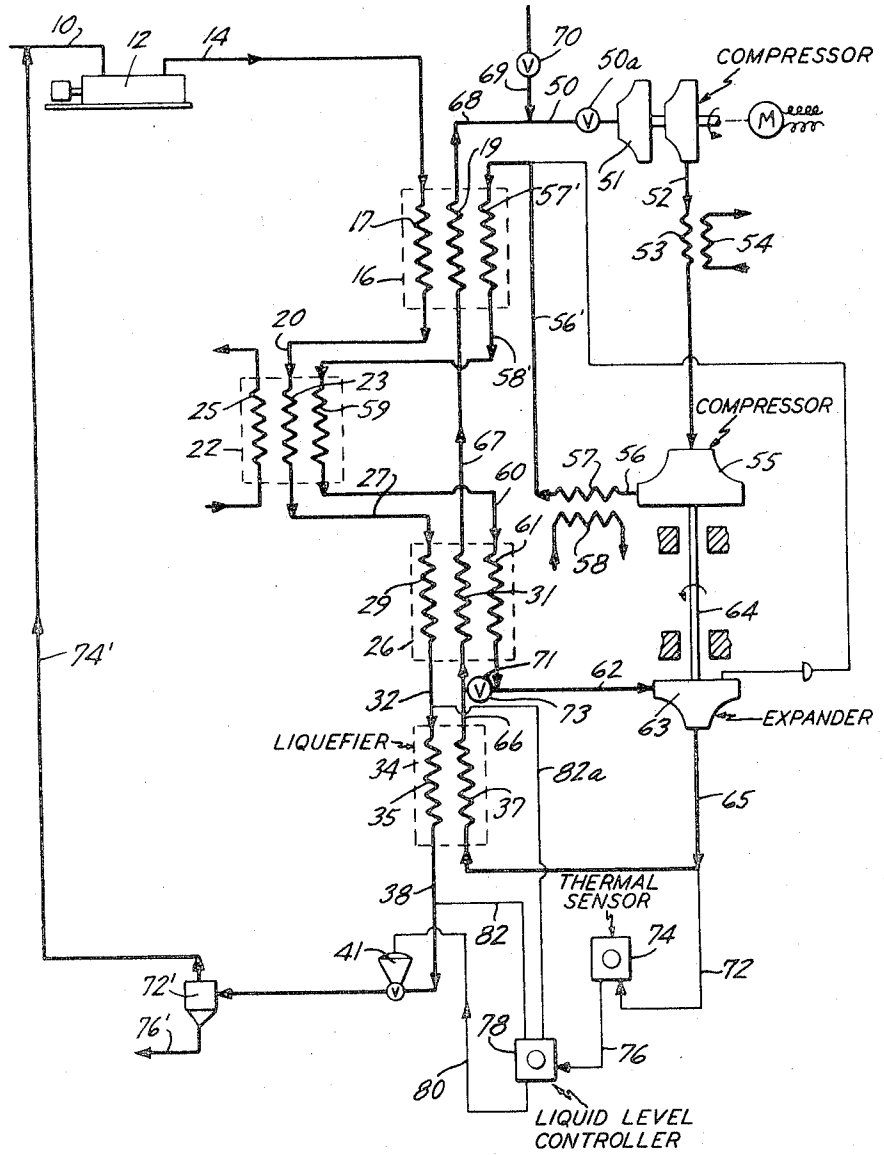
Jan. 31, 1967 R. R. CARNEY 3,300,991

THERMAL RESET LIQUID LEVEL CONTROL SYSTEM FOR THE LIQUEFACTION OF LOW BOILING GASES

Filed July 7, 1964 2 Sheets-Sheet 1

FIG. 1



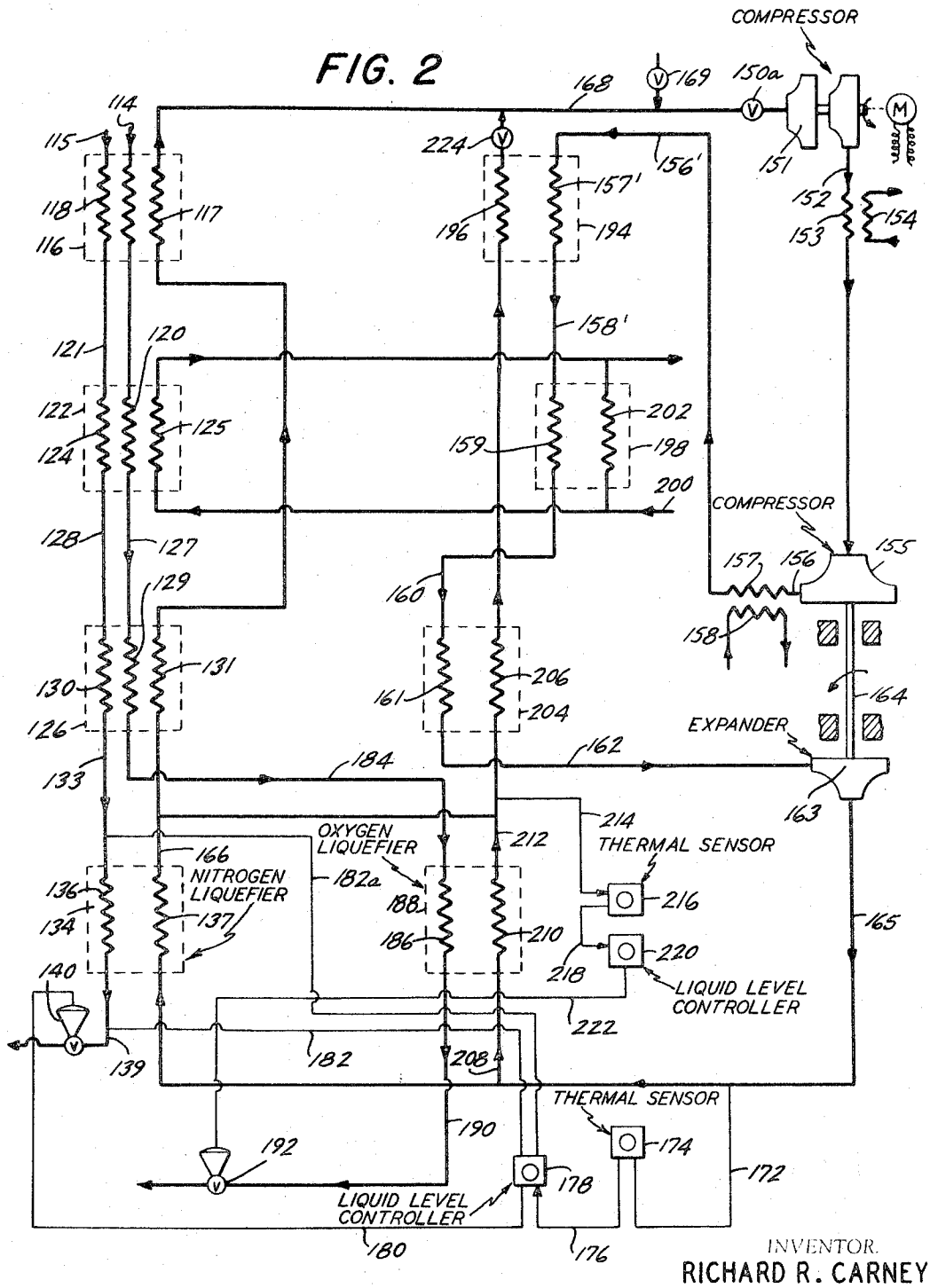
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THERMAL RESET LIQUID LEVEL CONTROL SYSTEM FOR THE LIQUEFACTION OF LOW BOILING GASES

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FIG. 2



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3,300,991

THERMAL RESET LIQUID LEVEL CONTROL SYSTEM FOR THE LIQUEFACTION OF LOW BOILING GASES

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19 Claims. (Cl. 62—9)

This application is a continuation-in-part of my copending application Serial Number 80,600 filed January 4, 1961.

This invention relates to an improved control system for the liquefaction of low-boiling gases, and more specifically to process and apparatus for the improved control of a refrigeration system for liquefying feed gases having boiling points below —80° C.

The consumption pattern of low-boiling industrial gases such as oxygen and nitrogen has changed appreciably in recent years, and created a pressing need for an economical system to liquefy such low-boiling material when available in the gas phase. This is due mainly to the fact that it is usually more economical to store and tansport such low-boiling gases in the liquid phase instead of the gas phase. Also, it is sometimes desired to employ low-boiling gases in liquid form for very cold refrigerative purposes, such as in food preservation or for cryogenic research requirements, and for propellants in rocket engines.

The prior art has proposed and employed numerous systems for cooling and liquefying low-boiling gases, but all of these systems have important disadvantages or limitations. For example, the power consumption of certain prior art liquefier systems is prohibitively high, and in others an inordinately large number of expensive heat exchangers are required. Also, some systems utilize high pressure equipment which is relatively complicated and difficult to maintain and control automatically, and in addition have limited rangeability.

In one system for liquefying feed gas by heat exchange with a refrigerant gas, the two gases having boiling points below —80° C. at atmospheric pressure, the feed gas is compressed to a pressure of at least 70 p.s.i.g., partially cooled in a first cooling step, further cooled in a second cooling step, liquefied, and withdrawn as a liquid product. The refrigerant gas is also compressed to a pressure of at least 70 p.s.i., after cooled to a temperature below about 40° C. and work expanded to sufficiently low pressure to develop power and cool the gas to a temperature below the condensation temperature of the feed gas. The work expanded refrigerant gas is consecutively passed through the liquefaction, further cooling and first cooling steps in countercurrent heat exchange with the feed stream to effect the cooling and liquefaction thereof. The warmed refrigerant is withdrawn from the warm end of the first cooling step and recirculated to the refrigerant compression step.

In the employment of this refrigeration system as disclosed above, it is first determined what product is to be liquefied and at what temperature this liquid product must be available consistent with the use to be made of it. For example, the low-temperature liquid product may be stored at various pressure levels or piped to some distant point, and a particular degree of subcooling will usually be desired to prevent flashing. Thus, the desired product temperature establishes the lowest required temperature of the refrigerant gas at the work expander exhaust. Furthermore, some consumers may require two different liquid products such as oxygen and nitrogen. Next, the refrigerant recycle stream is chosen and the desired head pressure is selected based upon its physical properties

and upon economic evaluations of investment and operating costs. Then, the product feed pressure is established to provide the optimum balance of the desuperheating, condensing and, if desired, subcooling requirements of the feed stream.

After these preliminary process features have been established, there remains the problem of controlling the overall process so as to meet a widely fluctuating product demand by operating the unit at high efficiency over a wide range of production rates.

The principal object of this invention is to provide such a control system. Additional objects and advantages of the invention will be apparent from a reading of the ensuing disclosure and the appended claims.

In the drawings,

FIG. 1 is a flowsheet of a system for cooling and liquefying low-boiling gases, illustrating the present control system; and

FIG. 2 is a flowsheet of a similar liquefaction system but having a modified control system according to the present invention for the simultaneous liquefaction of two products.

The conventional systems for controlling the liquefaction of low-boiling gases are based on adjustment of the refrigeration circuit to meet changing flow and temperature demands for the product liquid. For example, if more liquid was needed or a more deeply subcooled product required, the refrigeration production rate was increased. This for example would be accomplished by increasing the quantity of gas for work expansion, or work expanding to a lower pressure and temperature level. However, the refrigeration circuit consumes the greatest portion of the power requirements of the process, and frequent adjustments of the work expansion variables introduce serious inefficiencies. This is because expansion turbines are most usually designed for maximum efficiency while handling a certain quantity of gas, and for expanding such quantity through a particular pressure ratio and to a particular exhaust temperature which may be slightly above the condensation temperature of the refrigerant. Thus, it is desirable to operate the refrigerant circuit at substantially constant high efficiency conditions as much as possible.

The present invention is predicated on the discovery that the overall efficiency of the process may be appreciably improved by controlling the refrigeration-producing circuit under substantially constant high efficiency conditions, and adjusting the feed gas circuit to meet changing process conditions. The adjustment is effected by controlling the withdrawal rate of the liquefied feed, which may be subcooled if desired.

More specifically, a process aspect of the present invention contemplates a system for liquefying feed gas by heat exchange with a refrigerant gas wherein the feed gas and the refrigerant gas both have boiling points below —80° C. at atmospheric pressure and are compressed to the necessary pressure. The process has particular utility in regard to liquefying oxygen or nitrogen feed streams using nitrogen gas as the refrigerant. In such cases both the feed stream and the refrigerant gas stream might be compressed to pressures of at least 70 p.s.i.g. In other cases where low pressure cycles are used, such as for example, in small helium refrigerators and hydrogen liquefiers, much lower pressures are utilized.

In all of these processes, the refrigerant gas is compressed to the necessary pressure for subsequent work expansion to a lower pressure sufficient to develop enough power and cooling to cool the refrigerant gas to a temperature below the desired liquefied temperature of the feed gas. The work expanded and cooled refrigerant gas is then passed in countercurrent heat exchange with the

feed gas for cooling and liquefaction thereof. According to this invention, a refrigerant gas temperature from the work expansion step is sensed, and the withdrawal rate of liquefied feed is controlled in response to the sensed temperature so as to maintain the refrigerant gas work expansion exhaust temperature below the condensation temperature of the feed gas, and preferably slightly above the condensation temperature of the refrigerant. As used herein, the expression "work expansion step" includes the work expander inlet and exhaust streams, and may consist of one or more expansion stages with reheating therebetween if desired. It is preferred to sense the temperature of the exhaust stream of the last work expansion stage, as this provides an immediate indication of temperature and flow variations occurring during the entire work expansion step.

A prior art method of controlling work expander exhaust temperatures is to admix the gas stream just before work expansion wtih a warmer gas stream. However, with this method the overall efficiency of the cycle is reduced since this results in an irreversible thermodynamic mixing loss, and also causes a greater proportion of the required cycle refrigeration to be produced more expensively at low temperature by the work expander, instead of producing that portion at a higher temperature level by an external forecooling step. Furthermore, the control over a wide operating range is somewhat less precise because of the secondary effects arising from shifts in the refrigeration balance.

As stated above, the invention is particularly suitable for the liquefaction of nitrogen and oxygen, and will be described in detail with respect to these fluids. It is to be understood, however, that it may be advantageously employed for controlling the liquefaction of any low-boiling gas having a boiling point below about −80° C. For example, with a suitable refrigerant gas it is also applicable to ethane, ethylene, methane, argon, fluorine, carbon monoxide, neon, hydrogen, and helium.

Additionally, it is to be noted that the invention can be used to control processes for liquefying low boiling gases using such refrigerants as helium, hydrogen, etc.

The preferred refrigerants, particularly in regard to liquefying oxygen or nitrogen, are nitrogen and air, that is, a refrigerant wherein nitrogen is at least the principal constituent. However, other refrigerants, as mentioned above, may be used depending on the feed gas composition. The refrigerant gas may be of the perfect gas type, not relying upon special non-ideal properties to provide refrigeration at certain temperature levels by Joule-Thomson throttling. For example, when liquefying hydrocarbon gases such as methane, methane could also be advantageously used as the refrigerant gas.

Referring now more specifically to FIG. 1, the various fluid flows will be completely described followed by a discussion of the control system which constitutes the invention. The cycle will be initially described in terms of oxygen feed gas which is supplied to conduit **10** compressed to at least 70 p.s.i.g., and preferably to about 150 p.s.i.g. in compressor **12**. This pressure level is desirable to permit subsequent efficient liquefaction of the feed gas. The compressed oxygen feed gas is then passed through conduit **14** to aftercoolers and lubricant traps (not shown), and next to warm leg heat exchanger **16** as a first cooling step to about −46° C. The feed gas flows through passageway **17** and is cooled by countercurrently flowing refrigerant gas in passageway **19**.

The partially cooled oxygen feed gas is discharged from warm leg heat exchanger **16** into conduit **20** and preferably directed to externally refrigerated forecooler **22** for cooling therein to about −60° C. That is, the partially cooled oxygen feed gas in conduit **20** is directed to passageway **23** in forecooler **22**, and countercurrently cooled by an externally supplied refrigerant flowing through passageway **25**. The preferred external refrigerant is dichlorodifluoromethane, although monochlorodifluoromethane, ammonia, carbon dioxide, or nitrogen are also suitable. It is to be understood that the externally refrigerated forecooling step is preferred but is not essential to the present invention, and that the necessary cooling may alternatively be effected in warm leg heat exchanger **16** and cold leg heat exchanger **26**.

The forecooled oxygen feed gas is discharged from forecooler **22** into conduit **27**, hence to cold leg heat exchanger **26** for further cooling in passageway **29** by heat exchanging with the countercurrently flowing refrigerant in conduit **31**. The further cooled oxygen gas is discharged from cold leg heat exchanger **26** at a temperature of about −140° C. into conduit **32** and directed to liquefier **34** for flow through communicating passageway **35**, and liquefaction by countercurrently flowing gaseous refrigerant in passageway **37**. In liquefier **34**, the oxygen feed stream is cooled to saturation, totally condensed, and the product liquid is preferably subcooled to a temperature of about −186° C. It is withdrawn as a pressurized liquid product through conduit **38**, and passed through control valve **41** to storage means or consuming means as desired. One reason for subcooling the liquid product is to avoid excessive flashoff on throttling into a storage tank, preferably at a pressure of 0–15 p.s.i.g. If adequate subcooling is not performed, a vapor-liquid separator may be used to recover the vapor product. Any vapor generated from the product liquids downstream of control valve **41** from the pressure reducing step is preferably separated from the liquid in vessel **72'**, and returned through conduit **74'** to feed gas conduit **10** for reprocessing. The remaining low pressure liquid product is withdrawn from separator **72'** through conduit **76'**. It is to be understood however, that the liquid product may alternatively be stored at substantially the feed stream pressure if desired.

The refrigeration required by the oxygen feed stream is produced principally by the refrigerant recirculation stream. For this purpose, clean, dry nitrogen may, for example, be supplied at about 8 p.s.i.g. and 15° C. in conduit **50** with control valve **50*a*** therein, and pressurized in compressor **51**, preferably of the centrifugal type, to a pressure of at least 50 p.s.i.g. and preferably about 100 p.s.i.g. Alternatively, refrigerant gas inlet flow may be effected by inlet guide vanes (not shown) inside compressor **51**, instead of by valve **50*a***. The compressed nitrogen refrigerant gas is discharged into conduit **52** and aftercooled in passageway **53** to a temperature below about 40° C. by heat exchange with a suitable fluid such as water in thermally associated passageway **54**. The aftercooled nitrogen gas is then further compressed in the turbine loading booster compressor **55** to a pressure of at least 70 p.s.i.g. and preferably about 145 p.s.i.g., and discharged therefrom into conduit **56**. The further compressed nitrogen gas is then aftercooled in passageway **57** again to a temperature below about 40° C. by heat exchange with an appropriate fluid such as water in thermally associated passageway **58**.

The further compressed, aftercooled nitrogen is first directed to the warm end of warm leg heat exchanger **16** for cooling therein to about −46° C. by flow through passageway **57'** in countercurrent heat exchange relation with the refrigerant in passageway **19**. The partially cooled, compressed nitrogen gas is then preferably directed through conduit **58'** to passageway **59** in forecooler **22** for further cooling therein to about −60° C. However, as previously discussed, the forecooling step is not essential to this invention. The forecooled compressed nitrogen is then discharged into passageway **60** and directed to the warm end of cold leg heat exchanger **26** for flow through passageway **61** in countercurrent heat exchange relation with the refrigerant in passageway **31**.

The compressed nitrogen gas is cooled in cold leg **26** to a temperature of about −141° C., and discharged into conduit **62** for flow to a work expander such as turbine **63**. At this point, the nitrogen is expanded to a low

pressure preferably in the range of 6–10 p.s.i.g., although the discharge may be at subatmospheric pressure if desired to lower the condensing temperature of the refrigerant. Appreciable liquefaction of the refrigerant gas is purposely avoided to prevent reduced efficiency and possible erosion of the expander parts due to its handling mixed liquid-vapor flow, and to avoid two-phase flow in the heat exchangers with the resulting additional equipment such as entrainment separators, liquid levels and the like. However, the expander exhaust may contain up to about 5% liquid even though the temperature controller is set to normally maintain the exhaust temperature within the superheat range. The nitrogen gas is cooled to about —187° C. by virtue of such work expansion, and the power developed in the expansion turbine is preferably transferred directly to the highest pressure level of the refrigerant compression step. This is preferably accomplished by employing shaft **64** to connect turbine **63** with booster compressor **55**, to provide highly efficient transfer of the available power. High shaft speeds, which permit the most efficient and economic design of the turbine, can be most effectively utilized to absorb the power by centrifugally compressing an equivalent mass of a higher density gas stream at higher pressures and smaller volumes, rather than compressing a large volume gas stream at lower pressure, such as occurs in the first stage of compression. Alternatively, at least part of the work expander power may be absorbed by other means such as an electric generator (not illustrated), and used for reducing the net power requirements of the cycle.

The work expanded nitrogen is discharged from turbine **63** into conduit **65**, and passed from the cold end to the warm end of the feed gas heat exchange system to refrigerate the latter. More specifically, the work expanded nitrogen is first passed to the cold end of liquefier **34** for flow through passageway **37**, thereby desuperheating, condensing, and preferably subcooling the product oxygen stream in thermally associated passageway **35**. The nitrogen is simultaneously warmed to about —156° C. and thereafter directed through connecting conduit **66** to passageway **31** of cold leg heat exchanger **26** for further cooling of the partially cooled oxygen feed streams. Finally, the partially rewarmed nitrogen refrigerant gas is directed through communicating conduit **67** to warm leg heat exchanger **16** and passageway **19** therein for warming to near ambient temperature.

The resulting warmed nitrogen refrigerant gas is discharged from the heat exchange system through conduit **68**, and recirculated to connecting conduit **50** for return to the inlet side of compressor **51**. Makeup nitrogen gas from a suitable source is admitted to conduit **50** through conduit **69** and control valve **70** therein to overcome system losses through compressor seals, and the like. Any vapor arising downstream of control valve **41** from the pressure reducing step therein may be separated in vessel **72** and returned through conduit **74** to feed conduit **10** for reprocessing.

During the startup and cooldown phase of operation of the present liquefier, a condition will arise whereby the power developed by work expander **63** will exceed that which can be absorbed in booster compressor **55**, resulting in overspeeding of the latter. To alleviate this problem, bypass conduit **71** containing control valve **73** may be provided between conduit **62** processing the compressed and cooled nitrogen gas, and conduit **66** transporting the partially warmed work expanded nitrogen gas at the warm end of liquefier **34**. A sufficient quantity of gas is diverted from conduit **62** to conduit **66** to maintain the desired energy balance between work expander **63** and booster compressor **55**. Alternatively, this bypass line and valve may be located at the warm end of the heat exchange system between conduits **56'** and **68**. However, if expander **63** is loaded by an electric generator to maintain essentially constant speed, the bypass conduit is usually not advantageous.

The FIG. 1 system also may be employed to liquefy nitrogen instead of oxygen feed gas. It is to be noted that because of the difference in the normal boiling points of oxygen and nitrogen, it is necessary to compress the nitrogen feed stream to a higher pressure than oxygen to obtain the same cycle efficiencies. As explained previously, the feed gas stream must always be provided at sufficient pressure to permit its liquefaction at the lowest temperature level attained by the refrigerant stream.

For a particular combination of refrigerant and feed gas, optimum performance is obtained by carefully selecting the feed gas pressure which in combination with the refrigerant gas pressure and recirculation rate will provide small temperature differences within the heat exchangers and also permit maximum utilization of external forecooling if employed. The effect of increasing the condensing pressure of the feed gas stream is to reduce its latent heat and increase the degree of subcooling required. Thus, the feed gas pressure is selected to maintain optimum economy between the latent heat and subcooling requirements for providing a liquid stream preferably at essentially ambient pressure as will be understood by those skilled in the art.

It has been found that a temperature pinch occurs in the liquefier heat exchanger **34** at the point where condensation begins. That is, the temperature of the product feed stream being cooled in passageway **35** is reduced at that point to vary nearly the temperature of the expanded recycle refrigerant stream flowing countercurrently to it in passageway **37**. If the refrigerant recirculation ratio is reduced to about 7.2 cu. ft. (NTP) nitrogen circulated per 1 cu. ft. (NTP) oxygen liquefied, using external forecooling to —60° C., this temperature pinch becomes so severe as to limit the utilization of any additional refrigeration from the forecooler. However, optimum performance (with forecooling to —60° C.) is obtained with a recirculation ratio of about 8.5 cu. ft. (NTP) nitrogen recirculated per 1 cu. ft. (NTP) oxygen liquefied and subcooled. This ratio opens the temperature difference at the pinch point to about 6° C., and also opens the temperature difference at the warm end of liquefier **34** to about 16° C. to achieve efficient liquefier operation. While a recirculation ratio greater than 8.5 may be used, it results in more refrigeration being made available at the lowest temperature level than can be effectively utilized. This causes increased temperature differences within the liquefier heat exchanger and thereby permits less external forecooling to be used, thus causing reduced overall cycle efficiency.

Referring now to the control system, the expander exhaust temperature is sensed in conduit **65** by a thermocouple or other suitable means and transmitted by electrical conduit **72** to controller **74**. The latter then sends a signal, which may be either pneumatic or electrical, through conduit **76** to liquid level controller **78**. A signal is sent therefrom through conduit **80** to valve **41** which controls the rate of oxygen product liquid withdrawal through conduit **38**. Thus, if process variables or heat exchanger performance change, performance of the work expander **63** varies during operation, or a different level of subcooling is desired, the product liquid level in passageway **35** of liquefier **34** is changed to provide for it by varying the effective heat transfer area therein.

For example, if the inlet temperature to the expander **63** increases due to higher ambient conditions, or less efficient performance of the heat exchangers, or shutdown of the forecooler, or if the efficiency of work expander **63** decreases for any reason so that its exhaust temperature becomes warmer than desired, product liquid withdrawal valve **41** would close slightly to increase the liquid level of the condensing oxygen feed gas in passageway **35** of liquefier **34**. This would reduce the effective

heat transfer area between the condensing feed gas and the colder expanded refrigerant gas in conduit **37** of liquefier **34**. It will be recalled that both the compressed nitrogen refrigerant gas in passageway **61** and the compressed feed gas in conduits **29** and **35** are cooled by the work expanded nitrogen gas in communicating conduits **37** and **31**. Thus, raising the product liquid level in conduit **35** reduces the effective heat transfer area therein and shifts a greater portion of the total heat transfer load to cold leg exchanger **26**, which brings the expander inlet temperature in conduit **62** closer to the colder relatively fixed product liquid temperature, and thus lowers the expander inlet temperature. In this manner, the work expansion conditions may be readjusted to establish the desired optimum operating conditions.

Conversely, if the turbine inlet temperature is reduced and/or the turbine exhaust temperature lowers, a signal will be transmitted through the previously described control circuit to slightly open valve **41**. This has the effect of increasing or restoring the effective heat transfer area within liquefier **34** and thereby shifting to liquefier **34** more of the total heat transfer load between the condensing feed gas and the colder expanded refrigerant gas. This further warms the refrigerant gas which in turn raises the temperature of the work expander inlet stream in passageway **61** leaving the cold leg heat exchanger **26** by conduit **62**. Thus, adjustment of this liquid level adjusts the heat exchanger area to control the turbine inlet temperature and thereby to obtain the desired turbine exhaust temperature in the most efficient manner for a wide range of recirculation ratios and other operating variables.

The previously described turbine temperature control procedure is superimposed upon the normal liquid level control function in liquefier **34**. Thus, the liquid level controller instrument **78** senses the liquid level in passageway **35** by means of conduits **82** and **82***a*, and operates the product liquid withdrawal valve **41** to maintain it at a particular level as the "short range" control procedure which is accomplished in the conventional manner. The work expander temperature is the "long range" variable used to adjust the liquid level as required to maintain optimum efficiency in the work expansion step.

At steady feed gas conditions, the work expander exhaust temperature and the temperature "pinch" that occurs in the liquefier are both dependent on the refrigerant recirculation rate and on the heat transfer area available to desuperheat, condense, and subcool the incoming feed gas stream. By the present invention, the effective heat exchange area is varied by raising or lowering the product liquid level in the liquefier. The set point for the liquid level controller is preferably provided in terms of the expander discharge temperature. Adjustment of this liquid level adjusts the heat exchanger area to control the turbine inlet temperature in order to obtain the desired turbine exhaust temperature in the most efficient manner for a wide range of recirculation ratios and other operating variables. Also, the effectiveness of the turbine temperature control procedure is enhanced and the overall cycle efficiency improved by maintaining the refrigerant head pressure at the design value even for reduced recirculation rates. This is accomplished by variable area nozzles in the work expansion turbine, which operate to maintain the refrigerant head pressure into warm leg heat exchanger **16** at a selected level. For example, if the pressure of the further compressed nitrogen gas in conduit **56'** drops, this change is sensed and an electrical signal is transmitted through conduit **85** to slightly close variable area nozzles **87** located at the inlet to expansion turbine **63**. This maintains the desired enthalpy change through the work expansion step and helps preserve high overall efficiency of the cycle even at reduced load.

FIG. 2 illustrates another embodiment of the invention whereby two separate feed gas streams such as oxygen and nitrogen are simultaneously cooled and liquefied as products. The fluid flows are similar to those of the FIG. 1 embodiment, and FIG. 2 items corresponding to those of FIG. 1 have been given the same identification numeral plus one hundred. The compressed oxygen feed gas is introduced through conduit **114** and consecutively cooled in warm leg heat exchanger **116**, forecooler **122**, and cold leg **126**. The further cooled oxygen gas is discharged from passageway **129** into conduit **184** for liquefaction and subcooling in passageway **186** of second liquefier **188**. The oxygen product liquid is withdrawn from the bottom of second liquefier **188** through conduit **190** having control valve **192** therein. The other feed gas stream such as compressed nitrogen is introduced through conduit **115** and consecutively cooled, liquefied, and subcooled in warm leg **116**, forecooler **122**, cold leg **126**, and first liquefier **134**. The nitrogen product liquid is withdrawn from the lower end of passageway **136** through control valve **140**.

The further compressed and aftercooled refrigerant gas such as nitrogen is first directed through conduit **156'** to the warm end of second warm leg heat exchanger **194** for cooling therein to about −46° C. by flow through passageway **157'** in countercurrent heat exchange with the refrigerant in conduit **196**. The partially cooled, compressed nitrogen refrigerant is then directed through conduit **158'** to passageway **159** in second forecooler **198** for further cooling therein to about −60° C.

It is to be noted that the externally supplied refrigerant such as dichlorodifluoromethane is introduced to the system through conduit **200** and divided into two parts. The first part is directed through passageway **125** in first forecooler **122** for cooling of the compressed feed gases in passageways **120** and **124**, and the second part serves to further cool the nitrogen refrigerant by flowing through passageway **202** in second forecooler **198**.

The further cooled compressed nitrogen refrigerant is discharged from second forecooler **198** into conduit **160** and directed to second cold leg heat exchanger **204** for still further cooling in passageway **161** by refrigerant gas in thermally associated passageway **206**. The still further cooled refrigerant gas is then directed through conduit **162** to the inlet side of work expander **163** for expansion to a low pressure with simultaneous cooling and the production of external work. As in the FIG. 1 embodiment, the work expansion step may consist of one or more expansion stages with reheat of the partially expanded refrigerant therebetween if desired. The cold nitrogen gas exhausted from work expander **163** into conduit **165** is divided into two fractions. One fraction is passed consecutively through first liquefier **134**, cold leg **126**, and warm leg **116** for cooling, liquefying, and possibly subcooling of the compressed nitrogen feed. The second fraction of the work expanded nitrogen refrigerant is diverted from conduit **165** to conduit **208** for flow through passageway **210** in second liquefier **188** so as to liquefy and possibly subcool the countercurrently flowing oxygen feed in passageway **186**.

Referring now to the control aspects of the FIG. 2 system, the temperature of the expander exhaust stream is sensed and transmitted by means of conduit **172** to controller **174** which in turn is connected to liquid level controller **178** by conduit **176**. The latter instrument has liquid level sensing conduits **182** and **182***a*, and is connected by means of conduit **180** to valve **140** for controlling the rate of liquid nitrogen withdrawal. Thus, the liquefier liquid level of the larger feed stream may be regulated by its withdrawal control valve in response to the sensed refrigerant expander exhaust temperature as previously described. Likewise, the withdrawal control valve **192** for the smaller (or other equivalent) feed stream may be operated to maintain an equivalent liquid level by manually or automatically setting the liquid level controller **220** accordingly. However, the preferred method according

to the present invention is to operate the withdrawal valve for the smaller feed gas stream to match the temperature of partially warmed refrigerant gas leaving the warm end of the first and second liquefiers **134** and **188**. That is, the temperature of the refrigerant gas leaving second liquefier **188** in conduit **212** is closely matched to the temperature of the refrigerant gas leaving first liquefier **134** in conduit **166**.

The temperature of the expanded and partially warmed refrigerant gas in conduit **212** is sensed by suitable means such as a thermocouple and transmitted by conduit **214** to controller **216**. The latter is in turn connected by conduit **218** to liquid level controller **220**, having suitable liquid level sensing conduits (not shown). The latter transmits an electrical or pressurized gas signal through conduit **222** to liquid oxygen withdrawal valve **192** for control thereof.

It will be noted that the expanded refrigerant gas stream is divided downstream of the expander **163** and returned to the warm end of the heat exchange system as two separate streams. To properly balance flows in these two parallel paths, control valve **224** is employed in the refrigerant stream passing through the oxygen feed refrigeration system, such smaller stream normally having a lower pressure drop than the nitrogen feed refrigerant.

The liquefaction control process of this invention has been described in terms of a large flow volume nitrogen or oxygen liquefaction system. In such cases the feed gas and refrigerant gas (e.g., oxygen and nitrogen) would be generally compressed to about 70 p.s.i.g. or higher. The control process of this invention will also apply to other liquefaction processes wherein a refrigerant gas having a boiling point below —80° C. is first compressed and later work expanded to cool the refrigerant gas to a temperature below the desired liquefied temperature of the feed gas. More specifically, small, generally low output helium refrigerators or hydrogen liquefiers may be effectively controlled by the process here. These devices may use high speed turbomachinery and operate at relatively low head pressures, often only two or three atmospheres and in some cases the refrigerant gas may be at a subatmospheric pressure. Since the system is closed, the light refrigerant gases may be used at subatmospheric pressures, i.e., compressed to one subatmospheric presure and then expanded to a lower subatmospheric pressure for cooling.

An example of a cryogenic refrigerator of this type would involve supplying pressurized feed gas as at **14** in FIG. 1 and passing the feed gas through at least one heat exchange zone, e.g., heat exchanger **16**. The cooled feed gas is then directed to another heat exchanger, such as exchanger **26**, or passed directly to a liquefier, such as **34**, for liquefaction. The refrigerant gas, which might be hydrogen, helium, or other low boiling gases or mixtures thereof, is supplied to a compresser, such as **51**, for compression to a suitable pressure which may be only 2 or 3 atmospheres or lower. The compressed refrigerant gas is passed to the heat exchanger or heat exchangers, which are arranged in series if more than one is used, for passage therethrough (as shown in FIG. 1 in passages **57'** and **61**) in countercurrent heat exchange relation with the work expanded refrigerant in conduits **31** discharged from the liquefier **34**. The cooled compressed refrigerant gas is then work expanded, as at **63**, and the expander exhaust gas is passed from the cold end to the warm end of liquefier **34** through conduit **37** to liquefy and subcool the feed gas. According to this invention, the expander exhaust temperature is sensed in conduit **65** by a thermocouple and the reading transmitted to controller **74**. A signal is sent to valve **41** which controls the rate of liquefied feed gas withdrawal to vary the liquid level in the liquefier, and thus vary the effective heat transfer area as explained previously whereby the system is effectively adjusted to meet fluctuating product conditions.

Although preferred embodiments of the invention have been described in detail, it is contemplated that modifications of the process and apparatus may be made and that some features may be employed without others, all within the spirit thereof as set forth herein.

What is claimed is:

1. In a refrigeration process for cooling and liquefying feed gas by heat exchange with a compressed refrigerant gas wherein said feed gas and refrigerant gas both have boiling points below —80° C. at atmospheric pressure, and wherein the compressed refrigerant gas is work expanded to sufficiently low pressure to develop power and thereby cool the gas to a temperature below the desired liquefied temperature of said feed gas, and subsequently passed in countercurrent heat exchange relation with the feed gas in at least one liquefying zone and at least one heat exchange zone for cooling and liquefaction thereof, the improvement comprising sensing a refrigerant gas temperature from the work expansion, and controlling the withdrawal rate of liquefied feed from said liquefying zone in response to such sensed temperature to vary the liquid level in the liquefying zone and thereby vary the effective heat transfer area therein whereby a portion of the heat transfer load can be shifted to the heat exchange zone where compressed refrigerant work expansion feed gas can be regulated in temperature by heat exchange before work expansion so as to maintain the refrigerant gas work expansion exhaust temperature below said desired liquefied temperature of said feed gas.

2. A process according to claim **1** in which said refrigerant gas work expansion temperature is maintained at about the condensation temperature of said refrigerant.

3. A process according to claim **1** in which the liquefied feed gas is subcooled by said heat exchange with the refrigerant gas.

4. In a refrigeration process for cooling and liquefying feed gas by heat exchange with a refrigerant gas wherein said feed gas and refrigerant gas both have boiling points below —80° C. at atmospheric pressure and are compressed to at least 70 p.s.i.g., and wherein the compressed refrigerant gas is work expanded to sufficiently low pressure to develop power and thereby cool the gas to a temperature below the desired liquefied temperature of said feed gas, and subsequently passed in countercurrent heat exchange relation with the feed gas in at least one liquefying zone and at least one heat exchange zone for cooling and liquefaction thereof, the improvement comprising sensing a refrigerant gas temperature from the work expansion, and controlling the withdrawal rate of liquefied feed from said liquefying zone in response to such sensed temperature to vary the liquid level in the liquefying zone and thereby vary the effective heat transfer area therein whereby a portion of the heat transfer load can be shifted to the heat exchange zone where compressed refrigerant work expansion feed gas can be regulated in temperature by heat exchange before work expansion so as to maintain the refrigerant gas work expansion exhaust temperature below said desired liquefied temperature of said feed gas.

5. A process according to claim **4** in which said refrigerant gas work expansion temperature is maintained at about the condensation temperature of said refrigerant.

6. A process according to claim **4** in which the liquefied feed gas is subcooled by said heat exchange with the refrigerant gas.

7. In a refrigeration process for cooling and liquefying feed gas by heat exchange with a compressed refrigerant gas wherein said feed gas and refrigerant gas both have boiling points below —80° C. at atmospheric pressure and are compressed to at least 70 p.s.i.g., and wherein the compressed refrigerant gas is work expanded to sufficiently low pressure to develop power and thereby cool the gas to a temperature below the desired liquefied temperature of said feed gas and subsequently passed in countercurrent heat exchange relation with the feed gas in at least one liquefying zone and at least one heat exchange zone

for cooling and liquefaction thereof, the improvement comprising sensing the refrigerant gas exhaust temperature from the work expansion, and controlling the withdrawal rate of liquefied feed from said liquefying zone in response to such exhaust temperature to vary the liquid level in the liquefying zone and therefore vary the effective heat transfer area therein whereby a portion of the heat transfer load can be shifted to the heat exchange zone where compressed refrigerant work expansion feed gas can be regulated in temperature by heat exchange before work expansion so as to maintain said refrigerant gas exhaust temperature below said desired liquefied temperature of said feed gas and slightly above the condensation temperature of the refrigerant.

8. A process according to claim **4** wherein the refrigerant gas is nitrogen.

9. A process according to claim **4** wherein the refrigerant is air.

10. A process according to claim **4** wherein the refrigerant is nitrogen and the feed gas is oxygen.

11. A process according to claim **4** wherein the refrigerant and feed gas are nitrogen.

12. In a refrigeration process for cooling and liquefying a first larger quantity feed gas and a second smaller quantity feed gas with a refrigerant gas wherein the feed and refrigerant gases have boiling points below —80° C. at atmospheric pressure and are compressed to at least 70 p.s.i.g., and wherein the compressed refrigerant gas is work expanded to sufficiently low pressure to develop power and thereby cool the gas to a temperature below the desired liquefied temperatures of the first and second feed gases, and fist and second parts of the cold refrigerant gas are subsequently passed in countercurrent heat exchange relation with the first and second feed gases in a first liquefying zone and a heat exchange zone and in a second liquefying zone and heat exchange zone for cooling and liquefaction thereof, the improvement comprising sensing the refrigerant gas exhaust temperature from the work expansion, controlling the withdrawal rate of liquefied first larger quantity feed from said first liquefying zone in response to such sensed temperature to vary the liquid level in said first liquefying zone and thereby vary the effective heat transfer area therein whereby a portion of the heat transfer load can be shifted to a heat exchange zone to regulate the temperature of compressed refrigerant work expansion feed gas before work expansion so as to maintain said refrigerant gas exhaust temperature below said desired liquefied temperatures of the feed gas and slightly above the condensation temperature of the refrigerant; sensing the temperature of the partially warmed and work expanded second refrigerant gas part having liquefied and subcooled said second smaller quantity feed gas in said second liquefying zone and controlling the withdrawal rate of subcooled liquefied second feed from said second liquefying zone to vary the liquid level in the second liquefying zone and thereby vary the effective heat transfer area therein to regulate the temperature of the second part of the refrigerant gas in response to the sensed temperature of the first refrigerant gas part.

13. A process according to claim **12** in which said first larger quantity feed gas is nitrogen and said second smaller quantity feed gas is oxygen.

14. Refrigeration apparatus for cooling and liquefying feed gas by heat exchange with a refrigerant gas, said feed gas and said refrigerant both having boiling points below —80° C. at atmospheric pressure, comprising means for providing said refrigerant gas; means for compressing said refrigerant to a pressure of at least 80 p.s.i.g.; means for aftercooling the compressed refrigerant to a temperature below about 40° C.; means for work expanding the compressed and aftercooled refrigerant gas to sufficiently low pressure to develop power and thereby cool the gas to an exhaust temperature below the condensation temperature of said feed gas; means for providing said feed gas and compressing such gas to a pressure of at least 70 p.s.i.g.; first heat exchanger means for partially cooling the compressed feed gas; second heat exchanger means for further cooling the partially cooled feed gas; means for liquefying the further cooled feed gas; means for withdrawing the liquefied feed from the liquefier as a product; means for passing the work expanded refrigerant gas consecutively through said liquefier, second heat exchanger means, and first heat exchanger means in countercurrent heat exchange with the feed stream to effect such cooling, liquefaction, and subcooling thereof; means for withdrawing the warmed refrigerant gas from the warm end of said first heat exchanger means and recirculating such refrigerant to the refrigerant compression means; means for consecutively passing said compressed and aftercooled refrigerant gas through said first and second heat exchange means for further cooling therein by heat exchange with the expanded refrigerant gas withdrawn from the liquefier before passage of the compressed refrigerant gas to the work expansion means; means for sensing said exhaust temperature from the work expander; means for controlling the withdrawal rate of the liquid product from said liquefier in response to the sensed exhaust temperature of said work expander so as to maintain such exhaust temperature below the desired subcooled temperature of said liquefied feed gas withdrawn from said liquefier and slightly above the condensation temperature of the refrigerant.

15. Refrigeration apparatus according to claim **14** in which variable area nozzles are provided at the work expander inlet for maintaining the head pressure of said refrigerant gas substantially constant.

16. Refrigeration apparatus according to claim **14** wherein means are provided for supplying an external refrigerant, and a forecooler is provided between said first and second heat exchange means for further cooling said partially cooled feed stream, and means for introducing said external refrigerant to said forecooler to effect said further cooling.

17. Refrigeration apparatus according to claim **14** including means for throttling the product liquid from the feed gas pressure to a relatively low pressure thereby evaporating a portion of such liquid, means for separating the evaporated and liquid portions of the throttled fluid, and means for returning said evaporated portion to said feed gas for recompression therewith.

18. In a refrigeration process for cooling and liquefying a feed gas by heat exchange with helium gas wherein said feed gas has a boiling point below —80° C. at atmospheric pressure, and wherein the helium gas is work expanded to sufficiently low pressure to develop power and thereby cool the gas to a temperature below the desired liquefied temperature of said feed gas, and subsequently passed in countercurrent heat exchange relation with the feed gas in at least one liquefying zone and at least one heat exchange zone for cooling and liquefaction thereof, the improvement comprising sensing a refrigerant gas temperature from the work expansion, and controlling the withdrawal rate of liquefied feed from said liquefying zone in response to such sensed temperature to vary the liquid level in the liquefying zone and thereby vary the effective heat transfer area therein whereby a portion of the heat transfer load can be shifted to the heat exchange zone where compressed refrigerant work expansion feed gas can be regulated in temperature by heat exchange before work expansion so as to maintain the refrigerant gas work expansion exhaust temperature below said desired liquefied temperature of said feed gas.

19. In a refrigeration process for cooling and liquefying feed gas by heat exchange with hydrogen gas wherein said feed gas has a boiling point below —80° C. at atmospheric pressure, and wherein the hydrogen gas is work expanded to sufficiently low pressure to develop

power and thereby cool the gas to a temperature below the desired liquefied temperature of said feed gas, and subsequently passed in countercurrent heat exchange relation with the feed gas in at least one liquefying zone and at least one heat exchange zone for cooling and liquefaction thereof, the improvement comprising sensing a refrigerant gas temperature from the work expansion, and controlling the withdrawal rate of liquefied feed from said liquefying zone in response to such sensed temperature to vary the liquid level in the liquefying zone and thereby vary the effective heat transfer area therein whereby a portion of the heat transfer load can be shifted to the heat exchange zone where compressed refrigerant work expansion feed gas can be regulated in temperature by heat exchange before work expansion so as to maintain the refrigerant gas work expansion exhaust temperature below said desired liquefied temperature of said feed gas.

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