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(54) **Title**: METHOD AND DEVICE FOR COLLECTING SOLAR ENERGY

(57) **Abstract**: The present invention relates to the collection of solar energy with one or more one-dimensional linear tracking collectors, for instance but not only with a parabolic solar throughs and with the help of an appropriate number of gaseous heat carrying units. If the size of the solar field exceeds the dimensions appropriate to an efficient gas system, the hot gas may be transferred with the help of suitable heat exchangers to another medium which may be generally, but not exclusively, a liquid and which can transport the heat at further distances and eventually store with the help of thermal accumulation the energy of the solar field, in view of an appropriate application which could be electric energy production with conventional methods or other forms of industrial heat.

METHOD AND DEVICE FOR COLLECTING SOLAR ENERGY

DESCRIPTION

FIELD OF THE INVENTION

The present invention relates to the collection of solar energy with one or more one-dimensional linear tracking collectors, for instance but not only with a parabolic solar throughs and with the help of an appropriate number of gaseous heat carrying units. If the size of the solar field exceeds the dimensions appropriate to an efficient gas system, the hot gas may be transferred with the help of suitable heat exchangers to another medium which may be generally, but not exclusively, a liquid and which can transport the heat at further distances and eventually store with the help of thermal accumulation the energy of the solar field, in view of an appropriate application which could be electric energy production with conventional methods or other forms of industrial heat.

The method is especially applicable for the production of relatively high temperatures, typically in the range from 300 °C to about 600 °C.

STATE OF THE ART

As well known, since over a century, the parabolic linear solar collector has received a lot of attention. It generally comprises concentrating mirrors and linear receiver tubes arranged in a onedimensional focal point of a solar collector. Several different methods have been widely discussed in the literature, amongst which we indicate:

(a1) the so called parabolic through, in which mirror and receiver tube are directed with the help of a suitable assembly rotating around one axis, so that the direct solar radiation is constantly directed normal to the mirror plane and the radiation falling on the mirror is focussed to the receiver tube solidly held with the mirror.

(a2) the so called Fresnel lens, in which a series of horizontally laying, narrow and long mirrored segments, flat in the major dimension but parabolic in the shorter orthogonal direction are continuously but independently rotated along the major axis with the help of an appropriate assembly toward a parallel horizontal line, several meters above, where the receiver tube collects the light accumulated by the many mirror segments.

These mirrored units have generally a lateral width of several meters (typically from 4 to 8 metres for alternative (a1) and about 20 metres for alternative (a2)). Each receiver tube concentrating the solar light has approximately the same length as the mirror in the dimension orthogonal to the direction in which it provides concentration. The tube may or may not include a evacuated radiation sensitive inner tube and an outer tubular transparent jacket generally made of glass to ensure vacuum insulation. The surface of the tube must absorb strongly the incoming solar light but it should preferentially become reflecting at longer wavelengths, in order to reduce the amount of re-emitted infrared radiation. The tubes will reach a high temperature during operation and appropriate methods have to be introduced in order to compensate for the thermal elongation during operation.

As an example, according to the indicative values of the above mentioned parabolic through geometry (a1), a large power plant producing 300 MWatt electric at the peak of the solar direct radiation (DNI) may require the deployment of the order of 500 km of tubes. For tubes of 7 cm internal diameter, the indicated 500 km length corresponds to a volume of 1900 m3, to which one has to add the ancillary piping reaching to a total inner tube volume of maybe 3000 m3.

Until now, a number of different fluids have been used in order to ensure such a heat transfer. In particular they have included:

(b1) an appropriate organic liquid thermal fluid (oil), generally applicable up to temperatures less than 400 °C. Such organic liquids are highly flammable and considered toxic in many countries. In view of its considerable amounts, even a small fraction of accidentally spilled liquid may cause serious environmental consequences.

(b2) An alternative, based on superheated steam has also been developed. At the cold side of the tube, water is initially in a liquid form. Before the end of the path, it becomes steam and an appropriate separating unit performs the separation from liquid to vapour. A drawback of this solution is the complex thermal response of the steam and the necessity of high gas pressures, which may reach values of hundreds of bars for higher operating temperatures.

(b3) In order to overcome these problems, it has been also chosen to fill the whole piping with an appropriate molten salt, typically a eutectic mixture of Na and K or Na/K/Ca nitrates. Such medium is environmentally much more benign and not toxic. The highest operating temperature may be limited to around 600 °C due to the spontaneous decomposition into nitrites and a resulting increased corrosion of ordinary steel. The lowest operating temperature is limited by the solidification of the salt, at approximately 240 °C for the most common salt. An elaborate procedure is necessary at all times in order to maintain the fluid above solidification, in particular during the night-time and other off-periods. The most sensitive part of a molten salt system are undoubtedly the large and exposed assemblies of tubes running along the focal point of the light collecting mirrors and the manifolds between solar segments in the extended solar field. Any accidental, localized freezing of the molten salt will permanently block the flow of the coolant and the salt will have to be entirely liquefied again before restoring operation of the segment.

The variability of the solar yield can be decoupled from the energetic requirements of the user in simple, efficient and economical ways, with the help of the thermal storage of a previously accumulated heat, holding the solar energy up to several hours. Amongst the well known thermal energy storage methods we refer to the following:

(c1) A method with two large and thermally insulated tanks A and B, respectively at temperatures of the outlet T_A and inlet T_B of the solar field. The secondary liquid coming from the heat exchangers fills progressively the hot tank A with liquid from the cold tank B heated through the heat exchanger. The subsequent energy utilization process proceeds along the opposite path taking the liquid from T_A to T_B through the delivery of the produced energy. The process is fully reversible and nearly 100% efficient.

(c2) The so-called "thermocline" method in which a single thermally insulated volume is filled with pebbles, i.e. a large number of small (spherical or otherwise shaped) solid chunks imbedded in the running fluid, either gas or liquid. Three regions at distinct temperatures are as follows, in order of succession, during the heat accumulation process: a hot region in which the pebbles have reached about the same temperature as the thermally carrying fluid (either gas or liquid); a "thermocline" region in which the heat of the fluid is locally transferred to the pebbles and a cold region, all the way to the end of the stack, in which pebbles and the fluid are both cold. During the heat storage process, the thermocline region moves from the beginning till the end of the stack, at which point the storage is full. In order to recover the stored energy, the flow proceeds in the reverse order, from cold to hot. Although one tries to keep the lowest conductivity between the pebbles at different locations and the highest conductivity between pebbles and locally surrounding fluid, the process is not entirely reversible and a fraction of the heat may not be recovered.

(c3) In order to store thermal energy, materials which may undergo a reversibly a liquid/solid and a solid/liquid change at a given temperature are for instance same salts like NaNO3, KNO3, KOH, Salt ceramics

($NaCO_3$-$BaCO_3$/MgO), NaCl and so on.

DESCRIPTION OF THE INVENTION

The present invention relates to a process for high temperature solar thermal energy collection comprising the use of a pressurized gas, re-circulated by means of one or more blowers or compressors and heated by means of a number of solar one dimensional concentrating mirrors with a linear receiver tube in which the circulated pressurized gas absorbs heat from the concentrated solar radiation.

It is a goal of the present invention to provide a gas filled tube in order to collect the radiating solar heat with the help of a suitable array of tubes to a collecting manifold with the highest thermal efficiency. In general, the higher the temperature of the heat transfer gas, the more efficient the solar power system. Thus heat transfer gas and associated systems capable of withstanding higher temperatures are desirable.

It is a goal of the present invention that the easy operability of such a (inert) gas driven system ensures an unparalleled robustness of the necessarily very extensive and complex array of collecting pipelines and manifolds, for which reliability and ease of maintenance are of primary importance. In comparison with other known methods, because of the harmless nature of the heat carrying medium the over all environmental impact is strongly reduced.

It is another goal of the present invention that the choice of a suitable gas, because of the near constancy of the heat coefficient (cp) and the absence of physical changes when traversing the path across the tube, ensures the largest possible difference between the lower and higher temperatures of the receiver, fully exploiting the boundaries otherwise determined by the intrinsic performance of the tube, in conditions of remarkable uniformity of the thermo-dynamical behaviour.

It is another goal of the present invention to ensure a corresponding geometry of the inner tube which minimises the pressure drop and hence the pumping work required in order to ensure a smooth flow of the gas through the pipe. In particular the additional pressure losses of the joints and bents between the segments of the tube should be reduced to a minimum, for instance locating the tubes in a fixed position during the daily rotation of the mirror assemblies and introducing appropriate methods in order to compensate the considerable thermal elongation of the tubes during operation.

The tube is not illuminated uniformly, since the heat production is strongly oriented in the direction of the concentrating mirror. As a consequence of this asymmetry, also the gas temperature, due to the finite conductivity of the gas may become radially asymmetric. It is a further goal of the present invention to ensure that the lack of uniformity of the illumination of the sunlight produces the smallest temperature gradient across the cross section of the tube, thus making the temperature gradient across the tube as small as possible.

In order to further improve the uniformity of the gas temperature inside the tube, the heat transfer between the walls and the gas may be improved with an appropriate enhancement the exchanging surfaces, at the cost of some increases of the pumping power.

In order to further improve the uniformity of the gas temperature inside the tube, a slow rotational motion may be impressed to the gas with suitable means that rotate periodically the orientation of the gas with respect to the tube.

The choice of a gas, because of the already mentioned uniform heat capacity as a medium and the consequent possibility of a very wide temperature range increases considerably the specific heat capacity of the storage. As well known, the thermal energy storage capacity is increasing with the difference of temperatures between the inlet and outlet of the receiver segment, closely related to the lower and upper temperatures in the storing medium.

Any friendly gas can be used as proposed in the invention. Amongst widely available commercial gases, Hydrogen, Helium, CO_2, Dry Air, and Nitrogen have been found to be valid candidates with rather similar performances. We remark that Dry Air contains Oxygen, which might give rise to some corrosion at high temperatures and that Hydrogen is flammable. These are only indicative choices and a number of other gases may also be used.

The optimal gas can depend on the application parameters, site conditions or size of the plant, taking into account their different thermo-hydraulic properties, cost and management and corrosion constraints. The optimum temperature of the gas at the collector outlet will depend on many factors, including characteristics of the absorber and effects on user efficiency, but usually will be between 300 °C and 600 °C.

According to an advantageous embodiment, the optimum number of gas-cooled collectors in each closed circuit and their interconnection (in parallel, in series, combined) basically is determined by the minimization of the specific cost of the delivered thermal energy to what will be termed as the central user block, taking into account investment costs, pumping power, thermal efficiency, etc. The smaller circuit that makes sense would comprise two collectors in series, while the bigger circuit considered feasible could have up to hundreds of collectors, combining series and parallel interconnections.

Since the volumetric heat capacity of gases is much lower than the one of liquids, a tube beyond a certain length may introduce an excessive pressure drop with unacceptable auxiliary power consumption in the circulating blowers. In the present invention, in order to keep the pressure drops at an acceptable level, the problem has been solved introducing a modular approach with an appropriate number of gas driven units, each one of sufficiently modest pressure losses. Heat is then later collected with the help of additional heat exchangers and an appropriate secondary transferring fluid, for instance a molten salt or some other fluid. The heat transfer fluid can be any medium that has the capability to transfer heat with small losses and thermally maintain the heat in the medium such as a molten salt, a liquid metal or a metal alloy. Each of the closed gaseous circuits will include a heat exchanger where the pressurized gas is cooled, transferring heat to a molten salt or other suitable fluid, derived through a fluid piping network to a central user block, from which a flow of fluid is pumped back to each heat exchanger through a cold fluid piping network.

The present invention exploits the feature that while the widely exposed and unprotected sunlight receiving tubes along the collecting mirrors are operated with a compressed gas, the subsequent, thermally insulated secondary transferring fluid is maintained easily in a liquid form during all environmental conditions.

In spite of the limitations in size of the pressurized gas closed circuits employed for solar heat collection, the invention can be applied to practically any size of solar thermal high temperature application, from some hundreds of kWatt to some thousands of MWatt, thanks to the efficient heat transport system provided with the help of the secondary fluid, thus integrating in a single plant a very high number of gas circuits.

Acording to an embodiment of the present invention, the solar field for the collection and transport of solar energy of a larger installation to the central user block is composed of a series of closed circuits of pressurized gas, including gas-molten salt or other suitable fluid heat exchangers and gas blowers, and a secondary fluid piping system.

The central user block receives a flow of hot molten salt or other suitable secondary fluid from the solar field and delivers to this the required secondary fluid flow, in such a way that the central user block has available for any use a variable thermal power obtained from solar energy. This thermal power can be used for any process requiring heat, but the most typical application will be using it as heat source of a thermal cycle for electrical power generation.

Given the variability of the thermal power delivered by the solar field it is possible to include in association with the gas driven plant in the central user block also a thermal storage system, for instance according to one of the different alternatives already described in points (c1), (c2) or (c3) in the state of the art section. Thermal storage may be very efficient and it can be easily extended over several hours in order to cover a major fraction of the directly unavailable time. It allows decoupling thermal power use from thermal power delivery, what is especially interesting in the case of application to electrical power generation, in order to stabilize power production, follow the demand curve or deliver power during cloudy or night periods.

Thermal storage system may be driven either directly by the compressed gas or, if necessary, as it is the case for larger installations, by the already described secondary transferring fluid, for instance a molten salt or some other fluid. Based on the fact that molten salt is currently the best technical and economical technology for thermal storage, it may be based on the proven two-tank technology already described in point (c1) (hot tank/cold tank) or, in the near future, on the promising thermocline one-tank technology (point (c2)) or eventually on future materials which may undergo a liquid/solid phase change (point (c3)), either driven by the pressurized gas or by the secondary transferring fluid.

In the most typical application to electrical power generation, the practical lower size limit will be determined by the efficiency and cost of the power block, being estimated around 1 MWatt electric.

According to an example of the invention, this type of small power plants will be more feasible when combined with heat supply for district heating or industrial users or with desalination of water, as cogeneration plants. For this type of small generation or cogeneration plants, the power block can be based on the conventional steam cycle (or on other commercial alternatives like ORC cycles) with a Rankine cycle conversion system, but, given the high temperature of the available thermal power, it makes possible applying a regenerative Brayton cycle conversion system with low pressure ratio. This cycle, especially if includes an intercooling of the compression, can provide similar or higher efficiency than the steam cycle at small power levels, while providing a significant amount of heat at temperature levels around 100 °C and better options for unattended operation.

Finally the reduced mass of gas filled tubes allows a faster transition during operational thermal changes.

A first aspect of the invention is the process for collecting solar energy according to claim 1. A second aspect of the invention is the device for collecting solar energy according to claim 19. Both independent claims are included in the description by reference.

Dependent claims 2 to 18 and Dependent claims 20 to 23 and the combinations given by the dependencies are also included by reference in this description.

Claims 24 to 29 refers to a system having a thermal storage component, which are also included by reference to this description.

Still further embodiments and advantages will become apparent from the consideration of the ensuing description and accompanying drawings and specific examples are intended for purposes of illustration only and are not to limit the scope of the invention.

BRIEF DESCRIPTION OF THE TABLES

Table 1 and 2 are included at the end of the description.

Table 1. Indicative parameters for a parabolic trough configuration with a mirror lateral width of 5.75 m and Helium, CO_2, Air and Nitrogen at 20 atm and 30 atm. The tube length is 100 m and the inner diameter is 9 cm.

Table 2. Indicative parameters for a Fresnel lens configuration with a mirror lateral width of 24 m and Helium, CO_2, Air and Nitrogen at 15 atm and 25 atm. The tube length is 100 m and the inner diameter is 18 cm.

BRIEF DESCRIPTION OF THE DRAWINGS

The objects, features and advantages of the invention will now be illustrated in more detail with the aid of the following description of the preferred embodiments, with reference to the accompanying figures in which:

Figure 1 is the general layout of the light collecting tube, according to a first embodiment, receiving the concentrated solar radiation from the parabolic mirror. The pressurised gas is circulated along the collecting tube and it is closed through to an application dependent unit.

Figure 2 is the P-v diagram of the gas driven loop relative to the transfer of the heat from the light collecting tube to the application dependent heat exchanger. A mechanical compressor, isoentropically re-compressing the gas is required to circulate the pressurized gas.

Figure 3 shows three graphs (3a, 3b, 3c) of theoretical mechanical work due to the gas along the collecting tube, as a function of the tube length, the inner diameter and the initial gas pressure for different gases, starting from the main parameters of Table 2. The percent fraction of mechanical power in units of the nominal heat power delivered to the collecting tube is plotted. In figure 3a, the X-axis is the tube length, in figure 3b, the X-axis is the inner diameter; and, in figure 3c, the X-axis is the initial pressure. In 3a, 3b and 3c, Y-axis is the Mechanical fractional nominal work (%).

Figure 4 shows another embodiment wherein the hot gas is transferred prior to its utilization to another medium which may be generally a harmless liquid and which can transport the heat at further distances. As shown in FIG. 4, it may be advisable to segment the gas tubing arrays in a modular arrangement, transferring the accumulated heat to an appropriate heat carrying fluid with the help of a (modular) array of heat exchangers.

Figure 5 is the layout of the gas from the solar collector coupled to a heat storage unit made of a pair of thermally insulated vessels containing a heating fluid.

Figure 6 is the layout of the gas from the solar collector coupled to a heat storage unit made of a thermally insulated vessel filled with a large number of small size pebbles directly heated by the gas.

Figure 7 is a layout of the gas from the solar collector driven directly from air with a Brayton cycle conversion system. The use of a Brayton cycle eliminates the need for a steam Rankine conversion system reducing a significant amount of equipment, though at a significant reduction of the plant efficiency. An optional pebble bed storage system is included.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

The following description relates to illustrated, exemplified and preferred embodiments of the invention. As will be apparent, many of the particular parts of the invention are not necessarily to practice the invention in all its embodiments, and many of the particular embodiments that are described have equivalents that may be substituted to achieve the objects of the invention. Unless an element of the invention is specifically recited as essential to the invention, it is not required for all embodiments, and the present disclosure contemplates and encompasses a full range of equivalents for that part of the invention.

In one broad aspect the invention comprises solar concentrators having at least one concentrator unit, each unit including one reflective surface capable of directing and concentrating radiation to an absorbing surface wherein the absorbing surface includes means for converting at least some of the radiation into an appropriate energy generating unit capable of removing the heat from the surface with a suitable gas at an appropriate pressure.

In a related aspect the invention is able to comprise additional means capable of transferring the power absorbed by the gas to heat disposal means and back again with the help of one or more conduits, including eventually other transport mediums and appropriate heat exchangers. The heat collection comprises the use of one or more appropriate, eventually different, heat storage mediums of suitable volume in order to smooth out the supply of energy and/or eventually of a mixed operation backed up by an auxiliary power generator, operated whenever necessary for instance with a fossil or otherwise produced fuel.

Another aspect of the invention is the process for collecting the solar energy according to claim 1.

The objects, features and advantages of the invention will now be illustrated in more detail with the aid of the following description of the preferred embodiments, with reference to the accompanying Figures.

First, an example of a pressurized gas closed circuit comprising several gas-cooled troughs in a single modular unit will be described.

The basic circuit of the gas cooled layout is shown in FIG. 1. The circuit, eventually one of the many modular units, comprises a number of pairs of branches 2 (comprising two collectors 5, 6) of gas-cooled one dimensional, linear concentrating mirrors 1, where the focal point of the gradually heated gas focussing tube 3 is located.

The highly conductive main metallic collecting tube 3 is designed in order to withstand the gas pressure contained in its inside. The conductivity of the collecting tube ensures however that heat is transferred all along the circumference, though with some temperature gradient.

The outer walls of the collecting tubes 3 are preferably uniformly coated with a specific deposit that should be as absorbing as possible for the solar incoming light and as reflective as possible to the re-emitted infra-red radiation, thus minimising the radiant secondary losses toward the outside.

As shown in FIG. 1, the long collecting tube 3 is progressively increasing the temperature of the gas from its lowest inlet temperature T_{min}, gradually up to the highest temperature T_{max}. Already with a smooth inner tube 3 pipe the uniformity of the gas is generally excellent. Some fins for instance in the form of helixes of long lead may be added in the inner walls, in order to further improve the radial uniformity of the gas temperature across the collecting tubes 3, at the cost of some increases of the pumping power of compressor 7. In order to further improve the uniformity of the gas temperature along the collecting tube, a slow rotational motion may be eventually impressed to the gas with suitable means that rotate periodically the orientation of the gas with respect to the collecting tube 3.

The gas-cooled linear collectors 5, 6 may be generally connected in parallel half-branches 2, each with two or more parabolic linear collectors 5, 6 in series, according to a design optimized to minimize the piping interconnections and their corresponding thermal and pressure losses. A series of regulation valves 4, one for each tube 3 at the low temperature end of branch 2, control the flow of the gas. They are included as an optional accessory for balancing the outlet temperatures from the parallel branches 2. In FIG. 1 only a few half branches 2 are shown, but in practice either more or less sets may be chosen, depending on the application, where the gas flows from collectors 5 to collectors 6. Generally several additional collectors 5 are connected to a common well thermally insulated master inlet line 8 for the incoming cold gas. The outgoing hot gas from the corresponding collectors 6 is brought to the well thermally insulated master outlet line 9. A mechanical compressor (blower) 7 is inserted in the cold master inlet line 8, restoring the pressure losses in the circuit. The master gas loop is completed through the utilization application dependent power generating network N, where the produced heat is recovered and which will be discussed in detail in the subsequent Figures. After the heat has been removed in the generating network N from the hot master outlet line 9, the heat transfer gas is transported back to the cold master inlet line 8 for reuse.

The Pressure-volume idealized diagram of the gas circulation is schematically shown in FIG. 2. Starting from the point A, corresponding to the entry in collector 5 of focussing tube 3, the gas is gradually heated by the concentrator mirror 1 at a nearly constant pressure, the (small) losses being due primarily to the friction in the tube 3, until it reaches the highest temperature in point B. The gas is then loosing additional pressure in the well thermally insulated master outlet line 9 until it reaches the point C where a heat utilization network N, again at an almost constant pressure is recovering the heat with the appropriate secondary transfer line. The pressure losses in the segment CD are due to the inevitable friction through the heat utilization network N. At the end of the utilization network N, the gas is loosing additional pressure through the thermally insulated master inlet tube 8 until it reaches the point E, where the mechanical compressor (blower) 7 is isoentropically recompressing the gas in such a way that the loop is closed at point A.

The present embodiment uses a suitable dry gas. Amongst widely available commercial gases, Hydrogen, Helium, CO_2, Dry Air, and Nitrogen have been found to be valid candidates with rather similar performances. We remark that Dry Air contains Oxygen, which might give rise to some corrosion at high temperatures and that Hydrogen is flammable. These are only indicative choices and a number of other gases may also be used.

In order to compare quantitative results, indicative parameters have been chosen, corresponding to two typical configurations, summarized in Table 1 and Table 2:

Parameters of Table 1 correspond to the first embodiment represented in Figure 1, previously described. A parabolic trough arrangement along the lines already described in (a1) in the state of the art section, in which mirror 1 and receiver tube 3 are so that the direct solar radiation is directed normal to the mirror 1 plane and the radiation falling on the mirror 1 is focussed to the receiver tube 3 solidly held with the mirror 1. The thermo-dynamical values are primarily dependent on the structure of the parabolic mirror 1 and of the coating of the outer layer of the collecting tube 3 and are here taken as reasonable input parameters to the present configuration. The values (perfect gas approximation) in Table 1 are for an inner collector diameter of 9 cm, a length of L = 100 m and an effective lateral collecting tube width of 5.75 m normal to the solar incidence. The nominal incoming solar power on the solar tube 3 has been set to W0 = 362 kWatt and thermal variation of the gas flow is adjusted such as Tmin = 290 °C and Tmax = 550 °C. The fractional reradiated infrared power is 9.8 % for the nominal solar power.

Parameters of Table 2 correspond to a second embodiment very similar to the first embodiment of Figure 1, but substituting the parabolic troughs with a Fresnel lens arrangement along the lines described in (a2) in the state of the art section, in which a series of horizontal narrow and long mirrors 1, flat in the major dimension but parabolic in the shorter orthogonal direction are independently rotated along the major axis toward a parallel line, several meters above, where the receiver tube 3 collects the light accumulated by the many mirror 1 segments. The values (perfect gas approximation) in Table 2 are for an inner collector diameter of 18 cm, a length of L = 100 m and an effective horizontal lateral collecting tube width of 24 m. The nominal incoming solar power on the solar tube 3 has been set to W0 = 1.3 MWatt and thermal variation of the gas flow is adjusted such as Tmin = 290 °C and Tmax = 550 °C. The fractional re-radiated infrared power is 8.7 % for the nominal solar power.

Results for Helium, CO_2, Dry Air and Nitrogen are displayed for typical pressures for each corresponding concentrating method, of 20 and 30 Atm in Table 1 and of 15 and 25 Atm in Table 2. Gases are subdivided in three very similar groups, namely CO_2, (Nitrogen + Dry Air + Hydrogen), and (Helium + Argon), which exhibit growing gas speeds throughout the path along the collecting tube 3. The traversal time for the typical collecting tube length L = 100 m is very short, of the order of a few seconds.

The gas dependent massive flow rates in kg/s are, as expected, nearly independent of the gas pressure (they are exactly so for a perfect gas). The iso-entropic compressor 7 work WF due to friction in the tube 3 is typically a fraction of a percent of the nominal incoming solar power W0 on the tube 3. Additional and comparable losses, as shown in FIG. 1, are due to the connecting tubing (3, 8 and 9) and to the friction through the heat exchanger in the utilization network N, which are likely to about double the losses through the gas circuit of FIG. 2. Therefore the total fractional gas isentropic work for instance for a Fresnel configuration and Helium at 15 Atm may add up to about 1% of the nominal incoming solar power on the tube 3. Taking into account that this is electric power, an additional multiplicative factor of about three should be finally introduced in comparison with the solar power, with the conclusion that the gas dependent toll, for instance with respect to a liquid coolant, should be of the order of a few percent, smaller but comparable to the inevitable losses due to re-irradiated solar power from the tube 3, also indicated in Tables 1 and 2.

Assuming for instance the general parameters of Table 2 with the value of 0.5 percent for the theoretical iso-entropic mechanical work (1.5 ÷ 2 percent for the initial solar thermal work) we find that the corresponding pressures are 6.2 atm for Hydrogen, 13.9 atm for Helium, 16.6 for CO2 and 18.8 atm for Nitrogen and Dry Air. The value for Argon is much higher, well in excess of 45 atm and it should be generally excluded since it is a relatively heavy mono-atomic gas.

As expected, the isentropic work WF due to the friction through the gas is a rapidly varying function of the gas nature and pressure, the diameter of the tube d and of its length L. It also grows as the cooling massive flow grows in the case of a smaller temperature difference between the beginning and the end of the collecting tube. Typical results are shown in FIG. 3 for a number of different gases.

The best gas in this respect appears to be Hydrogen, which however is flammable and chemically reactive in some circumstances. The attractiveness of

CO2, Helium and Nitrogen as heat conducting gases lies in the facts that these gases are safe, are relatively easy to handle, are plentiful and cheap (with some small reservation for Helium). Dry Air contains a significant amount of Oxygen that may cause some chemical alteration of the inner walls of the collecting tube.

Gas coolant may be carriers of foreign particulate matter. The particulate matter, such as dust, salt and soot can be introduced or accumulated in the various elements of the closed gas loop. They can be trapped adding a suitable filter, which removes all but the smallest particles. The gas flow in the collector tube 3 is generally fully laminar (small Reynold's number) and these problems are generally not serious, except perhaps for the cases of accidentally ruptured elements.

As an empirical law, the fractional isentropic work of the tube for a given initial and final temperatures can be characterized as $WF/W0 \propto [\text{length}]^{2.81} \times [\text{diameter}]^{-4.81} \times [\text{pressure}]^{-2.0}$. In contrast with a liquid coolant, it is therefore evident that the compressed gas in order to be acceptable, demands very sensitive conditions for the parameterization of these variables.

In a number of possible applications, the acceptable limits of these variables are fully adequate for the realization of a single gas cooled arrangement. However in other circumstances it may be preferable, as shown in FIG. 4, to segment the gas tubing arrays in a modular arrangement, transferring the accumulated heat to an appropriate heat carrying liquid fluid with the help of a (modular) array of heat exchangers.

Therefore, if the size of the solar field exceeds the dimensions appropriate to an efficient gas system, the hot gas may be transferred to another medium which may be generally a harmless liquid and which can transport the heat at further distances, eventually combined with a thermal storage. Since the liquid circuit is thermally insulated everywhere, it can be easily kept at an appropriate temperature all the times. The liquid medium may for instance be an appropriate molten salt.

In FIG. 4 we exemplify modular arrays of gas driven units which transfer the gaseous heat into a secondary liquid fluid. For clarity a set of four units is shown, though a different number may be required, depending on the specific application. As already indicated in FIG. 1, each of the gas driven units is made of a number of concentrating mirrors 1, which focus the sunlight to the concentrating gas driven tubes 3. The gas loops are then connected from the mechanical compressors (blowers) 7 to an appropriate secondary transferring liquid fluid with the help of a set of local heat exchangers 10 (from gas to liquid). The resulting transferring liquid fluid array, which covers the full extent of the solar field, is finally connected with the help of a pair of thermally insulated network arrays of liquid filled tubes from the cold inlet 11 to the hot outlet 12. The circulation of the heat carrying fluid is ensured with the help of an appropriate mechanical pump 13.

The master secondary fluid loop is completed with components between inlet 14 and outlet 15 through the utilization application dependent power generating network N wherein only a heat exchanger 16 is represented. The cold inlet 11 and the hot outlet 12 of the secondary fluid network show more incomes and outputs indicating that more modules can be connected. The produced heat is recovered and the cold fluid is brought back to the modular heat exchangers 10 in the secondary fluid network. It will be discussed in more detail in the subsequent Figures.

In FIGS. 5, 6 and 7 we show more sophisticated arrangements which introduce the possibility of a thermal storage of the produced solar energy, offering a more flexible operation with a more substantial fraction of solar energy, in which the solar energy supply from the gas may be made available when needed by the user and not only by the solar conditions. To this effect the solar field and the energy production may be effectively decoupled with the help of a thermal storage device made of a sufficient amount of material which is heated whenever possible by the gas solar collector and later transferred to the utilisation plant whenever necessary.

In FIG. 5 we show a general layout in which there are two temperature insulated energy storage vessels filled with an appropriate fluid, respectively at low temperature T_{low} and at high temperature T_{high}. The specific energy stored is roughly directly proportional to the temperature difference $T_{high}-T_{low}$, which should be as high as possible, i.e. $T_{high} \sim T_{max}$ and $T_{low} \sim T_{min}$, respectively the temperatures of the inlet 14 and outlet 15 of the manifold as indicated in FIG. 4.

The fluid may be made with a suitable molten salt. Typical target values for a commonly used storage fluid made of 60 %NaNO3 - 40% KNO3 are T_{low} = 290 °C and

Thigh = 550 °C, with a thermal capacitance for unit volume of 0.731 GJ/m^3, i.e. about 5 m^3 of molten salt are required for each MWatth of thermal storage. The working temperature interval of the solar plant of FIG. 1 should be close to these values, once the temperature drops of the gas to liquid heat exchangers 10 are taken into account.

The cold and hot temperature insulated storage vessels are indicated with 17 and 18. This heat storage procedure is completely decoupled in time with the application dependent final heat generation. We can visualize three main modes of operation:

(a) Operation in the pure thermal storage mode. The secondary liquid fluid is accumulated in the temperature insulated storage vessels 17 and 18 and no heat is transferred to the utilization plant inlet 26. In this way, under the action of the solar heat from the gas through the inlet 14 and outlet 15, the hot liquid is stored in vessel 17 and the cold vessel 18 is progressively emptied. A series of regulation valves 19, 20 and 21, on the hot side and 22, 23 and 24 on the cold side control the flow of the secondary liquid. In this configuration valves 19 and 22 are opened and all the others (i.e. 20, 21, 23 and 24) are closed. The circulation of the heat carrying fluid out of vessel 17 is ensured with the help of an appropriate mechanical pump 25. The storage procedure may be continued until the hot storage vessel 17 is full and correspondingly the cold storage vessel 18 is empty.

(b) Transfer of the accumulated fluid to the utilisation plant. In this configuration there is no direct solar heat and the heat carrying fluid is extracted at temperature Thigh from storage vessel 17 and transferred from inlet 26 to the utilisation plant represented in this embodiment by the heat exchanger 27, from which it is extracted at outlet 29 at a temperature Tlow. The circulation of the heat carrying fluid out of the bottom of vessel 17 is ensured with the help of an appropriate mechanical pump 28. In this configuration valves 21 and 24 are opened and 19,20, 22 and 23 are closed. It is unlikely that the stored fluid can be directly used for the application. In this case an additional heat exchanger 27 must be used in order to transfer the heat from the molten salt to the different fluid that had been chosen for the application. For the typical application of energy production with a turbine and an alternator, the exchange fluid may generally be hot steam, but different choices are possible for this and other applications. The cooling system of the power cycle can transfer its waste heat directly to the ambient, in case of pure electrical generation, or to a heat user (heating, hot water, absorption chiller, desalination unit, etc.).

(c) Direct utilization of the fluid to the utilization plant without storage. In this simplest configuration, valves 20 and 23 are opened and 19,21, 22 and 24 are closed.

FIG. 6 refers to another orientative layout as another embodiment, in which the energy storage unit is made of a large number of very small pebbles or similar types of small chunks of a cheap, refractory material capable to withstand the frequent and large temperature excursions. The moderating material is made of a very large number of closely packed pebbles of simple geometrical shape, usually spherical, packed into a suitable thermally insulated and pressurised vessel 31 to form the heat exchanger unit. The unit can be fed either directly from the gas in a sufficiently small system as described in FIG. 1 or, alternatively, from the heat carrying secondary fluid as described in FIG. 4. The thermal conductivity of the system must be very low, but the contact area between pebbles and gas or fluid must be excellent.

FIG. 6 briefly illustrates the operation of the temperature insulated pebble bed storage vessel 30. In the case of operation with a gas the container must evidently be strong enough to withstand the pressure. There are three regions in the pressurized vessel 30: a hot pebble region 31 at the temperature of the hot gas, a cold pebble region 32 and a thermo-cline pebble region 33, in which the hot gas is rapidly cooled down, transferring the heat to the pebbles. In the cold pebble region 32 the gas has reached the temperature of the cold gas. No heat transfer occurs in hot pebble region 31 and cold pebble region 32, with time, the thermo-cline pebble region 33 moves gradually from the top to the bottom. When the bottom has been reached, the energy storage is full and no more energy can be accumulated.

In order to fill the heat storage from the gas array of FIG.1, the hot gas enters from master outlet tube 9 to the storage vessel 30 through the valve 34 which is open and after being cooled down exits through valve 35, also open, closing the loop through master inlet tube 8. Other valves 36, 37, 38 and 39 are closed. In order to extract the hot gas from the storage vessel to the utilisation plant represented by the exchanger 43, valves 38 and 39 are open and valves 34, 35, 36 and 37 are closed. The hot gas is extracted at temperature

Thigh from the storage vessel 30 with the help of a blower 40 and transferred from inlet 41 to the utilisation plant represented by the exchanger 43, from which it is sent back at outlet 42 and at temperature Tlow. Finally in the case of solar operation and no storage, valves 36 and 37 are open and 34, 35, 38, and 39 are closed.

If the operation is performed with a secondary fluid, according to FIG. 4, the structure of FIG. 6 is unchanged, except that the inlet and outlet are 14 and 15, as indicated in FIG. 4 rather than master tubes 9 and 8.

The present invention may also be based on air as a compressed gas. Since air is abundantly available, for instance in the specific application of mechanical energy generation, air may be operated as well as an open cycle, in which ordinary air from the atmosphere is heated through the solar field and used to drive a suitable turbo-compressor unit which rejects the used air back in the atmosphere. In this open configuration the system operates without specific heat exchanger, a configuration particularly interesting whenever coolant power is hard to obtain, like for instance in the arid and sunny desert areas, which are otherwise an ideal location for solar concentrators.

The above process has similarities with the familiar Brayton cycle used in gas-turbine or jet engine applications but where the solar field takes the place of the ordinary combustor. Referring to FIG. 7, it consists in a mechanically coupled compressor 44, a gas turbine 45 and a electric generator 46.

The cold air from the atmosphere enters the compressor 44 at point 47 where it is compressed iso-entropically, exiting at point 48. The gas then feeds the main solar master inlet tube 8, where according to FIG. 1 the solar field mirrors 1 are heating the tubes 3,with the gas merging at the collection master outlet tube 9. The hot gas is then entering the turbine 45 at the entry point 49, exiting at 50. The temperature in 50 is of course higher than the one in the ambient point 47 because of the inevitable losses due to Carnot and it may be used either for others thermal heat application or more simply directly exhausted in the atmosphere. The electricity produced by the generator 46 is available at 51.

As an additional facility, the solar plant may be complemented during dark periods, as previously, either by a fired backup heater-burner, also supplied with atmospheric air, or/and by a thermal energy storage whenever required. In FIG. 7 it is shown an optional pebble bed air driven thermal storage, already represented in FIG. 6. The hot gas from the solar field is exiting at the master outlet tube 9 and it is recovered in the master inlet tube 8, while compressor/turbine complex connects to the solar field either with or without thermal storage at points 48 and 49. Finally it may be convenient in some circumstances to introduce a recuperator, not shown, in which the air exiting at 50 from the turbine 45 is connected with a heat exchanger to the gas from the compressor 44 before it enters the solar field manifold through the master inlet line 8.

It is a general property of the gas driven system, as shown in FIG. 4 that the mechanical fractional nominal work from the solar plant is approximately proportional to the inverse of the square of the pressure in the tubes. On the other hand for a simple open loop solar Brayton cycle as in figure 7 the net cycle efficiency has a broad maximum in the ratio of the pressures $r_c=p(48)/p(47)$ after and before the compressor, nearly equal to the ratio $r_t=p(49)/p(50)$ namely $p(47)\approx p(50)$ and $p(48)\approx p(49)$ (i.e. neglecting the pressure drop across the solar field). In evaluating the net cycle efficiency the isentropic efficiencies for the gas turbine and the compressor must also be taken into account. Therefore the optimum efficiency for the cycle is roughly proportional to the inverse of the square of the inlet and outlet air pressures of the solar plant, $p(48)\approx p(49)$. A substantial benefit may result if these pressures are significantly increased with respect to the atmospheric pressure (gas entering in point 47 and exiting in point 50) and removing the produced heat with the help of an adequate heat exchanger to the open air.

Table 1. Parabolic trough configuration.

Mirror lateral width	5.75	m
Length tube	100.0	m
Inner diameter collection tube	9.00	cm
Temperature at inlet	290.0	C°
Temperature at exit	550.0	C°
Nominal solar constant at tube	630.0	W/m2
Nominal solar power to tube	362.3	kWatt
Sun linear specific power	3.62	kWatt/m
Radiated power from tube	31.45	kWatt
Fractional radiated power	8.68	%

Cooling medium	He	He	CO2	CO2	Air	Air	N2	N2	
Pressure of tube	20.00	30.00	20.00	30.00	20.00	30.00	20.00	30.00	atm
Cooling massive flow	0.2662	0.2662	1.647	1.647	1.387	1.387	1.340	1.340	kg/s
Min. gas speed	24.14	16.09	13.59	9.059	17.43	11.62	17.36	11.57	m/s
Max. gas speed	35.44	23.53	20.16	13.30	25.91	17.08	25.81	17.01	m/s
Gas pressure drop	0.1511	0.1005	0.3584	0.2377	0.3905	0.2589	0.4065	0.2694	Atm
Isentropic pump work	2.384	1.056	3.183	1.402	4.409	1.941	4.591	2.020	kWatt
Fractional pump power	0.6492	0.2879	0.8665	0.3832	1.211	0.5355	1.255	0.5547	%

gamma : Ar,He,=1.67 CO2=1.29 Air, N2, O2 1.40,

Table 2. Fresnel collector configuration.

Mirror lateral width	24.00	m
Length tube	100.0	m
Inner diameter collection tube	18.00	cm
Temperature at inlet	290.0	C°
Temperature at exit	550.0	C°
Nominal solar constant at tube	549.0	W/m2
Nominal solar power to tube	1318.	kWatt
Sun linear specific power	13.18	kWatt/m
Radiated power from tube	129.0	kWatt
Fractional radiated power	9.792	%

Cooling medium	He	He	CO2	CO2	Air	Air	N2	N2	
Pressure of tube	15.00	25.00	15.00	25.00	15.00	25.00	15.00	25.00	atm
Cooling massive flow	0.9683	0.9683	5.992	5.992	5.043	5.043	4.873	4.873	kg/s
Min. gas speed	29.27	17.56	16.47	9.885	21.14	12.68	21.04	12.63	m/s
Max. gas speed	42.86	25.63	24.29	14.46	31.19	18.56	31.07	18.48	m/s
Gas pressure drop	0.07383	0.04422	0.1747	0.1044	0.1903	0.1137	0.1981	0.1184	Atm
Isentropic pump work	5.644	2.026	7.508	2.685	10.39	3.715	10.82	3.867	kWatt
Fractional pump power	0.4228	0.1520	0.5633	0.2020	0.7873	0.2822	0.8157	0.2924	%

CLAIMS

1. Process for collecting solar energy comprising the use of a pressurized gas, re-circulated by means of one or more compressors (7) and heated by means of a number of solar one dimensional concentrating mirrors (1) with a linear receiver tube (3) in which the circulated pressurized gas absorbs heat from the concentrated solar radiation.

2. A process according to claim 1 in which the exposed external surface of receiver tube (3) is made opaque to visible light and reflective to the infra-red in order to minimize the amount of reirradiated solar radiation.

3. A process according to claim 1 wherein the walls of the exchanging surfaces of the receiver tube (3) have a surface treatment to improve the heat transfer.

4. A process according to claim 1 wherein the receiver tubes (3) have means to rotate the gas inside the tube (3) in order to further improve the uniformity of the temperature of the gas in said receiver tubes (3).

5. A process according to claim 1 wherein the concentrating mirrors (1) have a parabolic configuration.

6. A process according to claim 1 wherein the solar radiation is collected by Fresnel collectors.

7. A process according to any previous claim

wherein the pressurized gas is air, which is heated by the concentrated thermal energy from direct solar radiation (DNI).

8. A process according to claim 7, wherein the heat of the heated air is transformed in mechanical energy combining the use of a turbine (45) and a compressor (44), preferably by a Brayton cycle, rejecting the used air back in the atmosphere.

9. A process according to claim 8, wherein the compressed air after the compressor (44) is preheated before entering in the master inlet tube (8) with high temperature air exiting from the turbine (45).

10. A process according to claim 8 wherein the rejected air at high temperature is used to improve the mechanical conversion efficiency either with the addition of a conventional steam or organic Rankine cycle or to a heat user device.

11. A process according to any of claims 8 to 10 wherein the air is maintained in a pressurized closed loop and this loop has a further heat exchanger which dissipates the heat in the atmospheric air.

12. The process according to any of claim 1 to 11 wherein the gas is a first fluid and there is a further second fluid, so that the first fluid absorbs heat from solar radiation and transfers the absorbed heat to the second fluid that transports the collected heat to a single point for a further use.

13. A process according to claim 12 wherein the heat transfer from the gas to the second fluid is

carried out by means of heat exchangers (10) in two or more different distributed locations of the collecting field.

14. A process according to claim 12 wherein the gas is a widely available commercial gas, namely Hydrogen, Helium, CO_2, Dry Air, or Nitrogen.

15. A process according to claim 12 wherein the secondary fluid is pressurized steam and is used to generate mechanical energy with a conventional method.

16. A process according to claim 12, wherein the secondary fluid is an appropriate heat transfer medium capable of being heated to a temperature of at least 550 °C.

17. A process according to claim 16 wherein the second fluid is molten salt.

18. A process according to claim 16 where the heat transfer medium is a molten metal alloy.

19. Device for collecting solar energy comprising:

- one or more heat collecting units (5, 6), each unit comprising a primary circuit with gas impelled by one or more compressors (7) and,
- a secondary circuit with molten salt impelled by one or more pumps (13)

wherein the primary circuit and the secondary circuit are coupled by means of heat exchangers (10).

20. A device according to claim 19 wherein the heat collecting unit comprises one o more cylinder-

parabolic collectors (5, 6).

21. Device according to claim 19 wherein the secondary circuit comprises a thermal storage unit.

22. Device according to claim 19 wherein the secondary circuit transfers the heat to a central user block.

23. Device according to claim 22 wherein said central user block includes a power block, where heat flow available in the molten salt is converted to mechanical-electrical power.

24. The system of any preceding claim in which in order to decouple the energy supply from the availability from the solar conditions, a thermal storage of the produced solar energy is added.

25. A system according to claim 24 wherein this system comprises two further thermally insulated vessels (17, 18) that can be filled with molten salt so that, in operating mode, they are kept respectively at temperatures of the master outlet and inlet tubes (8, 9); and, when the thermal storage works storing energy, the fluid coming from the second cold vessel (18) heated through the heat source fills progressively the first hot vessel (17); when the thermal storage works returning the energy, the fluid takes the opposite path from the first vessel (17) to the second vessel (18) through the delivery of the energy.

26. A system according to claim 24 wherein the system further comprises a temperature insulated pebble bed storage vessel (30) so that, in operation mode, when

the hot fluid enters to the vessel (30), it goes through pebble transferring the energy to the pebbles, and when the flow is reversed in order to recover the thermal energy, the energy is transferred from the pebbles to the flow.

27. A system according to claim 26, wherein the fluid is gas or molten salt.

28. A system according to claim 24, wherein the heat storage is made with the help of pressurized gas with materials which undergo reversibly a liquid/solid and a solid/liquid change.

29. A system according to claim 24, wherein the heat storage is made with the help of the secondary molten salt with materials which undergo reversibly a liquid/solid and a solid/liquid change.

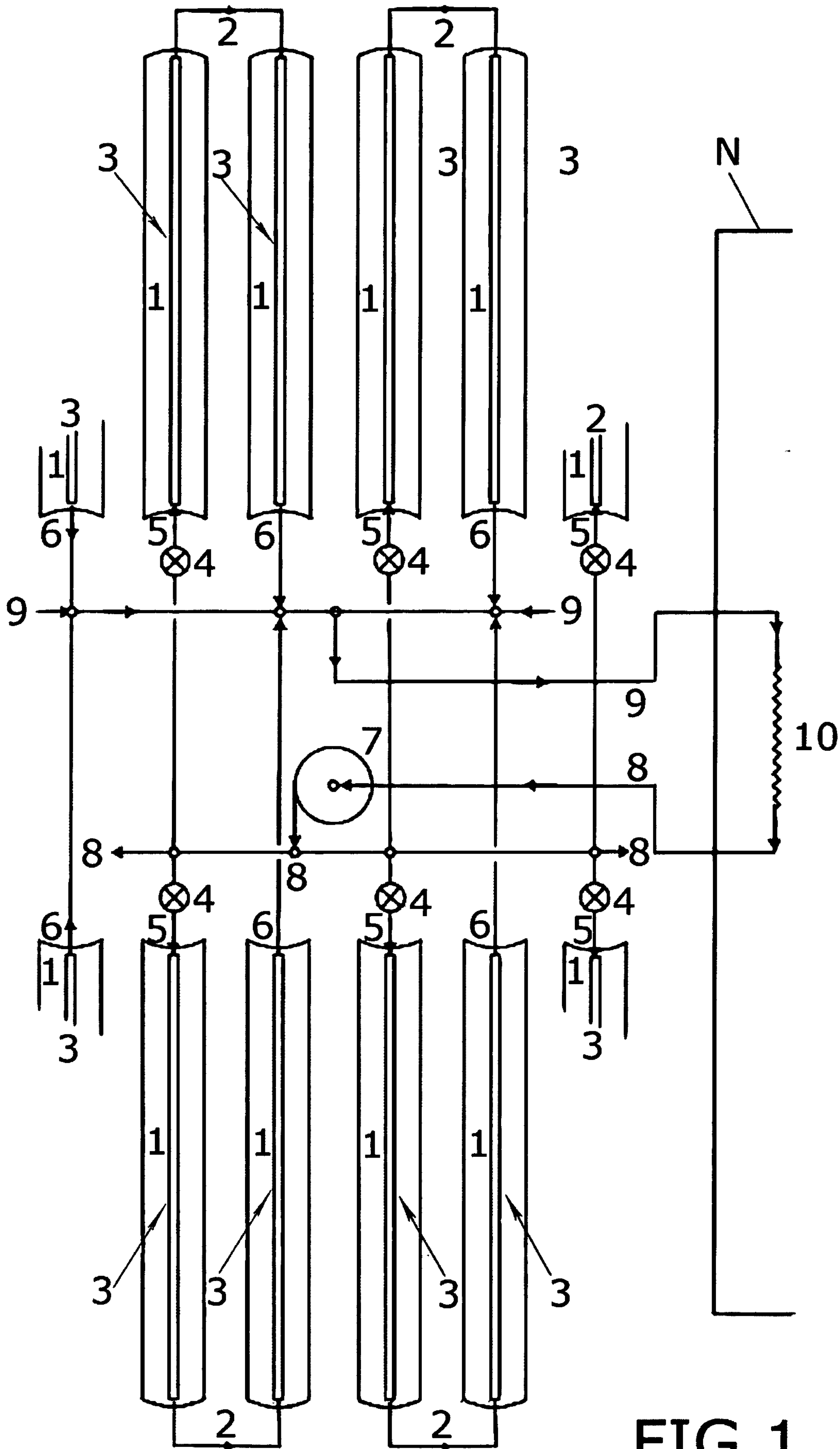
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FIG.1

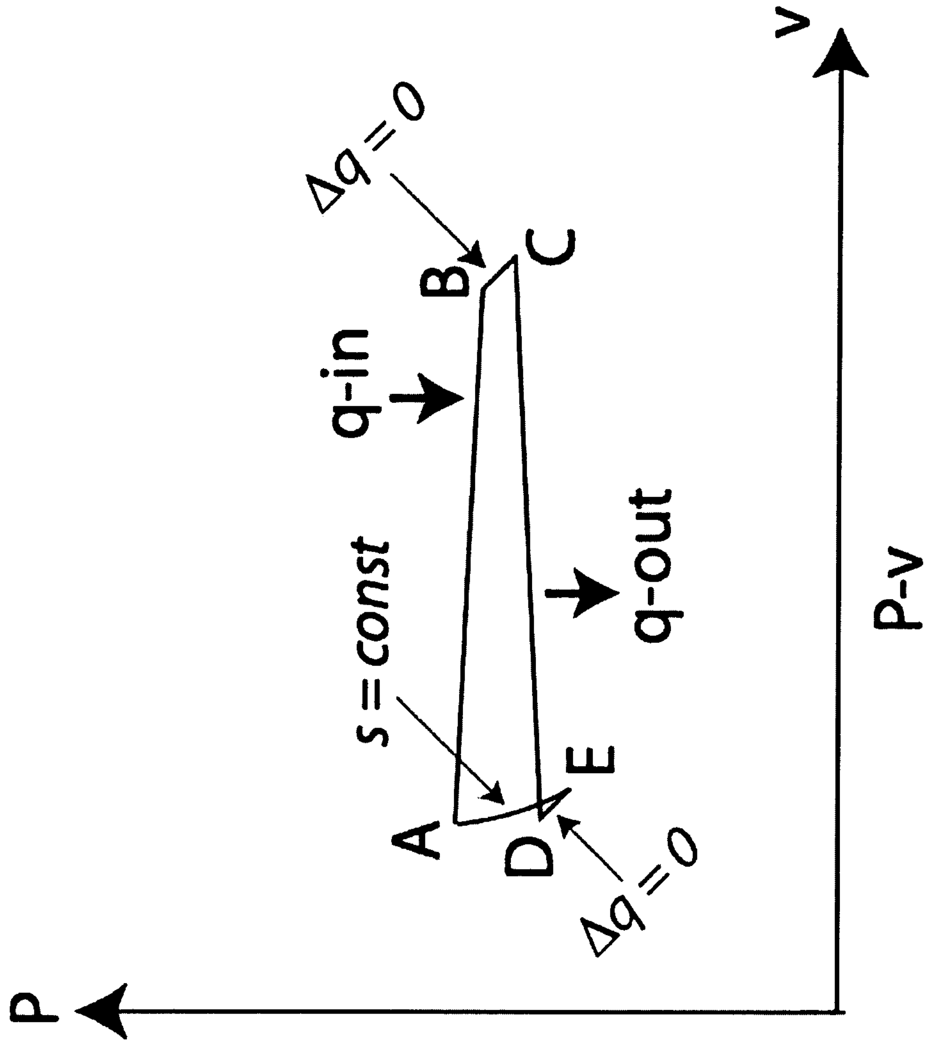

FIG.2

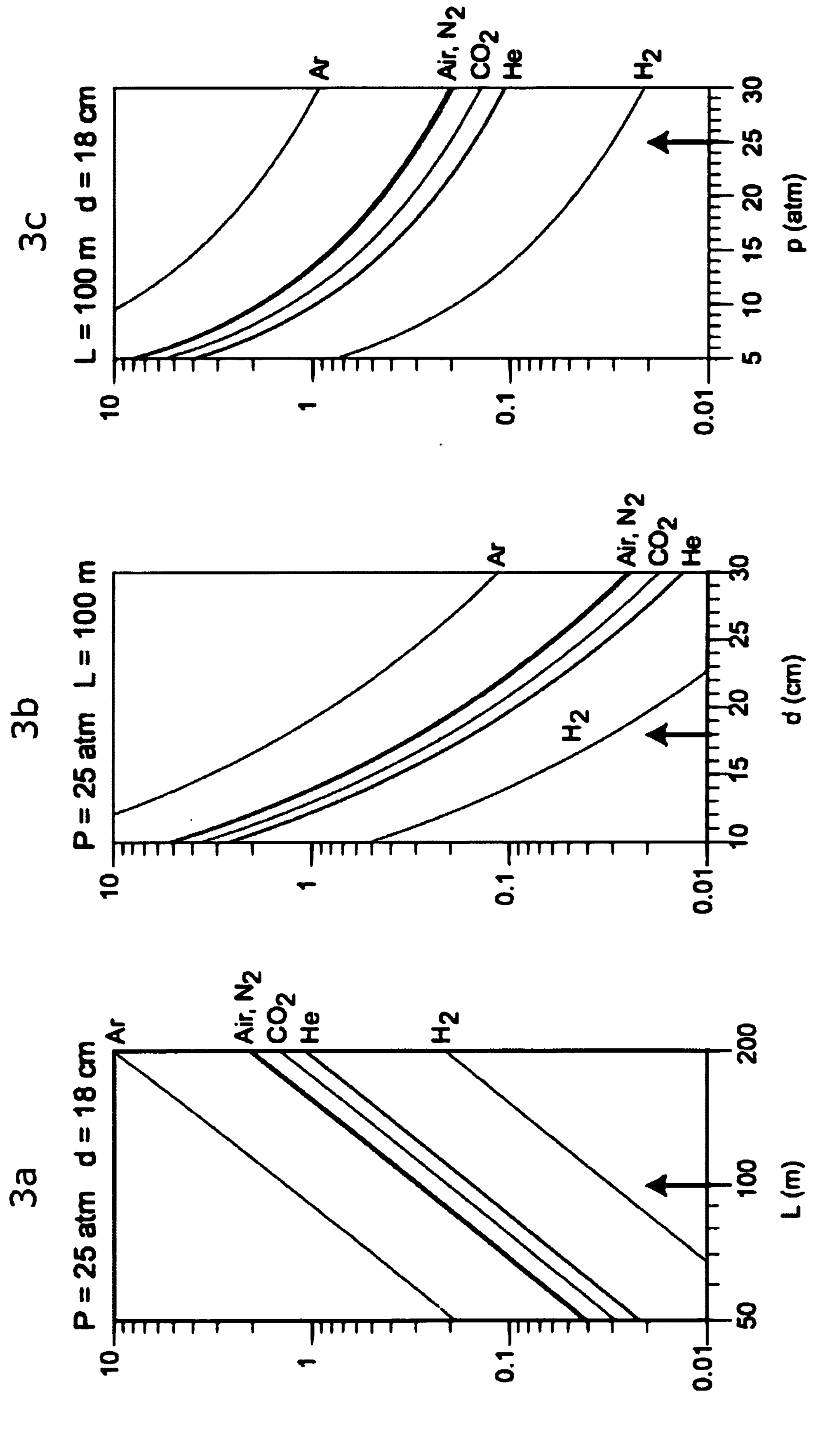
3a
P = 25 atm d = 18 cm
Ar
Air, N2
CO2
He
H2
L (m)
3b
P = 25 atm L = 100 m
d (cm)
3c
L = 100 m d = 18 cm
p (atm)

FIG. 3

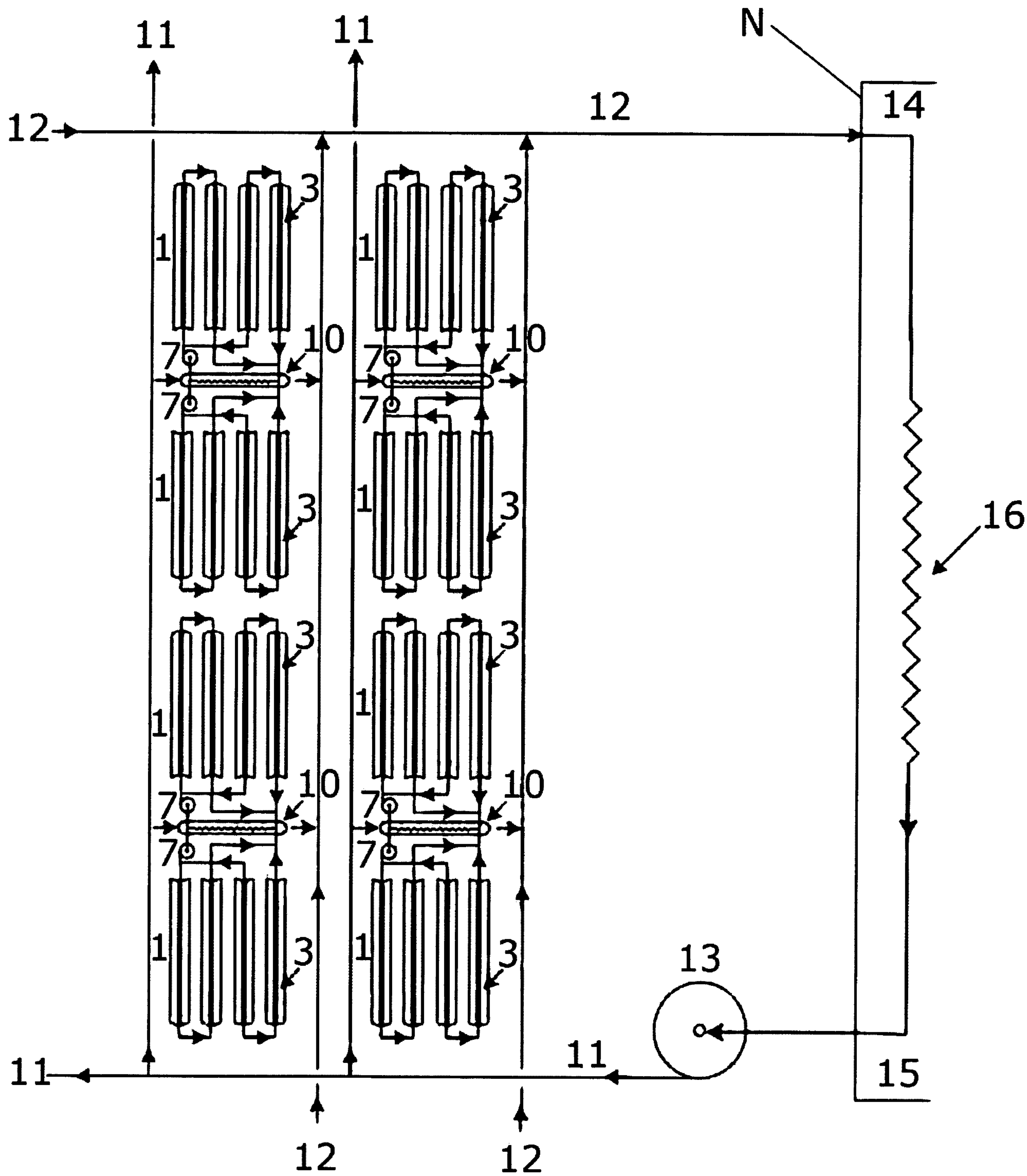

FIG.4

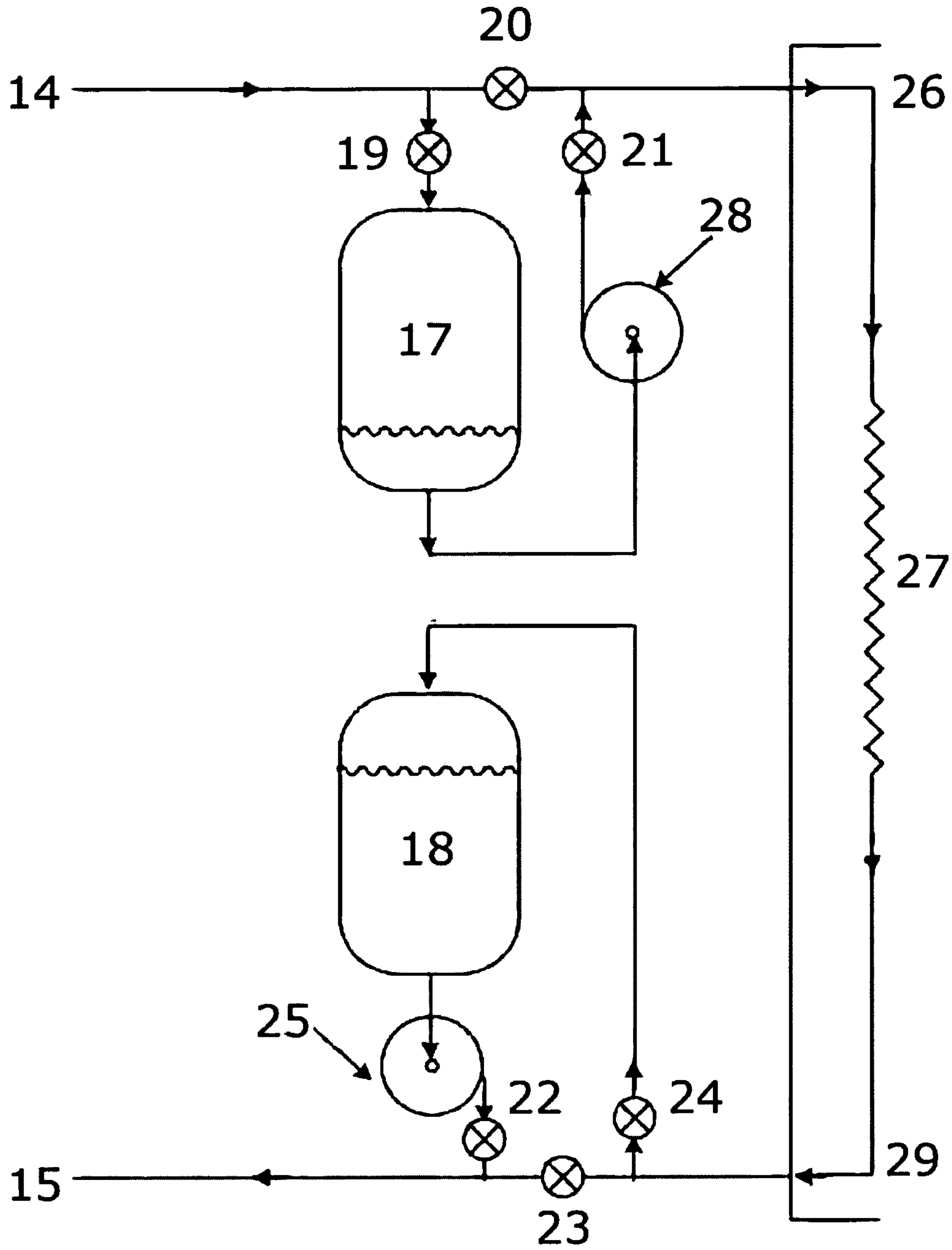

FIG.5

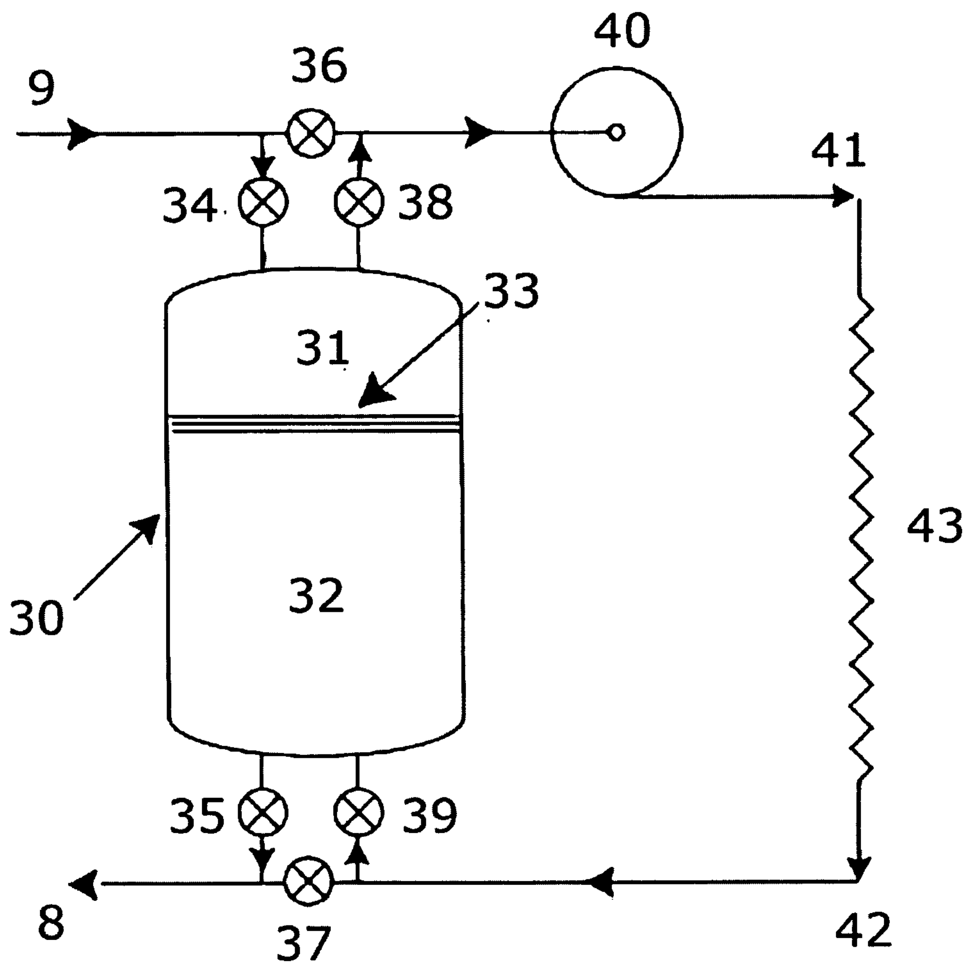

FIG.6

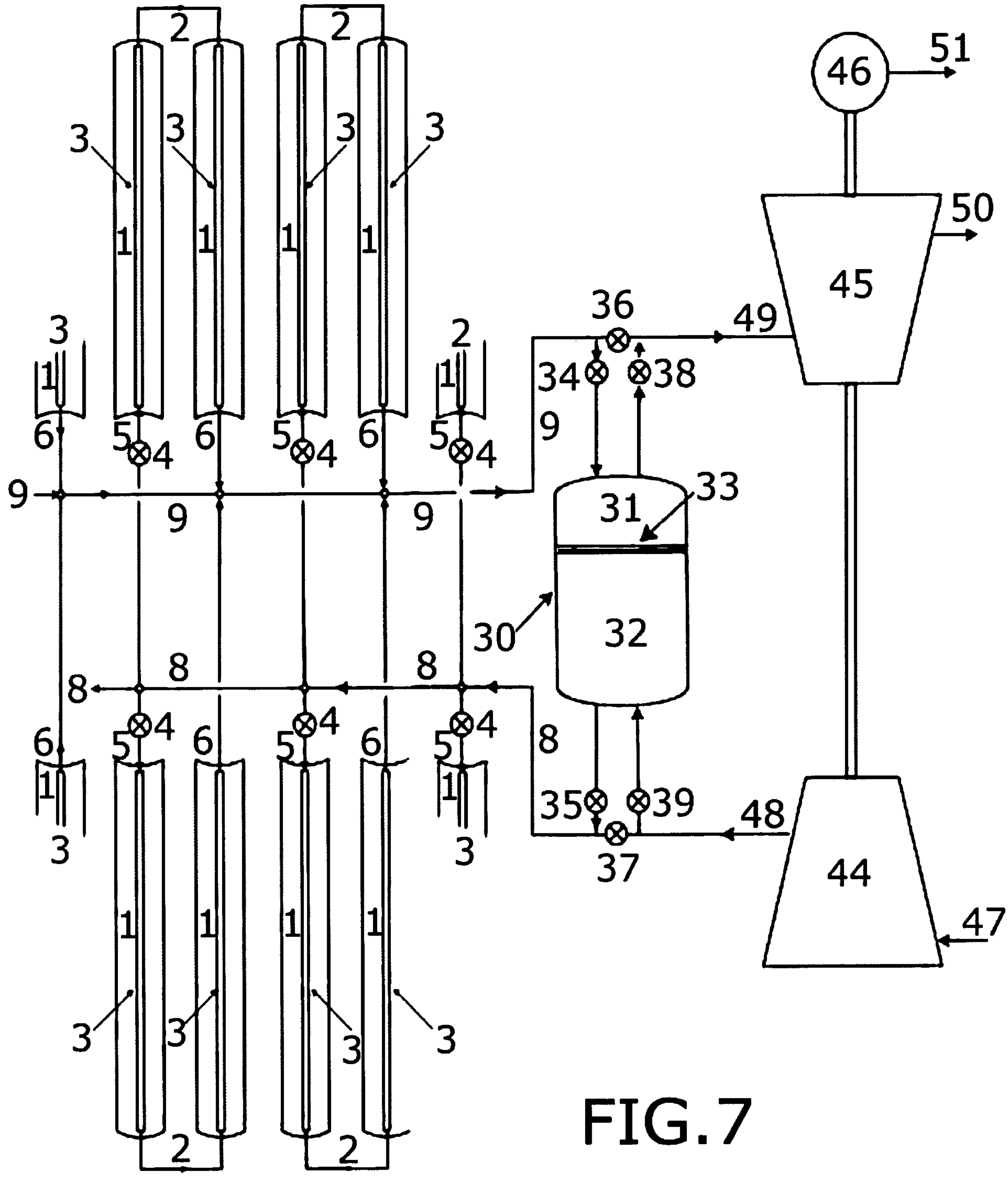

FIG.7