PUBLISHED
MAY 25, 1943.
BY A. P. C.

G. JENDRASSIK
APPARATUS FOR GAS TURBINES
Filed April 15, 1939

Serial No.
268,009
5 Sheets-Sheet 1

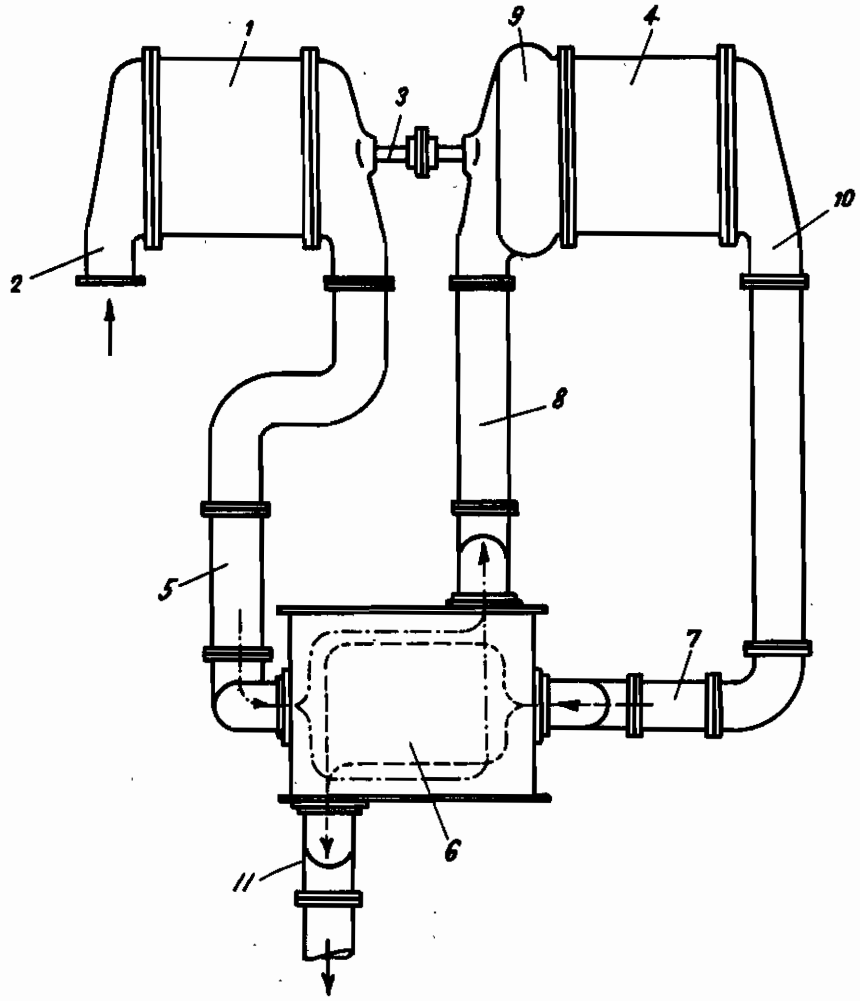

Fig. 1.

Witnesses:
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Edmund Tauner

Inventor:
George Jendrassik

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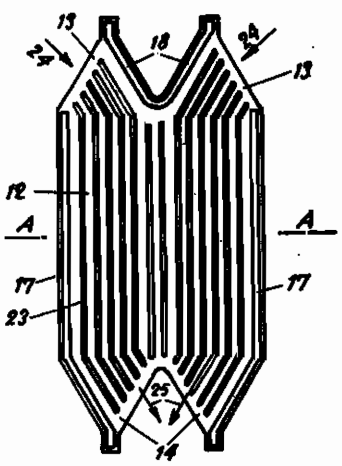


Fig. 2.

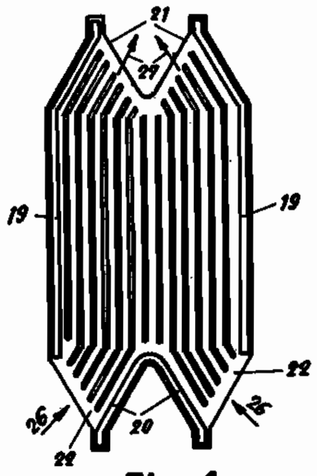


Fig. 4.

Fig. 3.

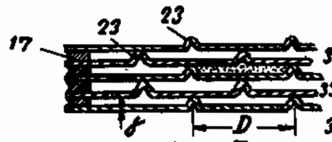


Fig. 5.

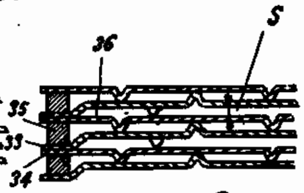


Fig. 6.

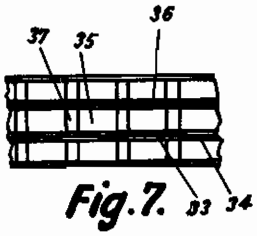
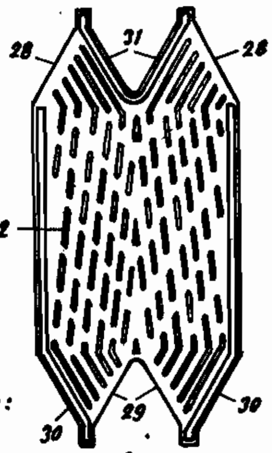


Fig. 7.



Witnesses:

Julius Farga
Edmund Tamas

Fig. 8.

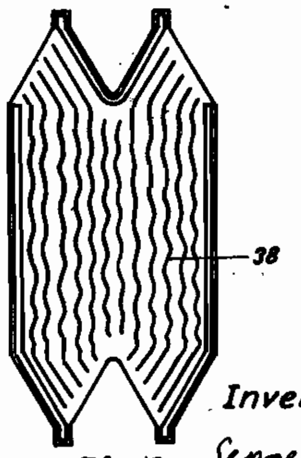


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Fig. 9.

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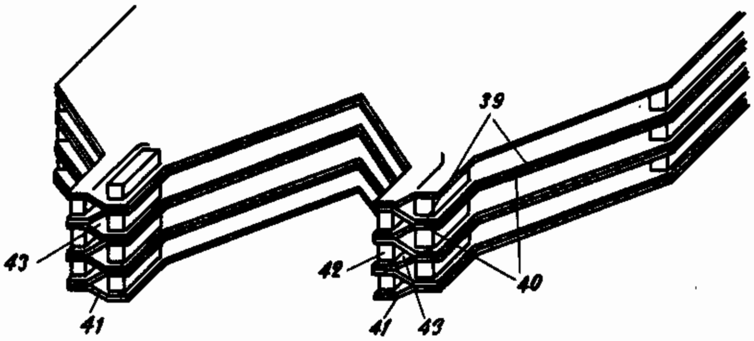

Fig. 10.

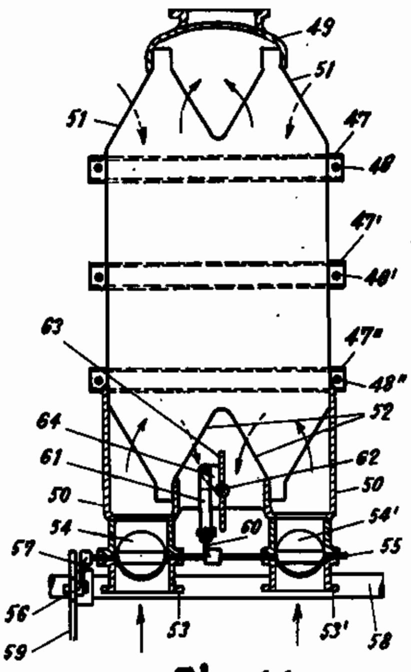

Fig. 11.

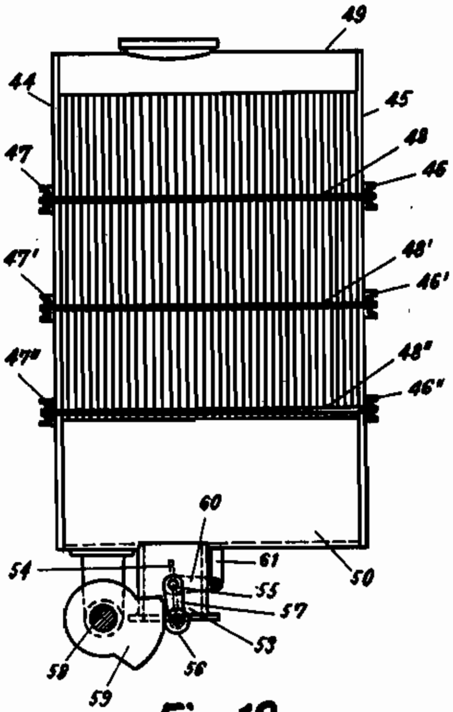

Fig. 12.

Witnesses:

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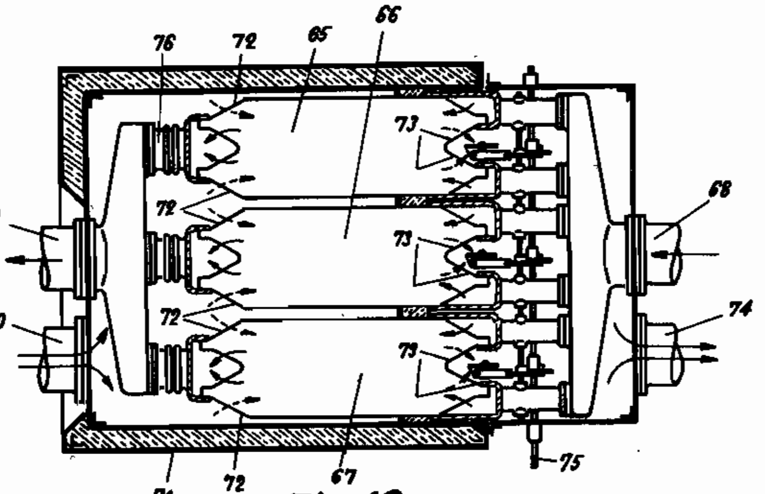

Fig. 13.

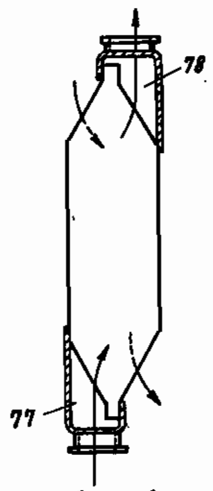

Fig. 14.

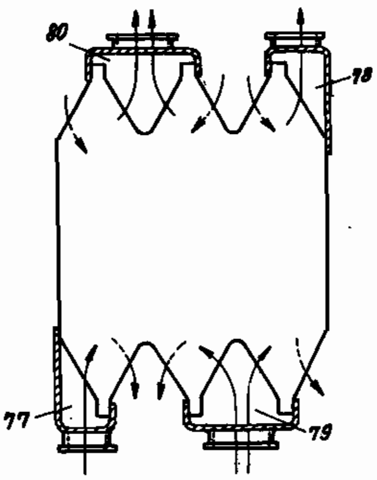

Fig. 15.

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Serial No.
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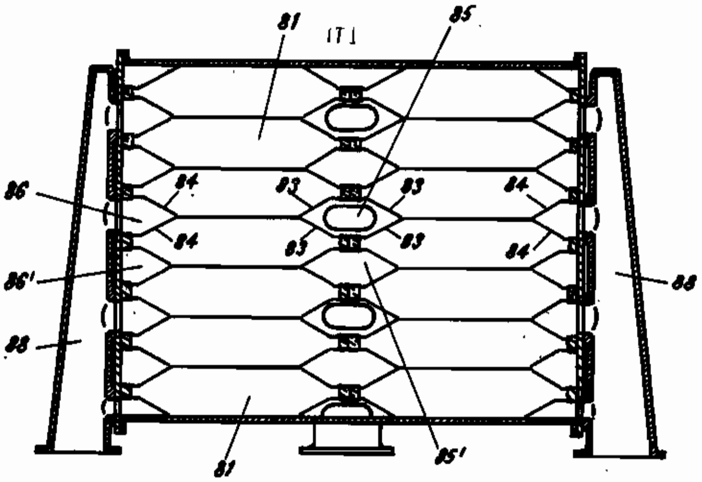

Fig. 16.

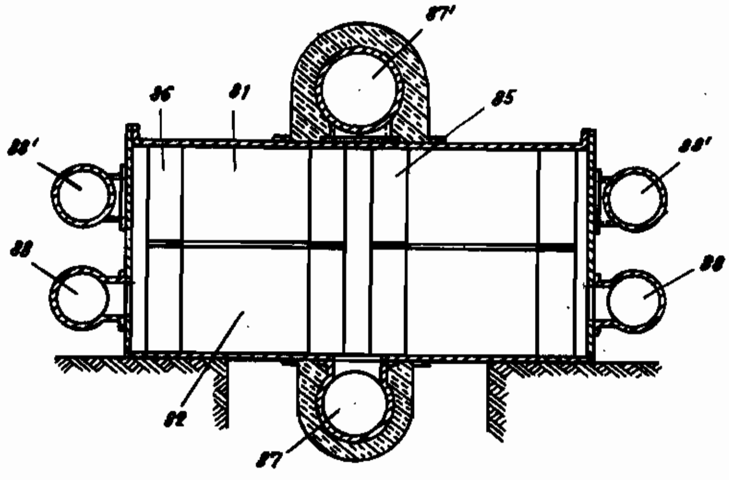

Fig. 17.

Witnesses:
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Edmund Tamas

Inventor:
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ALIEN PROPERTY CUSTODIAN

APPARATUS FOR GAS TURBINES

George Jendrassik, Budapest, Hungary; vested in the Alien Property Custodian

Application filed April 15, 1939

In sets of equipment composed of a gas turbine and of a compressor for the preliminary compression of the working fluid, in which in the turbine part of the set, in accordance with the kinds of constructional material known at present and suitable for being employed, only moderate maximum temperatures [500 to 600 deg. centigrade] are permissible, it is, in order to obtain high efficiencies, very important to recover the heat of the spent gases leaving the turbine. The recovery of such heat is preferably effected by means of heat exchangers in which the spent gases leaving the turbine are transmitting their heat to the fresh working fluid, already compressed by the compressor in such a manner as to ensure that the temperature of the fresh gas of higher pressure leaving the heat exchanger should approximate as far as possible the temperature of the spent gas of lower pressure entering the heat exchanger, such a heat exchanger being, accordingly, usually a necessary accessory of the equipment. In addition to low thermal loss, it is, of course, essential that the process of heat recovery should take place with as little mechanical loss as possible. This mechanical loss manifests itself in the first place by the loss of work corresponding to the pressure drop set up owing to flow friction during through-flow through the heat exchanger. In addition to all this, the heat exchanger should be designed so as to require little material of construction, this being necessary to prevent the heat exchanger to be expensive and of great weight, and to ensure that its space requirements also should not exceed certain limits.

The heat exchangers proposed in connection with the known gas turbine proposals are, partly, of a type the operation of which is periodical, their working space filled with heat-storing material being inserted alternately into the gas stream to be heated and into the gas stream to be cooled, and, partly, devices in which one fluid flows inside a system of tubes, whilst the other fluid flows outside the tubes of that system or in the tubes, surrounding the first-named tubes, of a second system, each working fluid thus flowing in a separate working space. As regards the devices belonging to the first group of design, they are, if they should be made so as to require a small amount of space and of material, of low efficiency, and further drawbacks of this type of devices are, on the one hand, that the changing-over members, requiring control mechanism also, involved by their alternate operation, are complicated and expensive, and, on the other hand, that the changes of pressure presenting themselves in the working space of the heat exchanger in consequence of the operations of changing over themselves are accompanied by a special loss in work.

In the case of the devices belonging to the group corresponding to the second type of design, the requirements of material are very substantial as compared to those of the first group of devices, seeing that in consequence of their type of design the coefficient of heat transfer of the heat transmitting surfaces is low, and consequently their space requirements are also very high, whilst on the other hand their cost of manufacture is high, and, at the same time, their efficiency also is unsatisfactory.

The working process and apparatus according to the invention relates to gas turbine apparatus equipped with a heat exchanger, and to the economical operation of such apparatus, in which the drawbacks mentioned can be eliminated, whilst the type of design of the heat exchanger employed consisting of bundles of sheets is such that the said heat exchanger is easy to manufacture, relatively inexpensive, possessing a high efficiency, and can, moreover, also be easily cleaned.

In order to enable the invention to be more readily understood, Fig. 1 illustrates the diagrammatical view, shown by way of example, of an apparatus according to the invention, Fig. 2 is a view of one of the sheets of the heat exchanger, whilst Fig. 3 is a cross-section of the sheet, and Fig. 4 of the view of the sheet adjaeent to the sheet shown on Fig. 2. Fig. 5 hows a detail [the ribbing] of the cross-section of a bundle of sheets, whilst Fig. 6 is a detail of the cross-section of a bundle of sheets made with a different kind of ribbing. Fig. 7 is a detail of the side elevation of the last-named bundle of sheets. Fig. 8 is a view of a sheet made with a ribbing of different type, whilst Fig. 9 is the view of a further variant. Fig. 10 is the view in perspective of a type of sheet bundle. Fig. 11 is a diagrammatical section of a bundle of sheets, equipped with a cleaning device and with junctions for inlet and outlet, whilst Fig. 12 is the side elevation of the same bundle of sheets. Figs. 14 and 15 are views of two further types of sheet bundles. Fig. 13 shows the diagrammatical section, shown my way of example, of the apparatus according to the invention, Fig. 16 shows the diagrammatical section of a heat exchanger equipment of a somewhat different type, and Fig.

17 shows the diagrammatical cross-section of the latter.

According to the arrangement shown on Fig. 1 it is into the compressor casing 1 that the compressor for the compression of the fresh working fluid entering through the inlet opening 2 is installed, the said compressor being driven through the shaft 3 by the turbine proper arranged in the turbine casing 4. The gas compressed to a substantial pressure passes, after leaving the compressor, through the duct 5 into the heat exchanger 6, and during its throughflow through the latter absorbs the heat of the spent gases leaving the turbine and entering through the pipe 7 into the heat exchanger, following which it leaves the heat exchanger through the pipe 7 into the heat exchanger, following which it leaves the heat exchanger through the pipe 8. Following this the working fluid is, at least in part, passed into the combustion space 9 in which it becomes heated owing to combustion, and then enters the turbine, in which it expands whilst performing work and leaves the working space of the turbine through the pipe 10. The pipe 10 joins on to the duct 7 already named through which the gases leaving the turbine are entering the heat exchanger. Here the spent gases are transmitting their heat to the fresh working fluid and are leaving the heat exchanger through the pipe 11.

The heat exchanger employed in this gas turbine apparatus serves according to the invention for heating gases of higher pressure by means of gases of lower pressure, and is composed of one or more bundles of sheets, stayed one against the other, preferably by close ribbing or in any other manner, which are in places mutually held together (clamped together, possibly soldered together or welded together) and which as far as their effective part is concerned are of mutually equal shape. One side of the sheets of the sheet bundle forms a lateral boundary surface of the throughflow space of the heat transmitting working fluid, whereas their other side forms a boundary wall of the throughflow space of the heat absorbing working fluid, whilst at the same time the spaces between the sheets are, by means of the tight closing (clamping together, soldering together or welding together) of the sheet edges in the direction of throughflow, separated from each other in such a manner, that by dispensing with the said closing along certain sections of the sheet edges, openings for the mutually independent inlet and outlet of the heat-transmitting and of the heat-absorbing working fluid are provided, it being possible to mutually unite the said openings, separately for each working fluid, at the inlet as well as at the outlet.

A sheet for such a heat exchanger sheet bundle is shown on Fig. 2, according to which the inlet and outlet extensions 13 and 14 of pointed arch or triangular shape are joining on to the oblong intermediate part of the sheet 12. On the longitudinal sides of the sheet, between each pair of adjacent sheets, the liner bands 17 and 16 are inserted, which are preferably fixed to the sheets, e. g. by means of welding. These liner bands are, either as extensions of the bands 17, or as independent bands 18, also running along an edge of both the inlet and outlet branches 13 and 14, and are only wanting on the two remaining sides of the said branches, notably either in such a manner as shown on Fig. 3 in connections with the bands 17 and 18 mentioned, or as shown on Fig. 4 at the outlet and inlet branches 21 and 22, respectively, in connection with the bands 19—20. The sheet bundles forming the heat exchanger are obtained by the alternate mutual superposition of the sheets fitted with liner bands arranged according to Fig. 2 and Fig. 4, respectively, if the liner bands and the sheets supported against them are mutually held together tightly by one of the methods mentioned. On the various sides of the inlet and outlet branches alternately inlet and outlet openings are obtained, one group of which will be open from one side of the said branches (e. g. according to Fig. 2) whilst their other group will be open from the other side (e. g. according to Fig. 4.) In view of the fact that the pressures of the heat-absorbing and heat-transmitting fluids are unequal, it is necessary to make provision for the absorption of the difference of pressure, so as to ensure that the mutual distance of the mutually adjacent sheets should not become altered during operation. This purpose is served by the system of ribs 23, visible on Fig. 3, showing the cross-section of the sheet illustrated on Fig. 2, the said system of ribs being preferably produced by stamping from the material of the sheet. In the type of design according to the invention the ribs for guiding the same working fluid at the inlet and outlet branches are substantially of mutually equal direction, in consequence whereof parts of fluid entering at different points of the inlet opening are covering equal lengths of travel between the sheets and the guiding ribs, and will therefore flow through the heat exchanger with a uniform distribution of velocities. The section, taken along the plane A—A, of the bundle obtained by the mutual superposition of the sheets is shown on Fig. 5, on which it is possible to see, on the one hand, the liner band 17 connected with the sheets by means of soldering or welding, and, on the other hand, the ribs 23, which, together with the liner bands mentioned, assure the staying of the sheets. In view of the fact that the heat transmission of the sheets depends in a high degree on their mutual distance great care should be bestowed on ensuring that the sheets should not be deformed by the difference of pressure mentioned. For this reason the mutual distance D of the ribs stamped into the sheets should preferably not exceed 50 times the thickness of the sheets. Considering the sheets shown on Figs. 2 and 4, the working fluid to be heated will, for instance, enter the bundle of sheets in the direction of the arrow 24 and will leave it in the direction of the arrow 25, whereas the working fluid transmitting heat will enter the bundle of sheets in the direction of the arrow 26, and will leave it in the direction of the arrow 27. Accordingly, such part of the sheets of the heat exchanger as represents the effective part proper is operating according to the counterflow principle, which arrangement enables very favourable heat transfer to be obtained. In this arrangement the inlet and outlet openings of the bundle of sheets are arranged in such a manner that two mutually non-communicating openings (i. e. two openings not belonging to the same working fluid, as, for instance, the openings 13 and 27, or 13 and 28) are situated at a less distance from each other, than two openings communicating with each other through the throughflow space situated between the sheets (e. g. the openings 13 and 14.)

Fig. 8 shows a type of design of the sheets in which, in a manner similar to the one in which the arrangement according to Figs. 2 and 4 operates, one working fluid flows into the sheet bundle along the edges 28, and leaves it along the edges 29, whereas the other working fluid flows-in along the edges 30 and leaves the bundle along the edges 31; in the case of this example, however, the ribs 32 are interrupted in the longitudinal direction, and are, moreover, forming an angle with the direction of the flow. The purpose of this arrangement is to produce eddies, the direction of the axis of which is identical with the direction of the flow, such eddies substantially increasing the heat transmission without increasing the resistance to flow in a measure which would exceed that corresponding to the increase of the heat transfer.

In addition to the arrangement described, the most varied arrangements are possible as to the staying, relatively to each other, of the sheets of the sheet bundles. In the embodiment shown by way of example in cross-section on Fig. 6 the sheets 33 and 34 are placed directly on each other, whilst between each two such pairs of sheets the liner band 35 is arranged, the ribs stamped into the sheets being constructed in such a manner that the sheets 33 and 36 are supported on each other by means of ribs stamped into each of the sheets in the direction towards the other sheet, whereas the sheets 33 and 34 are kept at a suitable distance from each other by the short supporting ribs stamped in places into the sheets. In the case of such an arrangement it is advisable to employ the space between the ribs 33 and 34 for the working fluid of higher pressure, whereas the working fluid of lower pressure occupies the space between the sheets 33 and 36. In the case of such an arrangement the sheets 33 and 36 will be well stayed against the higher pressure by the numerous ribs, whilst for the staying of the sheets 33 and 34 the ribs of small number and short length will be sufficient, because these sheets are not exposed to compression under pressure. In the case of this arrangement the liner bands 35 can be constructed, in accordance with Fig. 7, representing the side elevation of Fig. 6, in such a manner that the said liner bands, interrupted in places, will create the gaps 37. This can be done, as the liner bands are closing the space of the working fluid of lower pressure, the pressure of which is preferably equal to the pressure of the ambient, and as here, therefore, it is not necessary to make provision for any particularly careful packing. The advantage of the interruption in this manner of the liner bands is that this will enable the loss caused by the capacity of heat conduction of the liner bands to be avoided or substantially reduced, which loss manifests itself by a transfer of heat from the hot end of the heat exchanger towards its cold end.

Should—as will usually be the case with gas turbines—the pressure of the fluid to be heated and of the fluid transmitting heat be different, it will be preferable to choose the depth of the throughflow spaces serving for the two fluids, i. e. the mutual distance of adjacent sheets, at different figures, in consequence whereof the mutual distance of the sheets will be alternately greater and smaller. In general the mutual distance between the sheets forming the boundaries of the flow space of the fluid of higher pressure will be smaller than the mutual distance of the sheets forming the boundaries of the flow of the fluid of lower pressure.

In the longitudinal direction of the ribbing also, the ribbing of the sheets can be constructed in many different ways, regarding which an example has already been described with reference to Fig. 8. By way of a further example of construction a ribbing 38 of wave-line shape is shown on Fig. 9, the purpose of which is, similarly to the ribbing shown on Fig. 8, to increase heat transfer.

Fig. 10 is a sketch in perspective view, of another type of construction of the bundle of sheets. In the case of this arrangement the closing, at the inlet openings of the sheet bundles, of the sheet spaces not communicating with the inlet opening is effected by the direct fastening together, preferably by the welding together of the mutually adjacent sheets [39 and 40]. Thereby the gap widths of the inlet openings at the inlet into the sheet bundle gradually diminish owing to the gradual increase of the mutual distance of the sheets fastened together, and in consequence thereof the gas flow will not contract, which fact results in diminished flow losses. At the vertices 41 of the inlet and outlet branches of the bundle of sheets the terminal openings 43 of triangular shape formed between the sheets fastened together in pairs and the liner bands 42 must be packed [preferably by means of welding].

Fig. 11 and 12 are illustrating a bundle of sheets having no external closing sheathing, but fitted with outlet and inlet branches and with cleaning arrangements. The mutually superposed sheets are held together by the terminal sheets 44, 45 of greater thickness, and further by the reinforcing profile irons or ribs 46, 46′, 46″ and 47, 47′, 47″ and by the bolts 48, 48′, 48″ etc. by which the profile irons or ribs mentioned are drawn together. In addition hereto the sheets may be welded together to each other and/or to the liner bands. The discharge branch 49 consisting of a sheathing sheet fixed e. g. welded in the sheet bundle serves for the discharge of one of the working fluids,—preferably for the discharge of the working fluid of higher pressure—whereas the admission of the working fluid is effected by means of the inlet branch 50 fixed or welded on the sheet bundle likewise. In this case the working fluid of lower pressure enters the sheet bundle along the length of the inlet edges 51 and leaves it along the length of the edges 52. In view of the fact that it is possible for the dirt, soot, flue dust etc. to become deposited in the heat exchanger, it is advisable to make provision for the cleaning of the latter.

According to a further inventional discovery it is possible to effect the cleaning of the heat exchanger in such a manner that throttling or closing members are provided in the passages of the heat exchanger, preferably at its cold end, by means of which the various passages can be partly or entirely cut-out from the gas flow, in consequence whereof the velocity of the gas in the other passages will become substantially increased and will become suitable for blowing out the dirt from between the sheets of the heat exchanger. Such a cleaning device is visible on Figs. 11 and 12 according to which throttle valves 54, 54′ mounted on the shaft 55 are provided in the junction ducts 53, 53′ of the high pressure working fluid to be heated.

The shaft 55 further also carries the lever 57 fitted with the roller 56 keyed on it, which lever is supported on the cam 59 keyed on the shaft 58. It is similarly on the shaft 55 that the lever 60 is also keyed, which latter is, by means of the yoke 61 joining on articulately to it, connected, in an articulate manner likewise, to the lever 64 of the throttle valve 63 journalled so as to be pivotable around the shaft 62. The throttle valve 63 is arranged in the outlet cross-section of the heat transmitting fluid [i. e. of the fluid of lower pressure] and is suitable for throttling this cross-section. If a heat exchanger composed of at least two such heat bundles connected in parallel is employed in connection with the gas turbine, it will be possible, by deflecting the shaft 56 i. e. by deflecting, in the final result, the throttling members 54, 54' and 63, to throttle certain passages of the heat exchanger [in case of only two bundles connected in parallel the passages of one bundle], in consequence whereof, in those passages which are not throttled, the dirt is blown out by the flow of increased intensity.

In the case of the heat exchanger shown on Fig. 13, three sheet bundles [65, 66 and 67], connected in parallel with each other, are employed. The fresh working fluid to be heated flows into the heat exchanger through the pipe 68 and after having become heated, leaves it through the pipe 88. The heat-transmitting working fluid, the pressure of which is preferably equal to that of the surroundings, enters the insulating sheathing 71 of the heat exchanger through the pipe 70 and in the interior of the said sheathing enters through the inlet openings 72, into the various sheet bundles, which it leaves through openings provided along the length of the edges 73, following which it leaves the sheathing of the heat exchanger through the outlet pipe 74. This device is fitted with the cleaning equipment shown on Figs. 11 and 12, which can be operated by the deflection of a shaft 75 corresponding to the shaft 58 of the arrangement shown on Fig. 11. The operating cams and levers keyed on this shaft are keyed-on in such a manner relatively to each other as to ensure that the throttling of the passages should be preferably effected simultaneously in two sheet bundles, and that it is in one sheet bundle that the passages should remain free. By the gradual deflection of the shaft the place of throttling will be altered in such a manner that the throttling of the passages will take place periodically in a given order of sequence. It is preferable to make provision for the constant uniform alteration of these bundles which are to be connected to throttling, by means of the cleaning equipment, whereby it becomes possible to completely prevent any clogging by dirt of the heat exchanger. In view of the fact that in this manner the heat exchanger will assume temperatures varying during operation, it is necessary in order to avoid any detrimental stresses, to employ, at least on one place, a dilatation member in the ducts leading to the heat exchanger. Such a dilatation device [a pipe branch having the shape of bellows] is shown on Fig. 13, where the pipe branches 76 have been inserted into the duct of the working fluid of higher pressure.

The more detailed investigation of the heat exchanger reveals the fact, that a heat exchanger presenting small losses, and capable of being well utilized, can only be constructed with the aid of such sheet bundles in which the length of the path of travel of the gas is less than 1.2 m. Accordingly, in such cases, when the heating or cooling of a greater quantity of gas is concerned, it is necessary to employ a great number of sheet bundles mutually connected in parallel, in consequence whereof it is necessary to make provision for ensuring that the weight of the outlet and inlet pipe branches and similar fittings should not amount to a too large proportion of the total weight of the sheets.

In view hereof, it is, as far as possible, always advisable to unite the sheets—as has been shown on the preceding figures—at least in pairs, and even sheets united in groups of three can also be considered, whilst on the other hand, of course, in case of necessity, simple non-united sheets can also be employed.

Sheet bundles shown by way of example, composed of such sheets, and fitted with outlet and inlet branches are shown (with simple sheets) on Fig. 14, or (with sheets united in groups of three) on Fig. 15, the said bundles being, similarly to the arrangement according to Figs. 11, 12 and 13, fitted with the simple inlet and outlet branches 77—78, or respectively with the united inlet and outlet branches 79—80 for the high-pressure working fluid only. By thus dispensing with the inlet and outlet branches for the working fluid or lower pressure, and particularly by employing the uniting arrangements as shown on Figs. 16 and 17, the weight of the fittings now discussed can really be diminished to the lowest possible limit. As against this, it should be remarked, that in such cases when the pressure of the working medium of lower pressure also differs from the ambient pressure, inlet and outlet branches are of course required for this last-named working fluid also, which, however, can, on the analogy of the uniting arrangements to be described with reference to Figs. 16 and 17, also be simplified within a heat exchanger of greater size, so as to enable substantial savings in weight to be attained likewise.

Figs. 16 and 17, to which reference has been made repeatedly, are diagrammatically representing a method of assembly, shown by way of example, of such heat exchanger batteries of substantial size. According to these figures the individual sheet bundles 81—82 are, in order to obtain an as great simplification and unification as possible of the inlet and outlet branches, arranged in such a manner relatively to each other, that at the places of meeting of the inlet and outlet openings 83 and 84, respectively, of the sheet bundles, such parts of these bundles as belong to working fluids of identical pressure and temperature will meet, and will, for the purpose of the distribution of one or the other working fluid, leave the ducts 85, 85' and 86, 86', respectively at right angles to the sheets of the sheet bundle, open. These ducts are then—according to the various working fluids—joining-on to further ducts 87, 87' or 88, 88', respectively, which are effecting the united inlet and outlet of the working fluids.

It is an advantage, deriving from their type of design, of the embodiments described by way of example, that their heat-transmitting surfaces are operating with a high coefficient of heat transfer, and therefore the capacity of a heat exchanger intended for transmitting a given quantity of heat is small. This circumstance is of importance from the point of view of the regulation of the turbine, notably if at a part load the heat exchanger operates at a temperature different from that at which it operates at full load, the stationary thermal condition will in the case of a set of equipment fitted with a heat exchanger of such a type be arrived at more rapidly than in the case of heat exchangers of other types of design, and this is a circumstance advantageous also regarding the efficiency of the equipment.

In view of the fact that the abovementioned friction loss due to through flow will be all the greater, the greater the throughflow speed of the working fluid, it will be preferable to employ a low speed; in this case, however, a high coefficient of heat transfer, and thus, a good utilization of the sheets can only be obtained if the mutual distance of the sheets is small, which circumstance makes it necessary to employ very small distances between sheets. Notably, in the case of a purely laminar flow the coefficient of heat transfer between the flowing medium and the sheets is

$$\alpha \sim \frac{\lambda}{\delta}$$

in which formula λ is the coefficient of heat conduction in a condition of rest of the fluid, whilst δ is the thickness of the sheets. For this reason the mutual distance between the sheets should preferably be less than 2 mm, but it is possible to diminish this distance to as far as 0.5 mm, or even less. In order to prevent any excessive quantity of heat being conveyed by the sheets by conduction from the hot end of the heat exchanger towards its cold end, it is advisable to employ sheets of very small thickness. Accordingly, the thickness of the sheets shouuld be preferably less than 1 mm, e. g. 0.3 mm.

In order to ensure that in the interests of an as favourable efficiency of the heat exchanger as possible, the working fluid to be heated should be able to leave the heat exchanger at nearly the same temperature at which the fluid to be cooled enters it, it is necessary that on both sides of a sheet the weights of the fluid flowing in these parts of the heat exchanger should be mutually equal. The fulfilment of this requirement can be ensured by keeping the proportion of the gaps at both sides of the sheets

$$\left(\frac{s1}{s2}\right)$$

at a constant figure for the various sheets. Similarly it is important, that the condition according to which fluids of equal weight should flow on both sides of the sheet should be satisfied not only for each sheet, but also for each strip of the working parts of the sheets along the length of the flow. The fulfilment of this condition can be ensured if the two working fluids are handled in an equal manner from the point of view of guidance, i. e. if the resistance to flow is rendered equal along any line of flow of the sheet.

From the point of view of the operation at a favourable efficiency of the gas turbine plant is it most essential that the heat exchange device described should be used so as to ensure that the suitable conditions of operation of that device should be permanently maintained. In view of the fact that for the transmission of a given quantity of heat with a given difference of temperatures and with a given flow speed a smaller amount of friction work has to be performed in the case of a flow of laminar character, and as thus the quantity of loss of energy due to friction with which it is necessary to count will be substantially smaller in the case of laminar flow than in the case of a flow of turbulent character, it will be advisable to use the equipment according to the invention under conditions of service likely to assure this. The condition of the flow between the sheets of the heat exchanger being of a laminar character is that the so-called Reynolds figure characteristic for the flow

$$R = \frac{4F}{K} \cdot \frac{c\phi}{\eta}$$

should be lower than a certain critical figure amounting to about 2300, in which formula F is the cross-section of some flow separated in itself, K is the periphery of this cross-section [see the cross-section shown in dotted lines on Fig. 5], c is the average speed of flow, ϕ is the density of the flowing working fluid and η is the factor of viscosity of the fluid. It should be particularly emphasized in connection herewith that what is substantially decisive for the question whether the flow is of a laminar character or not is whether the condition raised above regarding the Raynolds figure is satisfied or not and not whether the flow itself is absolutely laminar in all its flow cross-sections or at any point of any flow cross-section, because the formula, in view of the average figures contained in it, will also be satisfactory in these cases when turbulences, gradually fading-out [abklingend], are set up in places in the flow. The flow will, if its characteristic figures satisfy the limit requirements fixed regarding the Reynolds formula, be of a laminar character also in spite of such phenomena of a turbulent nature, and in fact the creation of such local turbulences, which on their part do not modify the condition of the flow being of laminar character, is, from the point of view of improving the coefficient of heat transfer, not only permissible but directly desirable, regarding which fact an example has been stated in what precedes for the case of the employment of the sheets according to Figs. 8 and 9, respectively. Notably, such local turbulences without increasing on their part the friction resistance of the throughflow in any appreciable extent, are stirring and mixing the so-called boundary layer i. e. that layer of the flowing working fluid which is immediately adjacent to the sheets and which on the one hand, is moving at a diminished flow speed, whilst on the other hand, it acts in a certain extent as a heat insulating layer, and accordingly such local turbulences improve and facilitate heat transfer. As appears from the Reynolds formula, the satisfaction of the limit condition for the flow being of laminar character depends on the one hand on the type of design and suitable dimensioning of the heat exchanger, and on the other hand, on the working conditions which have to be chosen so as to be in conformity therewith [e. g on the figure of the average throughflow speed] and thus it will in any case be possible to ensure by means of the heat exchanger described as part of the invention, i. e. by the apparatus according to the invention, that it should be possible to create and maintain this working condition, essential and advantageous for the operation of the gas turbine plant, whilst at the same time employing a heat exchanger of a type which will be the most suitable from other points of view also.

In a gas turbine plant, assumed by way of example, which operates as constant pressure and in which the working fluid is compressed by the compressor from atmospheric pressure to a pressure of 2 atmospheres [absolute], and in which, further, the temperature ruling at the inlet into the turbine is 450 deg. Centigrade, a quantity of heat of 27,3 cal. has to be introduced during the combustion of each kg of working fluid, as against which the working fluid will absorb a quantity of heat amounting to about 64 cal. per kg in the heat exchanger. A heat loss of 10% in the heat exchanger will accordingly already increase the total quantity of heat to be introduced by 23.5%, and will thereby diminish the total efficiency in the same extent. In the case of this plant the useful work per kg of gas flowing through the turbine amounts to about 4270 mkg, and accordingly each $^1/_{100}$ of an atmosphere of pressure loss set up in the heat exchanger is liable to diminish the useful work by about 200 mkg, i. e. to diminish the efficiency by about 4.7%. These figures show, in what a great extent, with what degree of perfection exceeding that required by general technical practice the heat exchanger must operate advantageously, in order to prevent that it should diminish the efficiency of the gas turbine plant in an extent which would already render the creation of the gas turbine impossible from an economic point of view.

The gas turbine plant according to the invention, equipped with the heat exchanger described, will, particularly if the service condition outlined above is adhered to, or rather because it enables this condition to be kept easily, owing to its type of design, suit the requirements, difficult to be satisfied, raised, which no gas turbine plant fitted with any other kind of heat exchanger, suitable for any other kind of technical employment, would be able to satisfy. Notably, the heat exchanger of the type described will, not only because it renders throughflow of a laminar character possible, but also for other reasons, be suitable, in case of its being arranged and dimensioned in a suitable manner, for enabling the resistance to flow of the working fluid flowing through it, and therewith the loss resulting from such resistance and substantially influencing the total efficiency, to be reduced to a minimum, without deteriorating the coefficient of heat transfer and indeed whilst substantially improving the latter. In obtaining these advantages, on the other hand, a substantial role is played by the consideration, discussed above, according to which it should be possible to keep the working fluids, each separately, but if possible also in mutual dependence, flowing in the working spaces of the heat exchanger in such a manner that working conditions should be approximately mutually equal at all points of the flow. As appears from the above, this is rendered possible in the most satisfactory manner, and whilst ensuring a high coefficient of heat transfer, by means of a heat exchanger composed in the manner described of bundles of sheets. The heat exchanger described accordingly represents a substantial and advantageous necessary accessory of the gas turbine plant, by means of which the economy of the operation of the whole plane can be influenced in a decisive manner.

In the heat exchanger it is, in case the latter operates at high temperatures, advisable (at least in the vicinity of the hotter part of the latter) to employ sheets made of sutiable heat-proof material.

The compressor, the gas turbine proper, as well as the other accessories of the whole equipment can, of course, within the scope of the invention, be of any type of design corresponding to the present stage of development of the art of engineering, provided that they are on their own part also suitable, on the basis of present-day knowledge, for assuring a high total efficiency, and within one set of apparatus it is also possible to employ more than one compressor or turbine, which may possibly be mechanically independent of each other.

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