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Ness et al.

[15] **3,696,637**

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[54] **METHOD AND APPARATUS FOR PRODUCING REFRIGERATION**

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Related U.S. Application Data

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[52] **U.S. Cl.** **62/402**, 62/88, 62/38

[51] **Int. Cl.**.....**F25d 9/00**

[58] **Field of Search**.....62/86, 87, 88, 402

[56] **References Cited**

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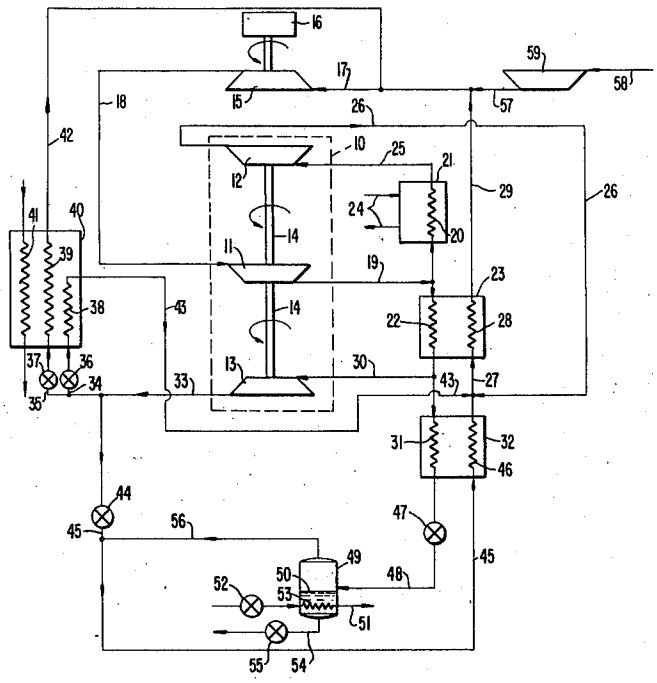
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Primary Examiner—William J. Wye
Attorney—Shanley and O'Neil

[57] **ABSTRACT**

Refrigeration system including a multi-stage compressor and two work expansion engines of the turbine type with the impellers of the expansion engines and the impeller of the final stage of the compressor being mounted on a common shaft. The work developed in the expansion engines provides the total power required for the final stage of the compressor and the final stage of the compressor provides the pressurized gas expanded in both expansion engines.

10 Claims, 4 Drawing Figures



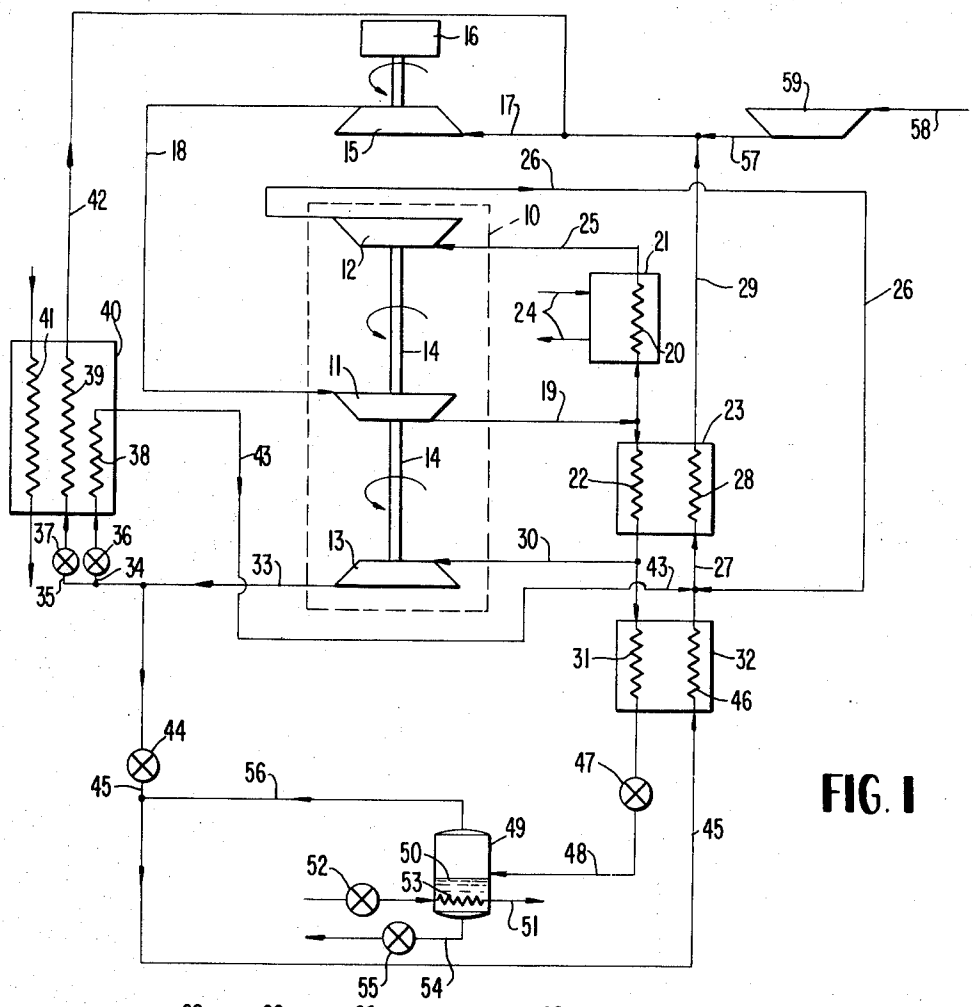

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FIG. 1

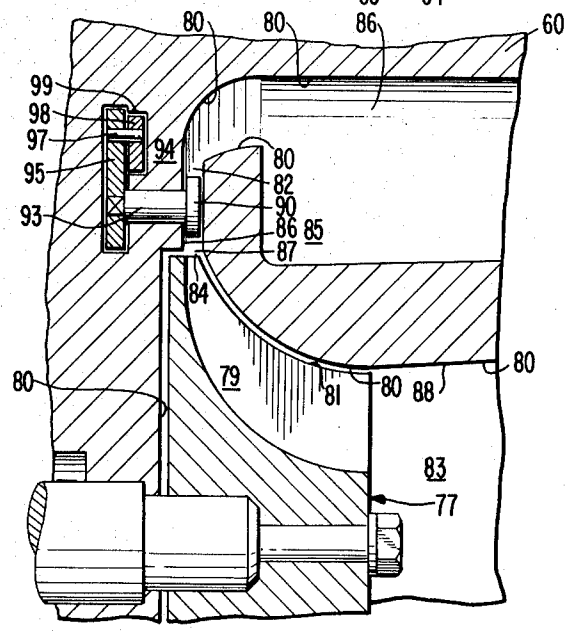

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FIG. 3

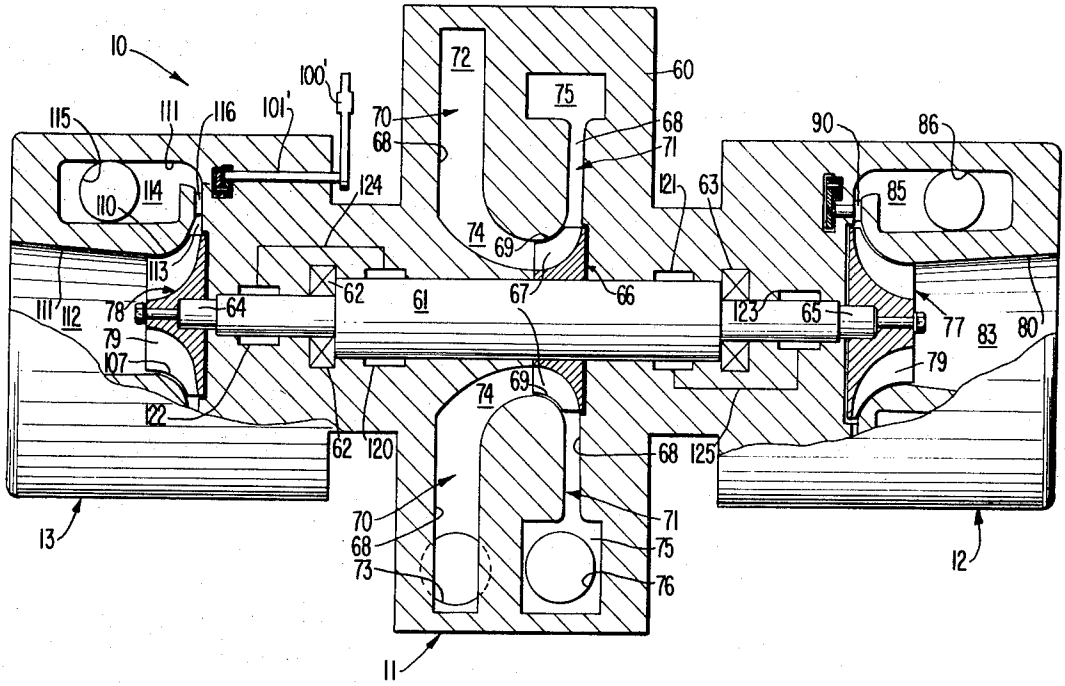

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FIG. 2

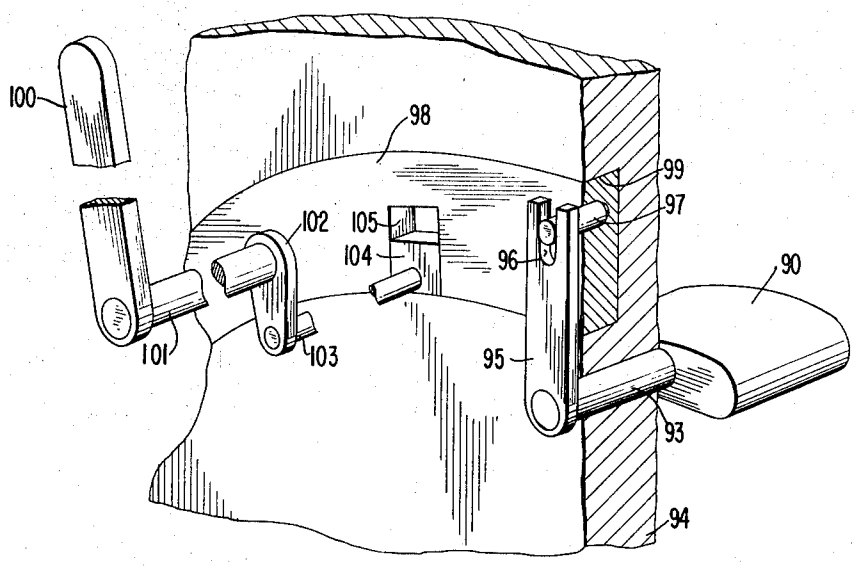

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FIG. 4

METHOD AND APPARATUS FOR PRODUCING REFRIGERATION

This a division of application Ser. No. 752,998, filed 8-15-68, now U.S. Pat. No. 3,657,898.

BACKGROUND OF THE INVENTION

This invention relates to improvements on refrigeration producing methods and apparatus particularly of the type employing a multi-stage centrifugal compressor and a plurality of work expansion engines of the turbine type.

It is known that the energy produced by work expansion of a pressurized gas in a refrigeration process may be used to provide a part of the power requirements of the process and it has been proposed in the past to directly couple the work expansion machine to a compressor which pressurizes the refrigeration gas. When the energy produced by the work expansion is employed to drive a stage of compression having power requirements substantially less than the energy of work expansion, it is possible to design the work expansion engine to operate at such a speed so as to obtain economically acceptable efficiency for existing parameters including pressure and temperature of the inlet gas and pressure and temperature of the effluent. It is known that the critical speed requirement of work expansion engines of the turbine type may be materially relieved by effecting the work expansion in two expansion turbines operating in series relation, that is, the effluent of the first expansion turbine being fed to the inlet of the second expansion turbine, and it has been proposed in the past to directly couple a pair of expansion turbines operating in series relation in the process to the final stage of the refrigerant gas compressor. In such an arrangement, the series operation of the expansion turbines makes it possible for the final compression stage to operate in a region of higher efficiency and thus utilize a greater percentage of the horsepower available from the work expansion as compared to the processes in which a single expansion turbine is coupled to a compression stage.

In another known refrigeration process, a pair of work expansion turbines operate in parallel relation and at materially different temperature levels; the turbines being fed with pressurized gas at substantially the same pressure and discharge the gas at substantially the same relatively low pressure. In such process, the critical speed requirements of each expansion turbine is in no way relieved, as in the process employing expansion turbines operating in series as discusses above, and it has not been possible heretofore to directly utilize at a high order of efficiency the energy developed from both parallel operating expansion turbines to drive the final stage of compression while maintaining high expander efficiency.

The present invention provides a novel method and apparatus for producing refrigeration employing multiple stages of refrigerant compression and two stages of refrigerant work expansion in which the horsepower developed by the work expansion stages is utilized to drive the final stage of refrigerant compression. The novel process employs and the novel apparatus includes a novel expander-compressor-expander combination in which both expansion turbines operate at different temperature levels under a substantially equal pressure ratio and at high efficiency and in which the

power produced from both expansion turbines is applied directly to the compressor operating as the final compression stage and at a high order of efficiency to thus usefully employ the total energy output of the expansion turbines within the normal range of efficiency obtained for low temperature machinery. The novel arrangement makes it possible to accomplish a greater percentage of the total work of compression in the final compression stage and more efficient compression of the gaseous refrigerant.

Other objects and advantages of the present invention will appear from the following detailed description when considered in connection with the accompanying drawings which disclose a preferred embodiment of the invention. It is to be expressly understood, however, that the drawings are designed for purposes of illustration only and not as a definition of the limits of the invention, reference for the latter purpose being had to the appended claims.

DESCRIPTION OF THE DRAWINGS

In the drawings, in which similar elements are identified by corresponding reference characters throughout the several views:

FIG. 1 is a diagrammatic presentation of a refrigeration cycle embodying the principles of the present invention;

FIG. 2 is a diagrammatic view, partly in section, of a novel expander-compressor-expander unit provided by the present invention;

FIG. 3 is a view in section of a part of the structure shown in FIG. 2; and

FIG. 4 is a three-dimensional view of another part of the structure shown in FIG. 2.

DESCRIPTION OF THE PREFERRED EMBODIMENT

The refrigeration cycle shown in FIG. 1 includes a novel expander-compressor-expander unit **10** which includes a compressor **11** of the centrifugal type and centripetal turbine expanders **12** and **13** connected to a common shaft **14** with the compressor **11**. The compressor **11** comprises the final compression stage of the refrigeration cycle which also includes a first stage compressor **15**, driven by any suitable prime mover **16**. The compressor **15** compresses refrigerant gas delivered to its suction inlet by conduit **17** to an intermediate pressure and refrigerant gas at the intermediate pressure is conducted by conduit **18** to the suction inlet of the compressor **11** from which the refrigerant gas is delivered in conduit **19** under relatively high pressure; the compressors **11** and **15** may be provided with conventional after coolers, not shown. The high pressure refrigerant gas is divided with a first part flowing through passageway **20** of heat exchange device **21** and a second part flowing through passageway **22** of heat exchange device **23**. The first part of the high pressure refrigerant gas is cooled to below ambient temperature in the heat exchange device **21** by heat interchange with an auxiliary refrigerant supplied by conduits **24** and then conducted by conduit **25** to the inlet of the turbine expander **12** wherein the refrigerant gas is expanded with the production of external work to a pressure corresponding substantially to the inlet pressure of the compressor

15 with concomitant cooling of the gas. The effluent of the turbine expander 12 is conducted by conduits 26 and 27 for flow through passageway 28 of the heat exchange device 23 and thereby warmed to ambient temperature and then conducted by conduit 29 to the conduit 17 for recycling to the compressor 15. The second part of the high pressure refrigerant gas is cooled upon flowing through the passageway 22 and is then divided with one part of the cooled high pressure refrigerant gas flowing through conduit 30 to the inlet of the turbine expander 13 and with a second part flowing through passageway 31 of the heat exchange device 32. In the turbine expander 13, the cooled high pressure refrigerant gas is expanded with work to a pressure slightly above the inlet pressure of the compressor 15 with concomitant further cooling to a relatively low temperature. The effluent of the turbine expander 13 may be utilized to provide refrigeration externally of the cycle or refrigeration for the cycle, or both. As shown, effluent of the turbine expander 13 may be passed by conduit 33 to conduits 34 and 35, provided with control valves 36 and 37, respectively, for flow through passageways 38 and 39, respectively, of heat exchange device 40, in countercurrent heat interchange with an external fluid flowing through passageway 41. The warm refrigerant gas leaving the warm end of the passageway 39 is conducted by conduit 42 to the conduit 17 for flow to the inlet of the compressor 15, while the refrigerant gas leaving the passageway 38 at a lower temperature is conducted by conduit 43 to the conduit 27 for flow through the passageway 28 and, hence, on to the inlet of the compressor 15. Also, all or part of the effluent may be passed through control valve 44 and conduit 45 for flow through passageway 46 of the heat exchange device 32 to provide refrigeration for the cycle as described below.

The second part of the cooled high pressure refrigerant gas is further cooled upon flowing through the passageway 31 of the heat exchange device 32 and then expanded in valve 47 to a pressure slightly above the suction pressure of the compressor 15 to effect its partial liquefaction, and then fed by conduit 48 to phase separator 49 where the liquefied refrigerant collects in a pool 50. The liquefied refrigerant may be employed to cool an external fluid flowed by conduit 51 having a control valve 52 through coil 53 immersed in the pool 50. Also, when the cycle is employed as a liquefier, liquefied refrigerant may be withdrawn from the phase separator through conduit 54 having a control valve 55. Gaseous refrigerant comprising unliquefied refrigerant gas fed to the phase separator and liquid refrigerant that may be vaporized in the phase separator is withdrawn from the phase separator through conduit 56 and merged with the refrigerant gas in the conduit 45 for flow through the passageways 46 and 28. Makeup refrigerant gas for the cycle, as may be required to compensate for liquid refrigerant withdrawn as product, for example, is fed to the suction inlet of the compressor 15 by conduit 57 and, when the makeup refrigerant gas is available at a pressure lower than suction pressure of the compressor 15, the makeup refrigerant gas is fed by conduit 58 through a suitable compressor 59 to the conduit 57.

The power developed by the work expansion of the high temperature high pressure refrigerant gas in the turbine expander 12 and the power developed by the work expansion of the low temperature high pressure refrigerant gas in the turbine expander 13 is applied directly through the common shaft 14 to the compressor 11 to provide the sole power source for the final compressor stage. As described below, the expander-compressor-expander unit 10 is characterized so that the expansion turbines 12 and 13, although rotating at the same speed and notwithstanding the material difference between their operating temperatures, both operate at a high order of efficiency to satisfy fully the work expansion refrigeration producing requirements of the cycle and both develop a high percentage of the potential power of the work expansion which is utilized to drive the final stage of compression which makes possible, although the final compression stage is rotating at the same speed as the expansion turbines, compression of the refrigerant gas to the required high pressure from a lower intermediate pressure thereby reducing the requirements of the first stage of compression and achieving a reduction in capital expenditure and operating power. It will be appreciated that the flow of high pressure refrigerant gas discharged from the compressor 11 is sufficient to satisfy the required flow of refrigerant gas to the expansion turbines and also provides high pressure refrigerant gas that is partially liquefied upon expansion in valve 47.

As shown in FIG. 2, the expander-compressor-expander unit 10 includes a casing 60, which may be made up of sections bolted or otherwise removably secured together, forming a housing for the compressor 11, the turbine expander 12 and the turbine expander 13. A shaft 61 is rotatably supported within the casing 60 by journal-thrust bearings 62 and 63, preferably of the tilting pad type, spaced axially of the shaft 61 and located inwardly of shaft ends 64 and 65. An impeller 66 of the compressor 11 is mounted on the shaft 61 for rotation therewith at a point intermediate the bearings 62 and 63 and the impeller 66 includes radial blades 67 which are of constant radius at its input. Internal walls 68 of the casing provide a chamber 69 within which the impeller 66 rotates and an inlet passageway 70 and an outlet passageway 71 communicating with the chamber 69. The inlet passageway 70 extends circumferentially about the shaft 61 and includes a portion 72 which extends from the outer regions of the casing 60, where it communicates with fluid inlet 73, radially toward the shaft 61 and a portion 74 which extends axially of the shaft and merges with the chamber 69 at the inlet of the impeller 66. The outlet passageway 71 extends circumferentially about the shaft 61 radially outwardly from communication with the chamber 69 at the discharge of the impeller 66. The passageway 71 functions as a diffuser which may be of the vaneless type and discharges into a volute 75 which communicates with discharge outlet 76. The turbine expander 12 includes an impeller 77 mounted on and secured to the shaft end 65 and the turbine expander 13 includes an impeller 78 mounted on and secured to the shaft end 64. The turbine expanders 12 and 13 are of the radial-inflow or centripetal type and the impellers 77 and 78 are provided with radial blades 79, preferably of the open type, having an axial dimension at the periphery of the

impellers corresponding substantially to the width of the nozzle inlet and a substantially greater dimension, measured in a plane perpendicular to the axis of rotation, at the exit of the impeller.

As shown in FIG. 3, internal walls 80 of the casing 60 define a chamber 81 within which the impeller 77 rotates, the chamber 81 being shaped to conform to the outer configuration of the blades 79. The internal walls 80 also define a fluid inlet passageway 82 and an exhaust passageway 83 which diverges in the direction of flow and is of circular cross section concentric with the axis of rotation of the impeller. The inlet passageway 82 extends circumferentially about the impeller 77 and communicates with the chamber 81 at the inlet 84 of the blades 79, and extends radially outwardly from the shaft into communication with a circumferentially extending chamber 85, also defined by internal walls 80 of the casing, into which high pressure gas to be expanded is fed by input conduit 86. A plurality of spaced vanes 90 are positioned in the passageway 82, in equally spaced relationship about the circumference of the impeller, to provide therebetween nozzles for directing the high pressure gas into the peripheral entries of the blades 79. The vanes 90 are spaced radially from the peripheral edge 84 of the blades 79 to permit the high pressure gas discharged from the nozzles to follow a path influenced by the position of the vanes 90 and the vanes 90 are adjustable in unison to vary the mass and the direction of flow of the gas into the impeller.

As shown more clearly in FIG. 4, each of the vanes 90 is mounted for pivotal movement about an axis parallel to the axis of rotation of the impeller by a shaft 93 supported in a wall 94 of the casing. On the side of the wall 94 opposite the passageway 82, the shaft 93 is connected to an arm 95 provided with a slot 96 which receives a pin 97 carried by a ring 98 positioned in concentric relation about the axis of rotation of the impeller and rotatably supported in a circular groove 99 of rectangular cross-section formed in the wall 94. A mechanism is provided for rotating the ring 98 relative to the wall 94 to simultaneously rotate each of the vanes 90 in the same direction and in the same degree about their respective supporting shafts 93. Such mechanism includes a crank arm 100 connected outside of the casing 60 to an end of shaft 101 journaled in the casing, the other end of the shaft 101 being rigidly connected within the casing to one end of a crank arm 102. The other end of the crank arm 102 is rigidly connected to shaft 103 secured to a rectangular block 104 received in a rectangular opening 105 formed in the ring 98. With this arrangement, movement of the crank arm 100 in one direction or the other effects rotation of the ring 98 in a clockwise or counterclockwise direction, as the case may be, relative to the wall 94 which in turn effects simultaneous rotation of the vanes 90 in a clockwise or counterclockwise direction about their supporting shafts 93.

The turbine expander 13 is constructed in a manner similar to the turbine expander 12 but is differently dimensioned as described below. The impeller 78 is a mirror image of the impeller 77 and rotates within a chamber 110 defined by internal walls 111 of the casing. The internal walls 111 also define a divergent discharge passageway 112 and an inlet passageway 113, the latter passageway communicating with an inlet chamber 114 to which high pressure gas is supplied through input conduit 115. Adjustable vanes 116 are located in the passageway 113 in a manner similar to the vanes 90 and a mechanism similar to the arrangement shown in FIG. 4 is provided to effect adjustment of the vanes upon movement of crank arm 100' connected to shaft 101'.

Seals 120, 121, 122, and 123, of the labyrinth type, are provided about the shaft 61. Seals 120 and 121 are located inwardly of the bearings 62, 63 on opposite sides of the compressor impeller 66, and function to impede the flow of high pressure gas along the shaft in a direction toward the turbine expanders, while seals 122 and 123, located between the bearing 62 and the impeller 78 and between the bearing 63 and the impeller 77, respectively, function to impede the flow of cold gas along the shaft inwardly toward the compressor impeller. In order to prevent the flow of cold gas inwardly toward the compressor impeller, high pressure gas is fed to the seals 122 and 123 by passageways 124 and 125 leading from the seals 120 and 121 to the seals 122 and 123, respectively.

In operation of turbine expanders of the foregoing type, the incoming compressed gas is accelerated and directed by the nozzles and enters the peripheral inlet of the radial blades with a large tangential velocity and a small radial velocity and, upon flowing through the impeller, the kinetic energy of the gas is transferred to the impeller to effect its rotation and the gas leaves the blades and enters the discharge passageway at a relatively low velocity. For efficient operation, the tangential velocity of the incoming gas and the angular velocity of the tips 84 of the radial blades should correspond within narrow limits. Hence, the maximum permissible tip speed of the blades is one of the controlling parameters for expansion turbines operating at cryogenic temperatures. The maximum angular velocity of the peripheral tips of the blades is usually selected to be equal to 0.8 of Mach. I. Within limits, the optimum speed of the tips of the blades for given flow of a specific gas at a given pressure and temperature may be established by correlating the speed of rotation of the impeller and the diameter of the blades at the inlet of the impeller. The provision of adjustable vanes, such as the adjustable vanes 90 and 116 of the turbine expanders 12 and 13, makes it possible to vary the mass and the direction of the high velocity gas discharged from the nozzles which, together with the provision of a radial space between the periphery of the blades and the discharge end of the nozzles, makes it possible to operate the expansion turbines at the designed efficiency over a wide range of different input flow rates for a specific gas at a given pressure and temperature.

Notwithstanding the limitations imposed on the designing of a low temperature expansion turbine, it has been discovered that two expansion turbines and a compressor may be designed as a unitary structure with all of the impellers mounted on a common shaft and rotating at a common speed, with the expansion turbines and the compressor operating at high efficiency and with substantially the total potential horsepower of the expansion being developed by the expansion turbines and utilized as the sole power source for the compressor which delivers the pressurized gas to both ex-

pansion turbines and excess pressurized gas for liquefaction. The foregoing has been accomplished, at least in part, by operating the expansion turbines at a constant input pressure or at a substantially equal pressure ratio, by operating the expansion turbines at substantially different temperature levels, by maintaining substantially equal flow to each of the expansion turbines, and by utilizing a constant input compressor. The feature of rotating the compressor and both expansion turbines at the same speed permits the impeller of the compressor and the impellers of the expansion turbines to be mounted on a common shaft to provide a compact assembly which offers special advantages in low temperature equipment especially in view of the high rotating speeds involved. The further feature provided by the present invention of mounting the impellers of the two expansion turbines on opposite sides of the compressor impeller aids in balancing the machine and permits counter application of the thrust developed by the expansion turbine impellers and partial compensation of the thrust developed by the compressor impeller.

An expander-compressor-expander unit constructed in accordance with the principles of the present invention has been operated for extended periods of time in a refrigeration cycle of the type shown in FIG. 1. The compressor was of the centrifugal type and included an open impeller with the radial blades at the periphery having a diameter of about 5.00 inches, the inlet nozzle and the diffuser being provided without vanes. The turbine expanders were of the centripetal type having impellers provided with open radial blades; the impeller of the high temperature turbine expander having a diameter at its inlet of 5.25 inches and the impeller of the low temperature turbine expander having a diameter of 4.50 inches at its inlet. The turbine expanders were provided with adjustable nozzles, as described above, and the nozzle control arms were ganged together for simultaneous control; however, means were provided for independent adjustment of the nozzles of each turbine expander. The following table provides data on the performance of the expander-compressor-expander unit utilizing nitrogen as the refrigerant, the data being collected at different times during a period of continuous operation which exceeded 5 days:

	Time 1	Time 2	Time 3	Time 4
Compressor				
Inlet pressure, psig	400	405	405	400
Inlet Temperature, °F.	81	83	82	80
Discharge pressure, psig	545	555	555	550
Discharge temperature, °F.	155	157	157	157
Flow, standard cubic feet/hour × 1000	1920	1900	1900	1920
Adiabatic efficiency, %	63.5	64.8	64	63
Horsepower	1025	1020	1030	1070
High Temperature Expansion Turbine				
Inlet pressure, psig	525	535	535	530
Inlet temperature, °F.	17	21	22	21
Discharge pressure, psig	69	70	70	68
Discharge temperature, °F.	−120	−123	−122	−124
Flow, standard cubic feet/hour × 1000	566	520	520	520
Adiabatic efficiency, %	66.5	67.5	68.6	69.3
Horsepower	499	472	480	488
Low Temperature Expansion Turbine				
Inlet pressure, psig	520	530	530	525
Inlet temperature, °F.	−178	−171	−173	−159
Discharge pressure, psig	75	71	71	70
Discharge temperature, °F.	−282	−283	−283	−272
Flow, standard cubic feet/hour × 1000	1075	990	990	990
Adiabatic efficiency, %	80.2	86.0	85.0	81.0
Horsepower	553	585	575	596
Shaft speed, rpm	35,500	36,000	35,500	36,000

During the foregoing operation, the cycle of FIG. 1 functioned as a closed system to provide sources of refrigeration for a low pressure air separation system to increase liquid producing capacity. Compressed air to be separated flowed through the passageway **41** of the heat exchange device **40** and the liquid-vapor nitrogen mixture in conduit **48** was introduced into the high pressure fractionating column and an equivalent mass of nitrogen vapor was withdrawn from the fractionating column by conduit **56**. Considering the data collected at Time 1 of the above table as being typical, the nitrogen gas was discharged from the compressor **11** at about 100°F. and under a pressure of about 545 p.s.i.g. and was divided with about 30 percent flowing through passageway **20** of heat exchange device **21** and with about 70 percent flowing through passageway **22** of heat exchange device **23**. The compressed nitrogen gas was cooled in heat exchange device **21** to about 17°F. in heat interchange with a freon refrigerant and was introduced into the inlet of the turbine expander **12** under a pressure of about 525 p.s.i.g. The major portion of the nitrogen gas was cooled in heat exchange device **23** to about −178°F. and then divided with about 80 percent being fed to the inlet of the turbine expander **13** under a pressure of about 520 p.s.i.g. The remaining 20 percent of the high pressure nitrogen gas was cooled to about −276°F. upon flowing through the passageway **31** of the heat exchange device **32**, then expanded in valve **47** to about 75 p.s.i.g. and then introduced into the fractionating column (phase separator **49**) at a temperature of about −285°F., partially in liquid phase. The effluent of the expansion turbine **13** at a temperature of about −282°F. and under a pressure of about 75 p.s.i.g. was subdivided with about 52 percent flowing through the conduit **45**, about 28 percent flowing through the passageway **38** of the heat exchange device **40** and about 20 percent flowing through the passageway **39** of such heat exchange device. The cold nitrogen gas in the conduit **45** was merged with the nitrogen vapor at a temperature of −285°F. withdrawn from the phase separator **49**, and the merged stream was warmed upon flowing through the passageway **46** of the heat exchange device **32** to about −179°F. and then combined with the nitrogen gas withdrawn from the passageway **38** at about −164° F.; the combined streams being warmed to about 75°F. upon flowing through the passageway **28** of the heat exchange device **23** and then returned to the inlet of the compressor **15** by way of the conduit **17**. The nitrogen gas flowing through the passageway **39** of the heat exchange device **40** was withdrawn from that passageway at about 84°F. and conducted by conduit **42** to the conduit **17** onto the inlet of the compressor **15**. The pressurized nitrogen gas from the compressor **15**, at about 80°F. and under a pressure of about 400 p.s.i.g., was fed by conduit **18** to the inlet of the compressor **11**.

Although only one embodiment of the present invention has been disclosed and described herein, it is to be expressly understood that various changes and modifications may be made therein without departing from the spirit of the invention as well understood by those skilled in the art. Reference therefore will be had to the appended claims for a definition of the limits of the invention.

We claim:

1. A compressor-turbine assembly comprising:
a. a rotatable shaft;
b. a compressor impeller secured to said shaft;
c. casing means forming a compression chamber surrounding said compressor impeller and forming inlet and discharge passageways communicating with said compression chamber;
d. a first turbine impeller secured to said shaft axially spaced on one side of said compressor impeller;
e. casing means forming an expansion chamber surrounding said first turbine impeller and forming inlet and discharge passageways communicating with said first expansion chamber;
f. a second turbine impeller secured to said shaft axially spaced on the other side of said compressor impeller;
g. casing means forming an expansion chamber surrounding said second turbine impeller and forming inlet and discharge passageways communicating with said second expansion chamber;
h. a first set of adjustable vanes located in the inlet passageway communicating with said first expansion chamber, and a second set of adjustable vanes located in the inlet passageway communicating with said second expansion chamber; and
i. radially and circumferentially extending spaces forming annular chambers between the periphery of each of said turbine impellers and the adjustable vanes associated therewith.

2. The compressor-turbine assembly as claimed in claim 1 wherein said inlet passageways communicating with said first and second turbine expansion chambers comprise radially extending passages communicating with the periphery of said turbine impellers.

3. The compressor-turbine assembly as claimed in claim 1 wherein said discharge passageways communicating with said first and second turbine expansion chambers comprise discharge passages extending coaxially with said shaft and directed to discharge the expanded gas in opposite directions away from said compressor casing means.

4. The compressor-turbine assembly as claimed in claim 1 wherein said compressor inlet passageway comprises a radially inwardly and axially extending passage, said compressor outlet passageway comprises a radially outwardly extending passageway, said first and second turbine inlet passageways comprise radially inwardly extending passages communicating with said vanes, and said discharge passageways comprise axially extending passages directed in opposite directions away from said compressor.

5. The compressor-turbine assembly as claimed in claim 1 wherein said first and second turbine impellers are of different size having different tip velocities, said first and second adjustable vanes and said annular chambers comprising means for matching the tangential angles of the inlet gases to the different tip speeds of the respective turbine impellers.

6. The compressor-turbine assembly as claimed in claim 5 including means for simultaneously actuating the first and second sets of adjustable vanes.

7. The compressor-turbine assembly as claimed in claim 5 including means for independently adjusting said first and second sets of adjustable vanes.

8. The compressor-turbine assembly as claimed in claim 5 wherein said compressor inlet passageway comprises a radially inwardly and axially extending passage, said compressor outlet passageway comprises a radially outwardly extending passageway, said first and second turbine inlet passageways comprise radially inwardly extending passages communicating with said vanes, and said discharge passageways comprise axially extending passages directed in opposite directions away from said compressor.

9. The compressor-turbine assembly as claimed in claim 8 wherein said compressor impeller includes blades having axially extending inlet portions and radially extending discharge portions, and said first and second turbine impellers include blades having radially extending inlet portions and axially extending discharge portions.

10. A compressor-turbine assembly comprising:
a. a rotatable shaft;
b. a compressor impeller secured to said shaft;
c. casing means forming a compression chamber surrounding said compressor impeller and forming inlet and discharge passageways communicating with said compression chamber;
d. a first turbine impeller secured to said shaft axially spaced on one side of said compressor impeller;
e. casing means forming an expansion chamber surrounding said first turbine impeller and forming inlet and discharge passageways communicating with said first expansion chamber;
f. a second turbine impeller secured to said shaft axially spaced on the other side of said compressor impeller;
g. casing means forming an expansion chamber surrounding said second turbine impeller and forming inlet and discharge passageways communicating with said second expansion chamber;
h. said compressor inlet passageway comprises a radially inwardly and axially extending passage, said compressor outlet passageway comprises a radially outwardly extending passageway, said first and second turbine inlet passageways comprise radially inwardly extending passages communicating with said turbine impellers, and said discharge passageways comprise axially extending passages directed in opposite directions away from said compressor.

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