the refrigeration being accomplished by evaporation of water in the evaporator at a low pressure wherein the heat of evaporation is extracted from the fluid to be cooled, said low evaporator pressure being maintained by removing the vapor from the evaporator through said connecting means whereby said vapor is captured by the jet of the primary nozzle of said ejector.

12. A steam jet refrigeration apparatus, comprising a source of relatively high pressure steam, a two stage compound ejector having in the first stage a nozzle, mixing chamber and diffuser, in the second stage a nozzle arranged around the stream ejected from the first stage ejector, a mixing chamber and a diffuser, a device operatively connected to said nozzle of the first stage ejector for reducing the steam pressure thereto, said nozzle of the second stage ejector receiving full steam pressure, a condenser, an evaporator connected to the condenser, means connecting the evaporator to said mixing chamber of the first stage ejector, the refrigeration being accomplished by evaporation of water in the evaporator at a low pressure whereby the heat of evaporation is extracted from the fluid to be cooled, said low evaporator pressure being maintained by removing the vapor from the evaporator through said connecting means whereby said vapor is captured by the jet of the nozzle of the first stage ejector wherein the jet stream of the secondary nozzle and the stream of mixture ejected from the diffuser of the first stage ejector form an acute angle.

13. A steam jet refrigeration apparatus for cooling fluids comprising an arrangement having a compound

ejector; a source of steam for said ejector; a condenser; an evaporator; a plurality of connections between the ejector, condenser and evaporator; a timer control means unit for said ejector whereby said ejector may be made operative or inoperative a constant number of

times per unit of time whereby the duration of the operative or inoperative periods of the ejector may be automatically varied to match the capacity of the apparatus with the demand.

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