This invention relates to new and useful improvements in turbo-expander-compressor units for high pressure natural gas streams.

The invention is particularly concerned with expander compressor units for use in conjunction with low temperature separation systems and is directed to lubrication systems for such units.

In a high pressure, high speed turbo-expander-compressor unit, there arise certain rather difficult lubrication problems in conjunction with the maintenance of adequate lubrication of the unit, in conjunction with conservation of the lubrication liquid, and in conjunction with avoiding marked pressure changes in the various portions of the lubrication system in order to prevent excessive foaming of the lubrication liquid due to gas dissolved therein under relatively high pressure. The problems of simplicity and effectiveness of design, adequate gas chilling, and positive circulation of the lubricating liquid are also encountered. It is the purpose of this invention to solve the above problems as well as such additional problems as are encountered in the operation and maintenance of this type of system.

It is, therefore, one object of this invention to provide an improved turbo-expander-compressor unit which may be operated either internally or externally of a low temperature separation vessel which includes an internal lubrication system providing adequate lubricant flow but avoiding marked pressure differentials within the lubrication system.

An additional object of the invention is to provide a system of the character described in which provision is made for minimizing, if not eliminating, the loss of lubricating liquid from the lubrication system.

A still further object of the invention is to provide an improved lubrication system for a low temperature expansion turbine of the character described in which means are provided for ensuring the positive flow of the lubricating liquid, but in which all portions of the lubrication system are maintained under pressures of the same magnitude.

Another object of this invention is to provide an improved turbo-expander-compressor unit having closely spaced turbine and compressor rotors mounted on one end of a shaft in a single pressure housing and a unit of the character described in which the shaft bearings and lubrication system may be confined in a single enclosure requiring only a very simple pressure seal.

A further object of the invention is to provide an improved turbo-expander-compressor unit having a novel shaft seal between the turbine and compressor rotors for minimizing loss of recoverable hydrocarbons to the compressor outlet.

Another object of the invention is to provide an improved turbo-expander-compressor unit having bearing means and means for supplying a liquid lubricant from a lubricant enclosure to the bearing means operating largely independently of the lubricant level in the enclosure.

Still a further object of the invention is to provide an improved turbo-expander-compressor unit which, for starting purposes, may be pressurized from the compressor outlet without causing the compressor rotors to spin in a reverse direction and possibly damage the unit.

Yet another object of the invention is to provide a lubrication system of the character described in which the lubrication liquid is constantly supplied to a chamber at a controlled rate and from which the lubrication liquid is withdrawn by a novel flowing means for positive supply to the turbine bearings.

An additional object of the invention is to provide an improved lubrication system for a turbo-compressor unit including lubricant pumping means having no wearing parts and a minimum of moving parts.

A still further object of the invention is to provide a turbo-expander-compressor having positive means for the removal of small accumulations of solids in the turbine nozzle ring.

Other and more particular objects will be apparent from a reading of the following description and claims.

A construction designed to carry out the invention will be hereinafter described, together with other features of the invention.

The invention will be more readily understood from a reading of the following specification and by reference to the accompanying drawings, wherein an example of the invention is shown, and wherein:

FIG. 1 is a diagrammatic view of a low temperature separation system constructed in accordance with this invention.

FIG. 2 is a vertical, longitudinal, sectional view of a low temperature separation vessel constructed in accordance with this invention.

FIG. 3 is an end elevational view of a turbo-expander-compressor unit and low temperature separator constructed in accordance with this invention.

FIG. 4 is a side elevational view of a turbo-expander-compressor unit constructed in accordance with this invention.

FIG. 5 is a plan view of the turbine unit.

FIG. 6 is a front elevational view of the turbine unit.

FIG. 7 is a vertical sectional view of the turbine unit.

FIG. 8 is a horizontal, cross sectional view taken upon the line 8—8 of FIG. 7.

FIG. 9 is a view in perspective of the compressor volute unit.

FIG. 10 is an enlarged, fragmentary, vertical, sectional view of the expander and compressor rotors, and

FIG. 11 is a view similar to FIG. 10 showing the lower portion of the turbine unit.

In the drawings, in FIG. 1, the numeral 10 designates a petroleum well producing petroleum fluids under relatively high pressures of the order of magnitude of several thousand pounds per square inch or considerably more. Under such pressurized conditions, the well fluids are substantially entirely in the gaseous phase, but by suitable pressure reduction and chilling of the well fluids, valuable liquid hydrocarbons may be removed and recovered therefrom while, at the same time, the effluent gas stream may be sufficiently freed of water and water vapor as to permit its delivery to a gas transmission pipe line. The present invention is directed toward effecting such liquid hydrocarbon recovery and gas dehydration while at the same time making provision for more effective cooling of the well stream and the efficient handling of the well stream at pressures lower than would otherwise be possible.

It is often desirable to heat the well stream prior to its further processing, and for this purpose, the stream is passed to a conventional heating unit 11 wherein the temperature of the well stream is raised to the desired level after which the stream is passed to a high pressure separator 12 wherein any liquids which may be present at this point. In the event any salt water is present in the well stream, it will be removed in the high pressure separator 12 and the operator assured that any remaining water will be in the vapor phase and thus will be fresh water.
Following discharge of the well stream from the separator 12, a suitable desiccant or hydrate inhibitor, such as calcium chloride, methyl or ethyl alcohol, the various ethylene glycols, and the like, may be injected through the injection point 13 prior to passage of the well stream into the heat exchanger 14. Within the heat exchanger, the well stream is cooled to the desired degree by indirect heat exchange with cold gas supplied from a subsequent point in the system, and further quantities of a hydrate inhibitor may be injected through the conductor 15 following the discharge of the well stream from the heat exchanger. From the heat exchanger 14, the well fluids are conducted to the inlet 16 of a turbo-expander-compressor unit 17 wherein the well fluids undergo an expansion or pressure reduction step, while doing work, and from which they are exhausted at a reduced pressure into the low temperature separator 18. Within the low temperature separator 18, the well fluids are resolved into cold gas which may be passed through a conductor 19 to the low pressure gas inlet of the unit 17 and therein compressed for passage through a conductor 20 to a three-way valve 21 or the cold gas may be by-passed around the unit 17 and carried through a conductor 22 to the valve 21.

The valve 21 is operated by suitable temperature responsive means and functions to supply the cold gas to the heat exchanger 14 for cooling of the well stream, or to bleed off and send all of the gas through the conductor 23 which extends around the exchanger 14 and joins into the outlet gas discharge conductor 24 of the exchanger. In this manner, the degree of chilling of the well stream passing through the exchanger 14 may be controlled as desired, the well stream usually being chilled to a temperature just above the point of gas hydrate formation as determined by the volumes of hydrate inhibitor which may have been injected.

The water separated from the well stream in the separator 18 is withdrawn through the outlet conductor 25 while the separated and recovered hydrocarbons are carried through a conductor 26 to a low pressure separator 27 from which somewhat stabilized hydrocarbon liquids may be withdrawn through the outlet conductor 28, or may be passed through the conductor 29 into a stabilizing or fractionating tower 30 for more efficient and effective stabilization of these liquids and removal therefrom of the gaseous portions which could not be maintained in the liquid phase under conventional storage conditions. The separated gas removed in the low pressure separator 27 is withdrawn through an outlet conductor 31 which leads into the vicinity of the bearing insert 44 below the clearance space 68, and a wide shallow groove 69 is cut in the exterior wall of the bearing insert, the construction with the low temperature separator disclosed in the U.S. patent to Jay P. Walker et al., No. 2,747,002, issued May 22, 1956. Accordingly, the low temperature separator will not be described in detail other than to point out to those skilled in the art the general nature thereof as indicated by dotted lines in FIG. 2, or interiorly of the low temperature separator, the high pressure well stream conductor 38 leading from the heat exchanger 14 or conductor 37 into the turbine inlet of the unit 17, while the turbine outlet discharges into the interior of the separator 18 through the conductor 39.

As previously described, the low pressure separated gas conductor 19 is connected into the unit 17 while the gas discharge conductor 20 leads therefrom.

In certain of the views of the drawings, the orientation of the several inlets and outlets to the turbine unit have been rotated through 90° in order to illustrate the structure, it being noted that the correct orientation is shown in FIG. 3 of the drawings in which the turbine unit is mounted exteriorly and laterally of the low temperature separation vessel upon a suitable stand or support in the manner illustrated in FIG. 2. The low pressure separator housing 41 into which the conductors 19, 20, and 38 are connected and which receives a cover plate 42 joined to the body 41 by suitable bolts and nuts 43 and through which the conductor 39 extends from the interior of the unit. The body 41 also contains the integrally formed boss 44 to which an oil cup 45 is joined as by welding or other suitable means. A bracket 46 depends from the boss 44 and receives a tubular support 47 extending laterally from the bracket for mounting an oil reservoir tank 48 at one side of the oil cup 45. The lubricant reservoir 48 comprises an elongate cylindrical tank extending horizontally and transversely with respect to the vertical axis of the turbine unit, there being provided an oil supply conductor 49 leading from the lower portion of the reservoir 48 through a filter 50 into the lower portion of the cup 45, while a return conductor 51 leads from the boss 44 to the upper portion of the reservoir 48. A suitable silt glass 52 constantly indicates the level of the lubricating liquid within the reservoir 48.

The housing 41 is provided with an axial cylindrical bore 53 extending downwardly through the boss 44 into the upper end of the cup 45 and opening upwardly into an enlarged, cylindrical compressor rotor chamber 54. The chamber 54 in turn opens upwardly into an enlarged, cylindrical, turbine rotor chamber 55 extending upwardly to the upper end of the housing 41. The separated gas inlet conduit 19 of the separator 18 is connected through a radial inlet passage 56, and a compressed gas outlet passage 57 extends radially from the housing to the opposite side of the chamber 54 and opens into the outlet gas conductor 20. The inlet gas conductor 38 opens into the radial passage 58. The upper lubricant conductor 51 is exposed to the bore 53 through a radial passage 59 provided in the boss 44.

The bore 53 receives a tubular bearing insert 60 carrying a radial, outwardly extending flange 61 at its upper end which seats upon the bottom of the chamber 54 and through which bolts 62 extend for securing the insert in position within the bore 53. The flange 61 is formed shortly below the upper extremity of the insert 60 so as to provide a short upstanding neck 63 on the uppermost end of the insert 60 carrying an annular, upwardly opening groove 64 receiving an O-ring or other suitable sealing element 65. An annular bearing seat 66 is formed internally of the insert 60 near its upper end and receives a ball-bearing or other suitable anti-friction means 67.

The external surface of the insert 60 has a snug fit within the bore 53 for a short distance below the flange 61 and is then reduced slightly in diameter to provide an ample clearance space 68 between the insert and the bore 53. Below the clearance space 68, a wide shallow groove 69 is cut in the exterior wall of the bearing insert, the
groove forming a lubricant mist circulation space registering with the passage 59. A grooved outwardly extending flange 70 is carried upon the insert below the groove 69 and receives an O-ring or other sealing element 71 for forming a liquid seal between the bearing insert and the bore 53. An integral seal 72 extends outwardly from the flange 71 and is formed with an internal bearing receiving bore 73 within which a lower ball-bearing or other suitable anti-friction means 74 has a snug fit.

It will be noted from FIG. 7 that the internal diameter of the cup 45 is smaller than the diameter of the bore 53 so as to form an upwardly-facing offset or shoulder 75 at the upper end of the oil cup. A thin-walled lubricant receptacle 76 is disposed within the cup 45, the receptacle 76 being formed with an outwardly extending flange at its upper end received upon the shoulder 75 for supporting the receptacle with its bottom wall 77 spaced a short distance above the bottom of the cup 45. The lubricant supply conductor 49 opens into the interior of the cup 45 through a radial passage 78, and thus, the lubricant is supplied to the cup 45 exteriorly of the receptacle 76, a small opening 79 is present in the bottom 77 of the receptacle and hence oil may flow upwardly through the opening at a controlled and relatively slow rate into the interior of the receptacle 76.

The turbine shaft is rotatably mounted upon the bearing 67 and 74 and includes a rather massive main body 80 disposed between the bearings and merging at its upper and lower ends into shouldered bearing faces, 81 and 82 respectively, received within the bearings 67 and 74. A tapered, tubular spindle 83 is screw-threadedly connected to the lower end of the turbine shaft and about the lower face of the bearing 74 so as to clamp the bearing into position upon the shaft. An axial opening 84 extends downwardly through the spindle 83 terminating at its open lower end in a convergent section 85 and communicating at its upper end with an axial passage 86 extending upwardly approximately midway of the shaft portion 80 and communicating at its upper end with radial passages 87 extending to the outer periphery of the shaft portion 80.

Above the upper bearing face 81, the turbine shaft is reduced in diameter as shown at 88 in FIG. 10 and receives a spacer sleeve 89 formed at its upper end with a reduced, upstanding neck 90. A labyrinth seal cap 91 encircles the spacer sleeve 89, extending from a point near the lower end of the sleeve to a point spaced slightly below the neck 90. The lower end of the cap 91 carries an O-ring and is engaged to the upper end of the bearing insert 60 by suitable bolts or screws 93, the O-ring 65 functioning to seal the joint between the cap 91 and the insert 60.

A circular bearing seat 94 is formed in the underside of flange 92 and receives the upper side of the bearing 67 whereby the bearing is clamped in position between the seats 66 and 94. Further, the cup 91 is provided in its under side with an annular recess 95 surrounding the lower end of the spacer sleeve 89 and registering with the bearing 67 for the passage of lubricant downwardly through the bearing from the recess 95. A brad簪 passage 96 extends laterally and radially from the recess 95 into alignment with a longitudinal lubricant passage 97 extending downwardly through the side wall of the bearing insert 60 and opening into the bore 53 exteriorly of the depending collar 72. In addition, a radial passage 98 is provided for the communication of the recess 95 to the bearing seat 94. The lubricant from the insert 60 into the annular groove 69 and communicating through the passage 59 with the lubricant conductor 51.

With the structure described, continuous and effective circulation of a lubricant to the upper and lower turbine shaft bearings is ensured it being pointed out that the supplying of the lubricant takes place in the form of the continuous passage of a lubricant mist or fog to both the upper and lower bearings. As the turbine shaft revolves at high speed, lubricating liquid flowing through the conductor 49 and passage 78 into the opening 79, and standing in the lower portion of the receptacle 76, will enter into the lower passages 87, and the action of the centrifugal forces created in the rapidly revolving, convergent lower portion 85 of the bore 84, will be forced to flow upwardly through the bore 84 into the bore 86 and outwardly into the lateral passages 87. The centrifugal force at the wall of the bore 84 is of the order of 1,000 times gravity. Thus the surface of the oil inside the bore 84 will be virtually cylindrical, deviating from a perfect cylinder only by a slope of the order of 1/1000 of an inch per inch of height. Hence, any oil that is introduced inside the convergent lower portion 85 will assume this cylindrical form against the walls of the rotating bores 84 and 86, and it will split through the passages 87 because the internal rims of the passages 87 are at a greater radius than the convergent lower portion 85. Thus, the only requirement of the lubricant to be raised through the bores 84 and 86 and delivered through the lateral passages 87 is to be introduced into the bore 84. This can be accomplished by the oil wetting the divergent lower portion 85 or by it merely being injected into the portion 85 by the action of orifice 79. The lubricant in the reservoir tank 48 is standing at a higher level than the orifice 79, and therefore there will take place some jetting action of the oil through the orifice 79 because of the hydrostatic head existing from the body of oil in the reservoir 48.

In this manner, a very effective lubricant pump is provided with a minimum of moving parts and an absence of bearing parts in the pump unit. The lubricant thus be thrown outwardly at high velocity into the space between the element 80 and the bore of the bearing insert 60 in the form of a lubricant mist or fog, and will be maintained in such dispersed condition due to the high speed of revolution of the element 80. Further, since the element 80 has an outside diameter greater than the revolving portions of the upper and lower bearings as well as the shaft thereabove and the spindle 83 therebelow, the pressure of the gas present within the annular space around the element 80 will be greater than the gas pressure above and below the bearings 67 and 74, resulting in a passage of the lubricant mist or fog outwardly through the passage 98 into the passage 97. From the passage 97, the lubricant mist passes upwardly through the lower bearing and also into the recess 95 for downward flow through the upper bearing. At the discharge end of the passages 87, and oil fog or mist is introduced into two internally circulating streams of gas. These internally circulating streams of gas pass through the bearings as part of their cycles and thus carry the lubricant to the bearings. They are caused to circulate by the actions of the tapered portions of the element 80 acting as centrifugal pumps. The inlet to each so-called pump is through the adjacent ball-bearing, and the discharge in each case is the annular space inside the insert 60. Of course, some of the lubricant fog or mist will flow through the passage 98 to the groove 69 and conductor 51 for return to the oil reservoir 48, while other portions of the lubricant will coalesce and return to the receptacle 76. Thus, there will be a constant flow of lubricating liquid from the reservoir tank into the lower portion of the receptacle 76, passage of this lubricant to the bearings in the form of a fog or mist, and the constant passage from the bearings into the reservoir tank through the conductor 51. To ensure additionally the return of some lubricant to the reservoir 48, the passage 98 is horizontally aligned with the passages 87 so that some portion of the lubricant thrown outwardly from the passages 87 will enter into the passage 98 for return to the reservoir tank, and thus, a slow but steady circulation of all of the lubricant co-
tained within the tank 48 to and through the lubrication system in ensured. Only a very modest rate of lubricant flow is required for adequate lubrication of the turbine shaft bearings, but by means of the described structure, the relatively large quantity of lubrication liquid contained within the vessel 48 is thus slowly but continuously employed for lubrication of the turbine shaft bearings and the entire body of lubricant employed for such lubrication in order to ensure a long useful life of the lubrication liquid and very infrequent need for replacement or renewal thereof. It is to be noted that all lubricant entering the groove 69 is prevented by the seal 71 from returning downwardly into the receptacle 76 and accordingly is directed into the passage 59 for return to the oil reservoir through the conductor 51. It is also to be noted that the receptacle 76 will at all times contain only a very small quantity of lubricant since the latter will flow into the receptacle only at a very low rate due to the smallness of the aperture 79 and because the capacity of the rapidly revolving compressor rotor 111 is so little, that the lubricant upwardly into the passage 86 is much greater than the capacity of the aperture 79 to admit lubricant into the interior of the receptacle 76. In this manner, the oil level will be maintained at or near the lower extremity of the spindle 83 and there will be no external loading of the turbine shaft bearings and the lower portion of the spindle 83 in a body of liquid lubricant. Such functioning of the lubricant circulation system also prevents the creation of a turbulent body of lubricant within the lower portion of the receptacle 76, as well as forestalling the rapidly revolving spindle 83 throwing the lubricant outwardly toward the walls of the receptacle and creating a lubricant mist or lubricant splash condition within the receptacle.

Wit in the chamber 54, there is disposed the compressor volute 99 illustrated in FIG. 9 which includes an outer, annular wall 100 having its lower edge joined to the lower edge of an inner, annular wall 101 by an inclined bottom wall 102. The outer wall 100 is provided at its upper edge with an outwardly extending flange 103 which is received in an annular seat 104 formed about the upper edge of the chamber 54. Suitable bolts 105 secure the volute 99 within the chamber 54 and clamp the flange 103 to the seat 104.

It is to be noted that the shortest portion of the outer wall 100 is placed in alignment with the passage 56 so that the gas passing through the inner wall 100, while the tallest portion of the outer wall 100 is provided with a radial opening 105 positioned in alignment with the passage 57 so that the latter passage is placed in communication with the space between the inner and outer walls of the volute. A circular groove 106 surrounds the opening 105 and provides an O-ring or other suitable seal 107.

The inner wall 101 is formed with an inwardly extending flange 108 defining a central gas passage opening 109 carrying labyrinth seal grooves 110. A compressor rotor 111 is disposed within the shaft bearing 88 in the upper portion of the chamber 54 above the volute 99 and formed with a depending entrance throat 112 having a cylindrical external surface 113 fitting closely within the labyrinth seal 110 to minimize gas passage downwardly through the opening 109 around the exterior of the entrance throat 112. The outlet of the rotor 111 of course communicates with the space between the inner and outer wall of the volute 99, and by rapid revolution of the compressor rotor, gas is drawn in from the conductor 19 and passage 56 and compressed in the rotor for discharge through the passages 57 and outlet conductor 20. In order to provide a diffuser space surrounding the rotor 111, an annular ring 116, having a rounded outer periphery 117, is secured to the upper end of the inner wall 101 by bolts 118, or in any other suitable manner, the ring 116 projecting from the wall 101 to ward the outer wall 100 to define a shallow annular, discharge space 119. The gas leaving the rotor 111 is moving at a high tangential velocity and has the equivalent high energy content which, by reason of the free vortex created in the space 119, is converted to energy due to the centrifugal force to which the rapidly whirling gas is submitted. At the same time the tangential velocity of the gas is reduced equivalently.

The upstanding neck 90 of the spacer sleeve 89 has a snug sliding fit with the axial bore 121 of the compressor rotor 111 through which the turbine shaft extends. Thus, the rotor 111 is held at a proper elevation upon the turbine shaft and prevented from moving downwardly. The upper face of the rotor 111 is formed with spiral threads 122 surrounding the bore 121, and similar and matching threads 123, formed upon the depending hub 124 of a turbine rotor 125 are received in and mesh with the threads 122 to lock the turbine and compressor rotor rotationally together. The turbine rotor 125 is provided with an axial bore 126 receiving the upper end of the turbine shaft, and a counter bore 127 of the same diameter as the bore 126 holds the lock of the turbine rotor 125, and thus, the compressor rotor 111, to the turbine shaft, while a nut 129 screw-threaded on the upper end of the turbine shaft holds both rotors against upward movement and locks the threads 122 of the rotor 111.

For isolating the turbine and compressor rotors, the chamber 55 receives a plate like partition member 130 having an upstanding marginal wall 131 which closely adjoining the inner periphery of the chamber 55. A recess 132 on the outer lower edge of the wall 131 accommodates an O-ring or other suitable sealing means 133, while a shallow, annular groove 134, formed in the adjoining upper surfaces of the wall 131 and the turbine body 41 receives a flat sealing ring 135 for closing off the upper juncture between the partition member 130 and the turbine body. The cover plate 142 carries a depend ing, annular rib 136 of rectangular cross-section which projects downwardly into the groove 134 and constantly urges the gasket 135 into sealing position under the clamping effect of the bolts 43.

A shallow conical recess 137 in the under side of the partition member 130 conforms closely to the upper side of the compressor rotor 111, while a short, up standing, cylindrical section 138 formed centrally of the upper side of the plate member 130 carries a shallow conical recess 139 conforming to the underside of the turbine rotor 125. A central opening 140 and the recesses 137 and 139 is provided with a plurality of labyrinth sealing grooves 140 conforming closely to the outer periphery of the hub 124 of the turbine rotor and preventing the passage of high pressure gas from beneath the turbine rotor to the discharge side of the compressor rotor. An annular, rectangular groove 141 is cut in the labyrinth rings 140 near the lower portion thereof and aligns with radial openings 142 extending through the hub 124. One or more gas discharge passages 143 extend from the inner ends of the passages 142 into the central portion of the turbine rotor 125 rather closely adjacent the axis thereof in communication with the openings 142.

A nozzle plate 144 overlies the central portion of the partition plate 130 and has a snug sliding fit within the inner periphery of the upstanding marginal wall of the partition plate. An O-ring seal 145 is provided between the nozzle plate and the partition wall while suitable bolts 146 clamp the nozzle plate to the partition plate. The high pressure gas sealing ring 147 surrounds the upstanding cylindrical section 138 of the partition plate between the partition plate and the nozzle plate and is forced by the bolts 146 against an annular gasket ring 148 which underlies the sealing ring 147 and is forced thereby into sealing engagement to prevent the passage of high pressure gas directly from the inlet.
passage 58 into the space between the turbine rotor 125 and the recess 139. The nozzle plate is provided with a central, shallow, conical recess 149 conforming closely to the conical upper side of the turbine rotor 125 and having an axial opening 150 provided with the multiplicity of labyrinth seal grooves 151 closely surrounding the upstanding discharge neck 152 of the turbine rotor 125. The opening 150 communicates upwardly through the upper surface of the nozzle plate 144 and aligns with an axial passage 153 cut in the cover plate 42 and registering with the low pressure side of the nozzle plate 144. As shown in FIG. 8, the turbine rotor 125 is of the centripetal or inward-flow radial type having a plurality of spaced, curved, primary turbine blades 154 alternating with a plurality of curved, spaced, secondary turbine blades 155. By their design, these blades are adapted to receive a radial flow of high pressure gas passing from the periphery of the turbine rotor to the central discharge neck 152 thereof and to extract work from the gas during such passage for high speed rotation of the turbine shaft and delivery of energy thereto to drive the fluid transport unit, radial flow type turbine rotor 151. For distributing the inlet gas properly to the turbine rotor, the nozzle plate 144 is formed with a nozzle volute 156 having an inlet groove or notch 157 overlying the inner end of the passage 50 and extending outwardly over the periphery of the turbine rotor 125. A discharge passage 160 extends upwardly from the cavity 159 to the upper surface of the nozzle plate and is closed by a screw-threaded valve member 161 suitably mounted in the cover plate 42 and which may be opened to permit discharge of material from the cavity 159 into the low pressure space present between the nozzle plate and the cover plate 42 and within the annular rib 136.

In the operation of the turbine unit, high pressure gas from the heat exchanger 14 or from the high pressure separator 12, or both, enters through the conductors 38 and flows through the passage 58 and the cutaway portion 157 into the nozzle 157. Approximately one half the total pressure drop in the incoming gas occurs in the nozzle 157, the result being the imparting of a high tangential velocity to the gas to correspond properly with the high peripheral velocity of the turbine rotor. The fluid stream passes between the blades 154 and 155 of the turbine rotor 125, expanding in the process and imparting work to the turbine rotor so as to drive the rotor and the compressor rotor 111 at high speed. Of course, in expanding while doing work, the fluid stream undergoes isentropic expansion so that the well stream is reduced in temperature to a greater extent than would occur if the well stream were passed through a simple expansion step in order to achieve cooling through the well known Joule-Thomson Joule-Kelvin effect. Thus, in addition to the cooling achieved through the Joule-Thomson effect, the well stream is also subjected to cooling in that it is caused to do work in the expansion step and accordingly loses heat in the form of rotational energy imparted to the turbine rotor.

The expanded well stream passes upwardly through the central throat of the turbine rotor into the passage 153 and the conductor 39 which leads to the interior of the low temperature separator 18, the well stream separating in the latter vessel into liquid and cold gas layers. The cold gas, at reduced pressure, is withdrawn from the separator, the occurrence of separation through the outlets conductor 19 and enters through the passage 56 of the compressor unit into the chamber 54 to flow upwardly through the throat 112 of the compressor rotor 111 and be discharged from the periphery of the compressor rotor into the space 119 and then into the annulus between the outer and inner walls 100 and 101 of the compressor volute 99. The gas subsequently flows through the passage 57 and the conductor 20 into the balance of the separation system. Of course, the work extracted from the fluid stream by the turbine rotor 125 is utilized to drive the compressor 111 and bring the separated gas from its relatively low pressure as discharged from the low temperature separator 18 to a pressure sufficiently high for further processing of the gas and flow thereof into the various gas transmission pipe lines. As set forth in detail in the above application, the concomitant advantages and beneficial results are enumerated in detail in the above application.

The flow of the lubricating liquid, in the form of a mist or fog to the bearings 67 and 74 has been set forth with particularity hereinafter, but it would be well to point out the structural features of the invention which bring the potential loss of lubricating liquid to a minimal level as well as those particular features which serve to isolate the various containing chambers operating at different pressure levels or render the flow of gas between such chambers of no operational consequence.

To begin with, the unusually long labyrinth seal collar 91 effectively prevents the flow of gas entrained lubricant particles or mist upwardly around the exterior of the sleeve 89, first because the labyrinth collar is a closed one and second, because the pressure existing within the lubricant circuit is the same as that existing within the inlet throat 112 of the compressor rotor 111. Thus, both because the lubricant system is a closed circuit and because it operates at the same pressure as the gas inlet to the compressor rotor there is not only no tendency for gas to flow into or out of the lubricant circuit but such flow is positively resisted and prevented by the closed arrangement of the lubricant circuit. The rapidly revolving enlarged section 80 of the turbine shaft presents a positive flow of gas along with gas and lubricant fog downwardly through the upper bearing 67, thus constantly withdrawing gas and lubricant mist or fog from adjacency with the lower portion of the labyrinth seal collar 91.

The quite appreciable height or vertical length of the collar 91 is made possible by extending the collar upwardly into the inlet throat of the compressor rotor, and yet, excessive height of the turbine case and excessive length of the turbine shaft, with its resultant problems of vibration and shaft balancing, is avoided through use of the unique compressor inlet volute mounted in a chamber of minimum height yet making ample provision for gas inlet and discharge. The fixed annular ring 116 is also of importance in this latter aspect in that it forms one wall of the diffuser space 119 in the relatively small or restricted outlet chamber between the walls 100 and 101. The space 119 forms what may be called a vanless diffuser which becomes of importance when the turbine unit is first placed in operation. It sometimes is desirable to pressurize the unit from the outlet end, and this type of diffuser prevents the compressor rotor from acting as a turbine when such pressurizing is being carried out and thus avoids possible damage to the machine.

With respect to the labyrinth seal 110 around the outer periphery 113 of the inlet throat 112 of the compressor rotor, gas leakage through the labyrinth seal 110 would be of no particular importance insofar as conservation of lubricating liquid or recoverable hydrocarbon liquids would be concerned since the gas on both
sides of the seal is lean, denuded gas differing only in pressure. The seal is important, however, for preventing excessive leakage into the low pressure side of the compressor maintaining the effectiveness and efficiency of the gas compression step.

The labyrinth seal 140 in the central portion of the plate-like element 130 functions to prevent the loss of high pressure gaseous fluids, and especially those gaseous fluids still rich in hydrocarbons, into the gas outlet conductor 20 by restricting, if not stopping, such gas flow and for causing such flow as may occur to pass into the conductor 39 so that any recoverable hydrocarbons which may be present in the gas will be separated and recovered in the low pressure separator 18. It is to be noted that the low pressure zone existent within the discharge throat 152 of the turbine rotor is exposed through the passages 143 to the ports 142 and the groove 141, and that the pressure above and below the labyrinth seal 140 is at level above that present within the throat 152. Accordingly, any gas escaping downwardly through the seal 140 and the port 142, the compressor rotor will be drawn into the throat 152, and similarly, any gas escaping upwardly from above the compressor rotor 111, will also be drawn into the throat 152. In this manner, such gas as may leak past the seal 140 will be commingled with the cold low pressure gas flowing to the low temperature separator and will be subjected to the remaining liquidifica tion and separation steps carried out therein. It is to be noted that the groove 141 is positioned in the lower portion of the seal 140 whereby the larger portion of the seal is employed for resisting the flow of high pressure gas from beneath the turbine rotor while the smaller portion of the seal is employed for resisting the flow of the gas from above the compressor rotor, such latter gas being at a somewhat lower pressure than the inlet gas or well fluids. It is also pointed out that the passages 143 and ports 142 are non-parallel to the axis of the turbine shaft which prevents any accumulation of foreign material in the passages and ports due to deposit under centrifugal force.

The labyrinth seal 151 surrounding the discharge portion of the turbine rotor of course functions to minimize or eliminate flow between the high and low pressure sides of the turbine rotor 125 in order to preserve the effectiveness and efficiency of the latter, but again, such small escape of well fluids as might occur at this point would result in the loss of recoverable hydrocarbon liquids or impaired dehydration of the well streams.

It is to be noted that the turbine unit is constructed for great ease of assembly and disassembly which facilitates the inspection and maintenance of the turbine unit, but which at the same time favorably affects the performance of the turbine unit. As pointed out hereinafter, the overall height of the turbine housing as well as the overall length of the composite rotor structure within the housing is held to a minimum whereby problems of vibration and rotor balancing are noticeably reduced. The utilization of a centripetal type turbine rotor position and the use of multiple balls are also possible the disposition of the lubrication system below the compressor rotor so that the lubricant system need only be seated from the relatively low pressure portion of the unit rather than the high pressure areas which exist around the turbine rotor 125. This arrangement also makes possible the alignment of only the single conductor 39 to the cover plate 42 so that loosening of the bolts 43 and disconnecting of the conductor 39, any by a suitable coupling or joint, makes possible the removal of the cover plate followed by the complete disassembly of the entire expansion and compression unit. Here again, maintenance and inspection of the unit is greatly facilitated.

Further, the placing of the lubricant and bearing chamber below the compressor rotor causes this chamber to be a stagnant one providing the best possible chance of maintaining the quality of the lubricant, keeping the bearings clean, and conserving the lubricant.

It is also to be pointed out that the unique spiral threads 122 and 123 formed upon the adjoining faces of the compressor and turbine rotors make the two rotors together and yet require no physical access in order to establish this effective means of connection. Thus, with the plate-like element 30 in position within the turbine housing, the turbine rotor may be lowered into position and positively connected to the compressor rotor without there being any requirement for access for the possible positioning of connecting keys or other connecting means.

The entire expander compressor structure is thus made readily available for assembly or disassembly through the open upper end of the housing 41 exposed by the simple removal of the plate 42.

It is further pointed out that the inlet or nozzle volute 156 formed in the plate 144 besides properly distributing the incoming well fluids also functions as a highly efficient centrifugal separator in that any particles of foreign matter which may be present in the incoming well fluids will be thrown to the outer wall of the volute and caused to flow through the passage 158 into the ramp 159 for periodic removal by opening of the valve 161. Otherwise this centrifugal action might cause foreign material to remain in the volute and continue to erode the parts of the turbine and the turbine rotors and any part of the system are thus protected from the erosive or abrasive action of such particles of foreign matter which are sometimes unavoidably present in the incoming well fluids. In addition, any slight wear or abrasion resulting from the presence of such particles is restricted to the plate 144 and the sealing ring 147 disposed below and enclosing the lower side of the volute passage 156, and since these two elements are relatively inexpensive and easily replaceable, wear due to erosion or abrasion is effectively handled such that maintenance on this portion of this passage 58 and the inlet notch 157 is not critical.

The foregoing description of the invention is explanatory thereof and various changes in the size, shape, and materials, as well as in the details of the illustrated construction may be made, within the scope of the appended claims, without departing from the spirit of the invention.

What I claim and desire to secure by Letters Patent is:

1. In an expansion unit having a high speed turbine rotor, a lubrication system including, spaced bearing means, shaft means for the rotor supported by the bearing means, a lubrication enclosure encasing a portion of the turbine rotor projecting from the shaft means into the enclosure, the extension having a convergent portion on its end, the shaft means having a passage leading from the hollow extension to the exterior of the shaft between the bearing means for supplying lubricant to the exterior of the shaft and establishing a zone of oil fog formation, the portion of the shaft means between the bearing means having a greater diameter than the portions of the shaft means outward of the bearing means for circulating gas streams circumferentially through the bearings and through the zone of oil fog formation, and means for returning lubricant from the outward portions of the shaft means to the enclosure.

2. In an expansion unit, a lubrication system including, a turbine rotor, spaced bearing means positioned on one side of the turbine rotor, shaft means for the turbine rotor supported by the spaced bearing means, a lubrication enclosure on the same side of said turbine rotor as the bearing means, the shaft means having a hollow extension projecting into the enclosure, the extension having a convergent portion on its end, the shaft means having a passage leading from the hollow extension to the exterior of the shaft means between the bearing means for supplying lubricant to the exterior of the shaft and establishing a zone of oil fog formation, the portion of the shaft means between the bearing means having a greater diameter than the portions of the shaft means immediately outward.
of the bearing means for circulating gas streams circuitously through the bearings and through the zone of oil fog formation, and means for returning lubricant to the enclosure from the zones surrounding the portions of the shaft means immediately outward of the bearing means.

3. A lubrication system as set forth in claim 2, and a support for the bearing means surrounding the shaft means between the bearing means, the support having passages communicating between the inner and outer sides of the bearing means.

4. A lubrication system as set forth in claim 3, wherein the support has a cross passage communicating between the passages extending between the inner and outer sides of the bearing means.

5. A lubrication system as set forth in claim 3, wherein the support has a cross passage communicating between the passages extending between the inner and outer sides of the bearing means and aligned with the outlet of the shaft means passage.

6. An expansion unit including, a housing, spaced bearings in the housing, a shaft rotatably mounted in the bearings, an enlarged portion on the shaft between the bearings, a stub projection extending from one end of the shaft, a turbine rotor in the housing and carried on the shaft stub, an extension projecting from the opposite end of the shaft, the extension having an axial bore convergent at the outer end of the extension, the shaft having a passage communicating between the extension bore and the periphery of the enlarged portion between the bearings, a lubricant enclosure into which the extension projects, a lubricant reservoir in communication with the lubricant enclosure, and an elongate labyrinth seal closely surrounding the stub projection between the bearings and the turbine rotor.

7. In an expansion unit, a lubrication system including, a turbine rotor, spaced bearings positioned on one side of the turbine rotor, a shaft for the turbine rotor supported by the spaced bearings, a seal surrounding the shaft between the bearings and the rotor, the unit having a pressure fluid inlet leading to the rotor and a fluid outlet leading therefrom, support means for the bearings having passages communicating between the outer sides of the bearings and between the inner sides of the bearings, a lubricant enclosure, a hollow shaft extension projecting from the shaft into the enclosure, means carried by the extension for flowing lubricant from the enclosure through the extension, the shaft having a lubricant passage leading from the interior of the hollow extension to the exterior of the shaft between the bearings for supplying lubricant to the exterior of the shaft and establishing a zone of oil fog formation, and means on the shaft for producing fluid flow between the inner and outer sides of the bearing means for circulating gas streams circuitously through the bearings and through the zone of oil fog formation, the lubrication system being completely enclosed except for the shaft means seal.

8. A lubrication system as set forth in claim 7, and a lubricant reservoir, a lubricant supply conductor leading from the reservoir to the enclosure, the bearing support means enclosing a chamber with its outer portion and having a port between said chamber and its inner side, and a lubricant return conductor leading from the chamber to the reservoir.

9. A lubrication system as set forth in claim 8, wherein the port is aligned with the outlet of the shaft passage between the bearings.

10. A lubrication system as set forth in claim 7, and a lubricant reservoir, a lubricant supply conductor leading from the reservoir to the enclosure, and a lubricant return conductor leading from between the bearings to the reservoir.

11. A lubrication system as set forth in claim 8, and a lubricant filter in the supply conductor.

12. A lubrication system as set forth in claim 10, and flow-restricting means between the supply conductor and the lubricant enclosure.

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