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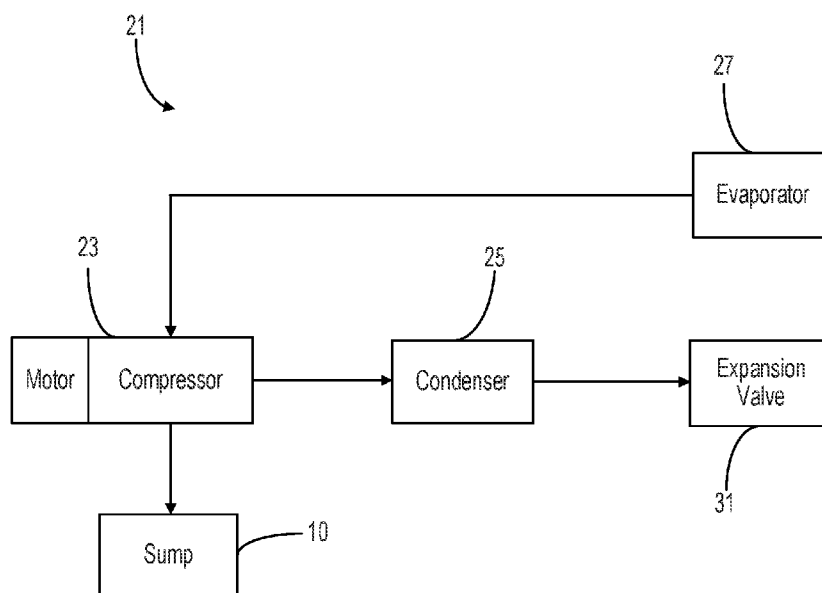


FIG. 1
Prior Art

(57) Abstract: Example implementations include a refrigeration system including a compressor having a rotor separating a low pressure side from a high pressure side, a sump having a first fluid connection to the low pressure side of the compressor and a second fluid connection to the high pressure side of the compressor, a vent pipe fluidly connecting the first fluid connection and the second fluid connection, an intermediate reservoir located within the second fluid connection between the high pressure side of the compressor and the sump, wherein the intermediate reservoir is configured to collect oil from the compressor, and a valve located within the second fluid connection between the intermediate reservoir and the sump, wherein the valve is configured to release at least a portion of the oil from the intermediate reservoir in response to a level of the oil in the intermediate reservoir exceeding a liquid level set point.

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REFRIGERATION SYSTEM WITH REDUCED COMPRESSOR THRUST LOAD

CROSS-REFERENCE TO RELATED APPLICATION(S)

- [0001] This application claims the benefit of U.S. Provisional Application Serial No. 63/414,228, entitled “REFRIGERATION SYSTEM WITH REDUCED COMPRESSOR THRUST LOAD” and filed on October 7, 2023, which is expressly incorporated by reference herein in its entirety.

BACKGROUND

- [0002] The described aspects relate to a refrigeration system, such as a chiller or a heat pump, using lubricated compressors.
- [0003] Refrigeration systems, such as chillers or heat pumps, include a compressor to compress refrigerant gas to a higher pressure, and a sump that collects lubricating oil used during the operation of the compressor. Some refrigeration systems include at least a second compressor or an auxiliary system, in communication with the compressor and the sump, which provide an intermediate pressure stage between the compressor and the sump to further depressurize the sump to achieve better oil properties for use in operation with the compressor. In such configurations, the differences in pressure on opposing sides of the impeller/rotor of the compressor can lead to increased thrust loads on the rotor supports and/or bearings, which in some operational conditions may lead to a failure condition.
- [0004] Thus, there exists a need for improvements in refrigeration systems.

SUMMARY

- [0005] The following presents a simplified summary of one or more aspects in order to provide a basic understanding of such aspects. This summary is not an extensive overview of all contemplated aspects, and is intended to neither identify key or critical elements of all aspects nor delineate the scope of any or all aspects. Its sole purpose is to present some concepts of one or more aspects in a simplified form as a prelude to the more detailed description that is presented later.
- [0006] In an aspect, a refrigeration system includes a compressor having a rotor separating a low pressure side from a high pressure side, a sump having a first fluid connection to the low pressure side of the compressor and a second fluid connection to the high pressure side of the compressor, a vent pipe fluidly connecting the first fluid connection and the second

fluid connection, an intermediate reservoir located within the second fluid connection between the high pressure side of the compressor and the sump, wherein the intermediate reservoir is configured to collect oil from the compressor, and a valve located within the second fluid connection between the intermediate reservoir and the sump, wherein the valve is configured to release at least a portion of the oil from the intermediate reservoir in response to a level of the oil in the intermediate reservoir exceeding a liquid level set point.

[0007] In another aspect, a compressor lubrication system includes a first fluid connection to between a sump and a low pressure side of a compressor, a second fluid connection between a high pressure side of the compressor and the sump, oil contained within the sump and configured to circulate, during operation of the lubrication system, from the sump via the first fluid connection to the low pressure side of the compressor, through the compressor, and from the high pressure side of the compressor via the second fluid connection to the sump, a vent pipe fluidly connecting the first fluid connection and the second fluid connection, an intermediate reservoir located within the second fluid connection between the high pressure side of the compressor and the sump, wherein the intermediate reservoir is configured to collect oil from the compressor, and a valve located within the second fluid connection between the intermediate reservoir and the sump, wherein the valve is configured to release at least a portion of the oil from the intermediate reservoir in response to a level of the oil in the intermediate reservoir exceeding a liquid level set point.

BRIEF DESCRIPTION OF THE DRAWINGS

[0008] The disclosed aspects will hereinafter be described in conjunction with the appended drawings, provided to illustrate and not to limit the disclosed aspects, wherein like designations denote like elements, wherein dashed lines may indicate optional elements, and in which:

[0009] FIG. 1 is a schematic of a prior art refrigeration system, including an oil sump.

[0010] FIG. 2 is a cross-sectional view of a compressor lubrication circuit and oil sump of a prior art refrigeration system.

[0011] FIG. 3 is a simplified schematic of the compressor lubrication circuit and oil sump of FIG. 2.

- [0012] FIG. 4 is a simplified schematic of the compressor lubrication circuit and oil sump of FIG. 2, but additionally including a pressure reducing device positioned between the sump and a compressor inlet.
- [0013] FIG. 5 is a simplified schematic of the compressor lubrication circuit and oil sump of FIG. 2, similar to FIG. 4, but additionally including a pressure reducing device located between the sump and a low pressure point of the refrigeration system.
- [0014] FIG. 6 is a simplified schematic of the compressor lubrication circuit and oil sump of FIG. 2, including an ejector pump, also referred to as a jet pump, implemented as the pressure reducing device associated with the sump.
- [0015] FIG. 7 is a simplified schematic of the compressor lubrication circuit and oil sump of FIG. 2, additionally including an auxiliary condenser implemented as the pressure reducing device associated with the sump.
- [0016] FIG. 8 is a partially cross-sectional side view of an example of an impeller/rotor, and corresponding supporting structures, of a compressor, according to some present aspects.
- [0017] FIG. 9 is a simplified schematic of the compressor lubrication circuit and oil sump of FIG. 2, similar to FIG. 5, but in which an additional valve is incorporated to control a pressure drop between a high pressure side of the compressor and the sump, according to some present aspects.
- [0018] FIG. 10 is a simplified schematic of the compressor lubrication circuit and oil sump of FIG. 2, similar to FIG. 9, but in which the gear cavity is vented through an additional oil reservoir and the additional valve, according to some present aspects.
- [0019] FIGS. 11A-11HH are sections of a composite diagram of an example heat pump.
- [0020] FIG. 12 is a simplified schematic diagram of the heat pump of FIGS. 11A-11HH, additionally including one or more valves to keep high pressure refrigerant in a condenser when the heat pump is stopped, according to some present aspects.

DETAILED DESCRIPTION

- [0021] Various aspects are now described with reference to the drawings. In the following description, for purposes of explanation, numerous specific details are set forth in order to provide a thorough understanding of one or more aspects. It may be evident, however, that such aspect(s) may be practiced without these specific details.
- [0022] The described aspects relate to a refrigeration system configured to increase a pressure in a gear cavity on one side of an impeller/rotor of a compressor, which results in lowering a thrust load on the impeller/rotor, and corresponding support components, during a

reverse thrust operation. Further, the described aspects are configured to maintain a sump connected to the compressor at a relatively lower pressure, as compared to pressure at the compressor, which allows the oil in the sump to have desired characteristics (e.g., viscosity, concentration, etc.). The described aspects include a valve, such as an expansion valve or any other valve in which internal pressure losses are controlled to produce a desired pressure drop, used to control a pressure difference between the gear cavity and the sump. Additionally, the described aspects include a venting pipe configured to vent the gear cavity to the compressor suction, along with an intermediate reservoir, between the gear cavity and the sump, that collects the oil and controllably releases the oil to the sump, via operation of the valve, based on a level of oil in the intermediate reservoir.

- [0023] In one example implementation, which should not be construed as limiting, a YORK® CYK Compound Centrifugal Chiller/Heat Pump system (“the CYK system”) may include a valve, e.g., an expansion valve, that controls a pressure difference set point between the gear cavity and the sump (typically 35°C saturated temperature in gear cavity and 20°C saturated temperature in the sump on the CYK system). Additionally, in this example implementation, the gear cavity is directly vented to compressor suction through a venting pipe, and oil from the gear cavity is collected in an intermediate reservoir equipped with a liquid level probe in communication with the valve. The valve monitors an oil level relative to a liquid level set point in the intermediate reservoir, controllably releasing the collected oil back to the sump. It should be understood that this configuration may be changed and still be within the principles of the described aspects. For instance, the gear cavity may instead be any low pressure area in the system, such as a vented evaporator, and the intermediate reservoir can be integrated into the gear cavity, or other low pressure area, rather than being a separate component.
- [0024] The described aspects allow for a reduced load on the compressor, and in particular the impeller/rotor and the corresponding support structures, during reverse thrust, which improves an efficiency and reliability of the refrigeration system.
- [0025] The various aspects discussed in detail below may be implemented in any of the systems described in FIGS. 1-12.
- [0026] Referring to FIG. 1, a typical refrigeration system 21 includes a motor/compressor 23 in fluid communication with a condenser 25 which is in fluid communication with an evaporator 27. Refrigerant gas is compressed to a higher pressure in compressor 23. The high pressure refrigerant gas, after flowing to condenser 25 is condensed to a high

pressure liquid via heat exchange, not shown. The high pressure refrigerant liquid is then sent to evaporator 27. An expansion valve 31 intermediate condenser 25 and evaporator 27 expands the high pressure refrigerant liquid to a mist, the mist being a mixture of gas and liquid at a lower temperature. In evaporator 27, the liquid refrigerant is evaporated, absorbing heat from a heat exchange fluid, as liquid refrigerant mist changes phase from liquid to gas. The cooled heat exchange fluid may be sent directly to a building environment or indirectly to an intermediate medium, such as a chiller for storage of chilled water until required. Refrigerant gas from evaporator 27, having undergone a phase change, is at a low pressure and serves as a refrigerant gas source for compressor 23. Also depicted in FIG. 1 is a sump 10, which collects the oil from operation of compressor 23 and is fundamental to proper functioning of compressor 23. Sump 10, as shown, is below compressor so that lubricating oil flows to sump 10 by gravity.

[0027] Referring to FIG. 2, a prior art centrifugal compressor 23 includes an associated sump system 10, wherein some lubricating oil is retained in an auxiliary oil reserve 32 that is intended to keep some oil supply during coast-down in the event of a power failure. Compressor 23 includes an inlet 34 which receives refrigerant gas from a low pressure source, typically an evaporator (shown in FIG. 1). The refrigerant gas is compressed by an impeller 36 before being delivered to volute 38. Lubrication is provided to lubricate shaft seal 40, main journal and thrust bearing 42, thrust collar 44, double bellows shaft seal 46, low speed gear rear bearing 48, pinion gear shaft bearing 50, thrust collar bearing 52 and low speed gear 54. Lubricant and refrigerant are in contact with one another as a small amount of pressurized refrigerant gas invariably leaks from impeller 36 into the various lubricated components described above. After lubricating the compressor components, the lubricant/refrigerant mixture drains by gravity through conduit 56 into sump 10. While settling in oil sump 10 before being re-circulated, refrigerant gas is released from the mixture in excess of the steady-state solubility, dependent upon the pressure and temperature conditions in the sump. Although the exact amount of refrigerant that may collect in sump 10 at any one instant of time is difficult to measure, it is estimated that the refrigerant that is absorbed by the oil and which should be separated in sump 10 is about 1-3% of the total flow of the compressor. To avoid an undesired oil viscosity as the oil cools once the compressor is stopped, an oil heater 57 is provided, heating or maintaining the lubricant within a predetermined temperature range so that it has the proper viscosity as soon as compressor 23 starts. Fluid is pumped from sump 10 by submersible pump 60 and sent to oil cooler 62, which is activated only when the oil is

above its predetermined operating temperature. The refrigerant gas that is separated from the oil in the sump is sent to compressor inlet 34 through a vent line 102 (see FIG. 3), while oil, which still may include miscible refrigerant gas, is sent to oil reserve 32 wherein it is metered to the compressor for lubrication purposes, after which the lubrication cycle repeats.

- [0028] In heat pump systems in which the evaporation pressure and temperature tend to be substantially higher than in water chillers, the oil temperature also should be set to a higher value in order to keep the oil dilution at an acceptable value. As a result of this higher temperature, the oil viscosity will be reduced if the same grade oil is used as in water chiller systems. An oil grade with higher viscosity can be used to compensate for the higher temperatures experienced in heat pump systems. But even with this compensation for the viscosity, the temperature elevation of the oil in such heat pump systems raises other issues. Among these is a risk of failure of the shaft seals and bearings if the oil temperature should become too high. Aspects described herein provide a system that compensates for some of the differences between operation of standard chillers and higher temperature heat pumps due to the temperature difference of operation that also affects oil temperature. These aspects should extend the range of application of current standard compressor systems used in chiller applications to heat pump applications, with minor, inexpensive modifications.
- [0029] Referring to FIG. 3, a simplified lubrication cycle schematic includes compressor 23 and sump 10, with lubricant and miscible refrigerant being drained from compressor 23 through conduit 56 to sump 10. Then, refrigerant gas at sump pressure is returned to the compressor inlet along gas conduit 102, while lubricant with miscible refrigerant is returned to compressor 23 along conduit 104.
- [0030] Referring to FIG. 4, a simplified lubrication cycle schematic includes compressor 23 and sump 10, and additionally includes a pressure reducing device 409 positioned between sump 10 and compressor inlet 34 to draw refrigerant gas from the sump while reducing the pressure of refrigerant gas in the sump. Although pressure reducing device 409 is shown as connected to the inlet of compressor 34 through connection 411, it is not so restricted, and, as will be recognized by one of skill in the art, pressure reducing device 409 can be connected to any low pressure point of the refrigeration circuit. Most often this low pressure point is the evaporator 27, but may be by any connection to the system between the evaporator 27 or an evaporator inlet and compressor inlet 34, including compressor inlet 34. Pressure reducing device 409 enables lowering of the pressure (and

temperature) of the refrigerant gas in the oil sump. As previously set forth, the lowering of the pressure of refrigerant gas in oil sump 10 has the beneficial effect of reducing the dilution of refrigerant in the oil, thereby mitigating the reduction of oil viscosity while providing proper lubrication of shaft seals and bearings. Lowering the refrigerant pressure in the oil sump initiates a “virtuous cycle” combining several combined benefits, one of which is the ability of refrigeration system 21 to operate at higher evaporation temperatures and pressures such as encountered in heat pump conditions. When operating at such heat pump conditions, the target for pressure reduction is to set the oil sump gas pressure at a value consistent with the validated range of the same compressor when operating as a water chiller. Thus, if a given type of compressor is validated, for example, for an evaporation temperature of 20° C. (68° F.) with a given refrigerant, the target will be to set the sump pressure corresponding to a 20° C. saturation temperature in heat pump operation, in order to set all the lubrication parameters at the same standard value as for chillers. Although all of the detail of the system as shown in FIG. 2 is not shown in the simplified version of FIG. 4, it will be understood that all of the detail of the system shown in FIG. 2 also may be in the simplified system of FIG. 4, except that the pressure reducing device 409 is included between sump and a low pressure point of the refrigeration system 21.

[0031] The pressure reduction in the oil sump can be achieved in different ways.

[0032] Referring to FIG. 5, for example, a simplified lubrication cycle schematic includes compressor 23 and sump 10 and additionally includes a pressure reducing device 509 located between sump and a low pressure point of the refrigeration system 21. Although all of the detail of the system as shown in FIG. 2 is not shown in the simplified version of FIG. 5, it will be understood that all of the detail of the system shown in FIG. 2 also may be in the simplified system of FIG. 5, except that a pressure reducing device 509 is included between sump and a low pressure point of the refrigeration system 21. In an aspect, the pressure reducing device 509 is a small additional “auxiliary” compressor 509 positioned between sump 10 and compressor inlet 34 to draw refrigerant gas from sump 10 while reducing the pressure of refrigerant gas in the sump. Auxiliary compressor 509 has its suction side connected to the gas volume of oil sump 10 and its discharge side connected, for example, to compressor inlet 34 of main compressor 23. In this implementation, the capacity of auxiliary compressor 509 is controlled in such a way that it keeps the refrigerant pressure in oil sump 10 at a pre-selected value as described above (e.g. corresponding to the saturated pressure of the refrigerant fluid at 20° C. in the above

example). As discussed above, the discharge of auxiliary compressor 509 can also be connected to any lower pressure point in refrigeration system 21, such as evaporator 27 or any point between evaporator 27 and compressor inlet 34 as shown in FIG. 1.

[0033] Referring to FIG. 6, in another example, a simplified lubrication cycle schematic includes compressor 23 and sump 10 and additionally includes an ejector pump 609, also referred to as a jet pump, implemented as the pressure reducing device associated with sump 10. Again, all of the detail of the system as shown in FIG. 2 is not shown in the simplified version of FIG. 6, and it will be understood that all of the detail of the system shown in FIG. 2 also may be in the simplified system of FIG. 6, except that ejector pump 609 is positioned between sump 10 and a low pressure point of the refrigeration system. In FIG. 6, high pressure gas from conduit 615, which is in fluid communication with condenser 25, after passing through an expansion valve (not shown), if required, is used to provide the energy to operate ejector pump 609. At the ejector outlet, the mixture of this high pressure refrigerant fluid from condenser 25 and the low pressure gas pumped from oil sump 10 is sent to a low pressure point in the refrigeration system, preferably the evaporator. Although shown in FIG. 6 as in direct fluid communication with compressor inlet 34 via conduit 611 (for consistency with FIGS. 4 and 5), the low pressure point may be at any intermediate location between compressor 23 and evaporator 27 that is at a low pressure, as previously discussed. An advantage of this implementation, using an ejector pump, is that it avoids moving parts such as found with the use of auxiliary compressor 509 of FIG. 5.

[0034] Referring to FIG. 7, in another example, a simplified lubrication cycle schematic includes compressor 23 and sump 10 and additionally includes an auxiliary condenser 709 implemented as the pressure reducing device associated with sump 10. Again, all of the detail of the system as shown in FIG. 2 is not shown in the simplified version of FIG. 7, and it will be understood that all of the detail of the system shown in FIG. 2 also may be in the simplified system of FIG. 7, except that auxiliary condenser 709 is included between sump 10 and a low pressure point of the refrigeration system. In FIG. 7, refrigerant gas from sump 10 is in fluid communication with auxiliary condenser 709 via conduit 713. Gas from sump 10 enters auxiliary condenser 709 where it is in heat exchange relationship with a cooling fluid flowing through cooling circuit 715. Cooling fluid in cooling circuit 715 cools the refrigerant gas, condensing it from a gas to a liquid, the liquid refrigerant being sent to liquid storage space 717 via conduit 730.

[0035] The auxiliary condenser 709 is selected to provide a condensing pressure equal to the desired refrigerant pressure in oil sump 10. In an aspect, the refrigerant gas in auxiliary condenser 709 is cooled by a cooling fluid at a temperature lower than the cold source of the heat pump. For example, if the desired condensing pressure in the auxiliary condenser 709 corresponds to a 20° C. (68° F.) saturation temperature, auxiliary condenser 709 preferably is cooled with water having an entering temperature of about 12° C. (about 54° F.) and a leaving temperature of about 18° C. (about 64° F.). The cooling water may be provided from any available chilled water source as well as from ground water within the desired temperature range. The condensing pressure in auxiliary condenser 709 may be controlled by varying the flow and/or temperature of the cooling fluid through cooling circuit 715 of auxiliary condenser 709 to maintain the desired gas pressure in oil sump 10. As depicted in FIG. 7, liquid storage space 717 for condensed refrigerant may be a separate vessel as shown, or may be a separate storage space integral to auxiliary condenser 709.

[0036] Per the principle of the system, liquid storage space 717 is at a lower pressure than the compressor inlet and the evaporator in the main refrigerant circuit. To avoid accumulation of liquid refrigerant in liquid storage space 717, refrigerant is pumped from storage space 717 back to refrigerant system 21 by pump 719 that is controlled by liquid level sensor 721. This pump 719 has its suction side connected to fluid storage space 717 and its discharge side in fluid communication with refrigerant system 21. To reduce the head and the absorbed power of the pump, it is preferred to set the pump discharge to a low pressure portion of the main refrigerant circuit 21. While this low pressure region may be compressor inlet 34, as previously discussed with regard to FIGS. 3-6, FIG. 7 depicts the low pressure region as the conduit between expansion valve 31 and evaporator 27, although refrigerant may be sent to the low pressure region at any convenient point, such as between expansion valve 31 and compressor suction 34. It is also normally desired to avoid sending refrigerant liquid directly into compressor suction 34 (inlet) from liquid storage space 717 to avoid liquid flooding of compressor 23. Therefore, a location along the conduit between expansion valve 31 and evaporator 27 is a desirable and preferred refrigerant input, as is supplying this liquid refrigerant to evaporator 27, such as at the liquid inlet of evaporator 27. More specifically, if evaporator 27 is of the dry-expansion technology (either shell and tube or plate heat exchanger), then it is desirable to discharge the liquid refrigerant into the main liquid line at the evaporator inlet. If evaporator 27 is of the flooded type, falling film or hybrid falling film, an alternative is to discharge the

liquid directly in the evaporator shell, at a location away from the suction pipe to avoid liquid carry-over to compressor inlet 34.

[0037] In an aspect, the system includes a mechanism to control the operation of liquid pump 719, wherein an example of such a mechanism is depicted in FIG. 7 as liquid level sensor 721. A desired arrangement is to have fluid storage space 717 located at the outlet of auxiliary condenser 709, allowing liquid refrigerant to flow by gravity from auxiliary condenser 709 into storage space 717. This volume can either be included in the same shell as the auxiliary condenser 709, or as a separate vessel. The liquid level in this storage space is sensed by a liquid level sensor which includes a control loop, depicted simply as liquid level sensor 721. This control loop portion of liquid level sensor 721 manages the operation of liquid pump 719 in order to keep the liquid level in the fluid storage space 717 within predetermined, pre-set acceptable limits. Liquid pump 719 can either have a variable speed drive, with the speed being controlled by the control loop of liquid level sensor 721, or it may simply have an ON/OFF operation sequence, also under control of the same control loop.

[0038] In another implementation, a conventional mechanical pump 719 may be replaced by a purely static pumping system. In a variation to this aspect, the static pumping system may utilize an ejector pump 609 powered by high pressure gas from main condenser 25. A mixture of pumped liquid from fluid storage space 717 and of high pressure gas from main condenser 25 is returned to evaporator 27. In still another variation to this aspect, two fluid storage vessels 717 may be located below auxiliary condenser 715, each having an inlet (A) connected to the discharge port of auxiliary condenser 709 to receive condensed refrigerant liquid, an inlet (B) connected to receive gas from evaporator or main condenser 25, and each having outlet (C) connected to evaporator 27. Each of these connections has an automatic valve that can be opened or closed. The system is operated in “batches,” being activated by a control circuit.

[0039] Any of these described aspects enable removal of refrigerant from oil in a lubricated compressor, and the present disclosure is not limited to use with a centrifugal compressor. These aspects may also find use with reciprocating compressors, scroll compressors and turbines as used in ORC systems, each of which requires lubrication. An auxiliary compressor 509 or ejector pump 609 may advantageously be used to remove refrigerant from oil in these units, as described above. An auxiliary condenser 709 has the further advantage of not requiring power to operate, assuming that water at the desired temperature is available, although it is implemented with liquid pump 719.

[0040] The basic pressure reducing devices described above with reference to FIGS. 4-7 to separate refrigerant from lubrication systems may also be adapted for use in refrigeration circuits to extend the operational limits of refrigerant fluid for cooling semi-hermetic motors. These pressure reducing devices 409 can advantageously be utilized in heat pump systems which typically operate at higher temperatures than chiller systems. These pressure reducing devices 409 extend the motor cooling capability of the refrigerant, permitting the use of chiller system equipment for heat pump applications. In these systems, refrigerant is utilized to cool the motor and the motor cavity from heat generated by operation of the motor. The pressure in the motor housing and in the coil surrounding the motor stator without such pressure reducing devices is nearly equal to or slightly higher than the pressure in the evaporator. But, pressure reducing devices are controlled to maintain the pressure in the motor cavity at a preset value below that of the compressor inlet and preferably lower than that of the evaporator so that refrigerant gas can be drawn through the housing. For a system operating in heat pump applications, it is desired to maintain the pressure in the motor cavity at a preset value below the pressure at the compressor inlet, for example, at a saturation temperature of 20° C corresponding to the desired pressure for a given refrigerant. These values typically correspond to the temperatures at which the compressor is validated when the system operates as a water chiller system.

[0041] Referring to Fig. 8, in operation of any of the above examples described in FIGS. 1 through 7, loading on impeller/rotor of the compressor 23, and corresponding supporting structures, may increase up to or beyond a failure condition, such as during a reverse thrust operation. For instance, in a refrigeration system with a single compressor, the sump pressure is close to the suction pressure of the compressor and the discharge pressure is generally much higher. The forward and reverse thrusts are designed for safe operation at design conditions. The described aspects can reduce the reverse thrust load when a refrigeration system, such as a YORK CYK-R134a system, is used at a highest pressure, where the reverse thrust margin is not so high. Also, on a heat pump using low pressure refrigerant (e.g., R1234ze, R1233zd), allowing the use of standard YK compressor (these compressors having generally lower reverse thrust maximum load, compared to High Pressure compressors) at HP stage of a CYK. Further, for example, in refrigeration/heat pump systems having auxiliary systems, there is an increase of the load on High Speed reverse thrust. For example, such increased thrust load may occur when pressure is lowered by compressor 509 (or by eductor 609, or by Heat exchanger 709 in

other aspects), as the force applied by gear internal pressure on the rotor support area) is decreased, the load on reverse thrust is then increased. At some extreme conditions, the maximum available load on the reverse thrust can be exceeded and the system can no longer operate.

[0042] Referring to FIG. 9, a simplified lubrication cycle schematic includes compressor 23 and sump 10 additionally includes a pressure reducing device 509 located between sump and a low pressure point of the refrigeration system 21 and a valve 902 to control a pressure drop between the gear box or high pressure side of the compressor 23 and the sump 10 to reduce thrust loading on the compressor 23. Although all of the detail of the system as shown in FIG. 2 is not shown in the simplified version of FIG. 9, it will be understood that all of the detail of the system shown in FIG. 2 also may be in the simplified system of FIG. 9, except that a pressure reducing device 509 is included between sump and a low pressure point of the refrigeration system 21 and that valve 902, such as an expansion valve, is included between the gear box of the compressor 23 and the sump 10. In FIG. 9, valve 902, such as an expansion valve or any valve in which internal pressure losses are controlled to produce a desired pressure drop, can control a pressure difference set point between a gear cavity on a high pressure side of the compressor 23 and sump 10. For example, typically there may be a 35°C saturated temperature in the gear cavity and 20°C saturated temperature in the sump, in a YK-R1234ze or CYK-R1234ze heat pump system. The valve 902 can be controlled to allow an acceptable load on reverse thrust.

Example 1

[0043] The above aspect was tested in an exemplary CYK auxiliary system for a R1234ze model.

[0044] The table below indicates thrust load calculated from this test, comparing the effect of the expansion valve on thrust load.

	100% 90°C Without Expansion valve	100% 90°C With Expansion valve	25% 90°C Without Expansion valve	25% 90°C With Expansion valve
Refrigerant	R1234ze	R1234ze	R1234ze	R1234ze
Tsat discharge (°C)	60	60	65	65
Tsat suction (°C)	35	35	50	50
Tsat gear (°C)	18	35	18	35
Tsat sump (°C)	18	18	18	18
F discharge (kg)	1483.1	1483.1	1810.3	1810.3
F rotor support (kg)	1043.2	1735.7	1043.2	1735.7
F pinion thrust (kg)	-815.7	-815.7	-254.9	-254.9
F suction (kg)	-2694.0	-2594.0	-4025.3	-4025.3
F oil (kg)	144.3	144.3	144.3	144.3
Total load (kg)	-839	-147	-1282	-590

Results From Example 1

- At 100% load:
 - o Without the expansion valve 902, the saturated temperature in the gear cavity is 18°C and the total load on thrust is 839kg.
 - o With the expansion valve 902, the saturated temperature in the gear cavity is increased to 35°C and the total load on thrust is 147kg.
- At 25% load:
 - o Without the expansion valve 902, the saturated temperature in the gear cavity is 18°C and the total load on thrust is 1282kg.
 - o With the expansion valve 902, the saturated temperature in the gear cavity is increased to 35°C and the total load on thrust is 590kg.
- In some modes of operation, using the CYK System without the expansion valve 902 may result in a thrust failure, as thrust load should be kept below 600kg for reliable operation. Consequently, use of the expansion valve 902 helps to reduce the thrust load and improve operation of the system.

[0045] Referring to FIG. 10, in another aspect, a simplified lubrication cycle schematic similar to FIGS. 5 to 7 includes the addition of expansion valve 902 and venting pipe 1005 through an intermediate reservoir 1010, which combine to operate to reduce thrust load on the compressor 23. The addition of valve 902, such as an expansion valve, and venting pipe 1005 through intermediate reservoir 1010, which acts as an additional oil receiver, can similarly control a pressure difference set point between the gear cavity on the high

pressure side of compressor 23 and the sump 10, allowing for an acceptable load on reverse thrust.

[0046] In this aspect, the gear cavity on the high pressure side of compressor 23 is directly vented to compression suction through venting pipe 1005. Venting pipe 1005 is connected to intermediate reservoir 1010, which collects oil from the gear cavity. Further, a liquid level set point 1015 is maintained in intermediate reservoir 1010 by valve 902. For instance, a sensor, such as a liquid level probe, may monitor an oil level relative to the liquid level set point 1015 and provide a signal to a control circuit that operates to open valve 902 when the oil level exceeds the liquid level set point 1015. Consequently, oil in excess of the liquid level set point 1015 in intermediate reservoir 1010 is sent back to sump 10. Sump 10 is stabilized in pressure by compressor 509, wherein the configuration allows compressed refrigerant gas to pass through venting pipe 511 to stabilize the pressure difference between gear cavity on the high pressure side of compressor 23 and sump 10.

[0047] In other words, the system of FIG. 10 may be implemented in a refrigeration system, such as described in FIG. 1, and may include a compressor 23 having a rotor separating a low pressure side from a high pressure side, a sump 10 having a first fluid connection to the low pressure side of the compressor 23 and a second fluid connection to the high pressure side of the compressor 23, a vent pipe 1005 fluidly connecting the first fluid connection and the second fluid connection, an intermediate reservoir 1010 located within the second fluid connection between the high pressure side of the compressor 23 and the sump 10, wherein the intermediate reservoir 1010 is configured to collect oil from the compressor 23, and a valve 902 located within the second fluid connection between the intermediate reservoir 1010 and the sump 10, wherein the valve 902 is configured to release at least a portion of the oil from the intermediate reservoir 1010 in response to a level of the oil in the intermediate reservoir 1010 exceeding a liquid level set point, and a compressor 509, allowing to reduce the pressure in the sump by sending refrigerant gas mass back to compressor 34.

[0048] In another implementation, the system of FIG. 10 may be implemented in a compressor lubrication system that includes a first fluid connection to between a sump and a low pressure side of a compressor, a second fluid connection between a high pressure side of the compressor and the sump, oil contained within the sump and configured to circulate, during operation of the lubrication system, from the sump to the compressor due to an oil pump (not represented), through the compressor, and from the high pressure side of the

compressor via the second fluid connection to the sump, a vent pipe fluidly connecting the first fluid connection and the second fluid connection, an intermediate reservoir located within the second fluid connection between the high pressure side of the compressor and the sump, wherein the intermediate reservoir is configured to collect oil from the compressor, and a valve located within the second fluid connection between the intermediate reservoir and the sump, wherein the valve is configured to release at least a portion of the oil from the intermediate reservoir in response to a level of the oil in the intermediate reservoir exceeding a liquid level set point.

- [0049] In some implementations, the vent pipe 511 can be connected to evaporator, or any low pressure area in a refrigeration or heat pump unit.
- [0050] Additionally, in some implementations, intermediate reservoir 1010 can be integrated into the gear cavity rather than being a separate component.
- [0051] Moreover, it should be understood that the aspects of FIG. 10, although described with reference to FIGS. 5 and 9, may be similarly applied to the aspects of FIGS. 6 and 7, and/or to a multi-stage compressor system.
- [0052] Referring to FIGS. 11A-11HH and 12, in another aspect, a heat pump 1100 is configured to allow high pressure refrigerant to be maintained in condenser 1120, even when the heat pump is stopped.
- [0053] In one example, heat pump 1100 includes a first compressor 1102 and a second compressor 1105, wherein the first compressor 1102 represents a relatively high pressure stage and the second compressor 1105 represents a relatively low pressure stage. Each compressor has a pressure regulation system with a sump (or oil reservoir), such as sump 1110 for compressor 1102, and sump 1115 for compressor 1105. Compressor 1102 outputs into condenser 1120, which in turn outputs into economizer 1130. Evaporator 1125 outputs into compressor 1105. The economizer 1130 also has outputs to both compressor 1102 and evaporator 1125.
- [0054] In the aspect depicted in FIG. 12, the example heat pump 1100 (of FIGS. 11A-11HH) has been modified with a check valve 1240, at the discharge of compressor 1102, and a fast action valve 1235, and a fast action valve 1199 at the outlet of the condenser 1120.
- [0055] The addition of these valves 1235, 1240, 1199 allows the refrigerant to be isolated in the condenser 1120 when the heat pump 1100 is stopped (including during a normal stop, or a security stop). After the stop, the heat (e.g. the high pressure/temperature refrigerant) in the condenser 1120 is then kept in the condenser 1120, and the pressure in the

evaporator 1125 is then lowered. This configuration makes the risk of relief valve discharge substantially reduced.

[0056] In some aspects, a solenoid valve may be on a primary side of an ejector, to further protect against relief valve discharge.

[0057] For example, the aspects of FIGS. 11 A-11HH and 12 may be implemented in a YORK YK or CYK heat pump system. In the CYK system, the first stage compressor is in general a standard air conditioning compressor having a Design Working Pressure (DWP) of 235psig and the second stage compressor is a high pressure compressor having a DWP that can be 415psig. The intermediate pressure at nominal conditions is generally chosen so that each compressor has a similar compression ratio; this allows the system to maximize the benefit of the economizer on the thermodynamic cycle. When the heat pump is stopped, refrigerant in the high pressure/temperature parts of the heat pump (mainly localized at the condenser) is mixed with the refrigerant in the lower pressure parts of the heat pump (economizer, evaporator, etc.) and the pressure then equalizes inside the heat pump. At some extreme conditions before a stop, this equalized pressure could be higher than DWP of the 1st stage compressor. In this case, the relief valves would open in order to protect some of the heat pump components. Recently, the need for such a heat pump to operate at higher evaporation temperature has increased. In the case of geothermal heat source applications, for example, evaporation temperature can be about 20°C at nominal conditions and 60°C at part load. Considering a condenser outlet of 90°C (for district heating rejection for example), the pressure could equalize at a saturation temperature close to 75°C. In these conditions, the risk of relief valve opening is high. Consequently, the CYK system may be modified to include check valve 1240 at the discharge of compressor 1105 and fast action valve 1235 and 1199 at the outlet of the condenser 1120, which allows the system to keep the high pressure refrigerant in the condenser when the heat pump is stopped.

[0058] Additionally, it should be understood that the aspects described above with respect to FIG. 9 or 10 for reducing thrust loading on a compressor may be implemented in the heat pump systems described in FIGS. 11A-11HH and 12.

[0059] Additional aspects of the disclosure are described in the clauses that follow:

[0060] Clause 1. A compressor lubrication system, comprising: a first fluid connection to between a sump and a low pressure side of a compressor; a second fluid connection between a high pressure side of the compressor and the sump, wherein the lubrication system is configured to circulate oil contained within the sump during operation, from the

sump, through the compressor, and from the high pressure side of the compressor via the second fluid connection to the sump; a vent pipe fluidly connecting the first fluid connection and the second fluid connection; an intermediate reservoir located within the second fluid connection between the high pressure side of the compressor and the sump, wherein the intermediate reservoir is configured to collect oil from the compressor; and a valve located within the second fluid connection between the intermediate reservoir and the sump, wherein the valve is configured to release at least a portion of the oil from the intermediate reservoir.

- [0061] Clause 2. The compressor lubrication system of clause 1, wherein the valve releases the at least a portion of oil from the intermediate reservoir in response to a level of the oil in the intermediate reservoir exceeding a liquid level set point.
- [0062] Clause 3. The compressor lubrication system of any of the previous clauses, further comprising a sensor for monitoring an oil level relative to the liquid level set point and provide a signal to a control circuit that operates to open the valve to release the at least a portion of the oil when the oil level exceeds the liquid level set point.
- [0063] Clause 4. The compressor lubrication system of the previous clauses, wherein the sensor is a liquid level probe.
- [0064] Clause 5. The compressor lubrication system of any of the previous clauses, further comprising an oil pump for circulating oil from the sump to the compressor.
- [0065] Clause 6. The compressor lubrication system of any of the previous clauses, wherein the vent pipe is in fluid communication with an evaporator of a refrigeration system.
- [0066] Clause 7. The compressor lubrication system of any of the previous clauses, wherein the intermediate reservoir is integrated into a gear cavity of a compressor.
- [0067] Clause 8. The compressor lubrication system of any of the previous clauses, further comprising the compressor and the sump so as to define a refrigeration system.
- [0068] Clause 9. The compressor lubrication system of any of the previous clauses, wherein the refrigeration system is a heat-pump refrigeration system.
- [0069] Clause 10. The compressor lubrication system of any of the previous clauses, wherein the refrigeration system further comprises an economizer.
- [0070] Clause 11. The compressor lubrication system of any of the previous clauses, wherein the compressor comprises a first stage compressor with the first fluid connection and the second fluid connection connected thereto and a second stage compressor.

- [0071] Clause 12. The compressor lubrication system of any of the previous clauses, further comprising one or more isolation valves for isolating refrigerant when the heat pump is stopped.
- [0072] Clause 13. The compressor lubrication system of any of the the previous clauses, wherein the one or more isolation valves are fast action valves.
- [0073] Clause 14. The compressor lubrication system of any of the previous clauses, further comprising a condenser, wherein the system is configured to allow high pressure refrigerant to be maintained in the condenser when the heat pump is stopped.
- [0074] Clause 15. The compressor lubrication system of any of the previous clauses, wherein the compressor comprises a first stage compressor and a second stage compressor, wherein a first compressor lubrication system according to any of the previous clauses provides lubricant to the first stage, and a second compressor lubrication system provides lubricant to the second stage.
- [0075] While the foregoing disclosure discusses illustrative aspects and/or embodiments, it should be noted that various changes and modifications could be made herein without departing from the scope of the described aspects and/or embodiments as defined by the appended claims. Furthermore, although elements of the described aspects and/or embodiments may be described or claimed in the singular, the plural is contemplated unless limitation to the singular is explicitly stated. Additionally, all or a portion of any aspect and/or embodiment may be utilized with all or a portion of any other aspect and/or embodiment, unless stated otherwise.

CLAIMS

What is claimed is:

1. A compressor lubrication system, comprising:
 - a first fluid connection to between a sump and a low pressure side of a compressor;
 - a second fluid connection between a high pressure side of the compressor and the sump, wherein the lubrication system is configured to circulate oil contained within the sump during operation, from the sump, through the compressor, and from the high pressure side of the compressor via the second fluid connection to the sump;
 - a vent pipe fluidly connecting the first fluid connection and the second fluid connection;
 - an intermediate reservoir located within the second fluid connection between the high pressure side of the compressor and the sump, wherein the intermediate reservoir is configured to collect oil from the compressor; and
 - a valve located within the second fluid connection between the intermediate reservoir and the sump, wherein the valve is configured to release at least a portion of the oil from the intermediate reservoir.
2. The compressor lubrication system of claim 1, wherein the valve releases the at least a portion of oil from the intermediate reservoir in response to a level of the oil in the intermediate reservoir exceeding a liquid level set point.
3. The compressor lubrication system of claim 2, further comprising a sensor for monitoring an oil level relative to the liquid level set point and provide a signal to a control circuit that operates to open the valve to release the at least a portion of the oil when the oil level exceeds the liquid level set point.
4. The compressor lubrication system of claim 3, wherein the sensor is a liquid level probe.
5. The compressor lubrication system of any of the previous claims, further comprising an oil pump for circulating oil from the sump to the compressor.

6. The compressor lubrication system of any of the previous claims, wherein the vent pipe is in fluid communication with an evaporator of a refrigeration system.

7. The compressor lubrication system of any of the previous claims, wherein the intermediate reservoir is integrated into a gear cavity of a compressor.

8. The compressor lubrication system of any of the previous claims, further comprising the compressor and the sump so as to define a refrigeration system.

9. The compressor lubrication system of claim 8, wherein the refrigeration system is a heat-pump refrigeration system.

10. The compressor lubrication system of claim 8 or 9, wherein the refrigeration system further comprises an economizer.

11. The compressor lubrication system of claim 9 or 10, wherein the compressor comprises a first stage compressor with the first fluid connection and the second fluid connection connected thereto and a second stage compressor.

12. The compressor lubrication system of claims 9-11, further comprising one or more isolation valves for isolating refrigerant when the heat pump is stopped.

13. The compressor lubrication system of claims 9-12, wherein the one or more isolation valves are fast action valves.

14. The compressor lubrication system of any of claim 9-13, further comprising a condenser, wherein the system is configured to allow high pressure refrigerant to be maintained in the condenser when the heat pump is stopped.

15. The compressor lubrication system of claim 8, wherein the compressor comprises a first stage compressor and a second stage compressor, wherein a first compressor lubrication system according to any of claims 1-7 provides lubricant to the first stage, and a second compressor lubrication system provides lubricant to the second stage.

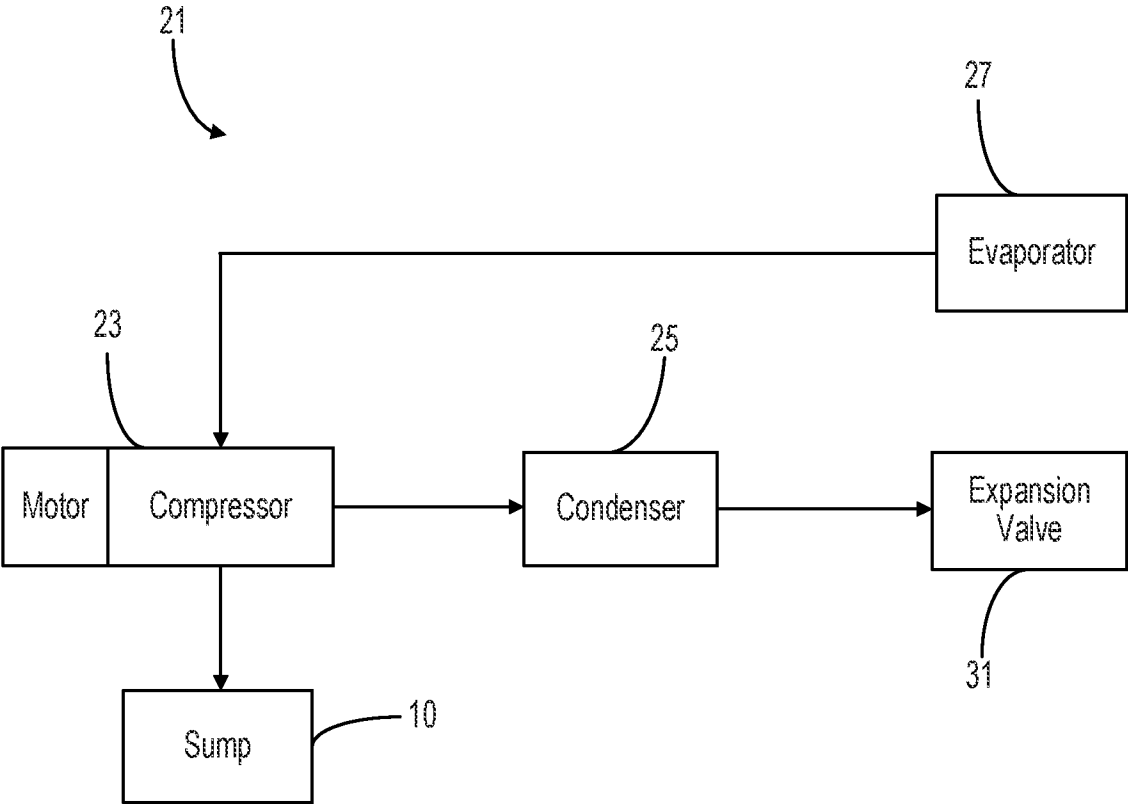


FIG. 1
Prior Art

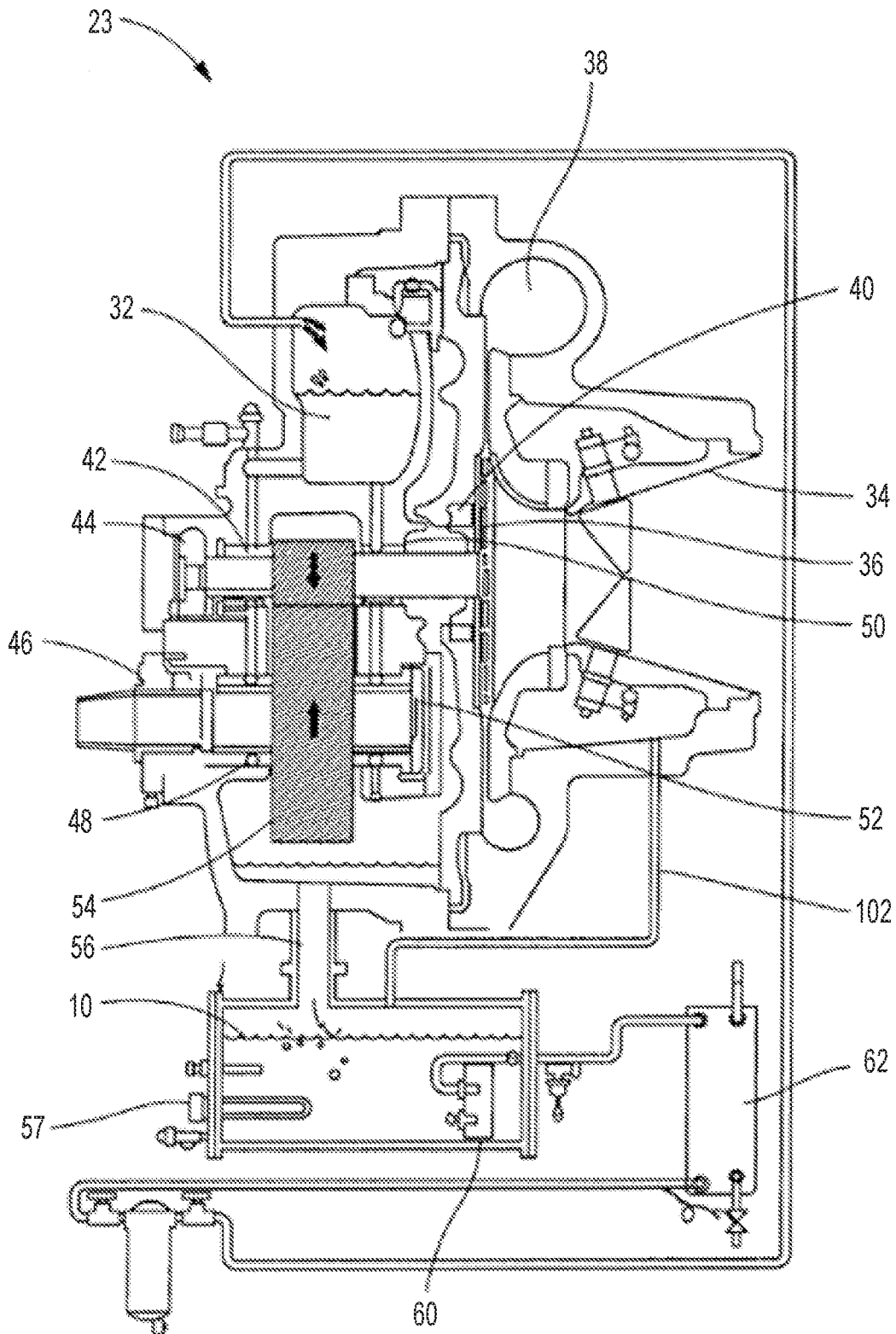
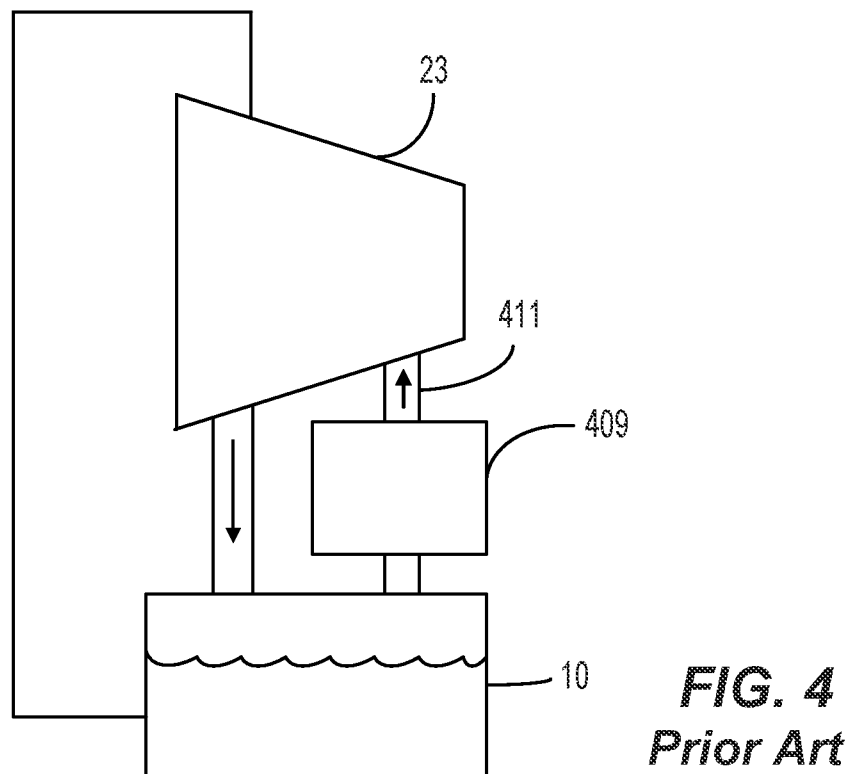
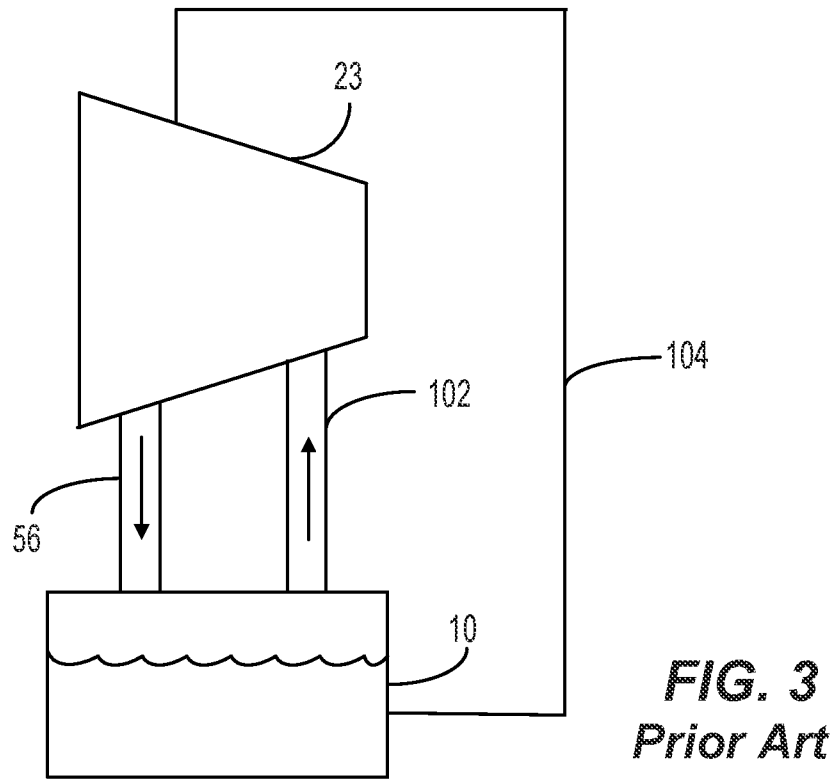


FIG. 2
Prior Art



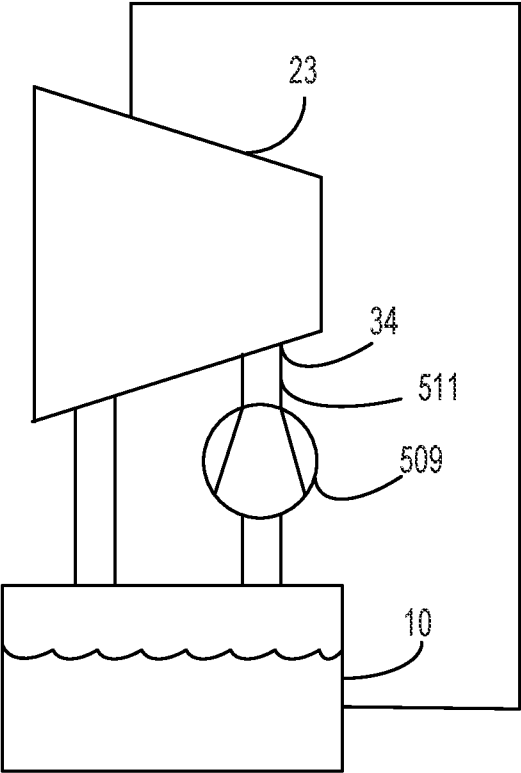


FIG. 5
Prior Art

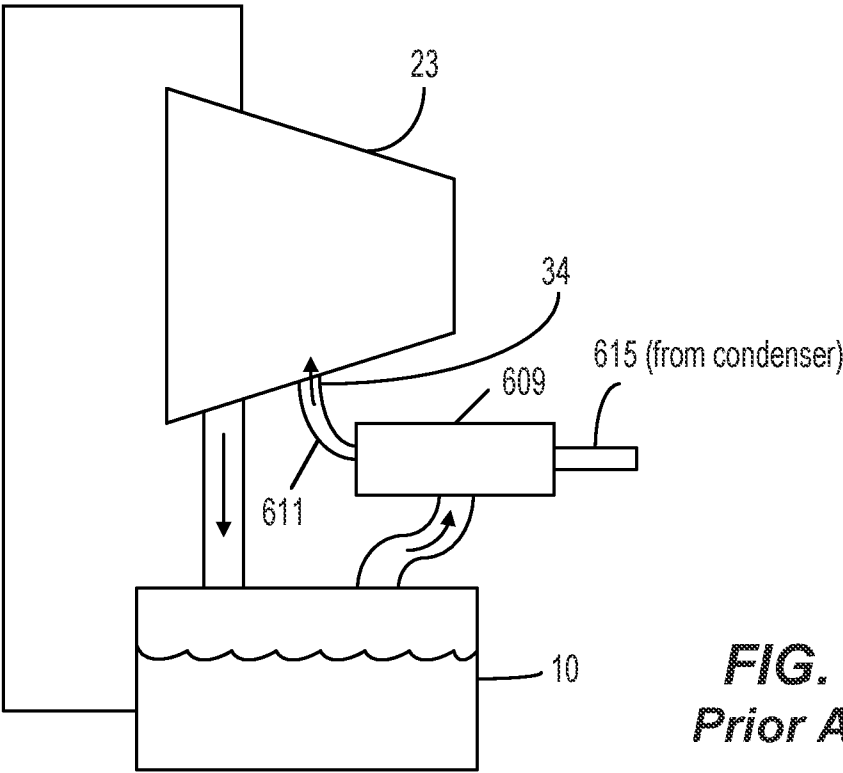


FIG. 6
Prior Art

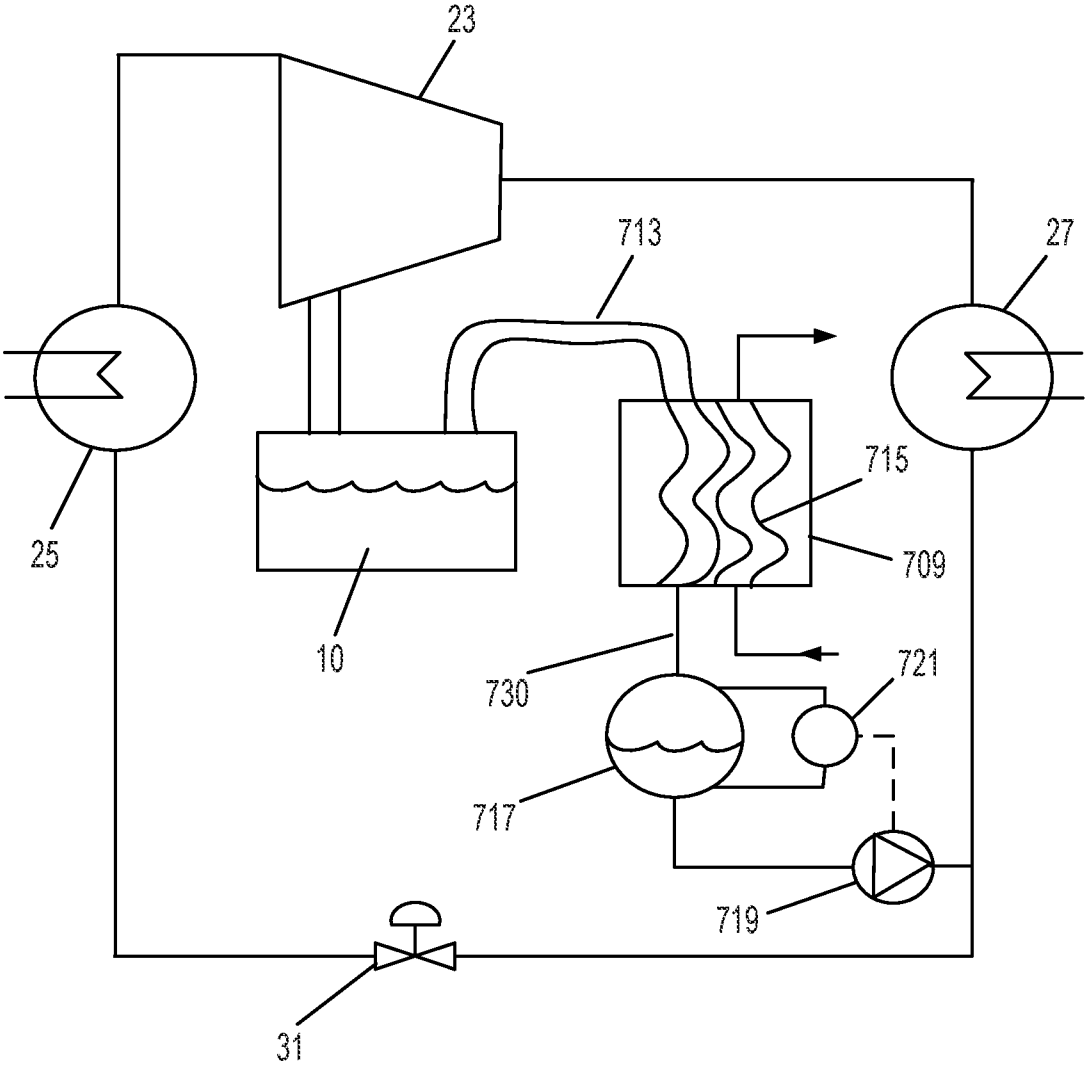
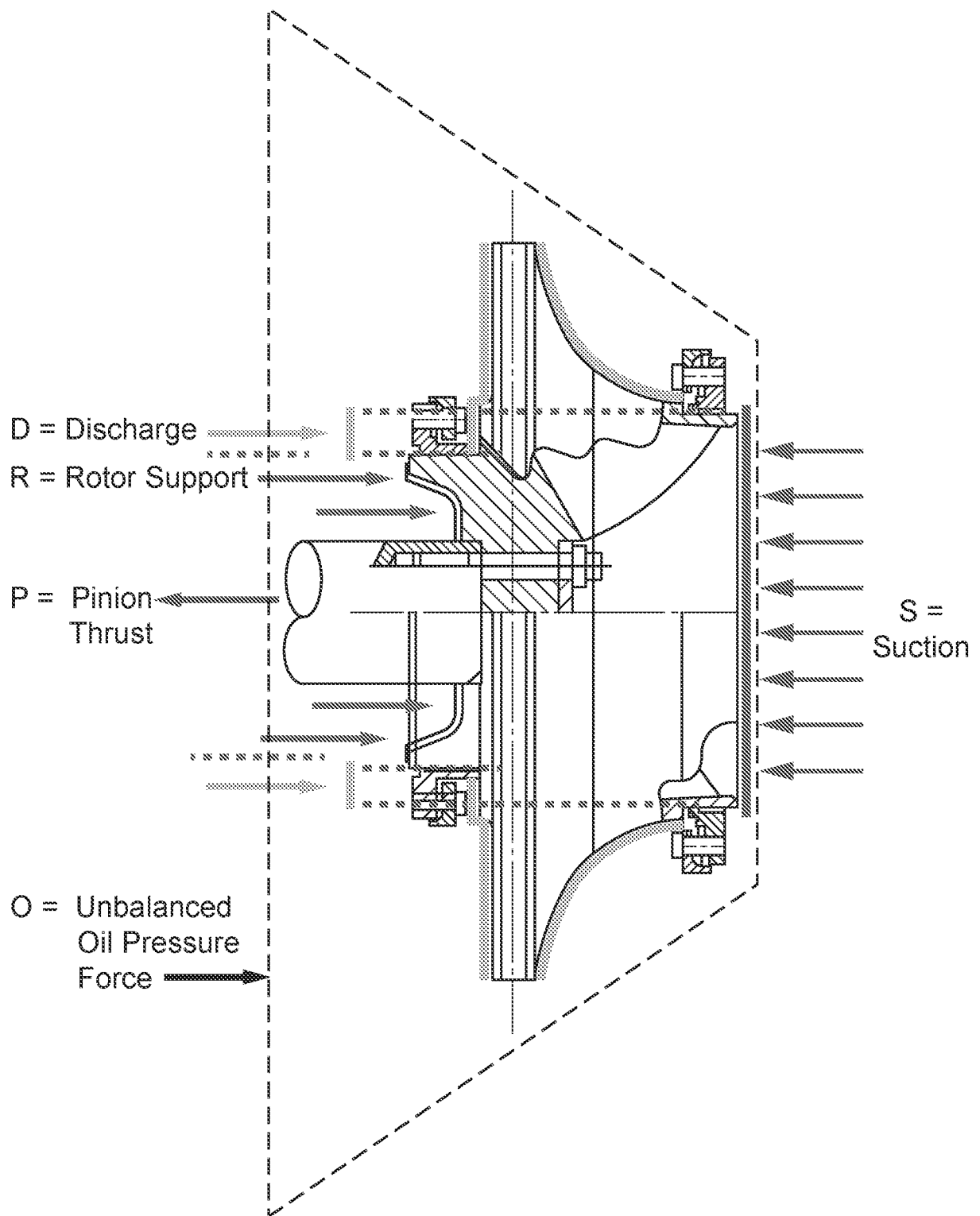


FIG. 7
Prior Art

**FIG. 8**

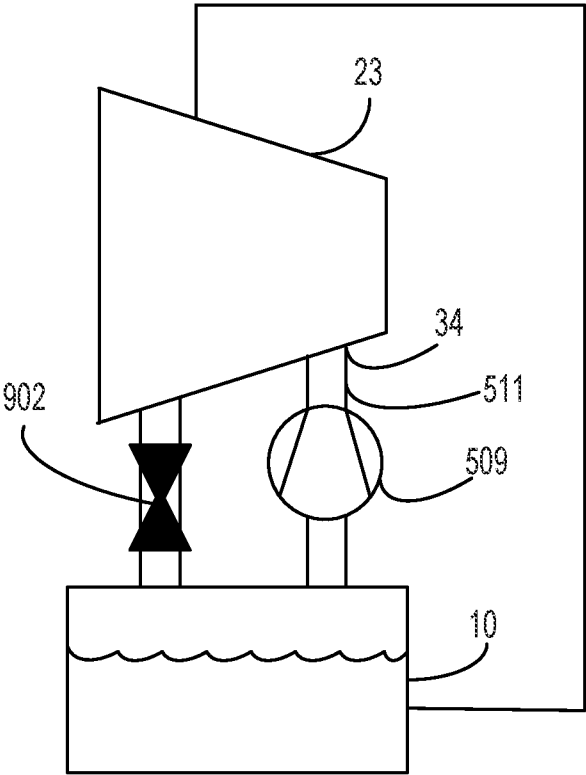


FIG. 9

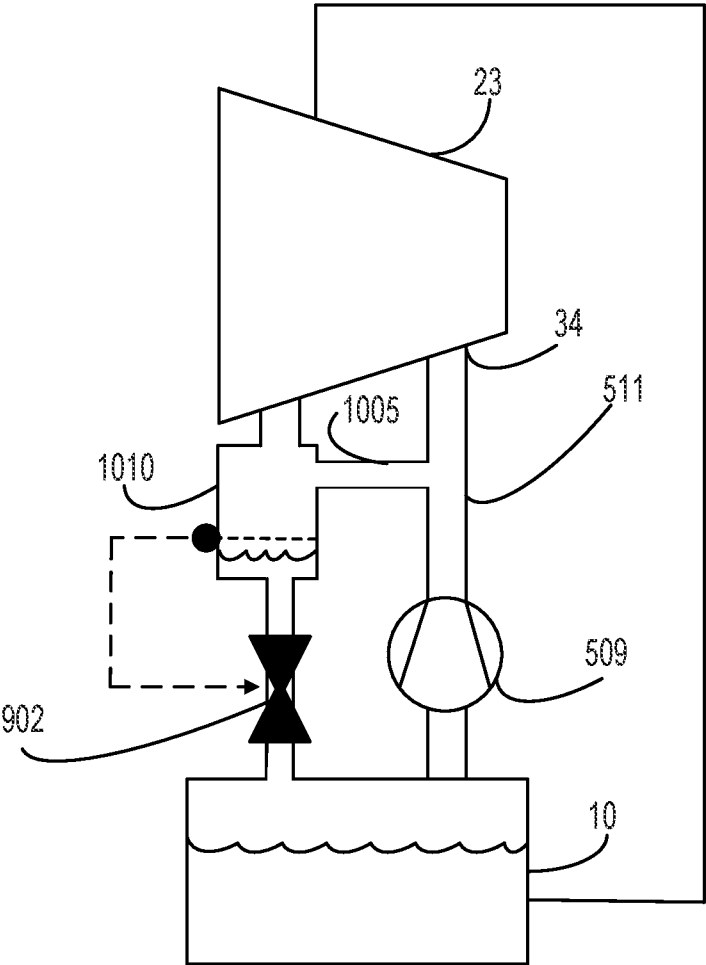


FIG. 10

FIG. 11A

TO FIG. 11B ➡

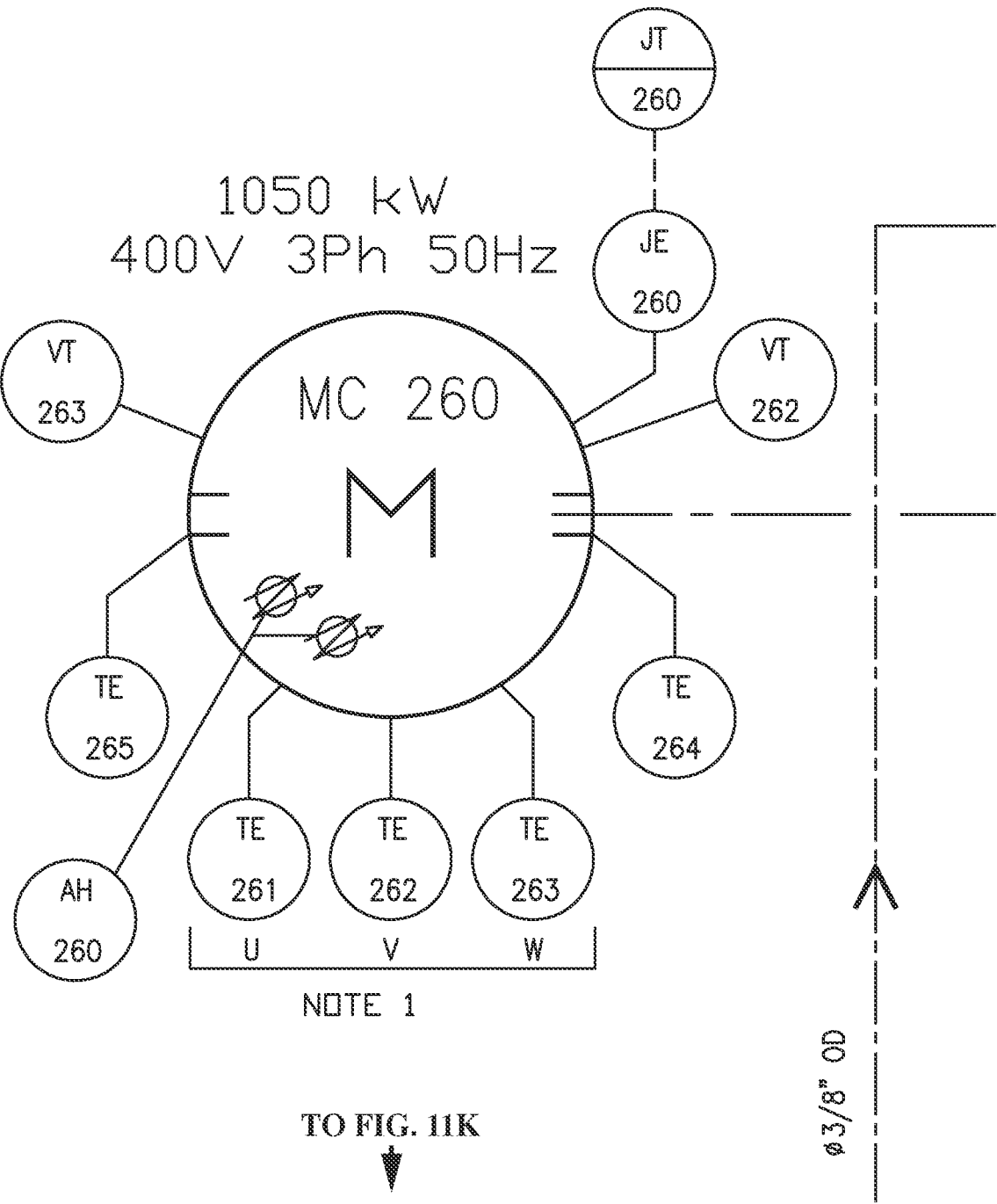


FIG. 11B

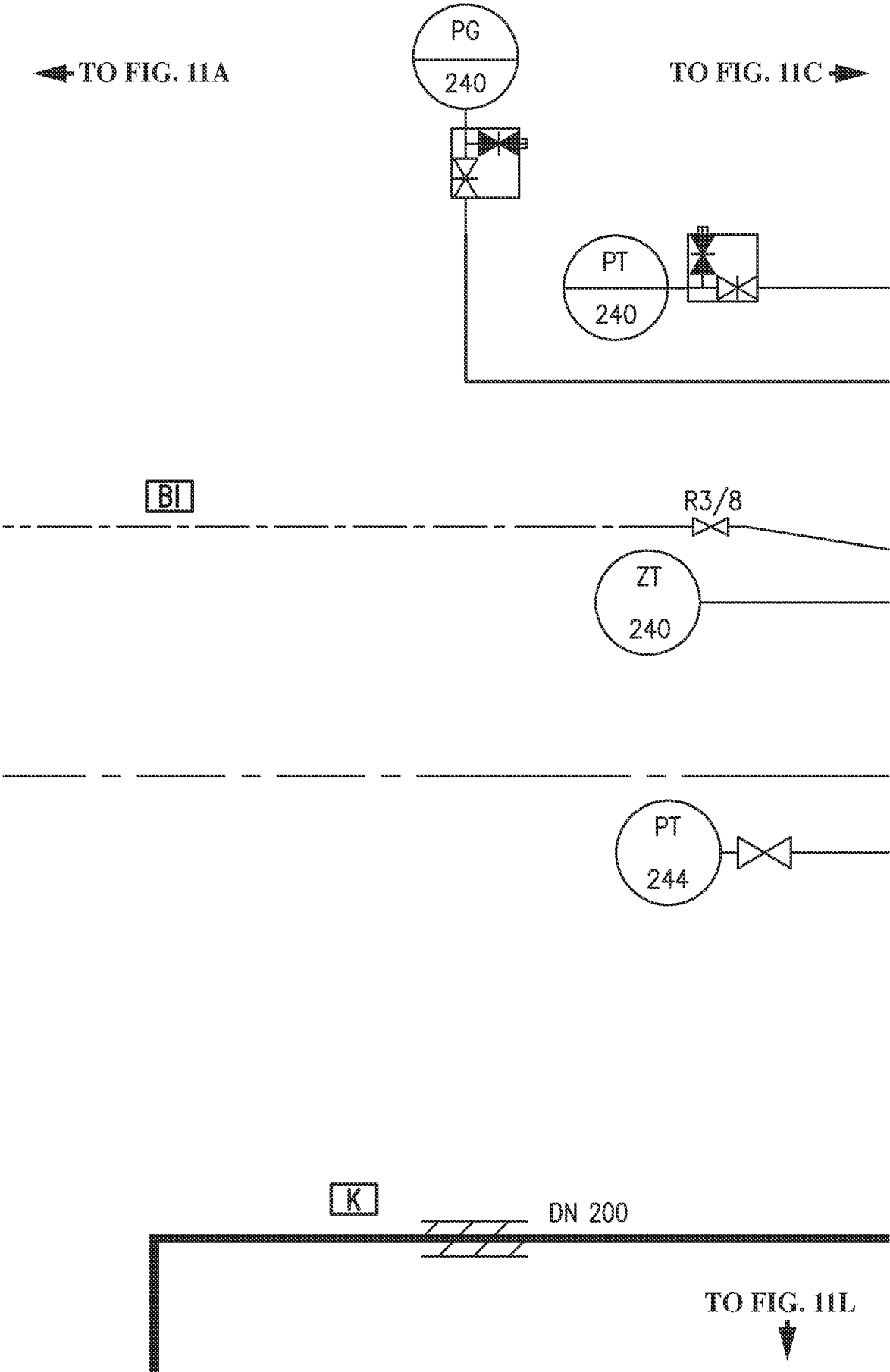


FIG. 11D

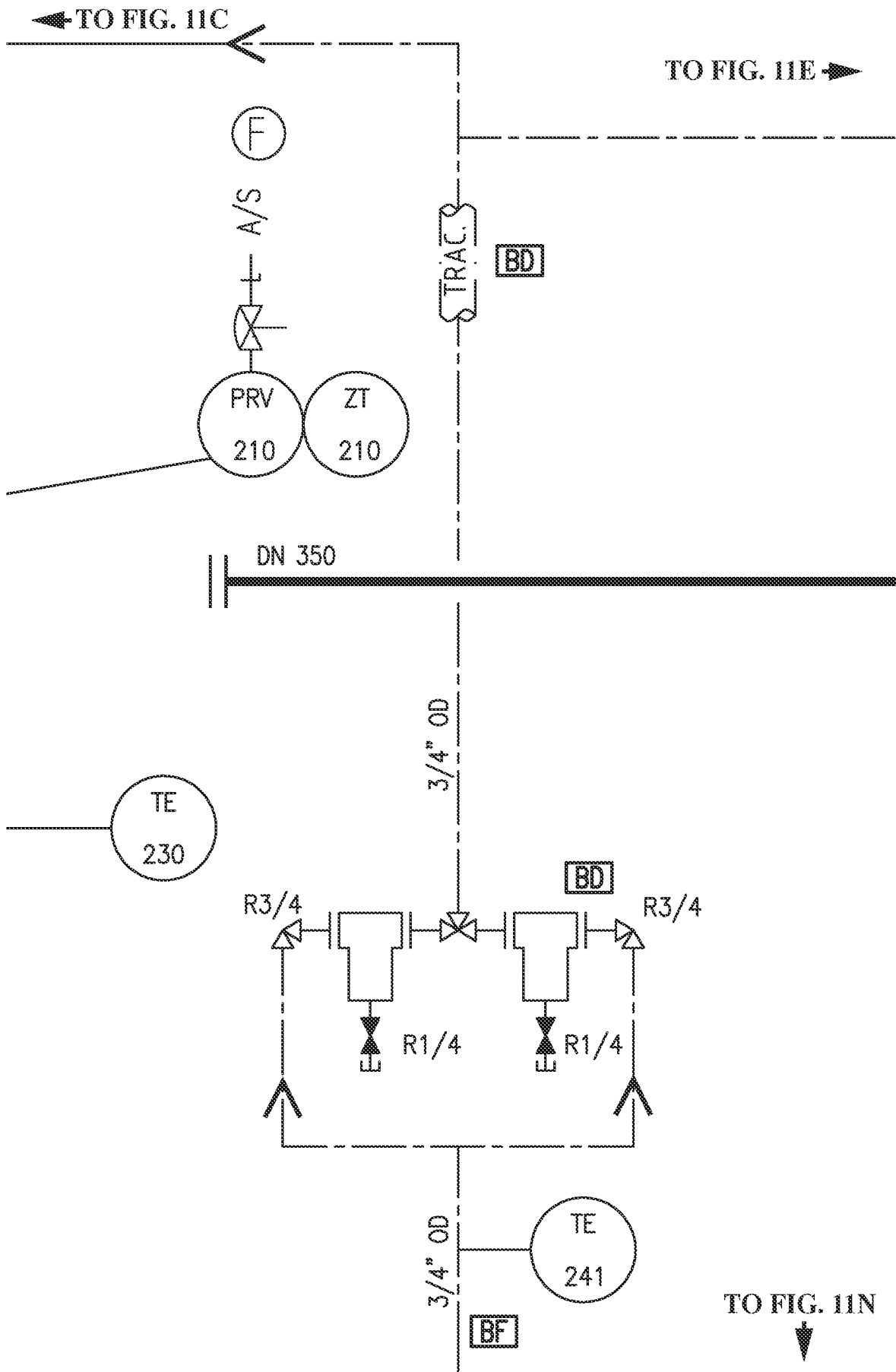


FIG. 11E

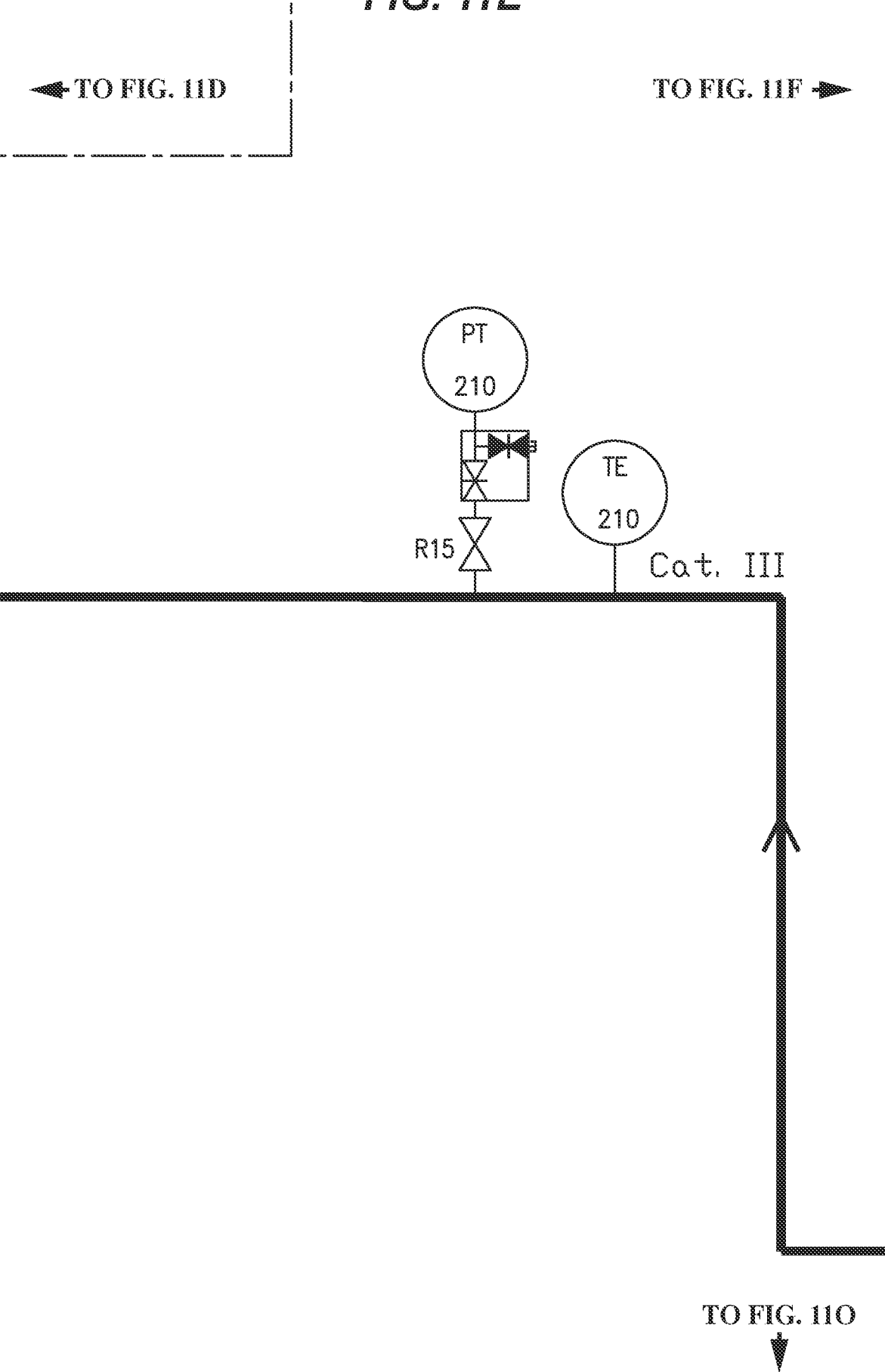


FIG. 11F

TO FIG. 11G ➡

➡ TO FIG. 11E

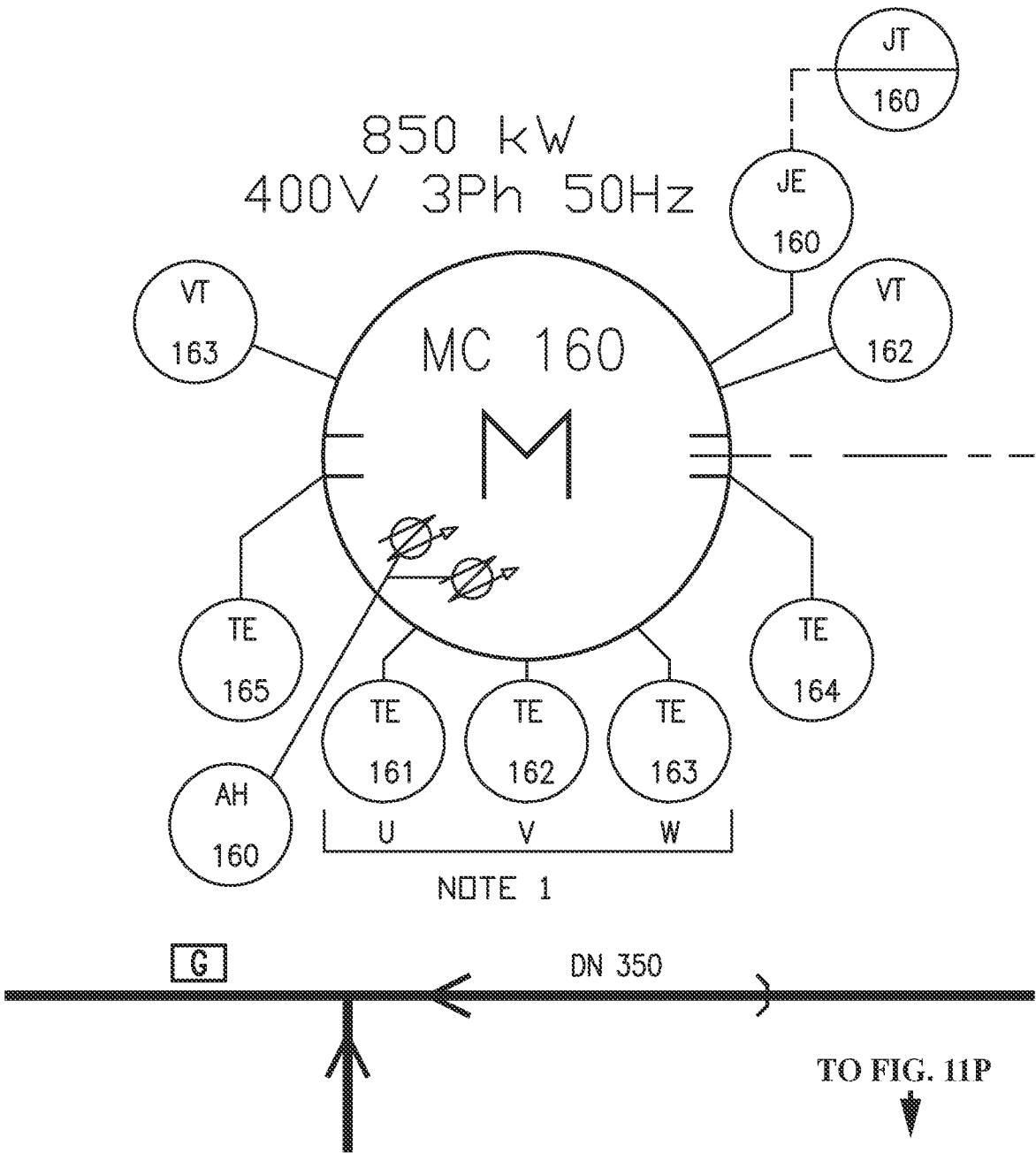
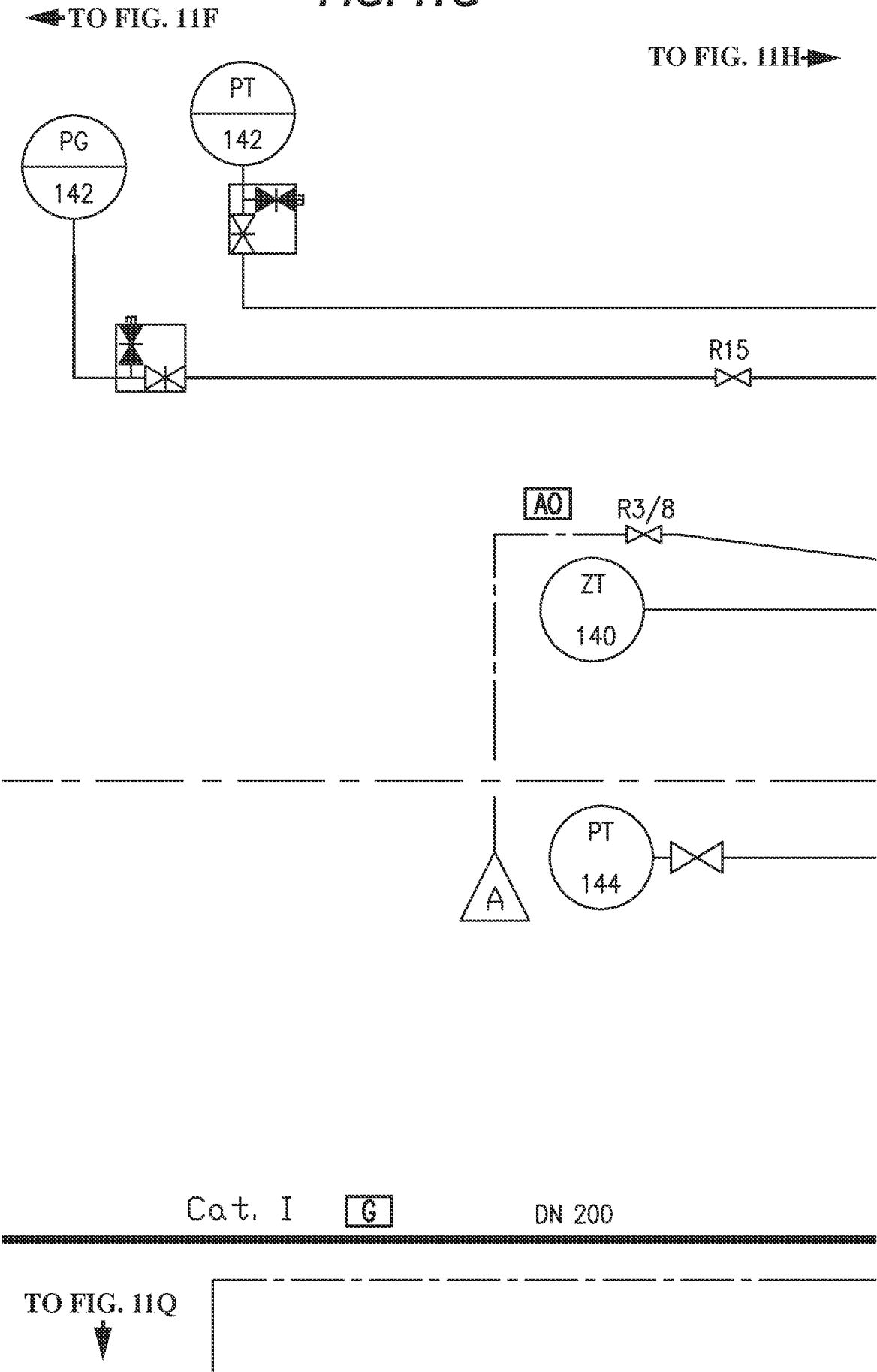
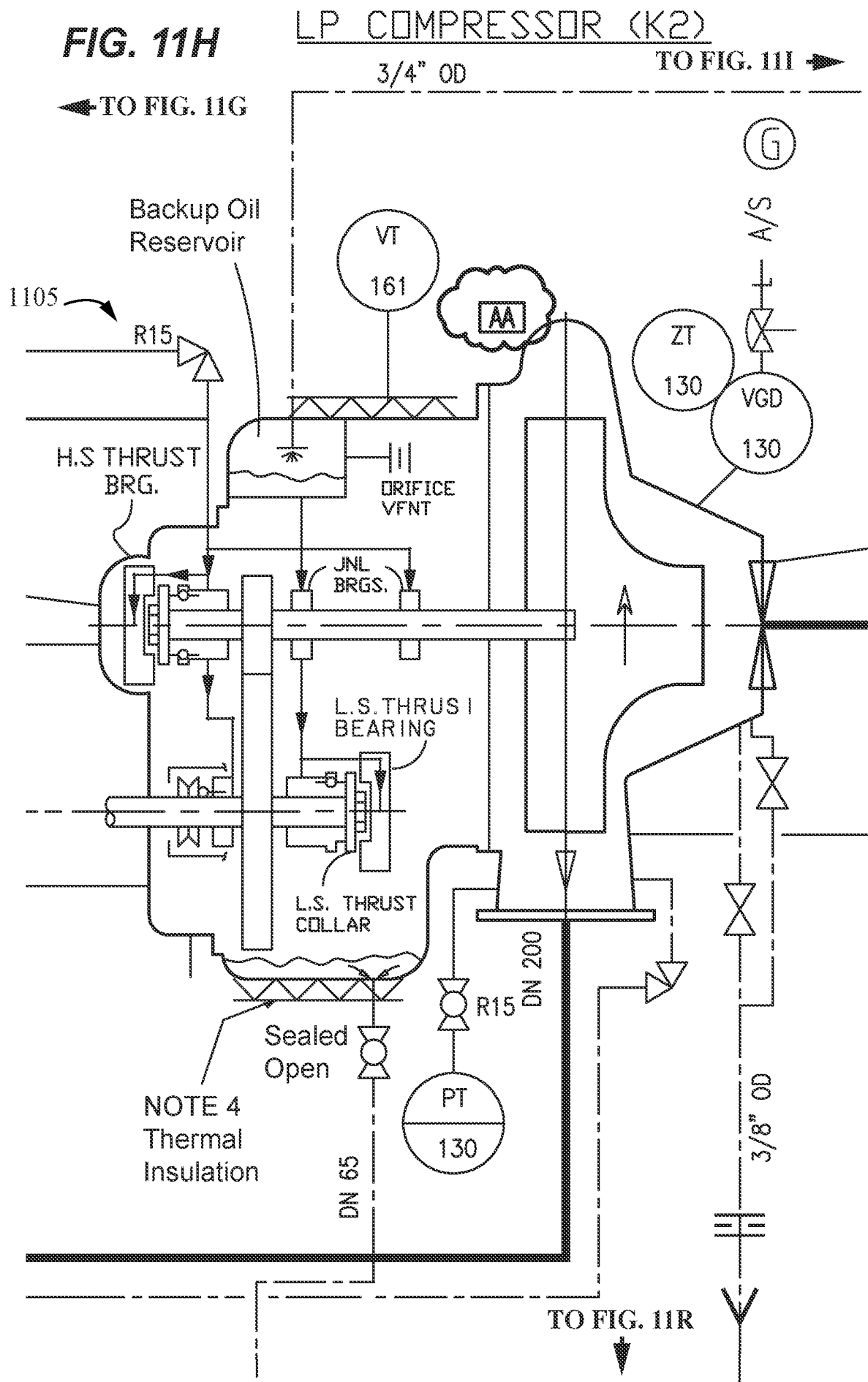


FIG. 11G





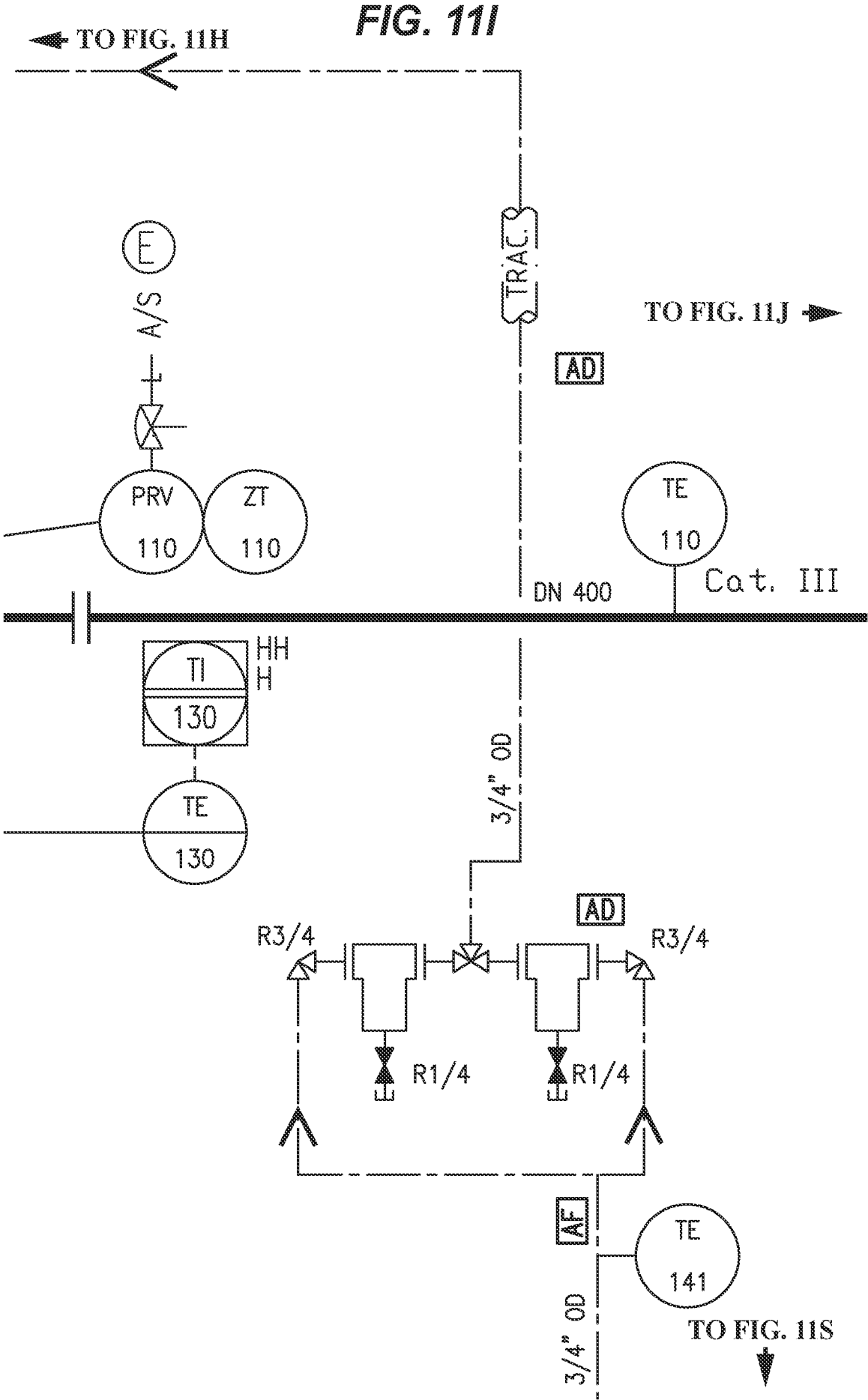


FIG. 11J

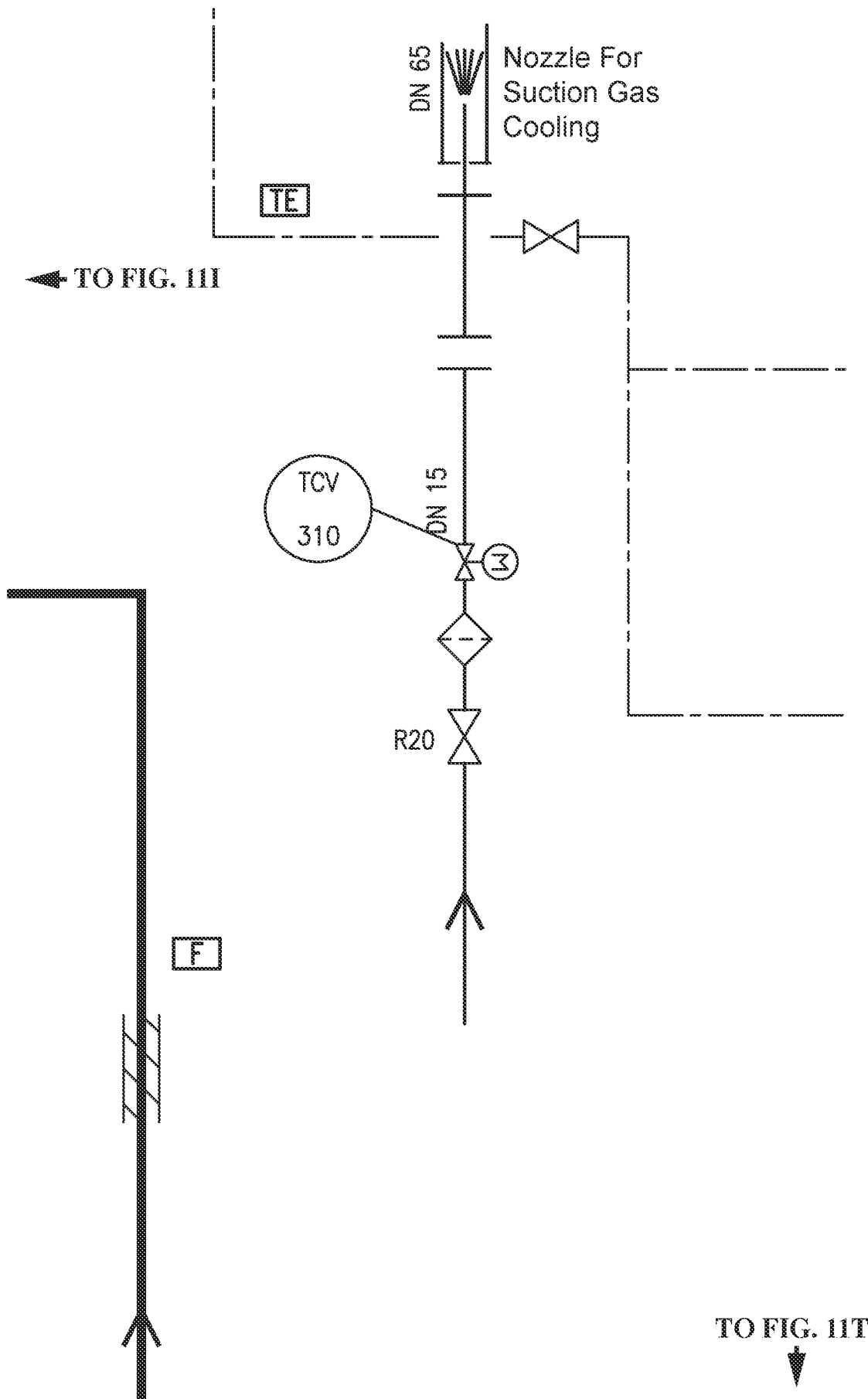


FIG. 11K

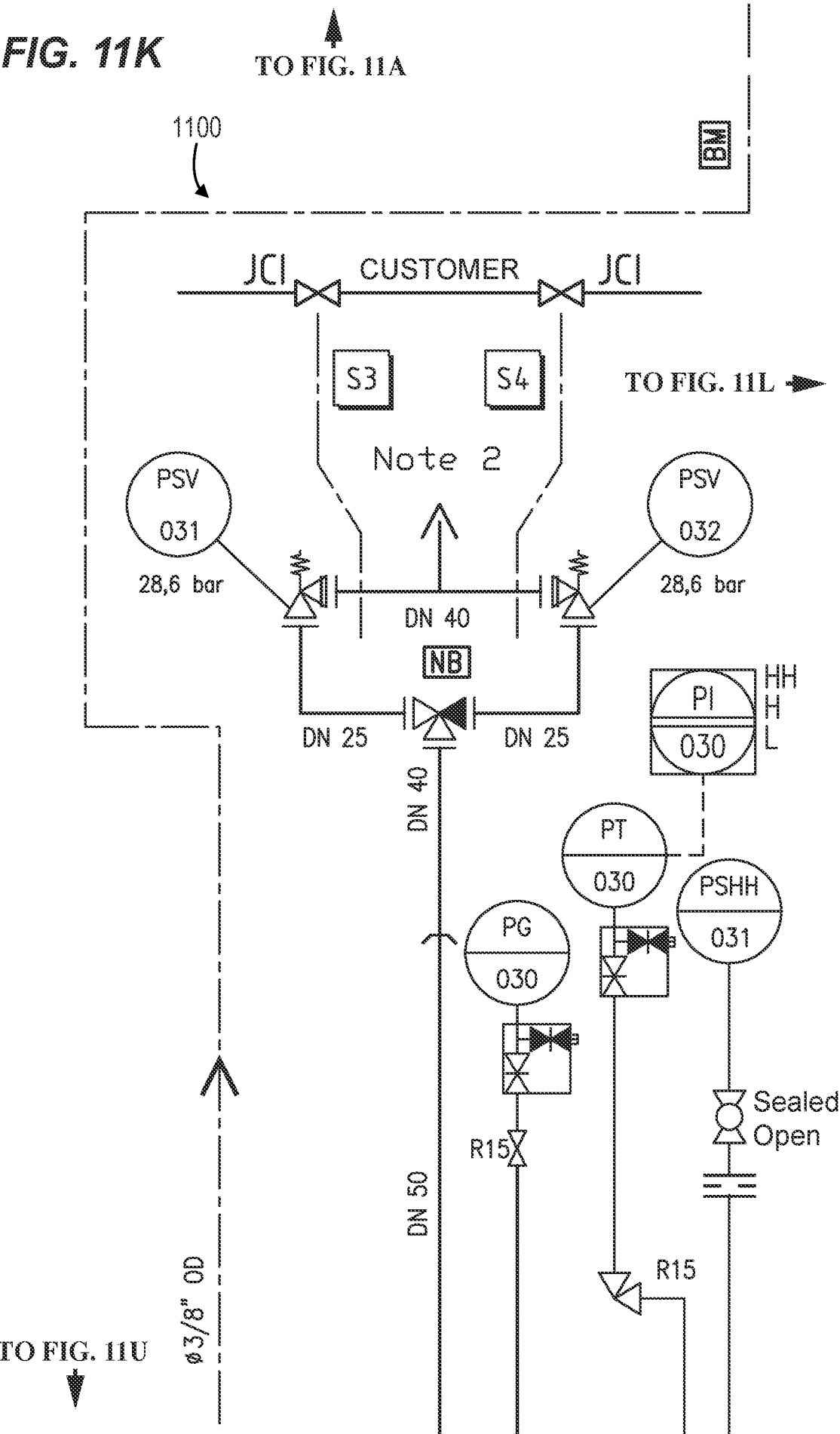


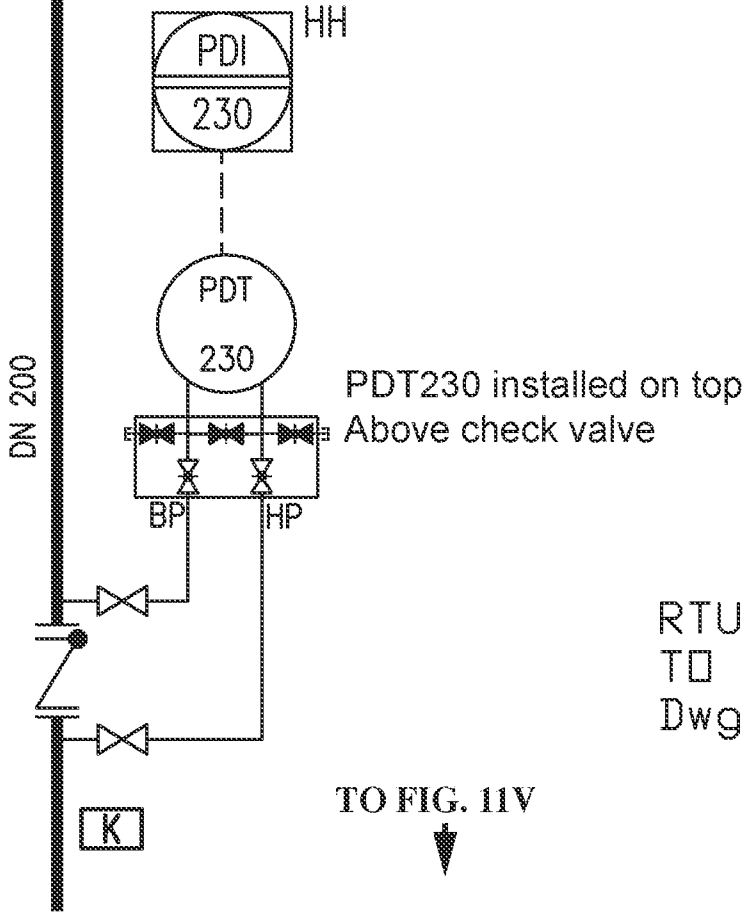
FIG. 11L

TO FIG. 11B

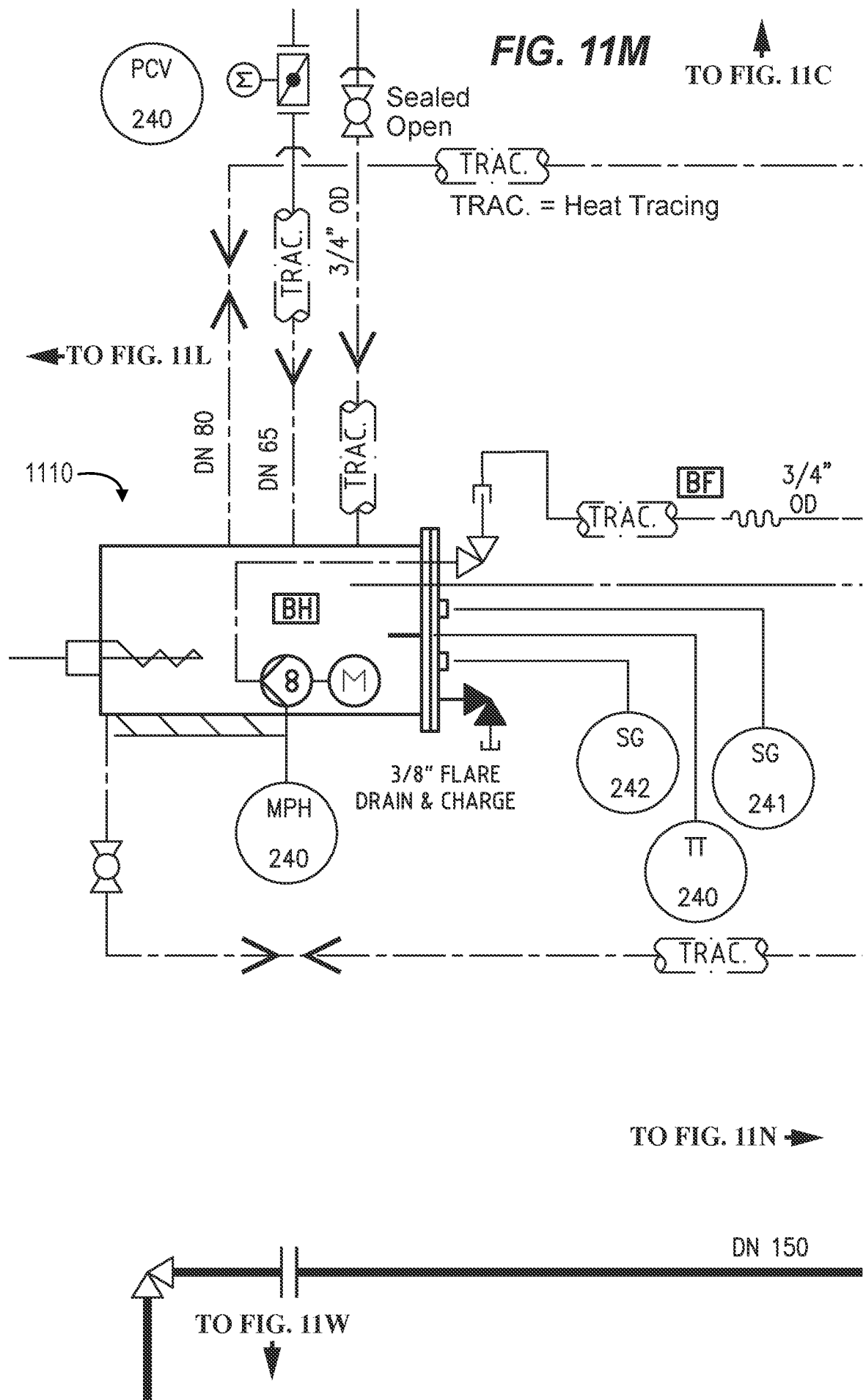
Ca.t. II
TO FIG. 11K

TO FIG. 11M

HP OIL
RESERVOIR



RTU (Gas)
TO RTU P&ID
Dwg 341162



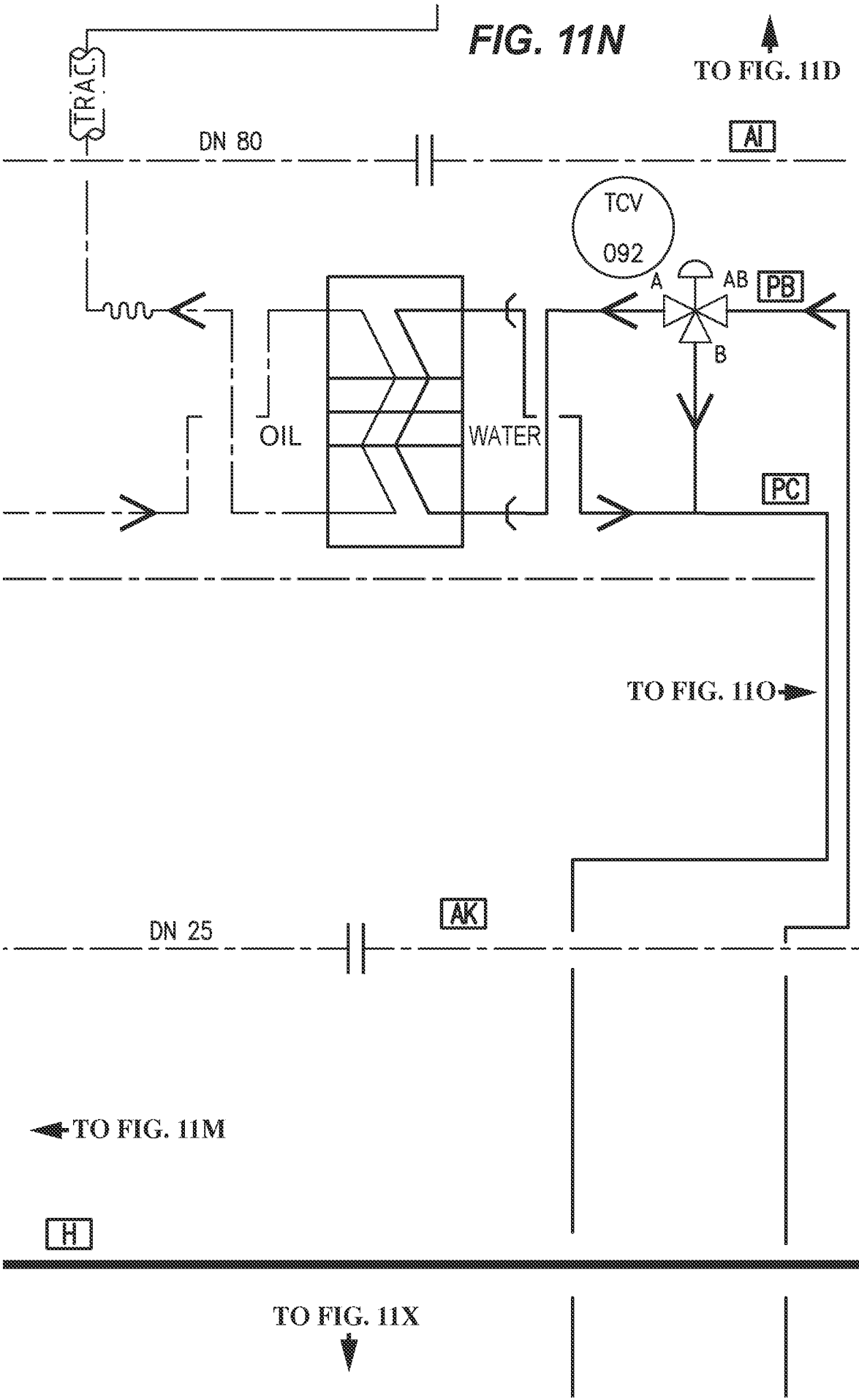


FIG. 110


TO FIG. 11E



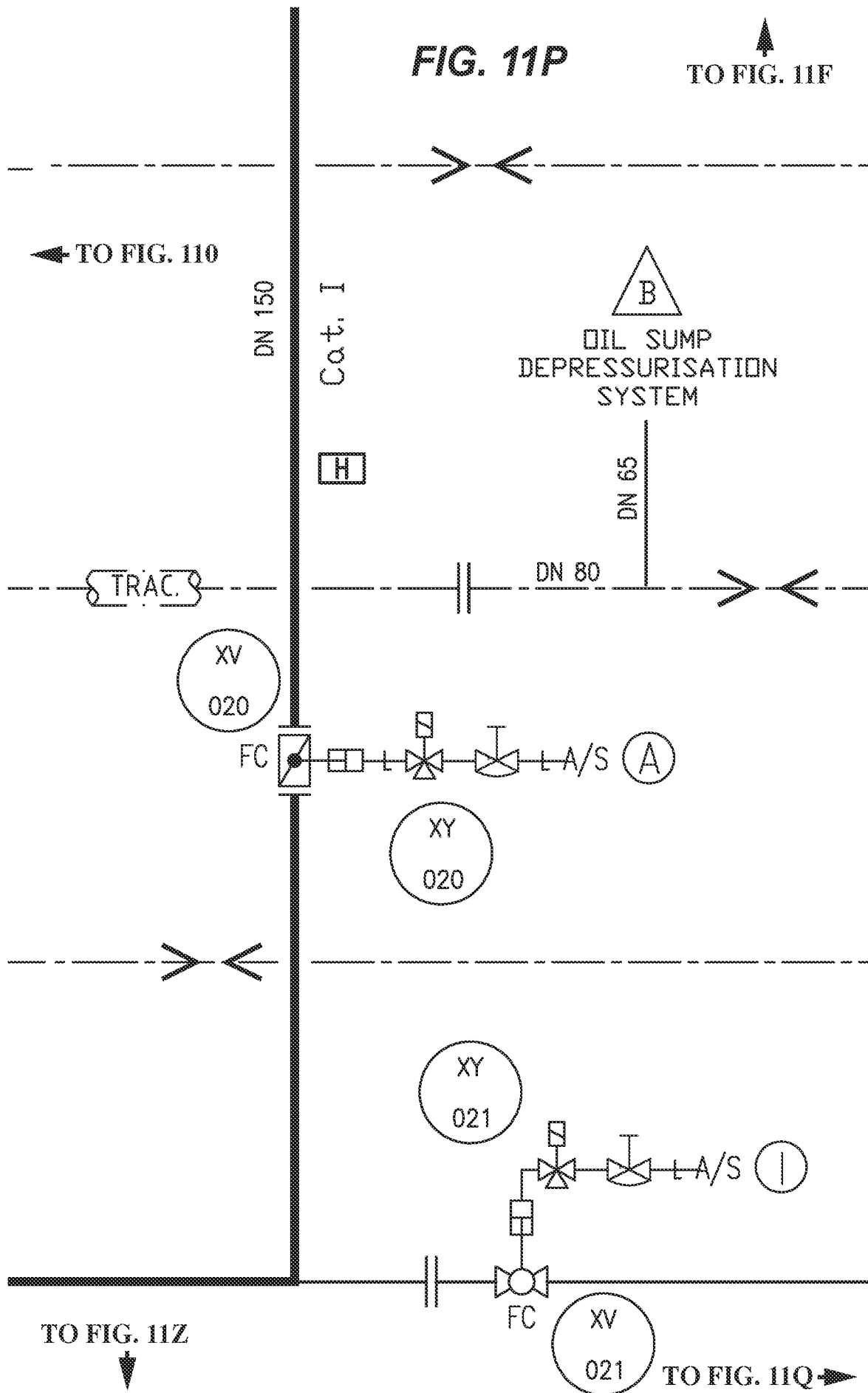
TO FIG. 11P 

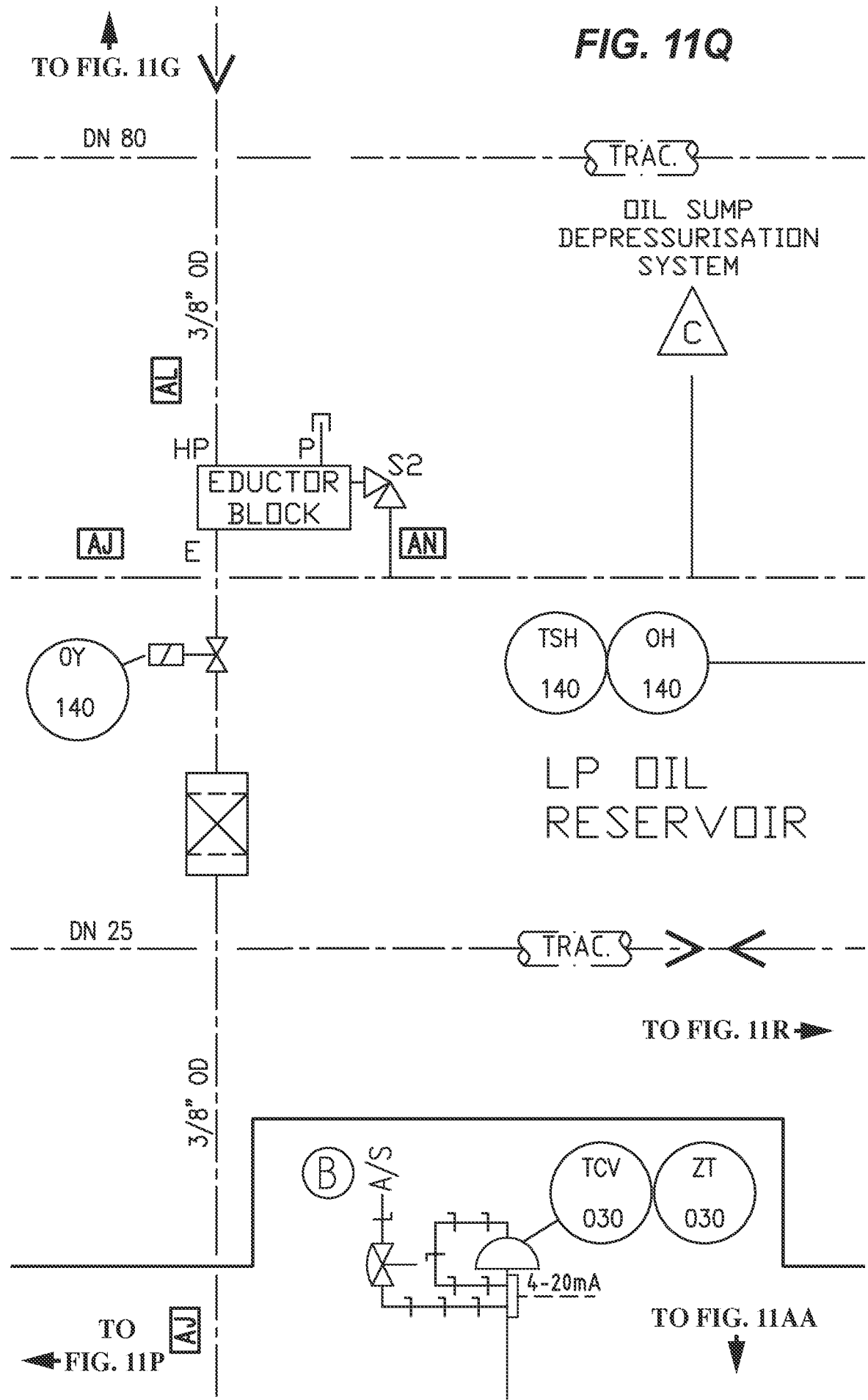


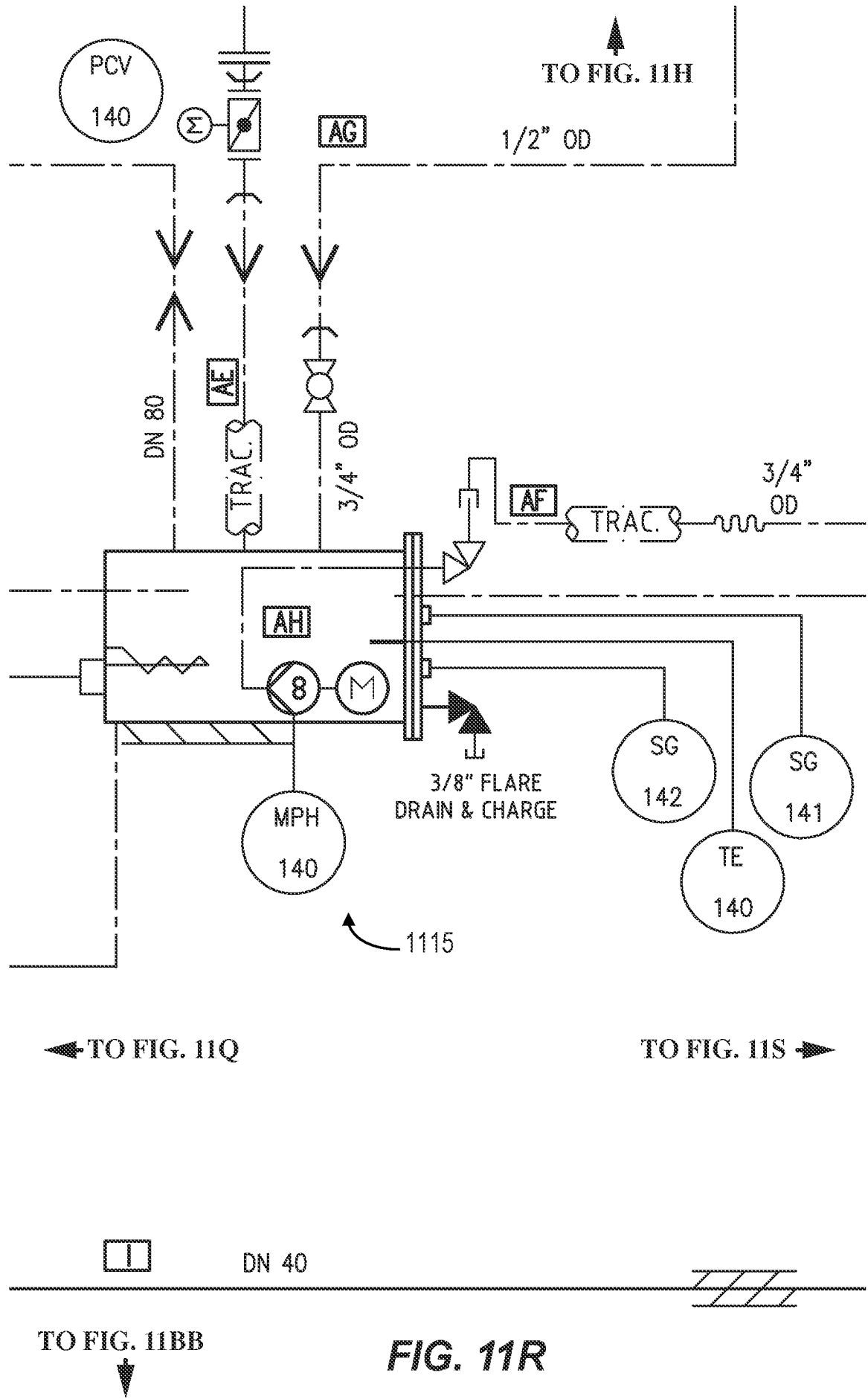
 TO FIG. 11N



TO FIG. 11Y





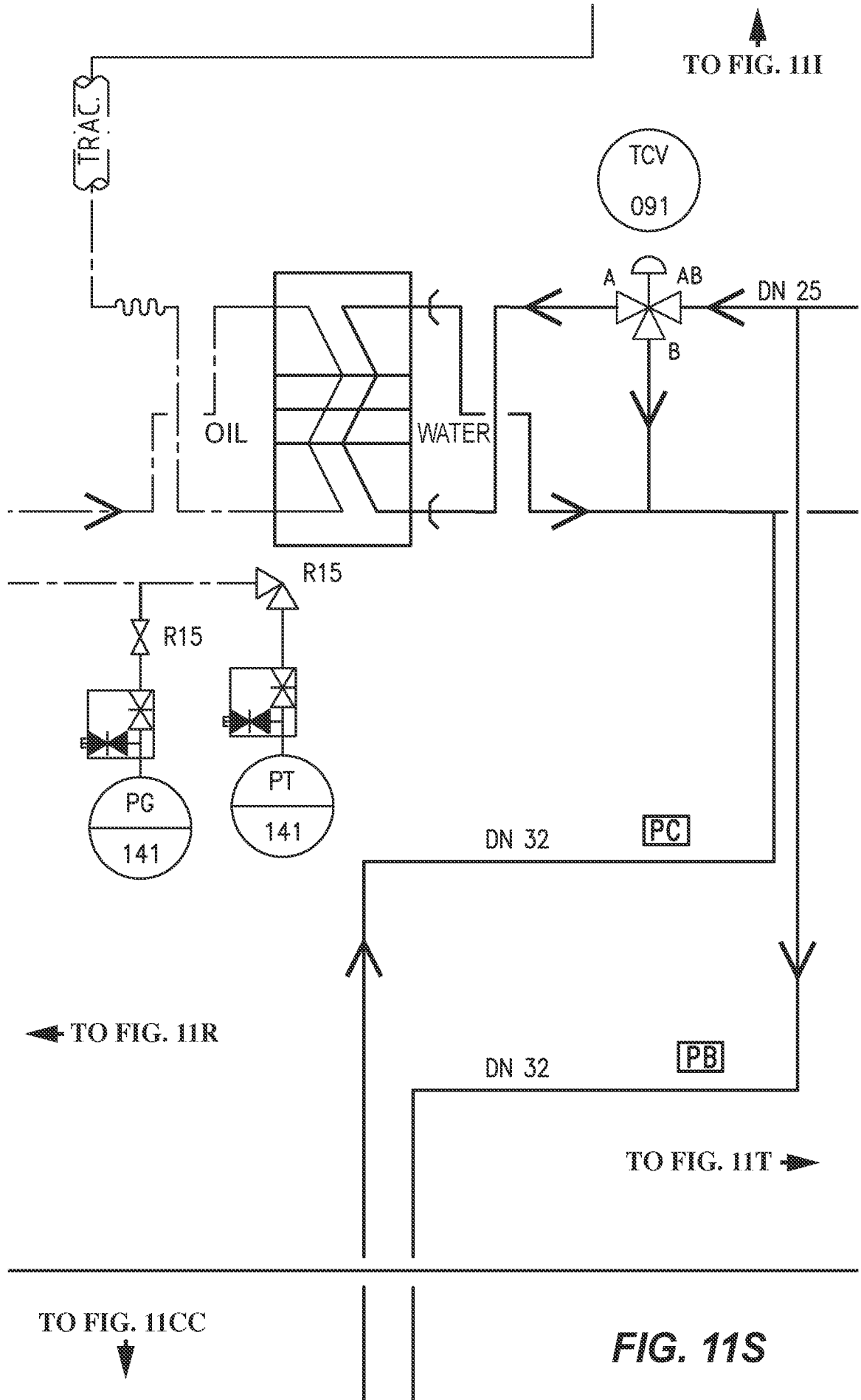
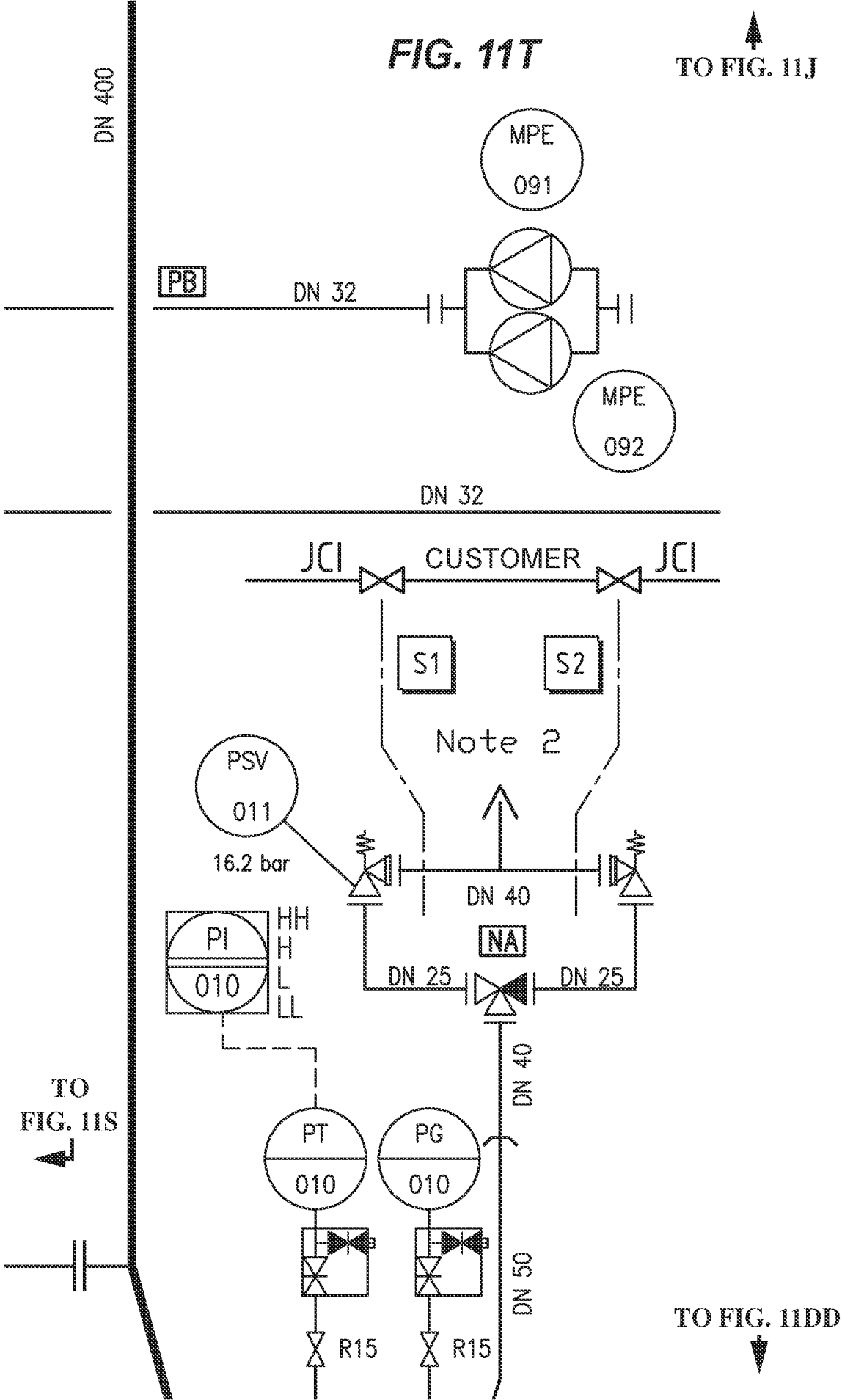


FIG. 11S



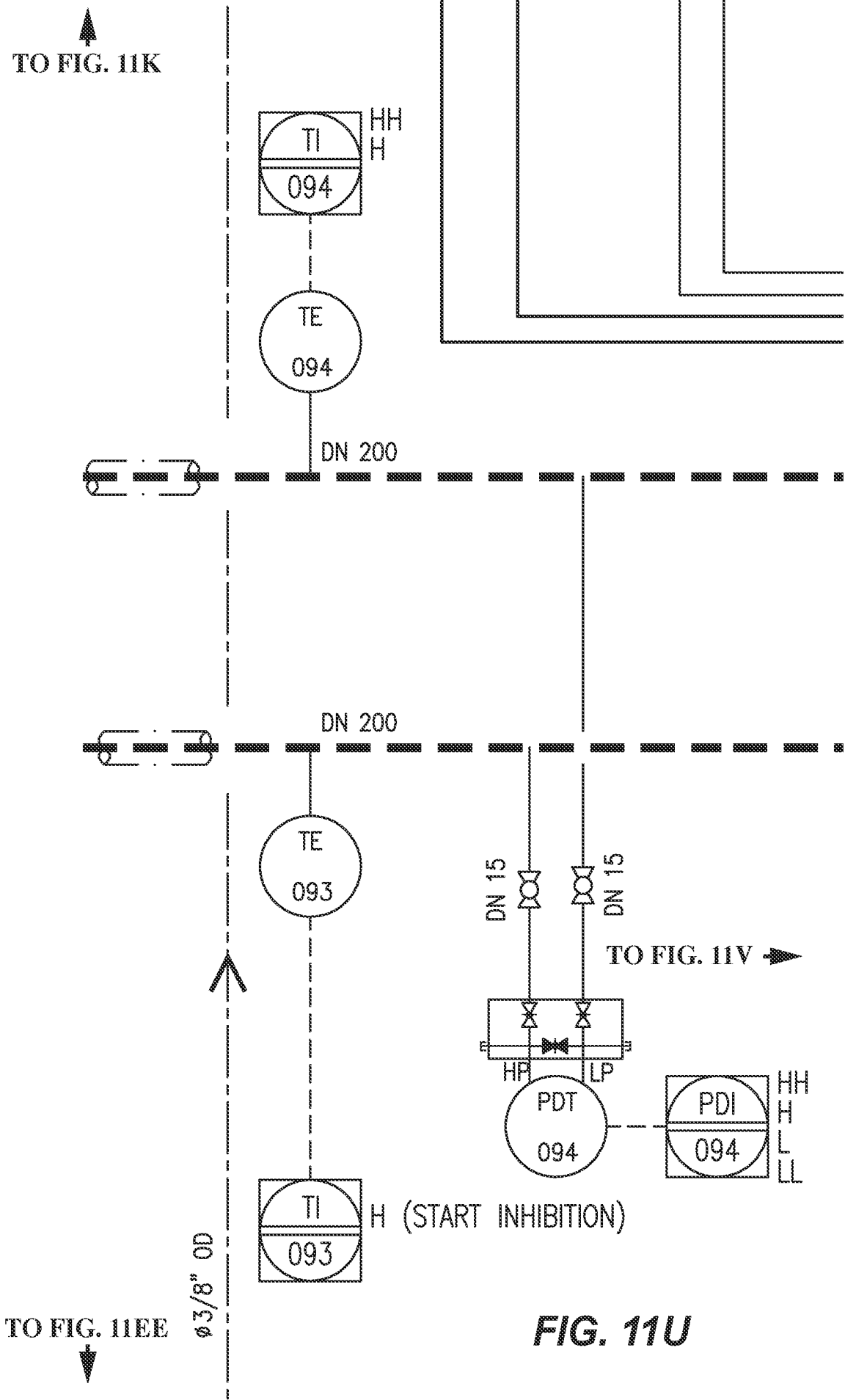
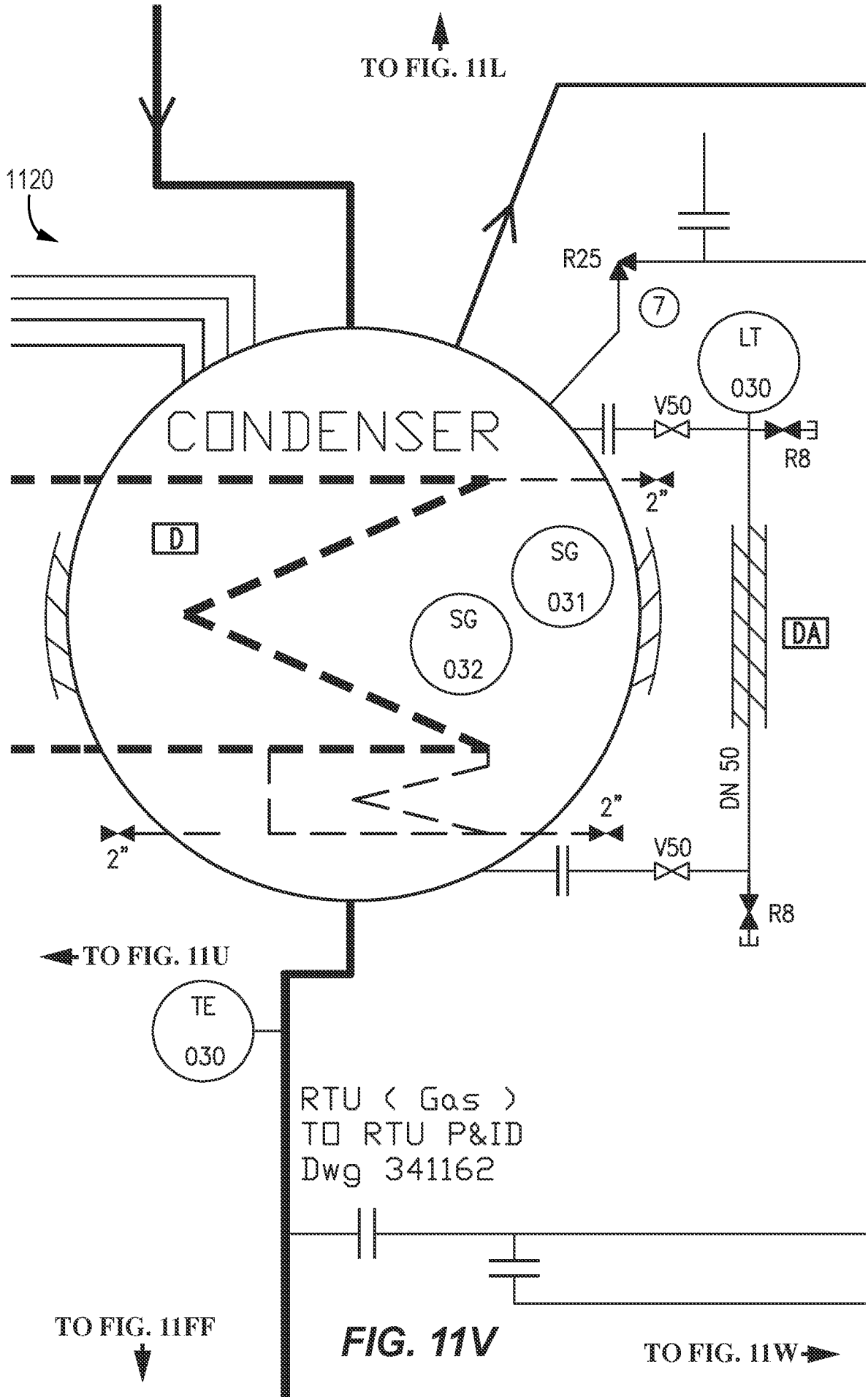
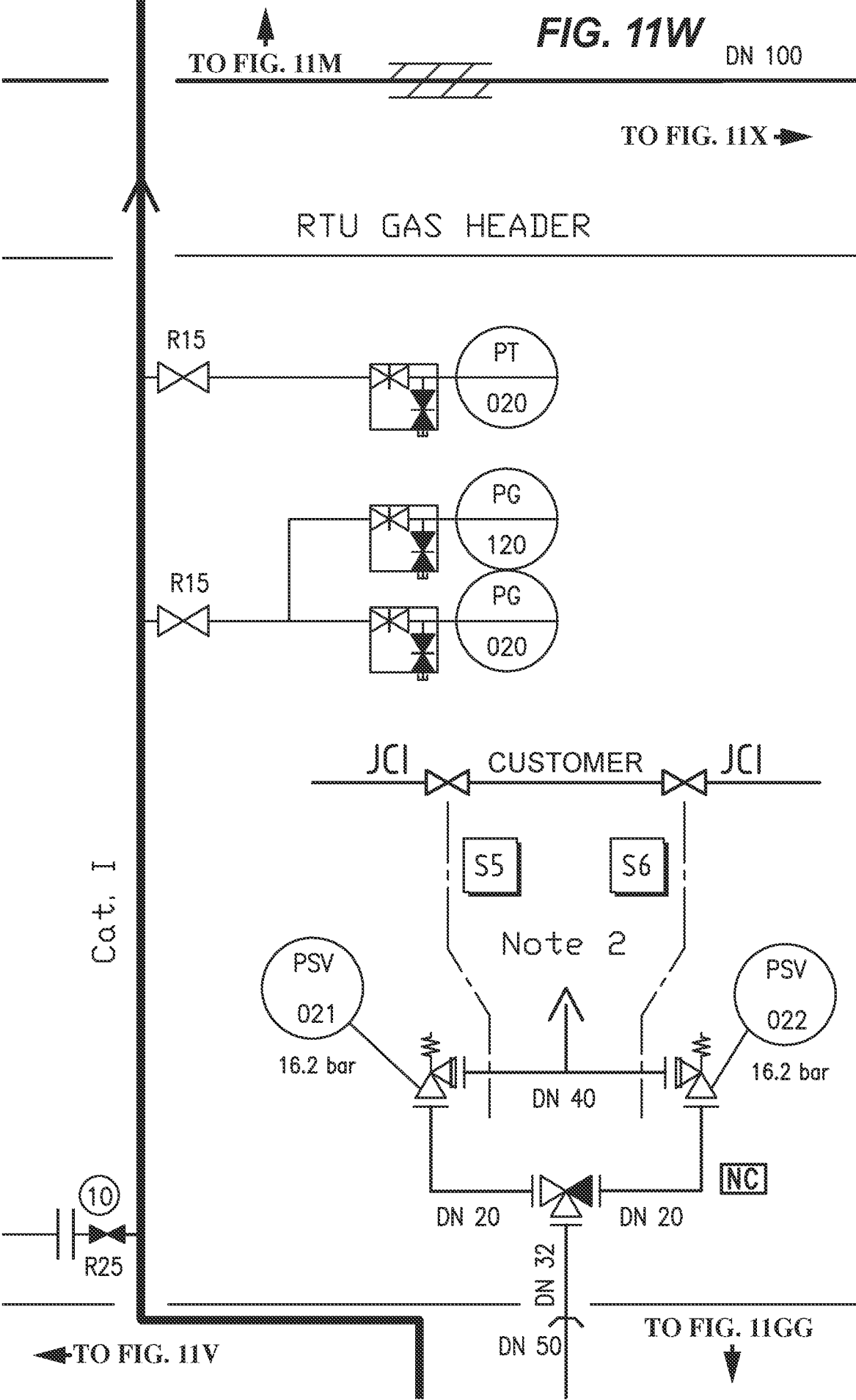
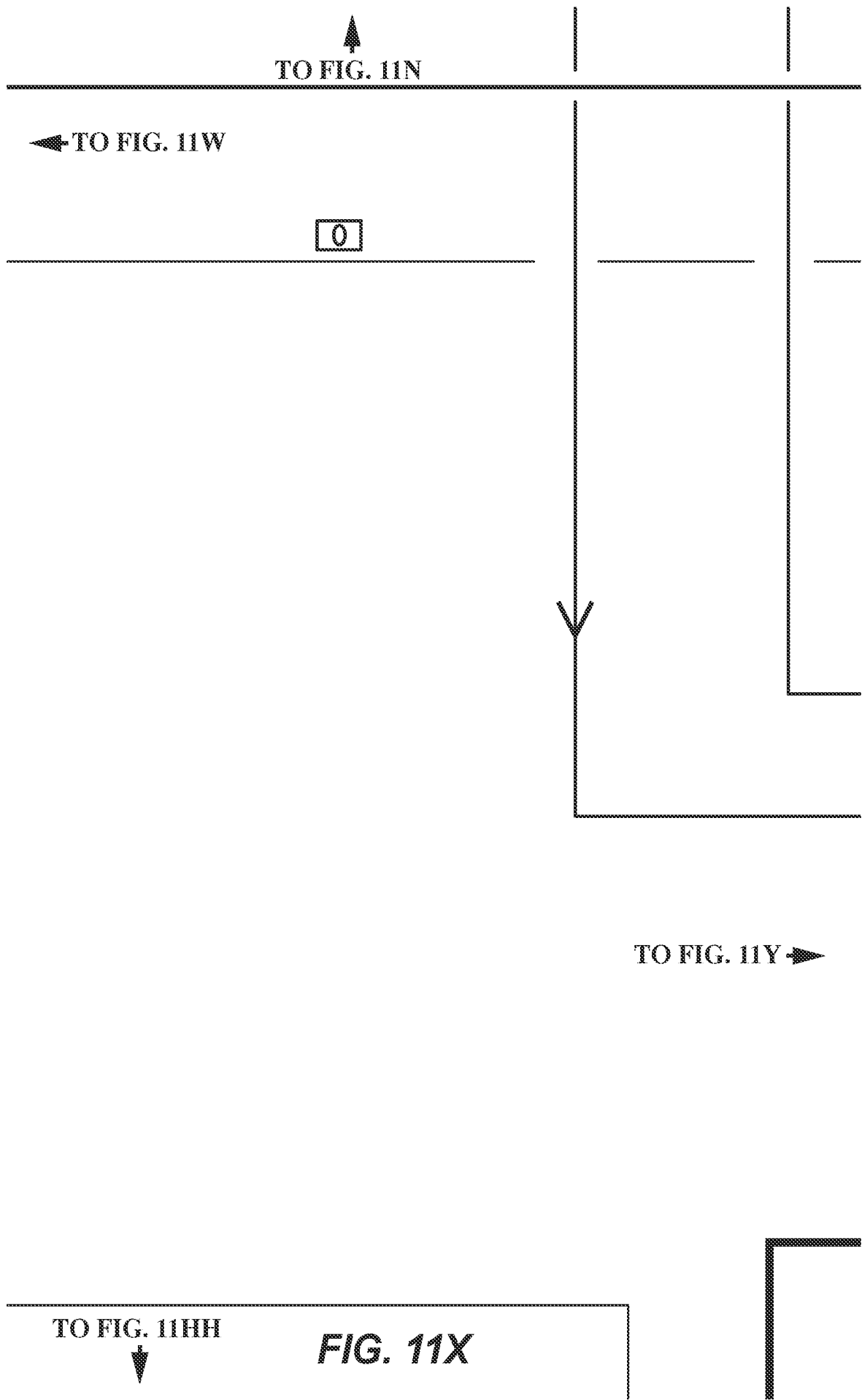


FIG. 11U









TO FIG. 11O



← TO FIG. 11X

TO FIG. 11Z →



FIG. 11Y

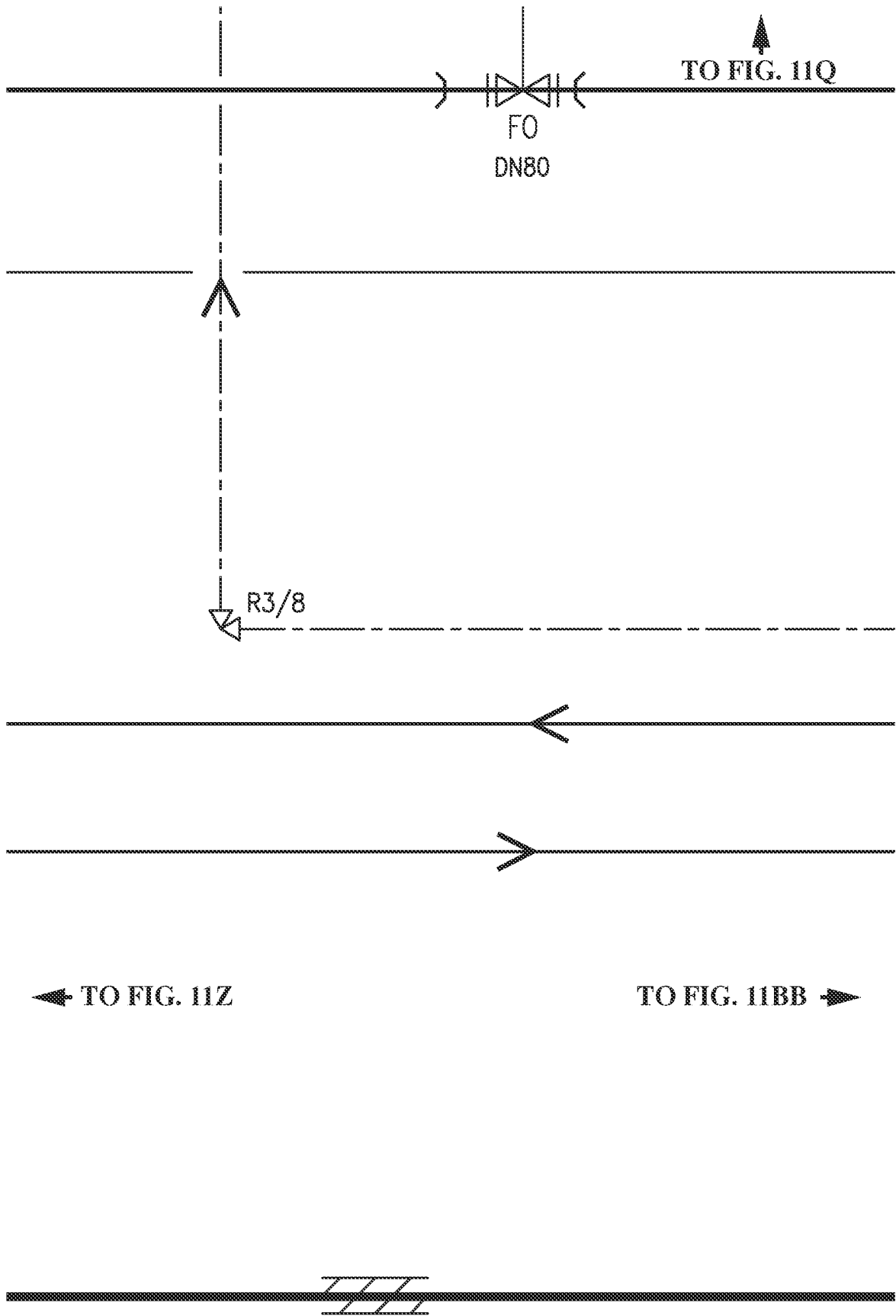
↑
TO FIG. 11P Cat. I J

← TO FIG. 11Y

TO FIG. 11AA →



FIG. 11Z



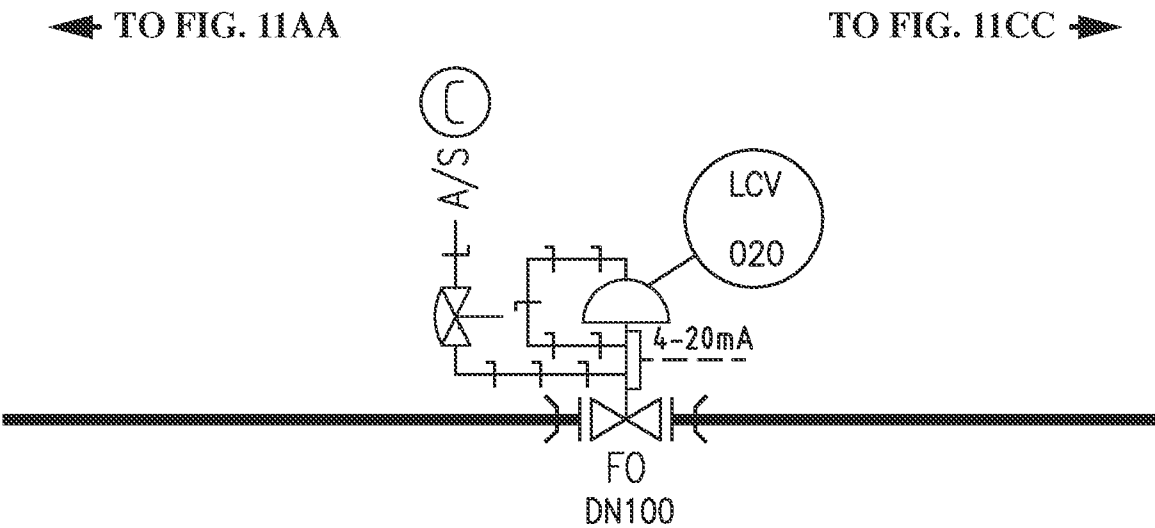
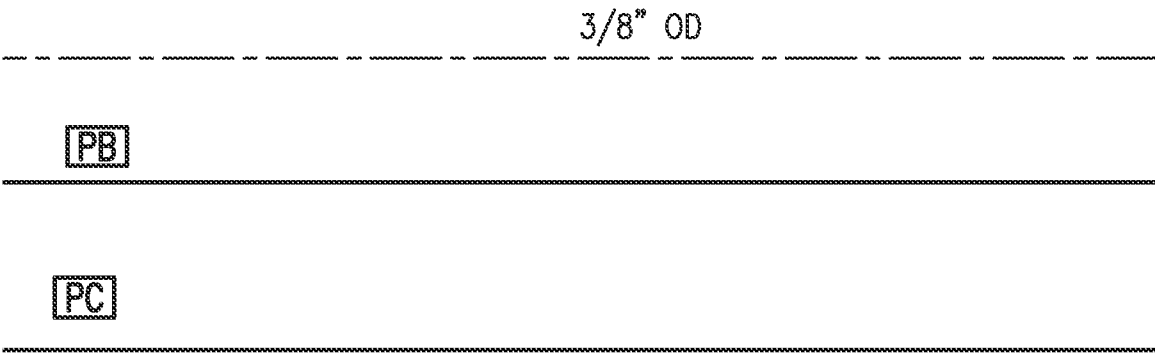
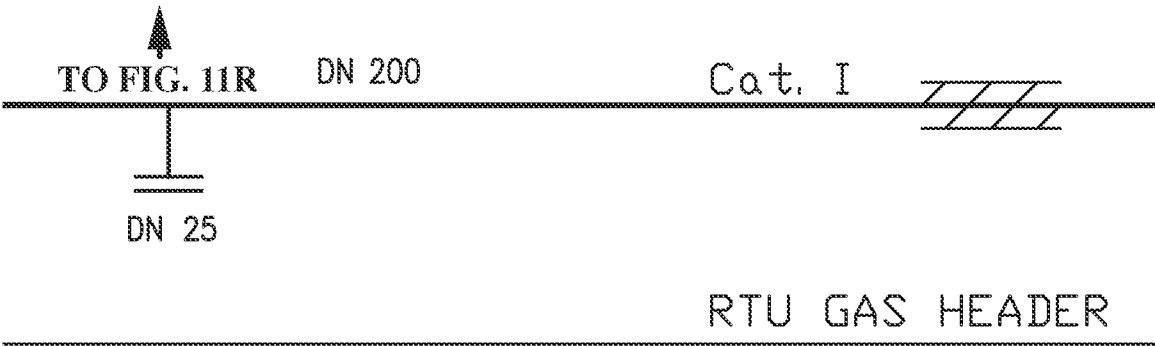


FIG. 11BB

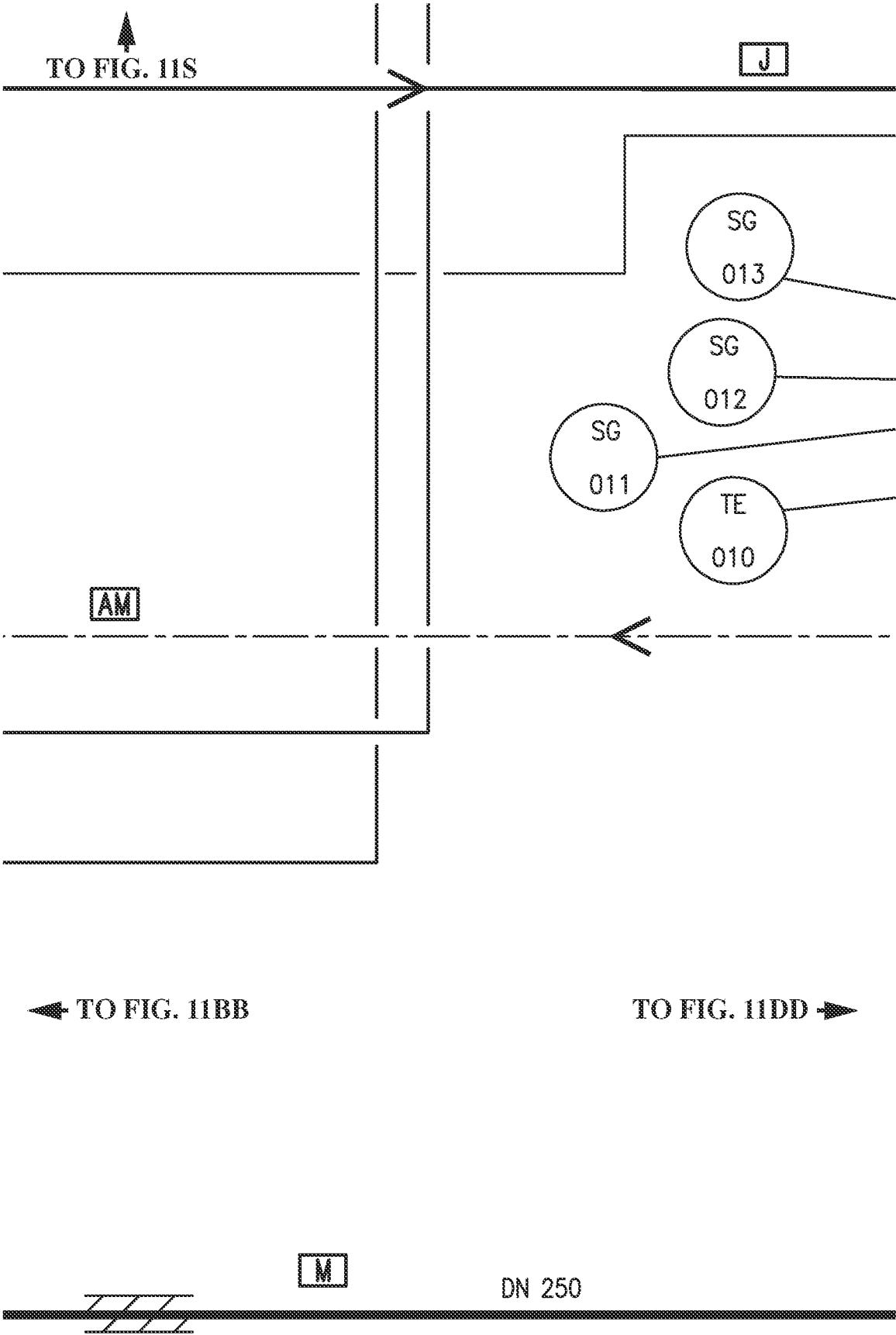


FIG. 11CC

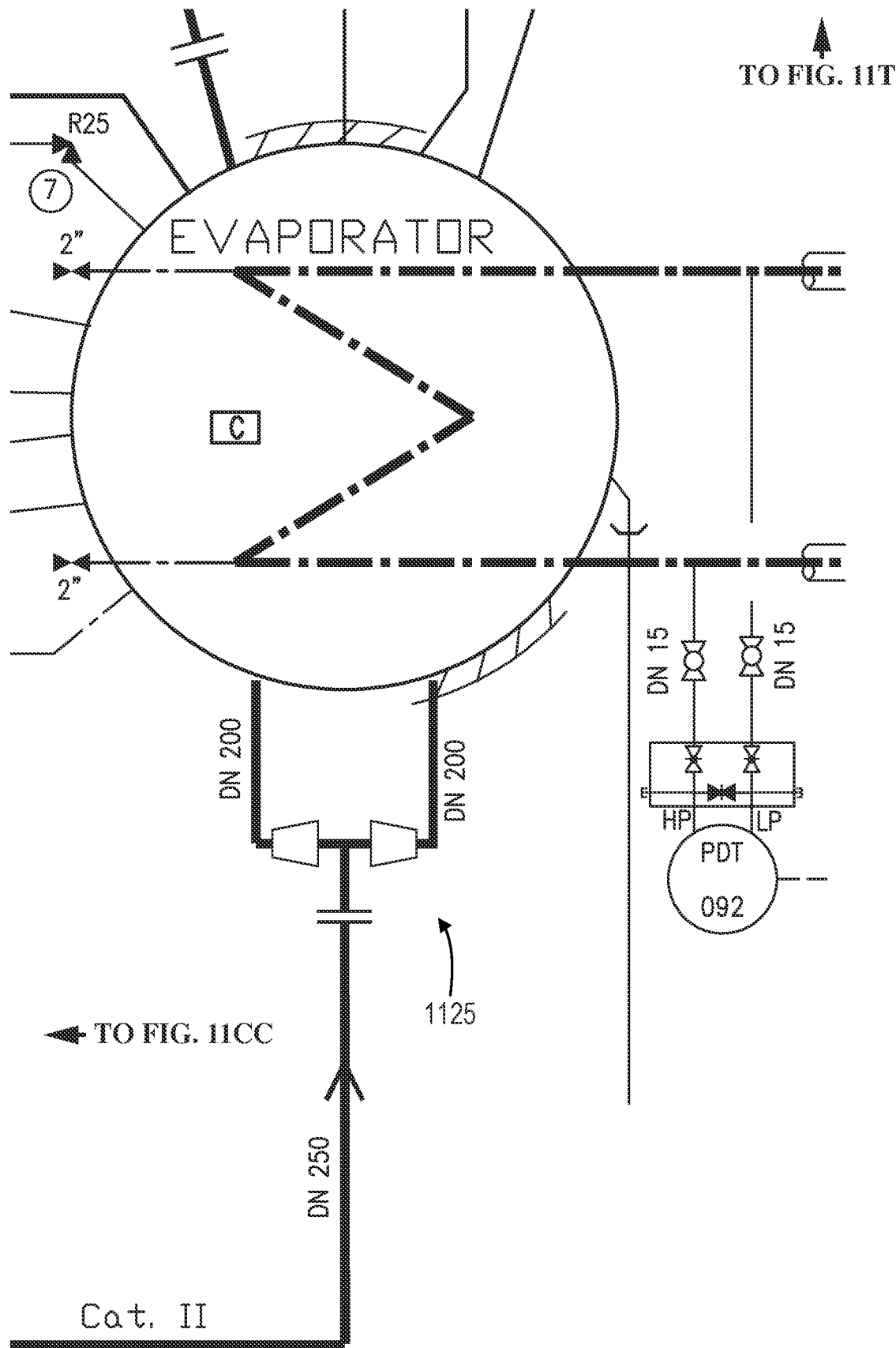


FIG. 11DD

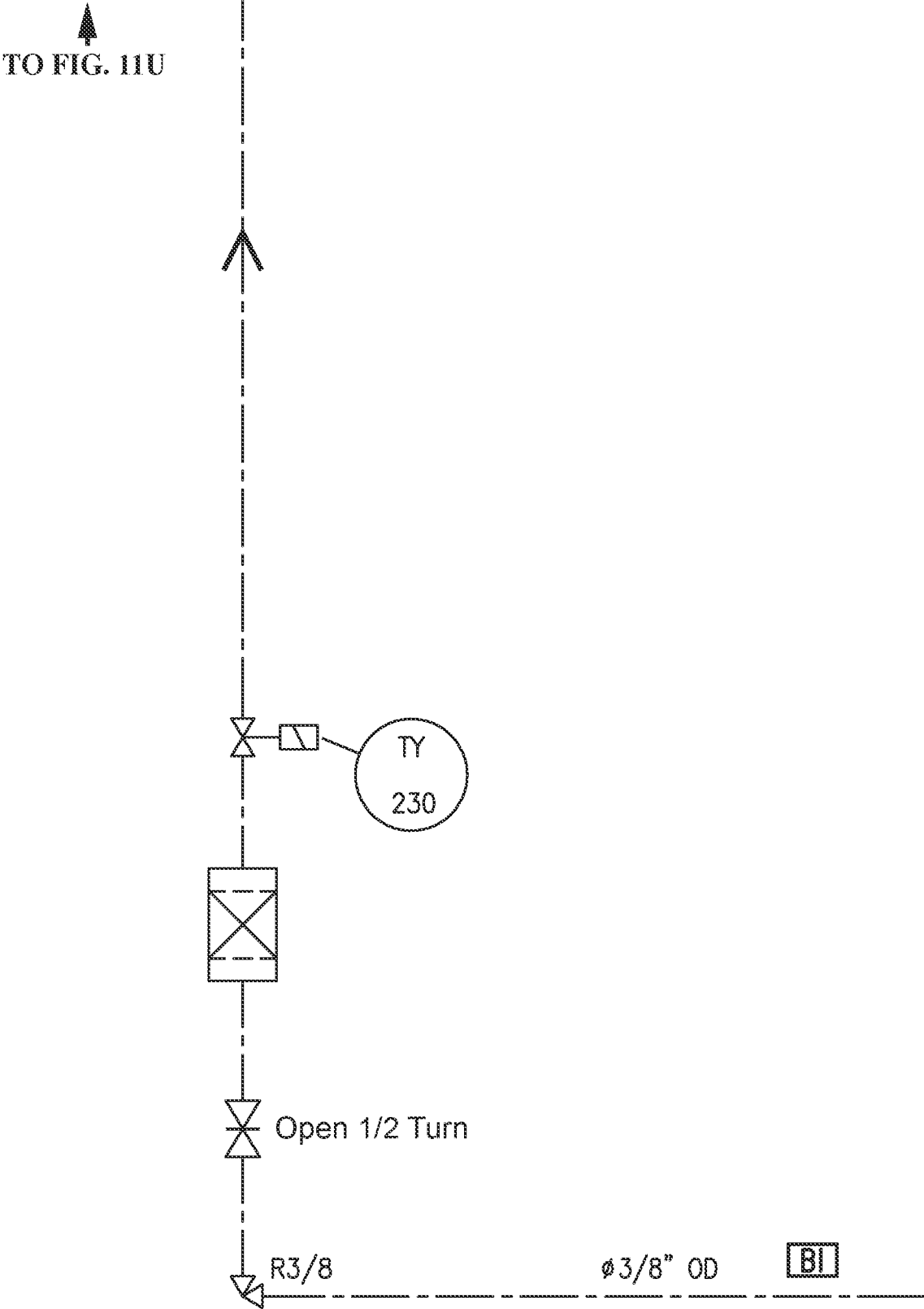


FIG. 11EE

TO FIG. 11FF

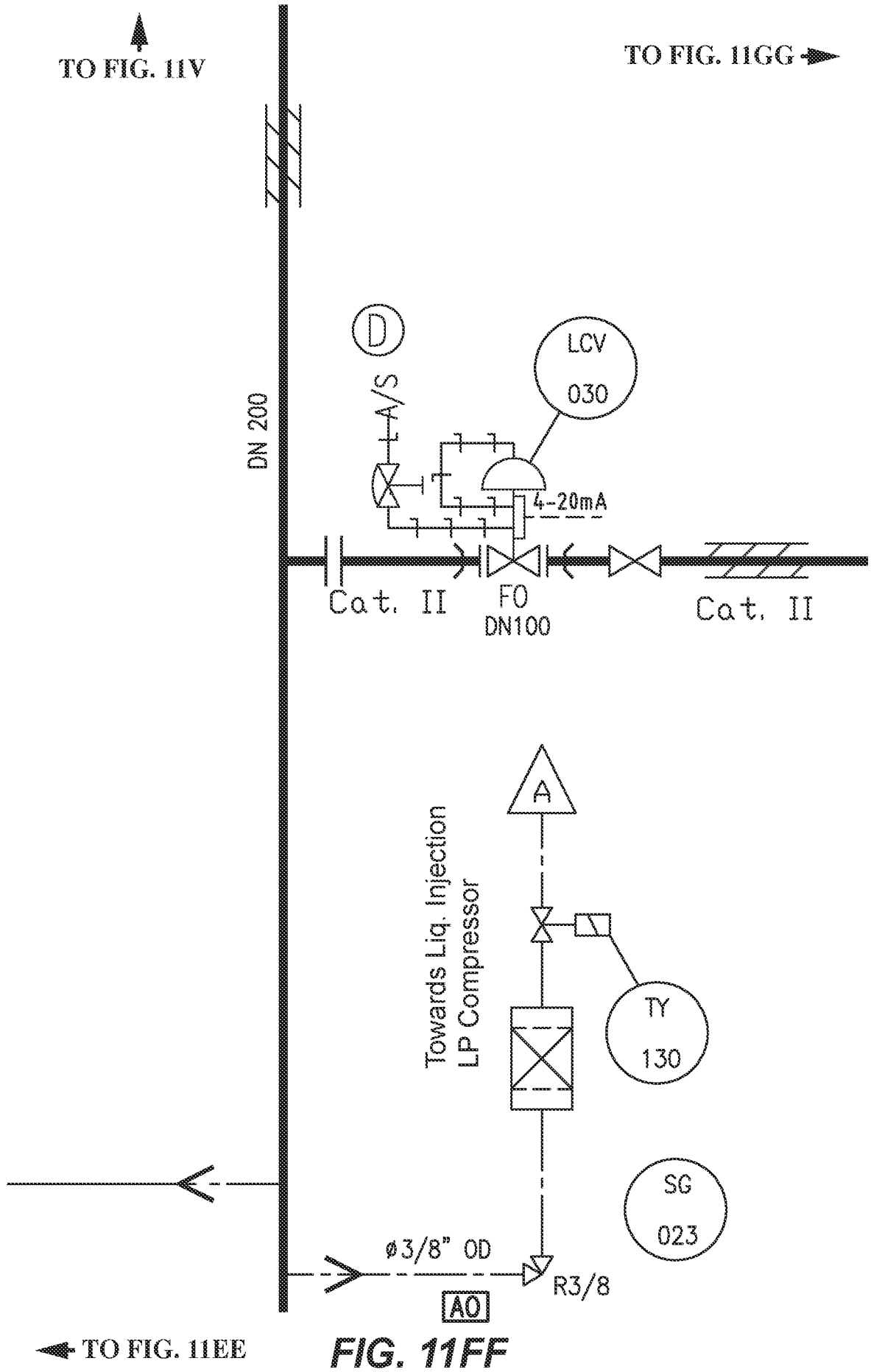
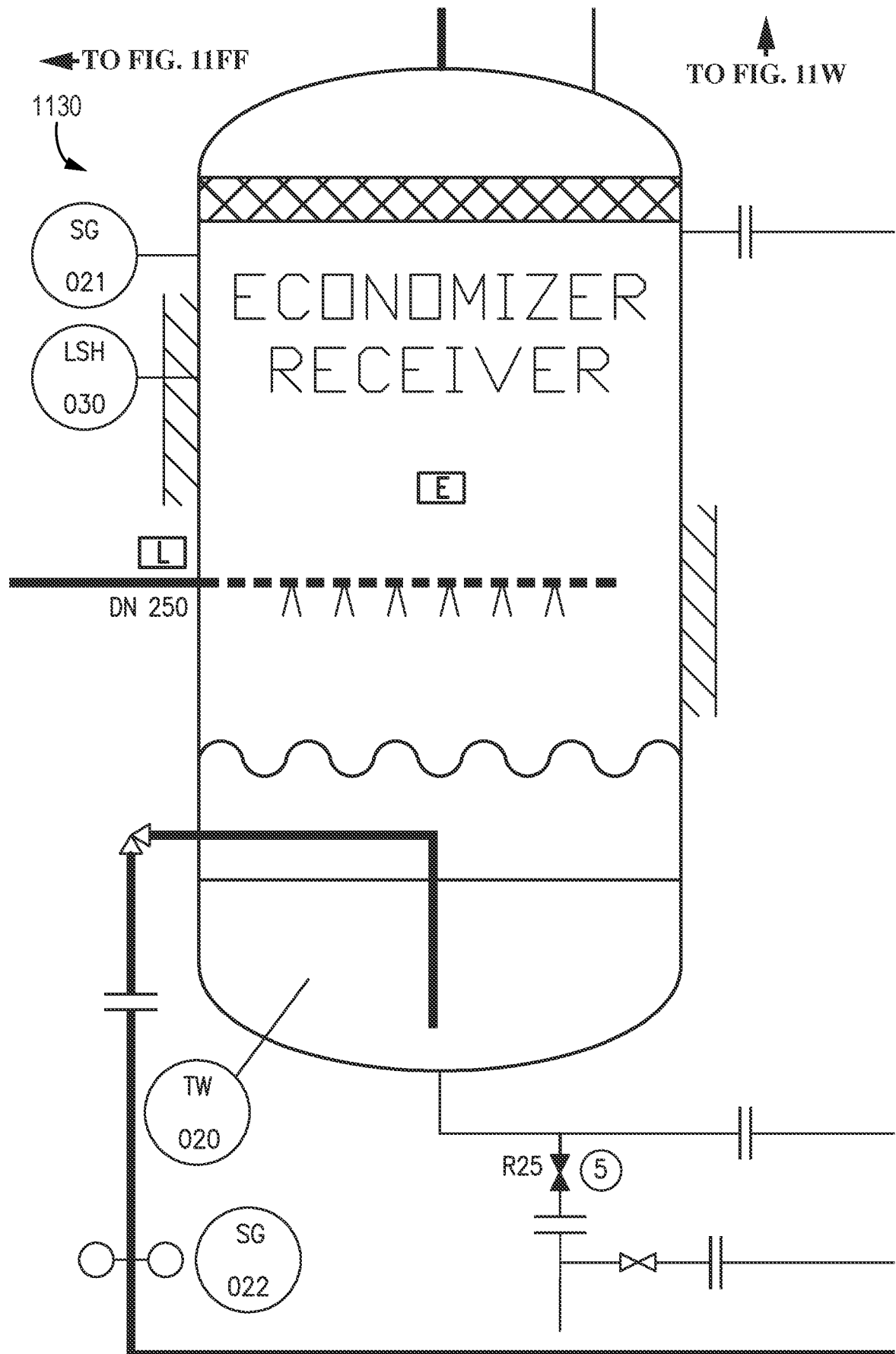
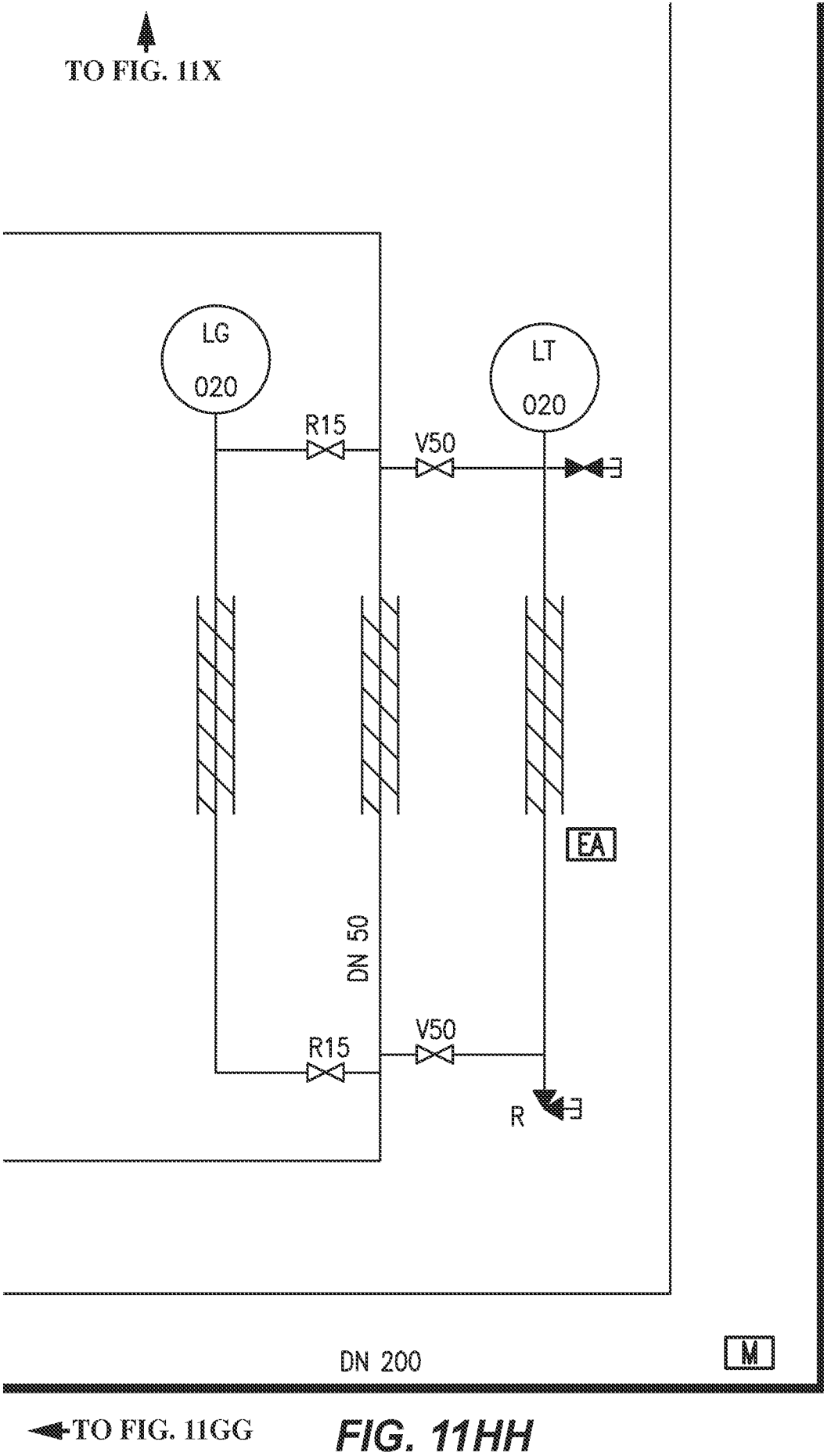


FIG. 11FF



TO FIG. 11HH



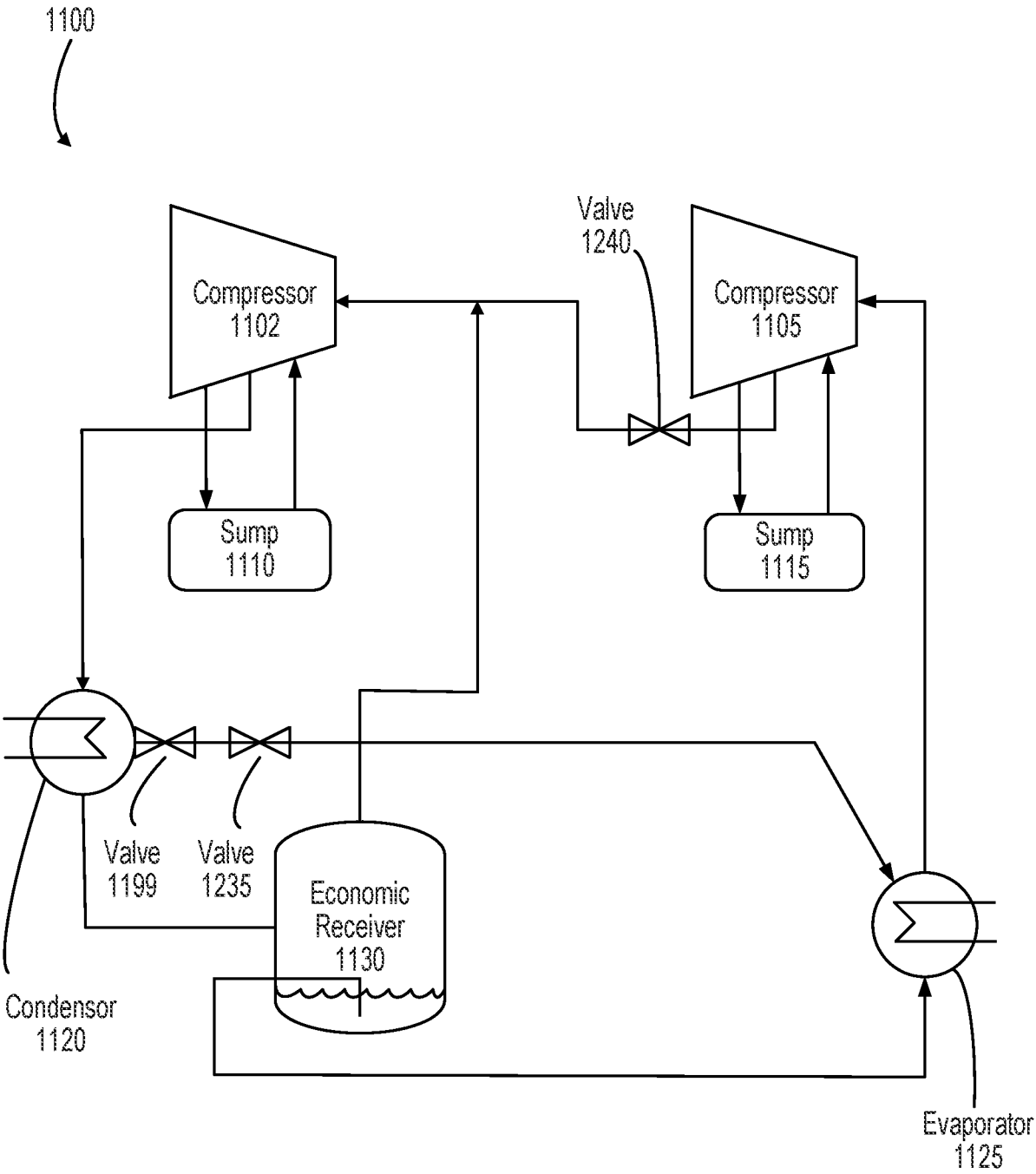


FIG. 12

INTERNATIONAL SEARCH REPORT

International application No

PCT/US2023/076259

A. CLASSIFICATION OF SUBJECT MATTER

INV. F25B31/00 F25B43/02

ADD.

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F25B

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

EPO-Internal, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 3 004 396 A (ENDRESS JAMES W ET AL) 17 October 1961 (1961-10-17) & associated description; figures 1-5 -----	1-15
X	US 3 620 038 A (MUENCH PAUL W) 16 November 1971 (1971-11-16) the whole document -----	1-15
X	US 2019/301778 A1 (ITO MASAHIRO [JP]) 3 October 2019 (2019-10-03) & associated description; figures 1-16 -----	1-15
A	US 10 197 316 B2 (JOHNSON CONTROLS TECH CO [US]) 5 February 2019 (2019-02-05) the whole document -----	1-15



Further documents are listed in the continuation of Box C.



See patent family annex.

* Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier application or patent but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

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INTERNATIONAL SEARCH REPORT

Information on patent family members

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