Office de la Propriété
Intellectuelle
du Canada

Un organisme
d'Industrie Canada

Canadian
Intellectual Property
Office

An agency of
Industry Canada

CA 2771812 A1 2011/03/03

(21) **2 771 812**

(12) **DEMANDE DE BREVET CANADIEN**
CANADIAN PATENT APPLICATION

(13) **A1**

(86) Date de dépôt PCT/PCT Filing Date: 2010/08/13
(87) Date publication PCT/PCT Publication Date: 2011/03/03
(85) Entrée phase nationale/National Entry: 2012/02/22
(86) N° demande PCT/PCT Application No.: CA 2010/001261
(87) N° publication PCT/PCT Publication No.: 2011/022810
(30) Priorité/Priority: 2009/08/24 (US61/236,171)

(51) Cl.Int./Int.Cl. *F01K 17/00* (2006.01),
F01K 17/06 (2006.01), *F01K 19/02* (2006.01),
F01K 27/02 (2006.01), *F22B 33/00* (2006.01),
F22B 33/14 (2006.01), *F22G 1/00* (2006.01)

(71) Demandeur/Applicant:
JANVIER, BENOIT, CA

(72) Inventeur/Inventor:
JANVIER, BENOIT, CA

(74) Agent: NORTON ROSE CANADA
S.E.N.C.R.L.,S.R.L./LLP

(54) Titre : PROCEDE ET SYSTEME POUR ENGENDRER DE LA VAPEUR A HAUTE PRESSION
(54) Title: METHOD AND SYSTEM FOR GENERATING HIGH PRESSURE STEAM

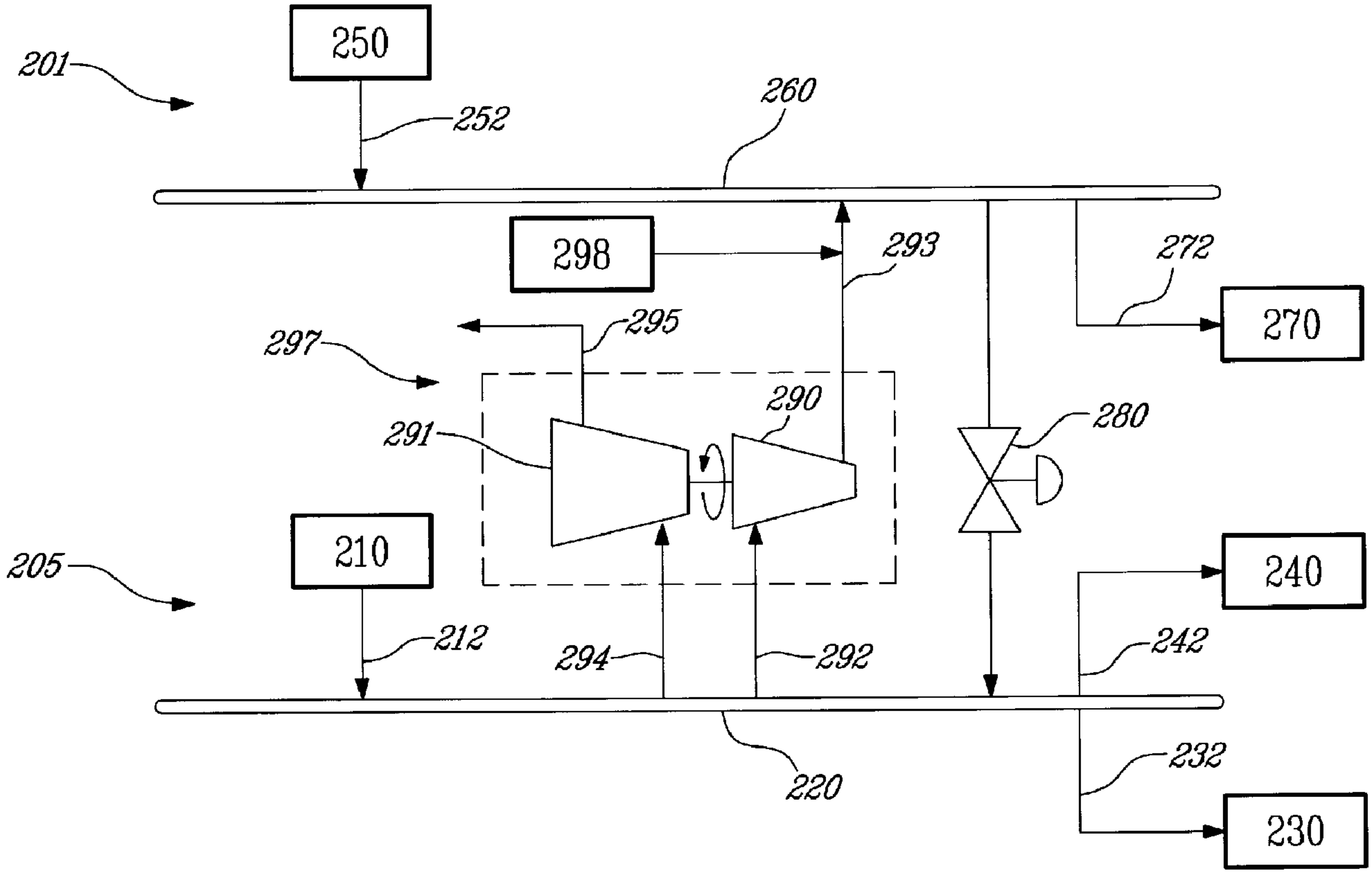

Fig-3

(57) Abrégé/Abstract:
A method and system of generating high pressure steam from a low pressure low energy steam is described and comprises providing a low pressure steam source; dividing the source into at least two streams. A first stream driving a turbine/expander

(57) Abrégé(suite)/Abstract(continued):
coupled to a steam compressor of a steam generator. The steam compressor is fed by the second stream of the low pressure steam source, and the mass flow rate of the first stream is sufficient to raise the pressure of the second stream to a desired pressure. At steady state operating conditions the first and the second stream respectively acts as the driving force for the steam compressor and the inlet feed to the steam compressor generating the high pressure steam. In another embodiment the steam driven generator comprises a thermocompressor and may include an organic thermal fluid Rankine cycle.

(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) **World Intellectual Property Organization**
International Bureau

(43) **International Publication Date**
3 March 2011 (03.03.2011)

PCT

(10) **International Publication Number**
WO 2011/022810 A1

(51) **International Patent Classification**:
F01K 17/00 (2006.01) ***F22B 33/00*** (2006.01)
F01K 17/06 (2006.01) ***F22B 33/14*** (2006.01)
F01K 19/02 (2006.01) ***F22G 1/00*** (2006.01)
F01K 27/02 (2006.01)

(21) **International Application Number**:
PCT/CA2010/001261

(22) **International Filing Date**:
13 August 2010 (13.08.2010)

(25) **Filing Language**: English

(26) **Publication Language**: English

(30) **Priority Data**:
61/236,171 24 August 2009 (24.08.2009) US

(72) **Inventor; and**
(71) **Applicant** : **JANVIER, Benoît** [CA/CA]; 4401 Coolbrook, Montréal, Québec H4A 3G1 (CA).

(74) **Agent**: **OGILVY RENAULT LLP/S.E.N.C.R.L., s.r.l.**; Suite 2500, 1 place Ville-Marie, Montreal, Québec H3B 1R1 (CA).

(81) **Designated States** *(unless otherwise indicated, for every kind of national protection available)*: AE, AG, AL, AM, AO, AT, AU, AZ, BA, BB, BG, BH, BR, BW, BY, BZ, CA, CH, CL, CN, CO, CR, CU, CZ, DE, DK, DM, DO, DZ, EC, EE, EG, ES, FI, GB, GD, GE, GH, GM, GT, HN, HR, HU, ID, IL, IN, IS, JP, KE, KG, KM, KN, KP, KR, KZ, LA, LC, LK, LR, LS, LT, LU, LY, MA, MD, ME, MG, MK, MN, MW, MX, MY, MZ, NA, NG, NI, NO, NZ, OM, PE, PG, PH, PL, PT, RO, RS, RU, SC, SD, SE, SG, SK, SL, SM, ST, SV, SY, TH, TJ, TM, TN, TR, TT, TZ, UA, UG, US, UZ, VC, VN, ZA, ZM, ZW.

(84) **Designated States** *(unless otherwise indicated, for every kind of regional protection available)*: ARIPO (BW, GH, GM, KE, LR, LS, MW, MZ, NA, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European (AL, AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HR, HU, IE, IS, IT, LT, LU, LV, MC, MK, MT, NL, NO, PL, PT, RO, SE, SI, SK, SM, TR), OAPI (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Declarations under Rule 4.17:

— *as to applicant's entitlement to apply for and be granted a patent (Rule 4.17(ii))*

— *as to the applicant's entitlement to claim the priority of the earlier application (Rule 4.17(iii))*

Published:

— *with international search report (Art. 21(3))*

(54) **Title**: METHOD AND SYSTEM FOR GENERATING HIGH PRESSURE STEAM

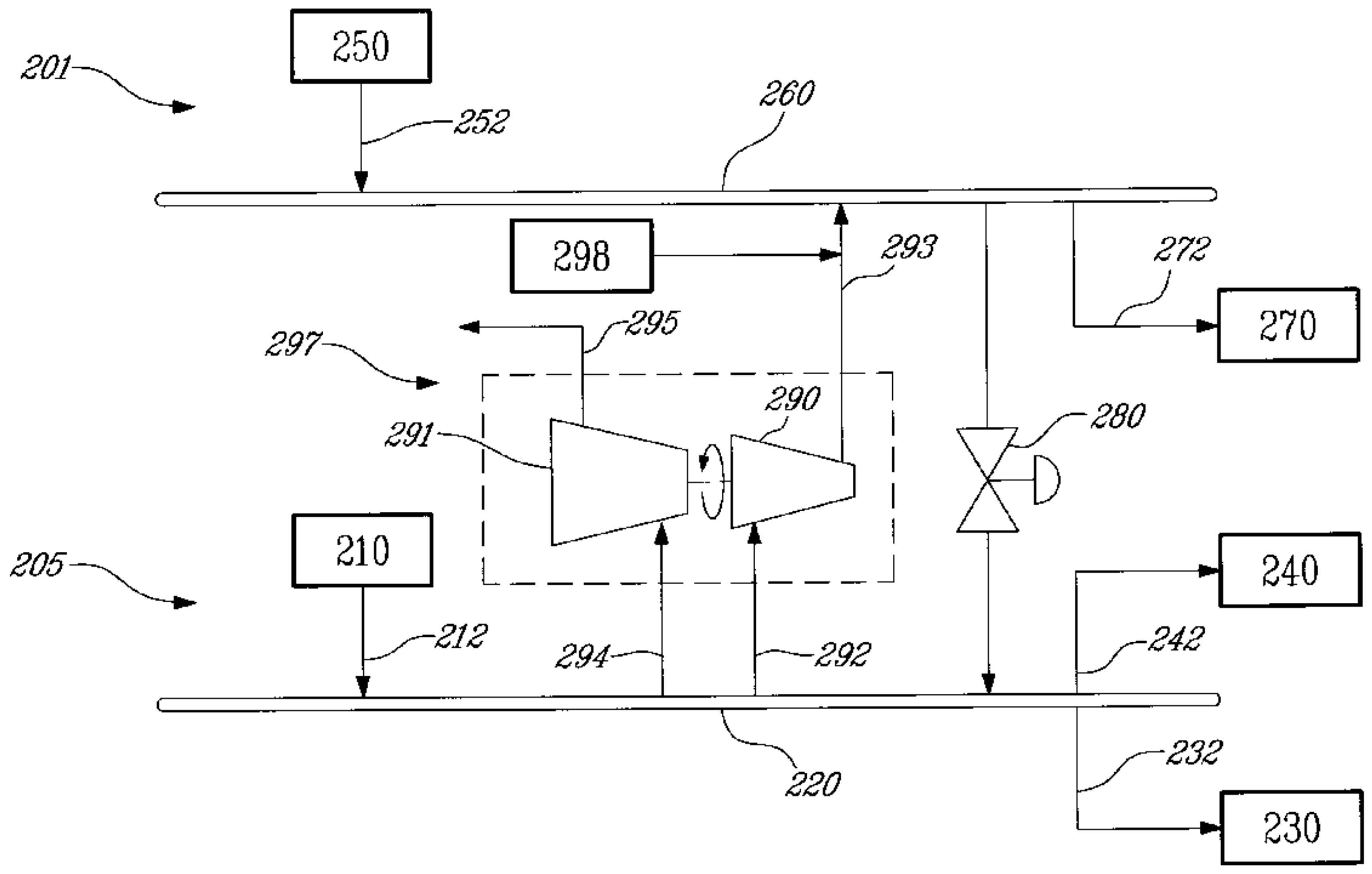

Fig-3

(57) **Abstract**: A method and system of generating high pressure steam from a low pressure low energy steam is described and comprises providing a low pressure steam source; dividing the source into at least two streams. A first stream driving a turbine/expander coupled to a steam compressor of a steam generator. The steam compressor is fed by the second stream of the low pressure steam source, and the mass flow rate of the first stream is sufficient to raise the pressure of the second stream to a desired pressure. At steady state operating conditions the first and the second stream respectively acts as the driving force for the steam compressor and the inlet feed to the steam compressor generating the high pressure steam. In another embodiment the steam driven generator comprises a thermocompressor and may include an organic thermal fluid Rankine cycle.

WO 2011/022810 A1

METHOD AND SYSTEM FOR GENERATING HIGH PRESSURE STEAM

FIELD OF THE INVENTION

The present invention relates to the generation of high pressure steam.

BACKGROUND ART

Today's process industries are being asked to be more cost and energy efficient and many are looking with increasing interest towards saving energy. These energy savings usually produce greener and more cost effective plants.

Some industries operate energy intensive processes or exothermic processes that generate a substantial amount of low grade steam during steady state operating conditions. This low grade and low pressure steam produced is used for low-value work such as, but not limited to, warming cold water streams but has generally been vented to atmosphere or condensed. However, within these same plants, some high temperature processes still require high pressure steam of 250 psig or more that is produced by dedicated high pressure steam boilers. In the past low pressure steam has been equated with low energy value.

In the past electric vapor recompression of low pressure steam has been used to generate high pressure steam from a low pressure steam source.

SUMMARY

The present invention concerns a method and a system where a low pressure steam source produces high pressure steam by compression of the low pressure steam, and substantially free of any electrical input or other external energy source when operating at steady-state conditions. The low pressure steam source serves as both the source of the high pressure steam and as the energy driver generating the high pressure steam. The present invention uses the low energy from a low pressure steam source that was thought to have little value.

Therefore in one aspect of the present invention there is provided a method of generating high pressure steam at a desired flowrate and a desired pressure from a low pressure steam source, the method comprising: providing the low pressure steam source, dividing the low pressure steam source into at least a first stream and a second stream, coupling the first stream to drive an expander/turbine, the expander/turbine linked to and driving a steam compressor of a steam generator, and feeding the second stream to the steam compressor wherein the steam compressor generating the high pressure steam, wherein the first stream has a mass flow rate sufficient to generate the energy required to raise the pressure of the second stream to the desired pressure, and wherein at steady-state operating conditions the steam generator the low pressure steam source acts as a driving force and steam input for generating the high pressure steam.

Therefore in another aspect of the present invention there is provided a system for generating a high pressure steam comprising a low pressure steam source divided into at least a first stream and a second stream, and a steam generator comprising a turbine/expander and a steam compressor driven by the turbine/expander, wherein the first stream is coupled to the turbine/expander driving the compressor, and the second stream is the input of the compressor, and wherein the first stream has a first mass flowrate and second stream has a second mass flowrate, wherein the first mass flowrate is sufficient to raise the steam pressure of the second mass flowrate to that of the high pressure steam.

BRIEF DESCRIPTION OF THE DRAWINGS

Reference will now be made to the accompanying drawings, showing by way of illustration a particular embodiment of the present invention and in which:

Fig. 1 is a process flow diagram of a high pressure steam generating system produced by a high pressure boiler (PRIOR ART);

Fig. 2 is a process flow diagram of a high pressure steam generating system that is produced at least in part by low pressure vapour recompression using an electrically driven compressor (PRIOR ART);

Fig. 3 is a process flow diagram of a high pressure steam generating system according to one embodiment of the present invention;

Fig. 4 is a process flow diagram of a high pressure steam generating system according to another embodiment of the present invention;

Fig. 5 is a process flow diagram of the high pressure steam generating system according to a further embodiment of the present invention comprising an organic Rankine Cycle; and

Fig. 6 is a Temperature vs. Entropy diagram of the Organic Rankine Cycle of Fig. 5.

DETAILED DESCRIPTION OF PARTICULAR EMBODIMENTS

Referring now to the drawings, Fig.1 represents a prior art system found in an existing process plant where high pressure steam of at least 250 psig is produced. The abbreviation "psig" is defined as: pounds per square inch gauge. The skilled person would understand that "psig" can be converted to "psia", pounds per square inch absolute by adding the 14.7 psi to the value of pressure in "psig".

The high pressure steam system 1 includes various components such as, at least one high pressure boiler 50. The boiler 50 is connected to a high pressure header 60 by at least one high pressure feed line or pipe 52. The high pressure steam header 60 is connected to the process units or users 70 in the plant by one or more steam lines or pipes represented by 72. Although only one pipe is represented by line 72 it is understood to represent numerous possible feed pipes, each transferring high pressure steam to the various users 70 in the plant.

The flow diagram of Fig. 1 includes a low pressure steam system 5 that includes a low pressure steam generator 10 connected to a low pressure steam header 20 by at least one process line or pipe 12. Any type of low pressure steam generator 10 may be envisaged that includes but is not limited to a quick-steam generator as described in PCT/CA2007/00874 and herein incorporated by reference. Pipes represented by 32 transfer the low pressure steam to various process users 30 in a process plant. In the process plants of the prior art there are also numerous low pressure steam vent pipes 42 used to discharge low pressure steam at any one of numerous vents 40 or steam condenser. These vents 40 although illustrated as deriving directly from the low pressure header 20, may also be found downstream of any one of the various process users 30 after use. A vent is also understood to be a heat sink. Although not illustrated in Fig. 1, vent pipes 42 may include a pressure regulating means such as a pressure regulating valve or steam trap.

In some process plants, the low pressure steam users 30 steam demand is lower than the low pressure steam producer 10. Furthermore, the high pressure steam from the header 60 (or other users 70) may have their pressure stepped down by means of a pressure regulating valve 80. In this way low pressure steam is provided to system 5 at higher temperature.

Fig. 2 represents a further development known in the prior art that includes electrical vapor/steam recompression. The high pressure steam system 101 of Fig. 2 may further include a high pressure steam boiler 150 connected to a high pressure header 160 by one or more high pressure steam lines or pipes 152. High pressure steam process users 170 are connected to the high pressure steam header 160 by one or more high pressure steam lines 172.

The low pressure steam system 105 of Fig.2 includes a low pressure steam generator 110, connected to a low pressure steam header 120 by one or more low pressure steam pipes 112. There are also various low pressure steam process users 130 and vents 140 connected to the low pressure steam header 120 respectively by pipes 132 and 142.

The system of Fig. 2 may also include steam pressure reducing valve 180 that reduces the pressure of the high pressure steam to generate low pressure steam.

The steam system of Fig. 2 also includes a compressor 190 that is operated by an electric motor 191. The compressor 190 is fed low pressure steam via an inlet pipe

192. The compressor 190 increases the steam pressure from the low pressure of header 120 to the high pressure of the steam header 160. The compressor will increase the steam temperature and the amount of superheat in line 193 may have to be controlled with the addition of cooling water 198. If cooling water 198 is added, the flow rate of the cooling water 198 is such that it reduces the temperature of the steam in leaving the compressor 190 in outlet 193 to that of the high pressure steam header 160. The optional addition of cooling water 198 at the outlet of the compressor 190 is typically, in the form of a fine mist or spray. The steam compressor 190 may also be a multi-stage compressor, where steam cooling in the form of heat exchange or direct water spray addition occurs between stages of compression. The addition of water 198 is adjusted to obtain a desired level of superheat in the high pressure steam system 101.

It should be noted that the reference numerals of Fig.1, Fig. 2, Fig. 3 and Fig. 4 are similar and comparable components all share the same last two digits, but in Fig. 2, Fig. 3 and Fig. 4 respectively, the reference numerals include an introductory first digit prefix 1XX, 2XX and 3XX, i.e. the high pressure steam header in Figs. 2 and 4 are respectively identified as 160 and 360.

A system generating high pressure steam according to one embodiment of the present invention is illustrated schematically in Fig. 3. This system includes features found in Fig. 1 and Fig. 2. Particularly, there is a high pressure steam system 201 that may include a high pressure boiler 250 with at least one outlet line or pipe 252 leading into a high pressure steam header 260. There are high pressure steams process users 270 that are fed via high pressure steam lines 272. High pressure steam is understood to be at least 35 psig and more preferably at least 400 psig. Although illustrated the high pressure steam boiler 250 is optional to the process.

The low pressure steam system 205 includes a low pressure steam generator or source 210 that may be any type of boiler and in a preferred embodiment is a quick-steam generator as described in PCT/CA2007/00874 and herein incorporated by reference. Low pressure steam is: at least 15 psig and more preferably at least 120 psig. The low pressure steam 210 source may have low pressure steam at various pressures.

The low pressure steam is transferred to a low pressure header 220 via a transfer line or pipe 212 from the low pressure steam boiler 210. This system includes numerous various users 230 and various vents 240 fed respectively by pipes 232 and 242. A system of generating low pressure steam from the high pressure steam by means of a pressure regulating valve 180 may also be present.

The system described in Fig. 3 uses a low pressure steam source 210 more efficiently with a steam driven generator 297. The high pressure steam generator 297 includes a steam turbine 291 coupled to and driving a steam compressor 290. The skilled person will understand that a turbine may be replaced by an expander that produces substantially the same result. The low pressure steam source 210, that is understood to be of low energy, serves as the motive force for the steam turbine 291 at steady state operating conditions, and is thus the main and possibly sole energy producer. Steady state operating conditions are understood to be operating conditions during regular operation and exclude start-up and shut-off operations.

It should be remembered that the low pressure steam source 210 may be one or more similar low pressure sources, possibly at different pressures that may be combined into a header 220 or vessel. Ideally, the low pressure steam source 210 is divided into at least two streams 292 and 294. A first stream 294 couples to and serves as a driving force used to propel the steam turbine/expander 291. A second low pressure steam stream 292 is the feed input that will go to produce high pressure steam at the desired pressure, and is fed to the inlet of the steam compressor 290.

The steam turbine 291 is designed to take an inlet steam flowrate from pipe 294 that is usually at least half of the inlet flow rate from pipe 292 into the steam compressor 290. The mass flow rate through the steam turbine 291 is at least 0.5 times that of the mass flow rate through the steam compressor 290, Preferably, the mass flow rate through the steam turbine 291 is at least 0.75 times that of the mass flow rate through the steam compressor 290. Most preferably, the mass flow rate through the steam turbine 291 is at least 1.0 times that of the mass flow rate through the steam compressor 290. The system and method presented in Fig. 3 takes advantage not only of the pressure of the low pressure steam but of the mass flow rate of the low pressure steam source 210 to move the steam turbine 291 coupled to the steam compressor 290.

It should be noted that because of internal steam decompression and the relatively high internal steam flow rate, the steam turbine 291 will typically be a larger unit than the steam compressor 290. The compressor 290 may over heat the outlet steam 293, and could require the addition of cooling water 298. If cooling water 298 is added, it is added at a flow rate that reduces the temperature of the steam leaving the compressor 290 in outlet 293 to that of the high pressure steam header 260. The addition of cooling water 298 at the outlet of the compressor 290 is optional, and is typically in the form of a fine spray. The steam compressor 290 may be a multi-stage compressor where cooling water 298 may be added as a fine spray between compression stages. This arrangement reduces the temperature after the first stage of compressor 290 and helps to reduce cost by minimizing higher temperature alloy material required for the compressor 290. In a preferred embodiment the steam compressor 290 is a multi-stage steam compressor and more preferably is a two-stage steam compressor.

Where the inlet low pressure stream 294 to the steam turbine 291 has a mass flowrate of 20 klbs/hr, and an inlet pressure of 200 psia that is allowed to fall to 15 psia within the turbine, 850 BTU/sec are transferred. Under these conditions, the steam compressor 290 can produce high pressure steam streams 298 at various flowrates and various pressures. Three possible scenarios are presented below in Table 1.

Table 1

For a steam turbine with 20 klbs/hr at 200 psia at the inlet and falling to 15 psia at the outlet the turbine, the following high pressure steam scenarios can be conceived:

	Compressor Inlet Steam Mass Flowrate (klbs/hr)	Compressor Inlet steam Pressure (psia)	Compressor Outlet Steam Pressure (psia)
Compressor 1	18	200	861
Compressor 2	40	200	381
Compressor 3	60	200	297

Fig. 5 includes an organic Rankine cycle system 500, that also uses the value and the quantity of low pressure steam to produce high pressure steam.

An organic Rankine cycle is based on the principle of using a thermal fluid with a lower boiling temperature than water. These thermal fluids include: refrigerant, hydrocarbons, organic fluids, ammonia-water mixture. Instead of using the thermal energy of steam directly, the steam is used to boil (and optionally superheat) the thermal fluid. The thermal fluid operates at a high pressure and high gas density (higher pressure than the steam that is being used as the energy steam).

The organic Rankine cycle (ORC) system 500 illustrated in Fig. 5 replaces the steam turbine illustrated in Fig. 3 with a closed loop system where the thermal fluid passes from liquid to vapour and back. ORCs have a very low energy-to-power conversion efficiency and as such are rarely considered for all but geothermal application. The efficiency is typically under 15% compared to modern power plants with +50% efficiencies. However, the goal here is not to efficiently recover energy but to extract the small amount of energy required to contact low-value steam to high-value steam.

The system 500 includes the various elements disclosed previously particularly there is a high pressure steam system 401 that may include a high pressure boiler 450 with at least one outlet line or pipe 452 leading into a high pressure steam header 460. There are high pressure steams process users 470 that are fed via high pressure steam lines 272. High pressure steam is understood to be at least 35 psig and more preferably at least 400 psig. Although illustrated the high pressure steam boiler 450 is understood to be optional to the process. The low pressure steam system 405 includes a low pressure steam generator or source 410 that may be any type of boiler and at least one line 412 to a low pressure steam header 420. The system 500 will also typically include low pressure users 430 via lines 432, and low pressure vents and line 440 and 442. The system may also include a control valve 480 between the high pressure header 460 and the low pressure header 420 that can be used to regulate the amount of steam in both headers.

Fig. 5 includes a Organic Rankine Cycle (ORC) 500 that includes a thermal fluid loop that drives the steam compressor 490. In Fig. 5, the high pressure generator 497 comprises a steam compressor 490 mechanically linked to a turbine (or expander) 491 that operates with the thermal fluid. As in previous embodiments, the compressor converts low pressure steam to high pressure steam, and may be a two-stage steam compressor.

At the outlet of turbine 491, there is stream 502 that is vaporized thermal fluid. The stream 502 may optionally be cooled in a heat exchanger 510 where a cooling medium 505 enters the heat exchanger via stream 504 and leaves via stream 506. The cooling medium 505 may be water although other alternatives are also possible, and will be further discussed.

The thermal fluid leaves the heat exchanger via stream 512 before entering a condenser 520. As the name suggests the condenser 520 takes the cooler vapour from the heat exchanger 510 and condenses it to a liquid state. The condenser may be of any type and a fan condenser may be used where as is the cooling medium. The condenser cooling fluid 526 may be any suitable fluid and in generally water. The condenser cooling liquid enters the condenser via line 527 and exits via line 528.

The cooled thermal fluid stream 522 from the outlet of the condenser 520 is transferred to the inlet of a thermal liquid circulation pump 530. This pump 530 generates sufficient pressure to circulate the thermal liquid through the ORC system 500 cycle.

At the outlet 532 of the pump 530 the liquid is optionally transferred to a preheater 540. That may be heated in a variety of ways. In one alternative there is a preheater 540 that heats stream 532 before evaporation with condensate stream 552 from the evaporator 550. In this case the heating may be done with steam condensate 552 from the evaporator. Other heating options include other heat sources such 535 via line 544 that may be a different steam source or possibly a second thermal transfer fluid. In a preferred embodiment units 510 and 540 are combined into a single heat exchanger, where cooling stream 505 is replaced by stream 532. This combined heat exchanger 510/540 therefore does not require condensate 552 to preheat the condensed thermal fluid.

The preheated thermal stream 542 enters the evaporator 550 where low pressure steam line 494 from the low pressure steam header 420 is used in an evaporator 550 to heat the thermal fluid to vapour, and this stream may be controlled with a control or manual valve 499.

Here the ORC begins again when the vapourized thermal fluid at the outlet 501 of the evaporator 550 enters the inlet of the turbine/expander 491. Thus, stream 494 is coupled with the turbine 491 to actuate the compressor 490. It is understood that the steam at pressure lower than the steam header 420 may also be used.

The control of the ORC of Fig. 5 may be conducted in a variety of ways. One such method is to control the organic-side evaporator 550 pressure by passing a varying quantity of organic vapors 501 into the turbine/expander 491 to increase power to the compressor 490. This may be accomplished with a PIC (pressure integral controller) or a pressure valve and increased heat transfer The organic level in the evaporator 550 is controlled using the level control valve 538. The steam pressure in the evaporator 550 will dictate the internal steam temperature in the evaporator 550 - when more energy is needed, the steam-side of the evaporator will be set at higher pressures. The outlet steam condensate valve 558 will open to control a level in the evaporator. This valve will be modulated to ensure that the evaporator is within its heat absorption capacity, where the water from the evaporator to the preheater should be cool enough to ensure no evaporation in the preheater 540. Preferred operation requires that most of the thermal fluid will dry when depressurized and the turbine 491 can have fluid at saturated conditions at the inlet 501 without having condensation in the back-end 502.

A second control strategy for the ORC is the organic-side evaporator 550 pressure is controlled by passing more or less organic vapors to the turbine 491 that once again increases power to the compressor 490. The organic level in the evaporator 550 is controlled using the level control valve 538. The steam pressure in the evaporator 550 will dictate the internal steam temperature in the evaporator, when more energy is needed, the steam-side evaporator will be at higher pressures. The outlet steam condensate valve 548 will open to control a level in the evaporator. This valve 548 will close to ensure evaporator 550 is within its heat absorption capacity. The water stream 552 from the evaporator 550 to the preheater 540 should be cool enough to ensure no evaporation in the preheater 540.

Fig. 6 is a temperature vs. entropy chart of an organic Rankine cycle for refrigerant R245fa where: between points a and b the organic fluid is heated in an optional preheater 540 or in the boiler 410; between the points c-d the thermal fluid is boiled and goes from saturated liquid to saturated vapor; between the points d-e the vapor is superheated; between e-f there is a decompression through a turbine 491 or expander; between f-i the gas leaving the expander/turbine 491 is cooled and condensed and part of the cooling can be used to heat the cycle between the points a-b.

In the optional case, where heat exchanger 510 and 540 are combined the flow of the ORC is directed from the stream 522 a subcooled organic liquid at 246 BTU/lb pressurized at 15 bars (pump 530 outlet); 532 is heated in the combined

510/540 heat exchanger with stream 542 leaving; Saturated organic liquid at 354 BTU/lb pressurized at 15 bars (before getting in the evaporator - with or with a preheater 510/540 the heating and boiling occurs in the evaporator 540. At point e the fluid is superheated, the superheating is optional set and the cycle may operate from point d at 500 BTU/lb to the organic turbine inlet 501. At the point f, represents the superheater organic fluid at 469.8 BTU/lb or the turbine (expander) outlet 502 to optional heat exchanger 510/540. Point i represents a saturated fluid at 249 BTU/lb at the 522 condenser outlet to pump inlet. The above cycle as the following technical attributes: High Pressure fluid at 15.5 bars, Low Pressure fluid at 2 bars; Inlet enthalpy of organic fluid in vapor generator 246 BTU/lb; Outlet enthalpy of superheated gas leaving generator 500.68 BTU/lb; and Outlet enthalpy of superheated fluid exiting expander 469 BTU/lb. The cycle represented in Fig. 6 with the without the optional heat exchanger 510/540 is 12%.

The methods and steam generating systems represented in Fig. 1, Fig. 2, Fig. 3 and Fig. 4, are compared in the following description and Examples.

EXISTING REFINERY - Fig. 1

An existing refinery has a system of high pressure and low pressure steam as illustrated in Fig 1. The low pressure steam source 10, that is a heat recovery steam generator, has a mass flow rate of 160 klbs/hr at a pressure of 120 psig (klbs = 1000 lbs) where the consumption of all low pressure steam users 30 is a total of 100 klbs/hr. The remaining 60 klbs/hr produced is vented completely in the process at 40 or may be used to heat low temperature streams of Fig. 1. It should be noted that in this refinery high pressure steam is not normally used to produce low pressure steam and as such the pressure regulating valves 80 is closed.

The required mass flow rate of high pressure steam for the refinery is 20 klbs/hr at a pressure of 900 psig for users 70. High pressure boiler 50 is meeting the demand by producing 20 klbs/hr of steam at 900 psig. The operating cost for producing 900 psig steam via the boiler 50 is approximately at 10US$ / klbs.

The estimated annual operating cost of the high pressure steam boiler in the refinery of Fig. 1 is in the order of:

20 klbs/hr X 10 US$/klbs X 24 hr/day X 300 days/yr = 1,440,000 US$.

EXAMPLE 1 - ELECTRICAL STEAM RECOMPRESSION

If we now consider including the electrical compressor 190 illustrated in Fig. 2 into the refinery of described above, at least 14.9 klbs/hr of the low pressure steam can be fed to an electrically driven compressor 190. The compressor 190 that increases the temperature and pressure of the low pressure steam source to the required 900 psig. The process of Fig.2, also uses 5.1 klbs/hr of cooling water 198, that is added into stream 193 at the outlet of the compressor 190. The amount of low pressure steam vented at 140 is reduced to 45.1 klbs/hr, from the 60 klbs/hr vented in Fig. 1 at 40. The electric compressor 190 used is a two-stage compressor. Thus with the combined flow of 14.9 klbs/hr of steam and additional cooling water 198, of 5.1 klbs/hr the total of 20 klbs/hr of 900 psig steam can be produced to meet the total high pressure needs of the plant.

The operating cost for the electric compressor 190 is:

939 BTU/sec X 3600 s/hr X 24 hr/day X 300 d/yr = 24,310 MMBTU/year X 293 kWh/MMBTU X 0.05 US$/kWh = or 356,000 US$/yr.

The value of 939 BTU/sec, used in the previous calculation, represents the ideal energy flow needed from the motor 191 to compress 14.9 klbs/hr of steam from 120 psig to 900 psig.

The system of Fig. 2, saves up to 1,080,000 $US/yr for the refinery as compared with the high pressure steam boilers of Fig.1.

EXAMPLE 2 - LOW PRESSURE STEAM DRIVEN GENERATOR FOR 900 psig HIGH PRESSURE STEAM PRODUCTION

This example proposes to include a steam driven generator 297 into the refinery of described above. The process of Fig. 3 uses even more of the surplus low pressure steam (i.e. 60 klbs/hr) than in Examples 1 and 2 above. Here, part of the provided low pressure low energy steam source 210 is divided into two streams that are fed to the high pressure steam driven generator 297. A first stream 294 of 25.9 klbs/hr of low pressure steam is sent to turn the steam turbine 291 via line 294. A second stream 292 from the low pressure steam source 210 is fed to the steam compressor 290 via line 292. The steam turbine 291 is coupled to the steam compressor 290 and compresses the 14.9klbs/hr of low pressure steam to high pressure low energy steam at 900 psig.

The process includes an addition of 5.1 klbs/hr of cooling water 298 after steam compression to reduce the temperature of the steam in line 293 to that found in header

260. Thus, the process of Fig. 3 reduces the amount of vented low pressure steam 240 to only 19.2 klbs/hr, a reduction of 25.9klbs/hr of steam mass flow vented when compared to Example 1 of an electrically driven steam compressor. The inclusion of the steam driven generator 297 that includes a low pressure low energy steam turbine 291 as a prime mover reduces the operating cost of high pressure steam on an annual basis to virtually zero and saves an additional 356,000 $US/year compared with the Example 1 describing electric vapor/steam compression.

It is understood that at steady-state operating conditions the low pressure steam source 210 serves as the sole or main greater than 50% driver or motive force for the steam compressor 290. It is however understood that a start-up conditions or when there is insufficient steam, the steam driven generator 297 may be driven partly or wholly by another prime mover such as an electric motor (not illustrated).

EXAMPLE 3 - LOW PRESSURE STEAM DRIVEN GENERATOR FOR 85 psia HIGH PRESSURE STEAM PRODUCTION

Example 3 describes a steam driven generator 297 of described above in of Fig. 3, in a second process plant having different process requirement and conditions. Example 3 illustrates the range of process conditions that can be employed using the method of the present invention.

The second process plant in Example 3, has high pressure steam in a header 260 at 85 psia. The high pressure steam user 270 has a steam flow requirement of 9.0 klbs/hr for the second process plant.

The low pressure steam source 210 produces some 86.2 klbs/hr of steam at a pressure of 45 psia. The process users 230 require 50 klbs/hr of low pressure steam. The process user 270 requires 9 klbs/hr of high pressure steam. Without the high pressure steam driven generator 297, the high pressure boiler 250 would supply 9 klbs/hr of steam to the user 270.

With an excess of low pressure steam, low pressure steam source 210 is once again divided into two streams from header 220 that are fed to the high pressure steam driven generator 297. A first stream in line 294, has a mass flow rate of 9.3 klbs/hr of low pressure steam turns the steam turbine 291. A second stream of 8.5 klbs/hr from the low pressure steam source 210 is fed to the steam compressor 290 via line 292. The steam turbine 291 is coupled to the steam compressor 290 and compresses the low pressure steam to high pressure steam at 85 psia. With low grade steam at 45 psia and high value steam at 85 psia, the compression ratio for compressing low grade steam to high pressure steam in this Example is 1.89. The system of Example 3 does not include a pressure regulating valve 280.

The process includes an addition of 0.5 klbs/hr of cooling water 298 after steam compression to reduce the temperature of the steam in line 293 to that found in header 260.

In Example 3, the compressor size is large enough to accommodate the steam demand (i.e 9 klbs/hr). Only a relatively small amount of power is transferred because the compression ratio necessary for the process is small and the energy required is small (45 psia to 85 psia).

With the high pressure steam driven generator 297, the high pressure boiler 250 will supply no steam to user 270 and the estimated annual operating savings for Example 3 are in the order of:

9klbs/hr X 10 US$/klbs X 24 hr/day X 300 days/yr = 648,000 US$

EXAMPLE 4 - LOW PRESSURE STEAM DRIVEN GENERATOR FOR 85 psia HIGH PRESSURE STEAM PRODUCTION INCLUDING A THERMOCOMPRESSOR

Fig 4 illustrates a further embodiment of the present invention and includes a steam driven generator 397 comprising a thermocompressor 396.

The process conditions for Example 4 are those of the second process plant, previously described in Example 3. In Example 4, the low pressure source 310 produces 86.2 klbs/hr of steam at 45 psia that enters header 320 via line(s) 312. The low pressure process users 330 via line(s) 332 require 50 klbs/hr, and the vented low pressure steam 340 via line 342 is 10 klbs/hr.

The required high pressure steam in header 360 is at a pressure of 85 psia, at a mass flow rate requirement is 9 klbs/hr via line(s) 372 to process users 370. Once again the overall compression ratio is 1.88.

The steam driven generator 397 includes at least three components: a steam turbine 391, a steam compressor 390 and a thermocompressor 396.

The low pressure steam source 310 is divided in Example 4 into three streams that are fed to the high pressure steam driven generator 397. A first stream of 9.3 klbs/hr of low pressure steam is sent to turn the steam turbine 391 via line 394. A second stream of 5 klbs/hr from the low pressure steam source 310 is fed to the steam compressor 290 via line 392.

Example 4 includes a third stream 396a at a flow rate of 4 klbs/hr that is fed to the thermocompressor 396.

The steam turbine 391 is coupled to the steam compressor 390 and compresses the low pressure steam to higher pressure steam of 138 psia. The outlet stream 393 from the compressor 390 serves as the motive steam for the thermocompressor 396 that draws and mixes with low energy steam from stream 396a into the thermocompressor. The sum of the outlet stream 393 and the inlet stream 396a will meet the required 9 klbs/hr of 85 psia high pressure steam that is transferred to the header 360 via line 399.

Thus, the steam compressor 390 of the steam driven generator 397 produces steam at a higher pressure steam than that in header 360. This higher pressure steam stream 393 has a smaller mass flow rate when compared to the overall high pressure steam requirement of users 370. The higher pressure steam entrains a low pressure stream 396a into the thermocompressor and in doing so increases the generator 397 overall compression capacity. The steam compressor 390 is therefore smaller and less costly than the steam compressor 290 of Example 3 and affords a further capital cost benefit to Example 4.

In Example 4, the compressor size is smaller than that of Example 3 but accommodates the steam demand (9 kpph) of the second process plant. The steam driven generator 397 having a combination of turbine 391, compressor 390 and thermocompressor 396 allows for a further reduction in capital cost of at least 20% where, instead of compressing all of the steam, the steam flow to the compressor 390 represents only a portion of the low pressure steam that is compressed and feeds line 399. The capital cost of the steam driven generator 397 of Example 4 are approximately 20% to 35% less than the steam driven generator 297 of Example 3. The larger capital costs savings occur are lower compression ratios around 1.2.

The embodiments of the invention described above are intended to be exemplary. Those skilled in the art will therefore appreciate that the foregoing description is illustrative only, and that various alternate configurations and modifications can be devised without departing from the spirit of the present invention. Accordingly, the present invention is intended to embrace all such alternate configurations, modifications and variances which fall within the scope of the appended claims.

CLAIMS:

1. A method of generating high pressure steam at a desired flowrate and a desired pressure from a low pressure steam source, the method comprising:

providing the low pressure steam source,

dividing the low pressure steam source into at least a first stream and a second stream,

coupling the first stream to drive a turbine, the turbine linked to and driving a steam compressor of a steam generator, and

feeding the second stream to the steam compressor wherein the steam compressor generating the high pressure steam,

wherein the first stream has a mass flow rate sufficient to raise the pressure of the second stream to the desired pressure, and

wherein at steady-state operating conditions the steam generator the low pressure steam source acts as a driving force and steam input for generating the high pressure steam.

2. The method of claim 1, wherein the steam compressor is at least a two-stage steam compressor.

3. The method of claim 1 or 2, wherein the low pressure steam source may be made up of several low pressure steam sources.

4. The method of any one of claims 1 to 3, wherein the low pressure steam source is produced by a quick-steam boiler.

5. The method of any one of claims 1 to 4, further comprising adding water to the high pressure steam to obtain the high pressure steam stream at the desired level of superheat.

6. The method of any one of claims 1 to 4, wherein coupling the first stream to the turbine comprises

feeding the first stream to the turbine to drive the compressor.

7. The method of any one of claims 1 to 4, wherein coupling the first stream to the turbine comprises

feeding the first stream to an evaporator,

vaporizing a thermal fluid within the evaporator,

feeding the vaporized thermal fluid to the turbine to drive the compressor.

8. The method of claim 7, further comprising

recirculating the thermal fluid to a condenser, condensing the thermal fluid to liquid,

pumping the condensed thermal fluid to the evaporator.

9. A system for generating a high pressure steam comprising

a low pressure steam source divided into at least a first stream and a second stream, and

a steam generator comprising

a turbine/expander and

a steam compressor driven by the turbine/expander, wherein the first stream is coupled to the turbine/expander driving the compressor, and the second stream is the input of the compressor, and

wherein the first stream has a first mass flowrate and second stream has a second mass flowrate, wherein the first mass flowrate is sufficient to raise the steam pressure of the second mass flowrate to that of the high pressure steam.

10. The system of claim 9, wherein the steam compressor is at least a two-stage steam compressor.

11. The system of claim 9 or 10, further comprising a quick-steam generator.

12. The system of any one of claims 9 to 11, further comprising several low pressure steam sources connected to the low pressure source.

13. The system of any one of claims 9 to 12, further comprising a water source at the outlet of the high pressure steam compressor.

14. The system of any one of claims 9 to 13, wherein the first stream is coupled to the turbine through an evaporator feeding a vaporized thermal fluid to the turbine to drive the compressor.

15. The system of claim 14, further comprising

a thermal fluid condenser connected to the outlet of the turbine,

a pump connected to the condenser returning the thermal fluid to the evaporator.

16. The method of claim 1, wherein the steam generator comprises a thermocompressor, using the high pressure steam from the steam compressor as a motive force.

17. The system of claim 9, wherein the steam generator comprises a thermocompressor using high pressure steam from the steam compressor as a motive force.

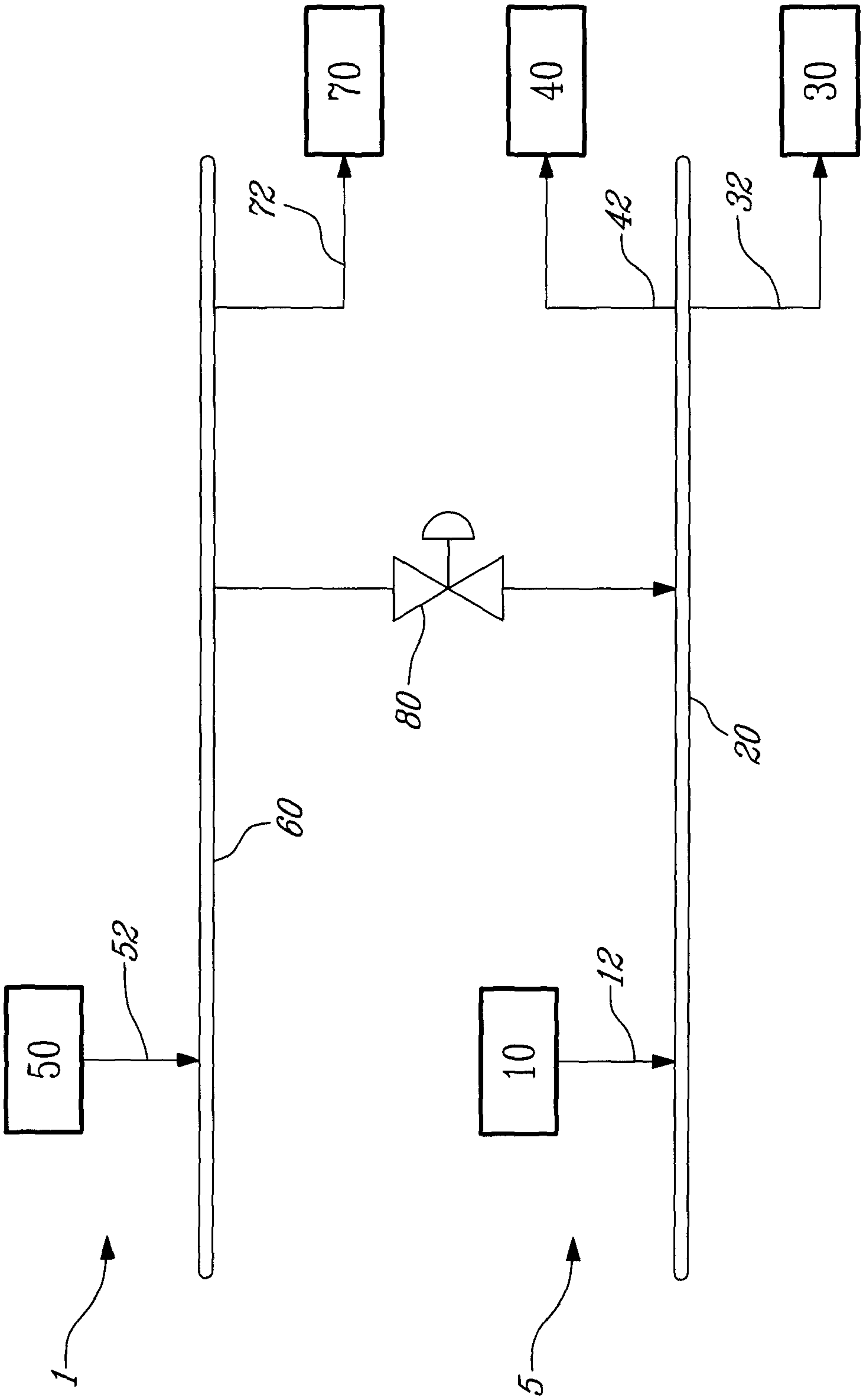

Fig-1 PRIOR ART

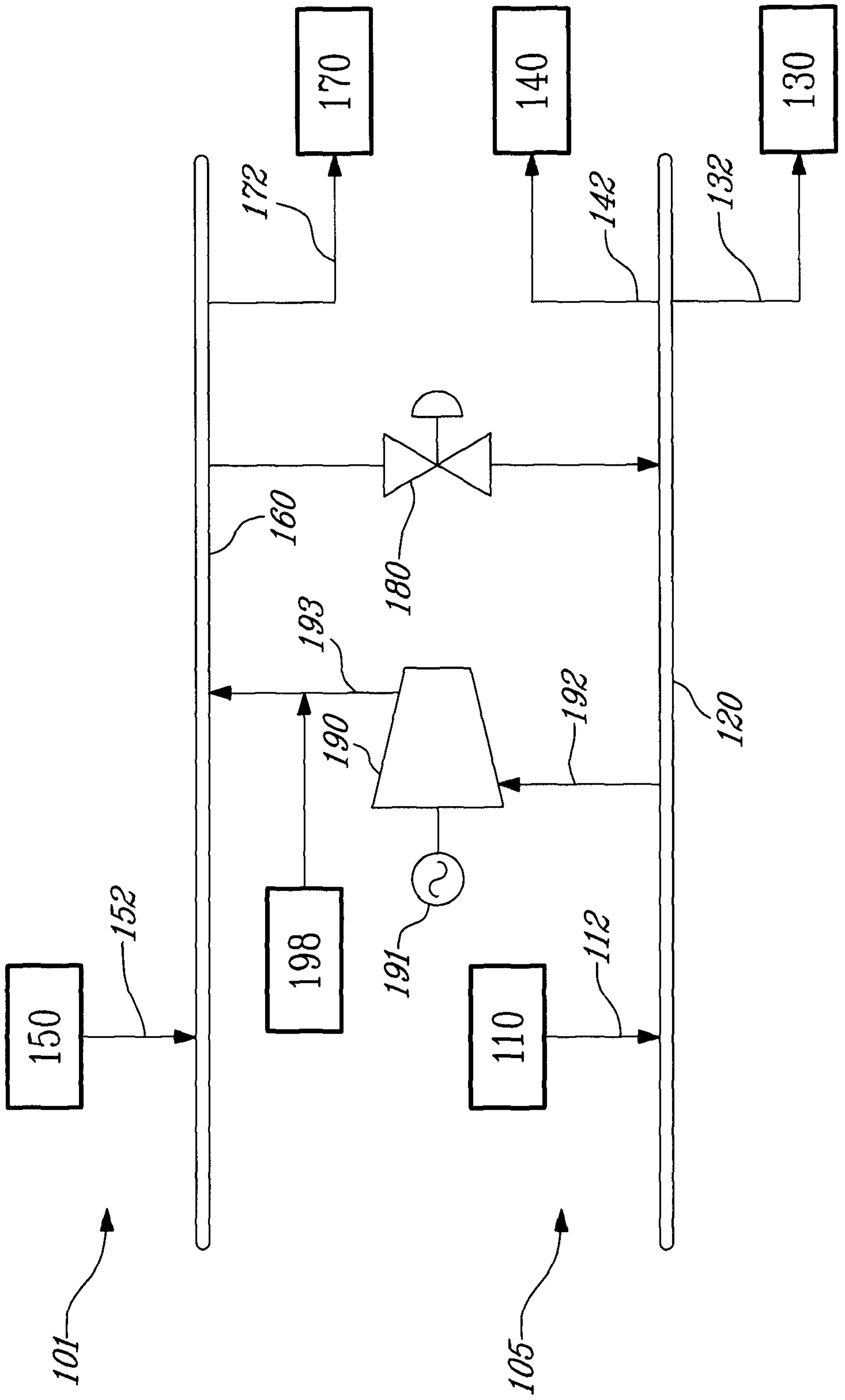

Fig-2 PRIOR ART

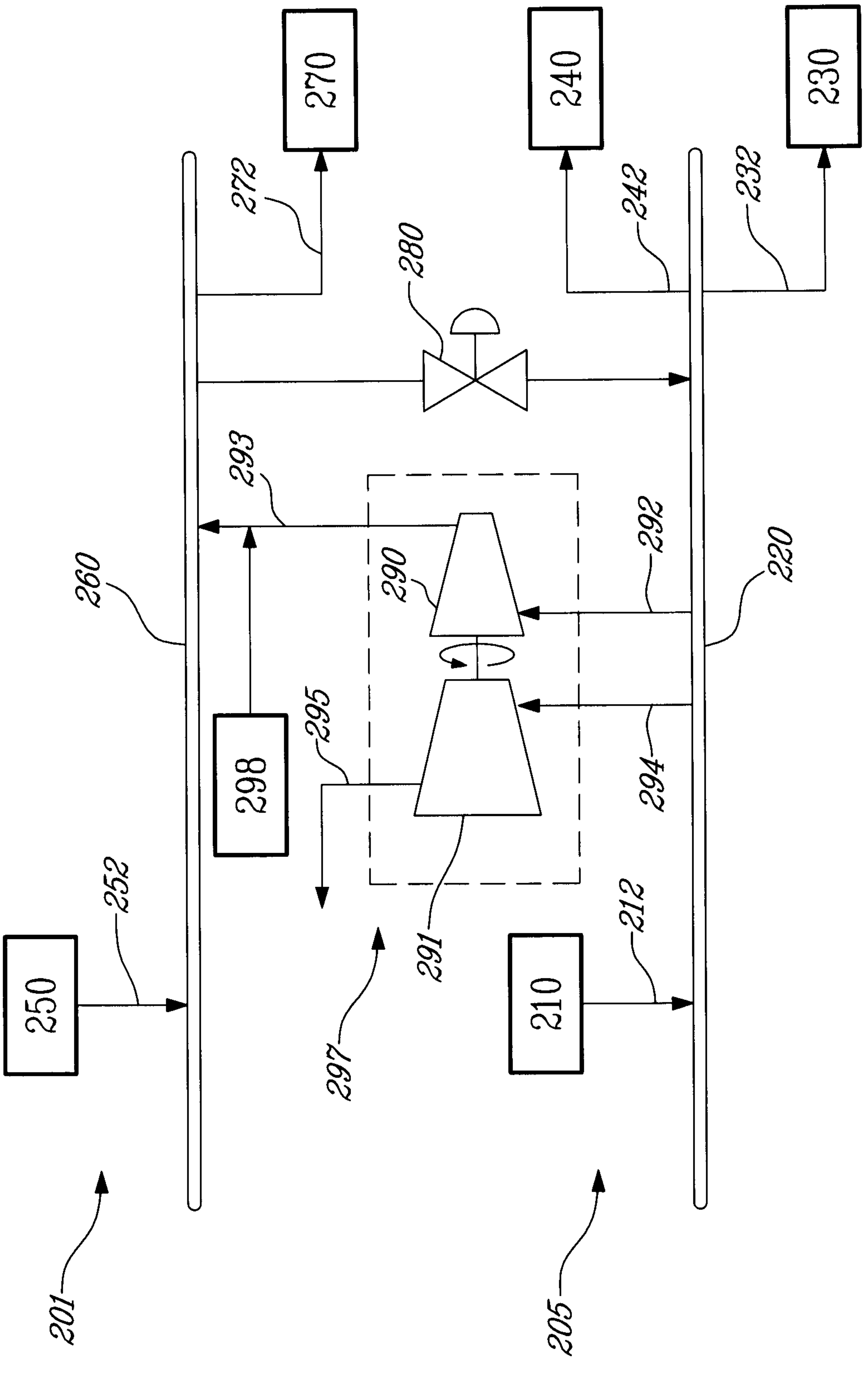
270
240
230
272
242
232
280
293
292
290
260
220
298
295
294
291
252
212
250
210
297
201
205

Fig-3

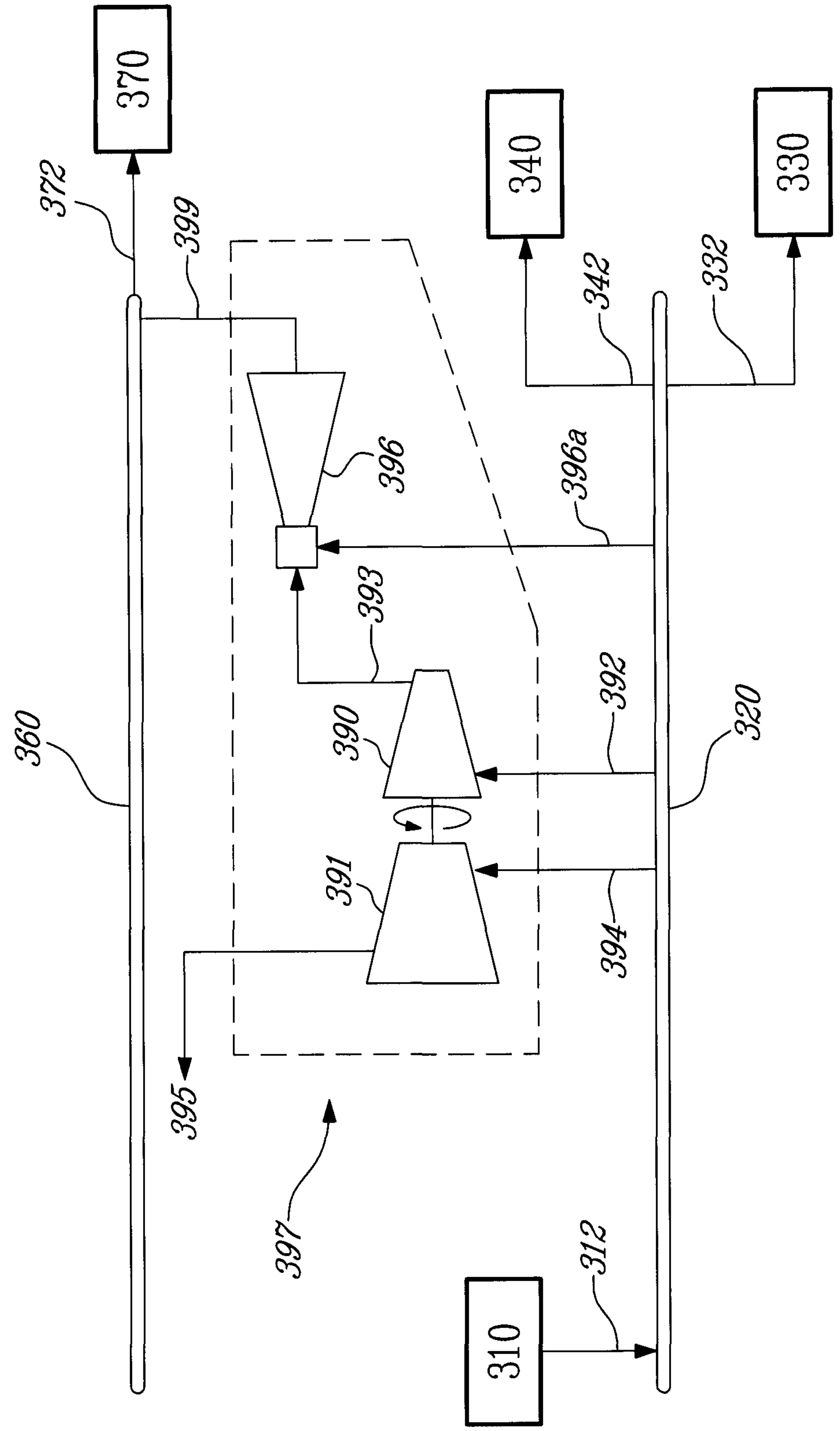
370
372
399
340
330
342
332
396
396a
393
392
390
320
360
391
394
395
397
312
310

Fig-4

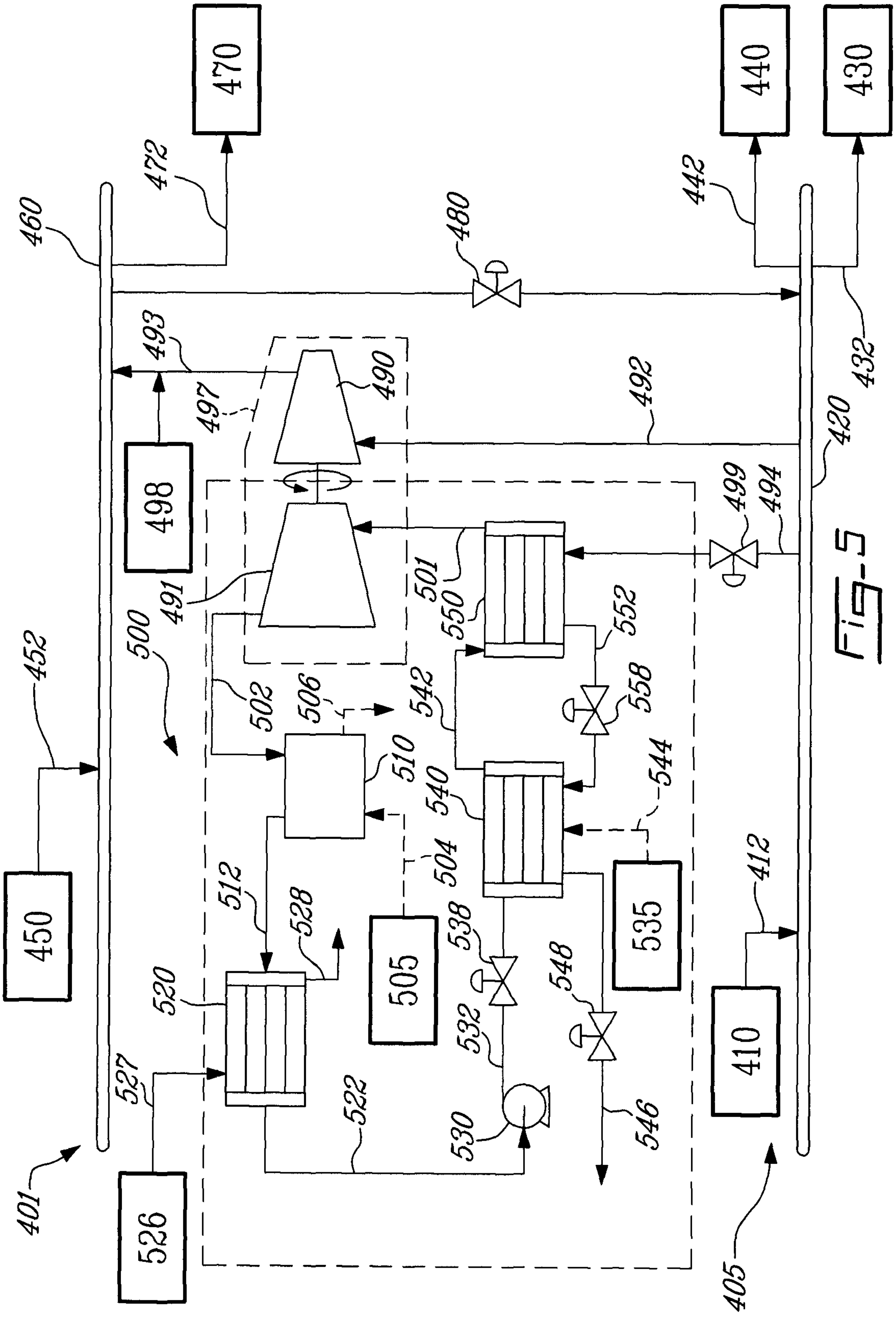
470
472
460
480
442
440
430
432
420
493
490
497
492
498
499
494
491
501
550
552
500
452
502
506
542
558
510
540
544
504
450
512
528
538
535
520
505
548
532
412
410
527
522
530
546
401
526
405

Fig-5

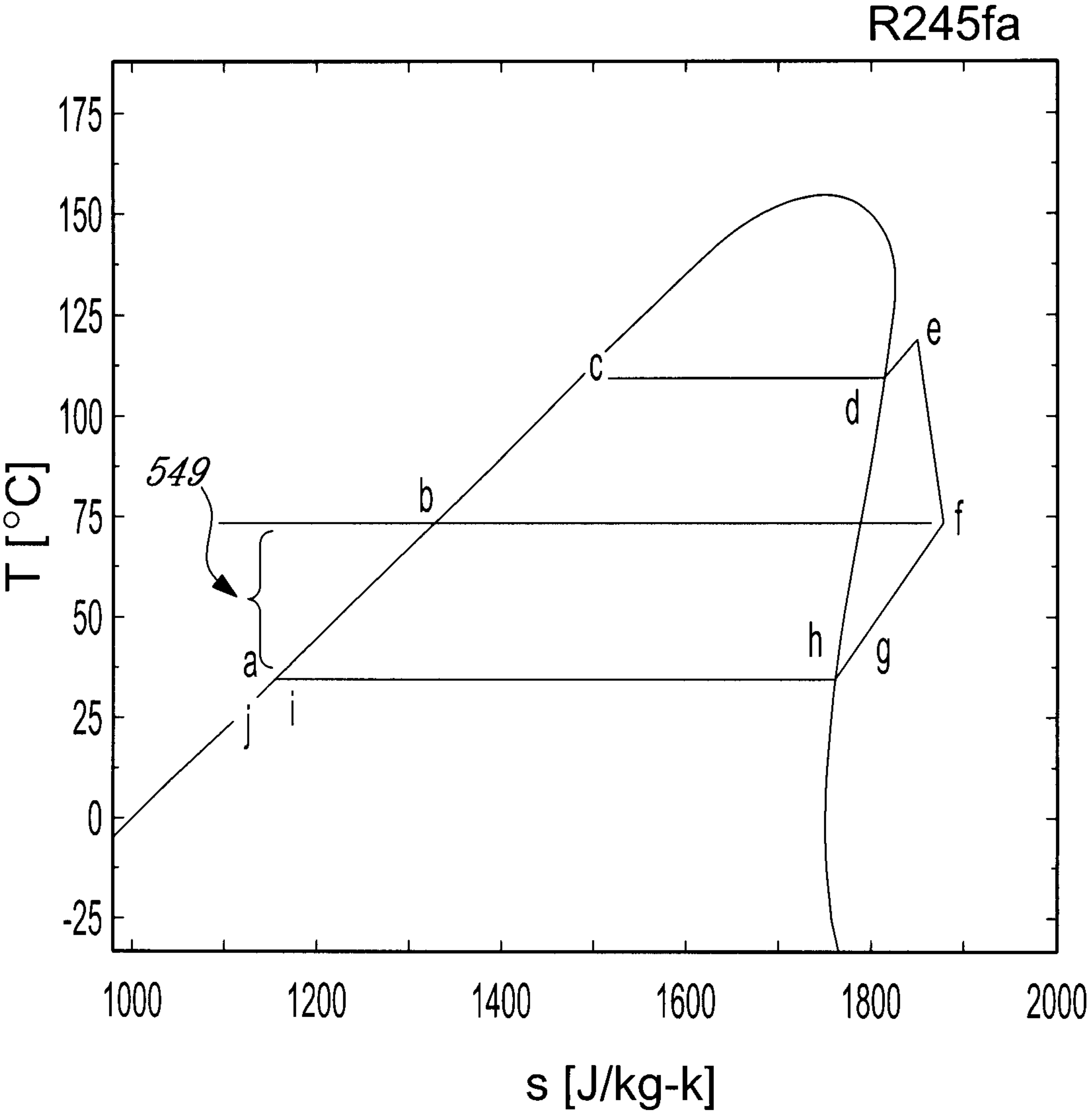
R245fa
T [°C]
s [J/kg-k]
175
150
125
100
75
50
25
0
-25
1000
1200
1400
1600
1800
2000
549
a
b
c
d
e
f
g
h
i
j

Fig- 6

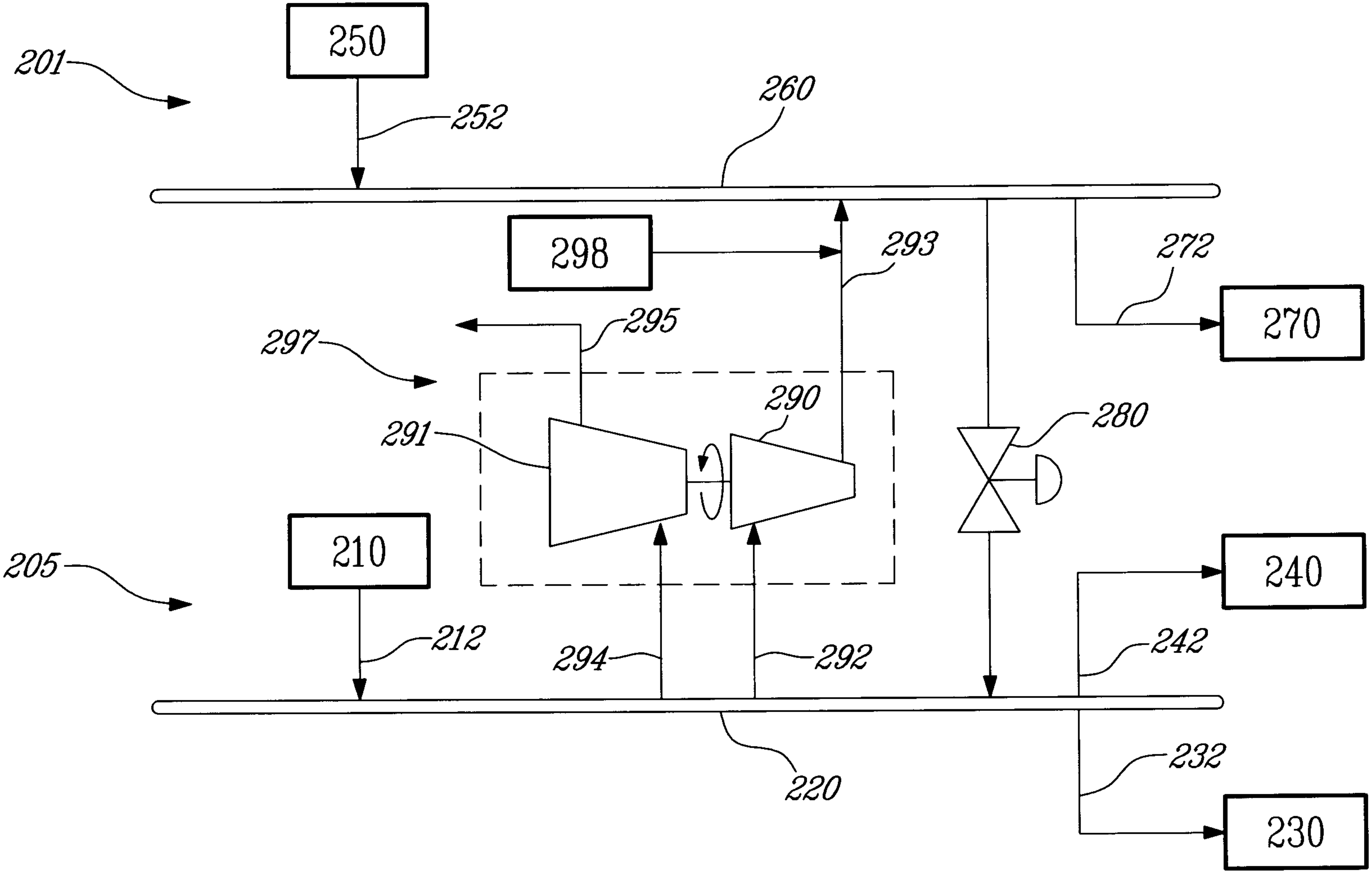
250
201
252
260
298
293
272
270
297
295
290
280
291
210
240
205
212
294
292
242
232
220
230

Fig-3