Sept. 8, 1959 W. LENZ 2,902,830

STEAM POWER PLANTS

Filed June 28, 1956 4 Sheets-Sheet 1

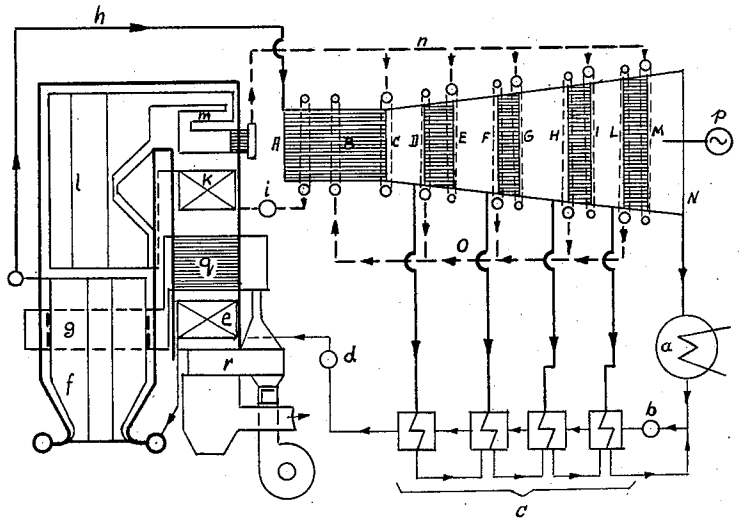



FIG. 1

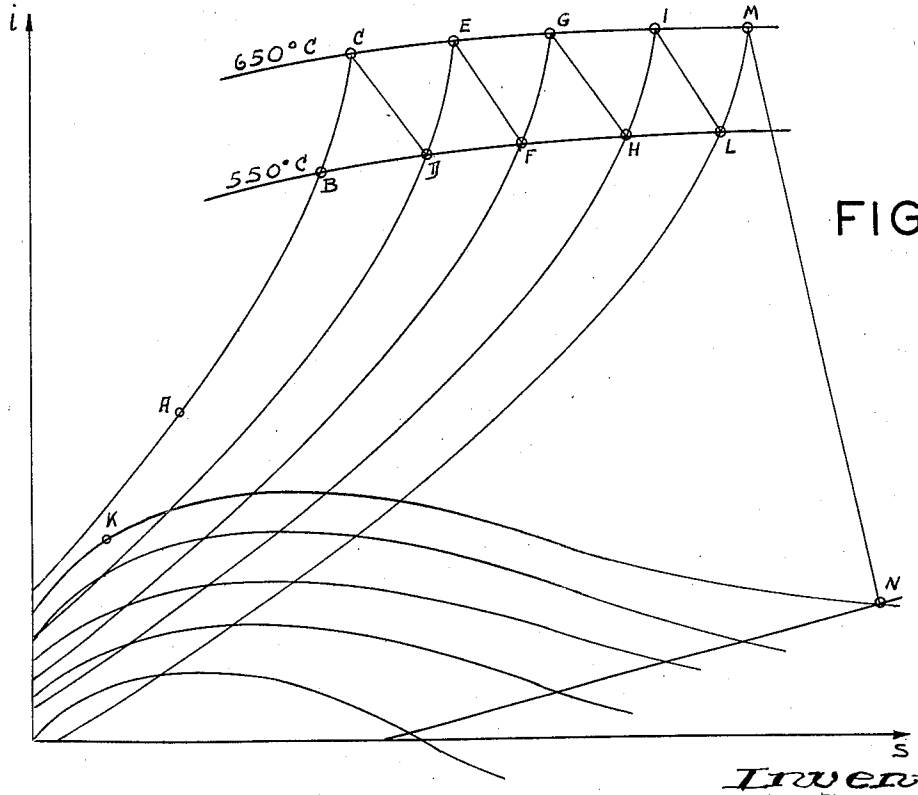



FIG. 2

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STEAM POWER PLANTS

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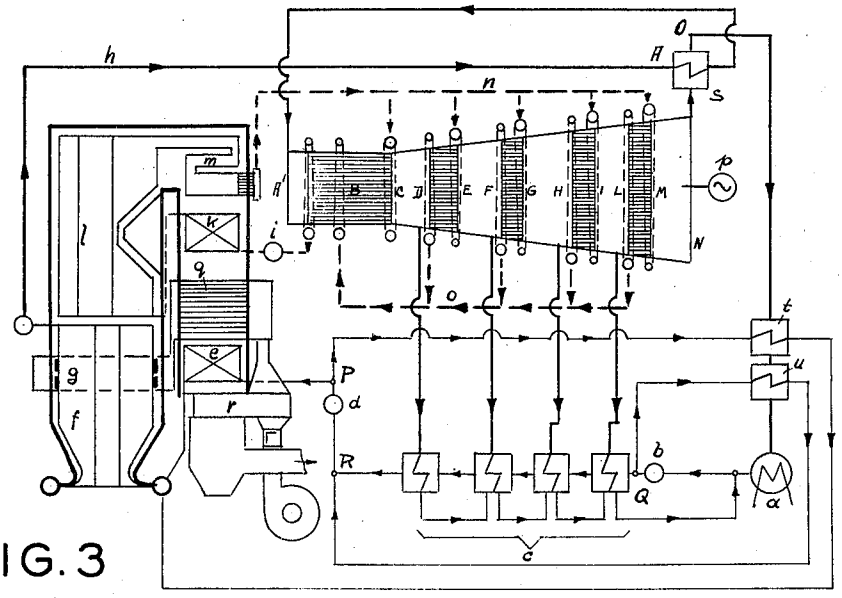



FIG. 3

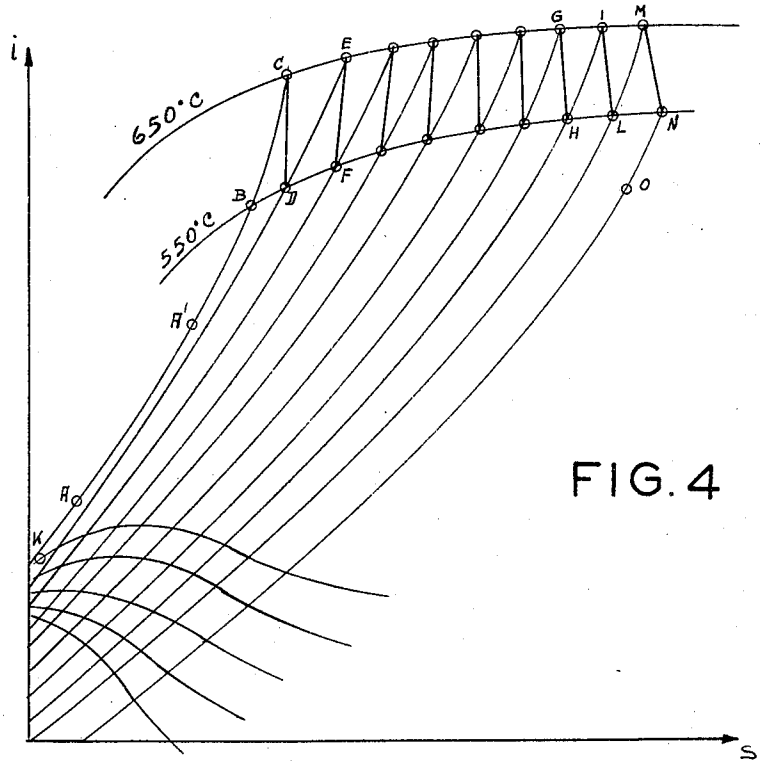



FIG. 4

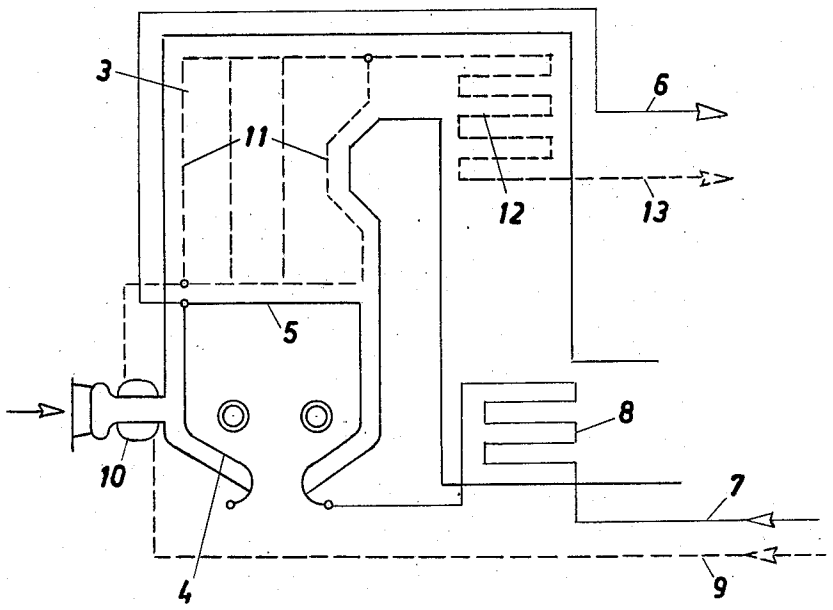
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STEAM POWER PLANTS

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FIG. 5




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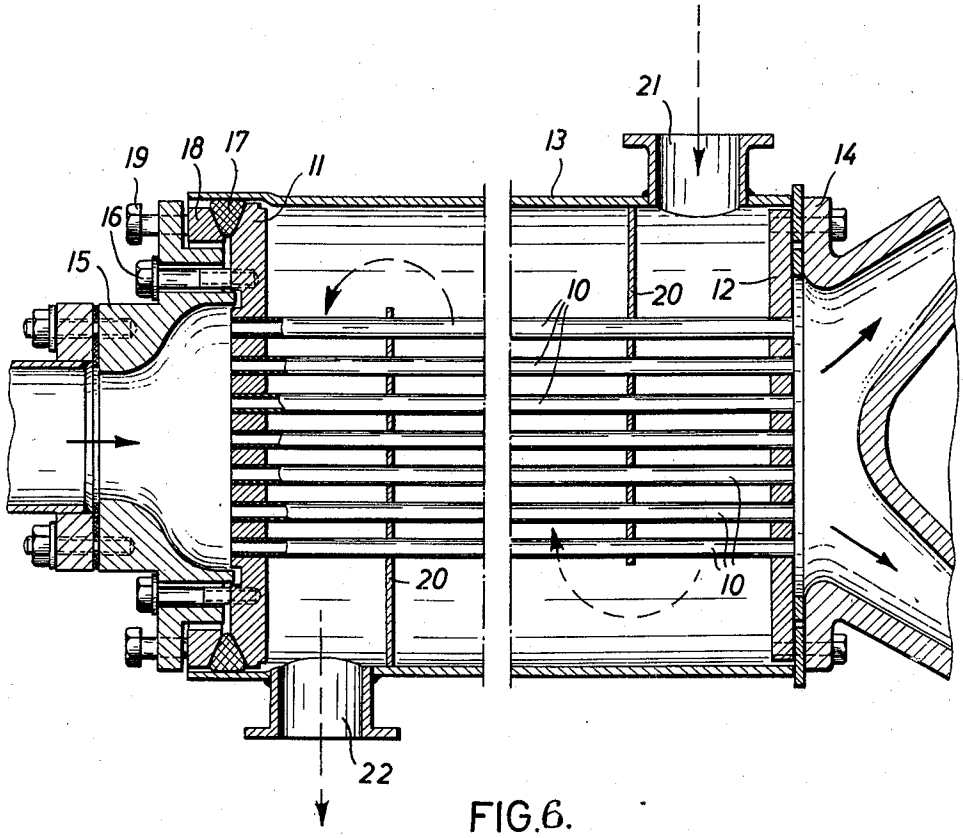



FIG.6.

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United States Patent Office 2,902,830
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2,902,830

# STEAM POWER PLANTS

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Application June 28, 1956, Serial No. 594,535
Claims priority, application Germany July 2, 1955
11 Claims. (Cl. 60—73)

It is known that the thermal efficiency of a steam circuit between a steam generator and a steam turbine can be substantially increased if the average temperature of the energy conversion is kept as high as possible. For this reason isothermal expansion is aimed at a temperature which is equal to the highest temperature arising in the circuit. It can be approximated by multiple intermediate superheating during the expansion, as represented by a sawtooth curve in the entropy-total heat diagram. Technically, however, a repeated return of the steam from the turbine to the boiler and from the latter back to the turbine would cause so high a pressure loss in the steam that a large part of the thermal gain would be lost. Also the plant would be made much more expensive and its construction would be so complicated and so difficult to supervise that its operation would be made much more difficult.

Furthermore, a high starting pressure is also necessary for multiple intermediate superheating. If now the steam is superheated to the final temperature in the boiler difficulties are met with in the last part of the superheater and in the pipeline from the superheater to the turbine because these are at the same time exposed to high pressure and temperature stresses. In addition, in the case of superheaters heated by flue gases difficulties arise due to the high temperature of the flue gases which lead to high temperature stresses in the tube walls, while the great wall thicknesses which the additional temperature stresses require may lead to non-uniform heating of the heat transfer surfaces. For these reasons, in order to achieve adequate cooling of the superheater tubes high steam speeds are necessary which in turn again result in disadvantageous increased pressure losses.

To overcome the first mentioned disadvantage it has already been proposed to effect the intermediate superheating not in the steam generator itself but in a heat carrier which is brought to a high temperature in the steam generator and which transfers the heat to the steam in the intermediate stages, the intermediae superheater being mounted near the turbine.

According to the invention, the advantages above set out of isothermal expansion in as finely divided as possible a sequence of stages are ensured by using for all the superheating steps divisional currents of heat carrier working in parallel, which carrier is used also for heating the steam to the turbine entry temperature immediately before it enters the turbine. A heat carrier is used of which the boiling point at normal pressure is above its highest operating temperature.

Such fluid heat carriers are in themselves known. To increase the heat transfer, they are circulated by a pump and as well as having a low vapour pressure they have a melting point which is low enough to ensure that they still remain fluid at the lowest temperature occurring in the circuit. Further, they have as good as possible a heat conductivity in order to achieve high values of heat transfer, a not too high specific weight in order to keep



the pressure losses low and also have no chemical affinity for steel at the temperatures arising, of which material the heating surfaces, pipelines, intermediate superheaters and circulating pump consists. Also they must not react with water, as contact between the heat carrier and water can easily occur where there are leakages or where damage occurs.

Molten metals such as lead or a lead bismuth or lead magnesium alloy have been found best, as such heat carriers, but molten salts can also be used. The pressure losses in the circuit which have to be provided by the circulating pump can be kept very low, as with liquids only relatively low speeds are necessary in order with good heat conductivity to attain a high figure of heat transfer. For this reason and on account of the small volume of the liquid, the energy requirements of the pump are substantially smaller than the saving in relation to the energy loss involved with multiple intermediate superheating with direct heating. Also on the steam side the pressure losses in the intermediate superheaters heated by the heat carrier can be kept low, because on account of the good heat transfer, the heating surfaces can be relatively small and the intermediate superheaters can be arranged close to the turbine.

With such an arrangement of the steam circuit the above mentioned advantage is obtained that the superheating of the steam by flue gases does not have to be carried up to the turbine entry temperature but only to a temperature which lies below the lowest temperature of the heat carrier. This, again, is determined by the temperature rise in the intermediate superheating. In this way a considerable separation of the highest pressure stresses and the highest temperature stresses in the tubes is obtained, because the highest temperature of the steam arising in the steam generator is relatively low and in spite of high pressure this can be resisted with heating surfaces and pipelines of low alloy material with moderate wall thicknesses, while at the high temperatures to which the heat carrier must be raised there is practically no raised pressure, so that here also the wall thicknesses can be kept small, temperature stresses in the heated tubes can be avoided and also the pipeline for the heat carrier from the steam generator to the turbine can be constructed with relatively thin walls. Wall temperatures of the final heating surfaces for the heat carrier are then, it is true, higher by a certain amount than the highest temperature of the heat carrier itself and this again by a certain amount higher than the highest steam temperature taking part in the production of power, but in spite of this owing to the separation of the locations of pressure and temperature stresses, the mastering of these maximum wall temperatures is easier than in superheaters heated by flue gases for taking the steam up to its maximum temperature.

The steam is brought to the full superheat temperature (600°/650° C.) directly before the first guide blade ring of the turbine by a superheater heated by a heat carrier. Only in the last part of this superheater do high pressure and temperature stresses arise together in the tubes carrying the steam. Here, however, a tube diameter substantially smaller than in the steam generator can be used and the tubes can be placed closer together because there is no risk of deposits. For this reason, again, the wall thickness and therefore the temperature stresses can be kept low.

As all the intermediate superheaters according to the invention are arranged in parallel between the pipeline for the hot heat carrier and the return pipeline for the cooled heat carrier, so that each intermediate superheater is traversed by a divisional current of the heat carrier, only a single pipeline for the hot heat carrier and a single

return pipeline for the cooled heat carrier is necessary. If the intermediate superheaters are traversed by the steam and the heat carrier in countercurrent, it is then possible to superheat the steam each time to the inlet steam temperature. Countercurrent flow can be obtained by providing a steam-carrying pipe with a concentric pipe about it and leading the heat carrier in the intermediate space in the opposite direction to the steam or by letting the heat carrier flow in a cross direction over the steam-carrying pipes but directing it by walls in such a way that overall the heat transfer takes place in countercurrent.

In order to keep the pressure loss on the steam side as low as possible the steam carrying pipes are made straight and the exit speed of the steam from the last row of rotor blades before each intermediate superheating stage is transformed with as little loss as possible into pressure by suitably shaped guide blades. In this way the speed is reduced to the value necessary for the transfer of heat. These guide blades at the same time take care of the distribution of the steam between the individual tubes of the intermediate superheater. The supply of the heat carrier is effected as uniformly as possible over the whole periphery of the turbine in spirals, and the outflow of the heat carrier can take place in a similar manner at the outer periphery.

Further, from a particular value of steam pressure on, no further intermediate superheating is effected. At higher pressures, on the steam side the value of heat transfer is so high on account of the great density of the steam that only relatively small heating surfaces, that is, short tubes, are needed in order to transfer the heat necessary for the intermediate superheating, so that only small pressure loss is suffered. The stated steam pressure is chosen either so that during the expansion from this pressure down to the condenser pressure the saturated steam curve is reached or even downwardly crossed or so many intermediate superheating stages are provided that the exhaust steam of the turbine only has a certain degree of superheat.

The above mentioned temperature at which the steam leaves the steam generator can be substantially lower than the lower temperature limit of the intermediate superheater stages. Heating from the exit temperature from the steam generator up to the lower temperature of the intermediate superheating stages can be effected in this case in two ways. One possibility is to carry the whole return flow of the cooled heat carrier from the intermediate superheaters to the first superheater at a place at which the superheater heated by the heat carrier has the same temperature as the subdivided current of the heat carrier, and making this superheater such a size that the collected total return flow of the heat carrier heats the steam from the exit temperature of the steam generator at which it enters the steam pipeline leading to the turbine up to the lower temperature of the intermediate superheaters. The other possibility is that with intermediate superheating effected at so low a pressure that the steam at condenser pressure is still superheated, the superheat of the steam leaving the turbine is used to heat the live steam and the rest of the heating up, up to the lower temperature of the intermediate superheating stages, is effected by the total return flow of the heat carrier. The remainder of the superheat in the exhaust steam is taken up in a water preheater traversed by the whole or part of the feed water.

The steam generator necessary for supplying the steam circuit according to the invention is a heat transferring device which consists only in part of water or steam cooled heating surfaces and in part of about the same size of heating surfaces which are cooled by the liquid heat carrier. The rest of the heat transferring surfaces in the steam generator is taken by an air preheater for an air temperature of at least 400° C. In order not to make the heating surfaces of the air preheater too large, a two-part construction of the air preheater is advisable between



the parts in which a small water preheater is incorporated. Particularly with high pressures, the residual evaporating and superheating heat only suffices for a part of the heat take-up in the combustion chamber. The rest of the heating surfaces of the combustion chamber is formed by heating surfaces over which the heat carrier flows and by the contact and radiation heating surafces between the combustion chamber and the air preheater.

It is known that the heating surfaces of steam generators and therefore similarly as regards the heating surfaces used in the invention for heating of the heat carrier, can be considerably reduced if pulsating combustion is used.

As pulsating combustion takes a very short time, as is in itself known, very high combustion temperatures are reached with small burner volumes but it requires an intensive cooling of the burner nozzles. Hitherto burners have been cooled by tubes which are under full boiler pressure.

On account of the small diameter of the burners and the necessity to cover them on the inside with a heat insulating layer, in order to be able to draw off the ash in molten condition, it is very difficult to rectify any leakages which may develop in these cooling tubes.

According to the invention the burners are cooled by the heat carrier, which is not under pressure. As the heat carrier not under pressure is cooled the burner can simply consist of a double jacket of sheet metal on the inside of which the heat insulating layer is provided. Such a burner can not only be made more simply and be more reliable in operation than a burner constructed of cooling tubes but it is also more convenient for dismounting and simpler to repair if a leakage should develop, the likelihood of which is, however, much less than in the case of a burner for pulsating combustion provided with cooling tubes under full pressure.

The following are, by way of example, data for the circuit:

| | | |
|---|---|---|
| Temperature of the live steam and of intermediate superheating | ° C. | 650 |
| Lower temperature of the intermediate superheating | ° C. | 550 |
| Live steam pressure | At. | 300 |
| Temperature of the steam leaving the boiler | ° C. | 400 |
| Feed water temperature | ° C. | 200 |
| Flue gas offtake temperature | ° C. | 150 |
| Highest temperature of the heat carrier | ° C. | 700 |
| Temperature of the heat carrier leaving the intermediate superheaters | ° C. | 600 |
| Lowest temperature of the heat carrier | ° C. | 500 |
| Efficiency of the process | percent | 45 |

Figure 1 of the accompanying drawings is a diagram showing the layout of the plant for the case in which the intermediate superheating is only carried to such a value that at the end of the expansion the steam is no longer superheated.

Figure 2 is the corresponding entropy-total heat chart.

Figure 3 shows diagrammatically the layout of another plant in which the exhaust steam from the turbine is at the lower temperature of the intermediate superheating stages.

Figure 4 is an entropy-total heat chart corresponding to Figure 3, and

Figure 5 is a diagrammatic section of a boiler used in the plant according to the invention.

Fig. 6 is a section through a heat exchanger for superheating or reheating the steam.

The stages marked with capital letters in Figures 1 and 3 have the same capital letters indicating the corresponding state of the steam in the chart in Figures 2 and 4. In the charts, K indicates the critical point for water vapour. From the fact that the live steam isobars are above the point K, it will be seen that the live steam is above the critical condition. In Figures 1 and 3 the water circu-

lation is indicated by the thin solid line, the steam circulation by the thick solid line and the heat carrier circulation by the thick broken line.

In Figures 1 and 2, at A the steam enters with low superheat the first superheater heated by the heat carrier. The first divisional current of the heat carrier at high temperature enters this superheater at C, while at B the total return flow of the heat carrier from the intermediate superheaters enters this preheating superheater. The entire return flow of the heat carrier flows through the first superheater from B to A, where it leaves, after having raised the steam temperature to the lower temperature of the intermediate superheating stages. From A the heat carrier returns to the steam generator. From A to B the steam is raised to the lower temperature of the intermediate or reheater superheating stages, from B to C to the top live steam temperature and is then expanded from the top live steam pressure at C to the pressure of the first intermediate superheating or reheating stage at D, where it has reached the lower temperature limit of the intermediate superheating stages. In the first intermediate superheater or reheater between D and E it has its temperature again raised to the live steam temperature by the second divisional current of heat carrier, it is expanded further to F, where the second intermediate superheating stage by means of the third divisional current of the heat carrier commences and is completed at G and so on. Finally, at N the condenser pressure is reached. Before each intermediate superheating stage, advantageously a bleed-off for feed water preheating is provided.

The heat exchangers, as shown in Fig. 6, have straight extending-throughout steam tubes **10** which are welded between plates **11** and **12**. One of the plates **12** is rigid ly screwed with the outer mantle **13** of the heat exchanger and pipe connection **14** of the turbine. To the other plate **11** a connecting member **15** is joined by screws **16**. The resilient seal between the plate **11** and the exterior mantle **13** is brought about by a ring seal **17** of about triangular profile which can be tensioned by means of a pressure ring **18** and screws **19**.

The liquid heat carrier flows through the heat exchanger owing to the built-in guiding walls **20** in coil shape and is delivered or drained by connections **21** and **22**. Because of the straight steam tube **10** the resistance to flow of the heat exchanger for the steam is very slight.

The condensate formed in the condenser *a* is forced by the condensate pump *b* through a bleed-off preheater *c*, is brought to full pressure in the feed pump *d* and is forced through the feed water preheater *e* into the section of the boiler provided with the steam generating heating surfaces *f*. These advantageously lie in that part of the furnace in which the burners *g* are located. The steam then flows through the steam pipe *h* to the entry A of the above mentioned first superheater heated by the heat carrier.

The return flow of the whole of the heat carrier is effected by a circulating pump *i* which directs it through a preheater *k* in the walls *l* surrounding the upper part of the furnace and then through further heaters having radiation and contact heating surfaces *m* and thence through a pipe *n* to the entry to the intermediate super heaters. The divisional currents flowing out of the intermediate superheaters are collected in a pipe *o* and led at B into the first superheater. It will be understood that the steam expansions above referred to take place in a turbine and the output of the turbine is transformed into electrical energy in a generator *p*. The flue gases are reduced to the desired discharge temperature in a high temperature part *q* of an air preheater and a low temperature part *r*, between which the feed water preheater *e* is located.

Figure 3 shows the layout and Figure 4 the corresponding entropy total heat chart for the case in which the exhaust steam from the turbine is still at the lower temperature of the intermediate superheating stages. The intermediate superheating stages coming between the first two and the last two are omitted from Figure 3 and reference letters are therefore omitted from the corresponding stages in Figure 4. Here the expansion from M to N is down to condenser pressure but only down to the lower temperature of the intermediate superheating stages. The first part of the superheat of the exhaust steam is used in a heat exchanger *s* to bring the live steam from condition A to condition A', at which it now enters the first superheater. The exhaust steam then flows through a heat exchanger *t* in which the whole or a part of the condensate after outflow from the bleed-off preheater *c* is further heated. It is advantageous, in order to obtain the same temperature differences in the flue gas heated feed water preheater *e* as in the first example, only to branch off a part of the feed water at the point P and after preheating this in the preheater *t* to send it direct into the boiler part *f* without passing through the preheater *e*. The more rapid heating up of the other part of the feed water by the preheater *e* gives less desirable temperature differences for the preheater *e* and the air preheater *q* and therefore requires larger heating surfaces for this, but the same outflow gas temperature can nevertheless be obtained as in the former example. On account of the preheating of the feed water to as great an extent as possible in the bleed-off preheaters, only a part of the residual superheat and exhaust steam can be extracted with the preheated feed water. Accordingly at Q part of the condensate is branched off and is preheated in a third heat exchanger heated by the exhaust steam up to about the same temperature at which the other part comes out of the bleed-off preheaters, whereupon it is added at R to the bleed-off preheated part. Less bleed-off steam can then indeed be used, while more steam does its full work in the turbine and the superheat in the exhaust steam is fully utilised. The pressure loss in the exhaust steam in the three heat exchangers *s*, *t*, *u* results in a certain energy loss but this is less than the heat recovery.

Between the two limiting cases in which the exhaust steam has no superheat and in which it has the lower temperature of the intermediate superheating stages, any desired intermediate stage is possible. The expansion in the turbine is then carried after one of the intermediate superheating stages down to condenser pressure and for example, the heat exchanger *s* is made smaller or entirely omitted while the heat exchanger *t* for feed water may also be reduced in size or even omitted.

Figure 5 diagrammatically illustrates in vertical longitudinal section a boiler plant with pulsating combustion provided with burners cooled in accordance with the present invention, in which a heat carrier not under pressure is used. In Fig. 5 the path of the liquid heat carrier is indicated by dash lines and leads over the cooling mantle **10** of the pulsating burners.

The shaft form furnace **3** with a conical slag discharge hopper **4** is covered in its lower part with cooling tubes **5** in which steam is generated under the prescribed pressure and at a temperature which is somewhat above the corresponding saturated steam temperature. Through a pipe **7** the condensate reaches the preheater **8** and thence is returned into the boiler. The fluid heat carrier not under pressure which is cooled down in the intermediate superheaters through a pipe **9** reaches the burner cooling jackets **10** and thence passes into the radiation heated section **11**, the contact heated section **12** and a pipe **13** to the intermediate superheaters, where it gives up a part of its heat content to the steam and heats the latter to the turbine inlet temperature. Differing from this arrangement, the burner cooling can also constitute the final stage in the heating up of the heat carrier, if the higher temperatures arise in the burners themselves.

I claim:

1. A steam circuit comprising piping, a steam boiler having a furnace, a multi-stage steam turbine, intermediate superheaters for superheating the steam in the circuit

between one turbine stage and the next, a first superheater at the first turbine stage for superheating live steam from said boiler to the first turbine stage, means for circulating a fluid heat carrier through said intermediate superheaters arranged in parallel to effect the heating thereof and through said boiler to heat the heat carrier, the heat carrier being under low pressure and having a boiling point at that pressure which is above the highest temperature employed in the plant, the parts of said boiler in which water is heated and steam is generated surrounding at least part of said furnace and the remainder of the heating surfaces of the boiler serving to heat said heat carrier, said boiler including further heat exchange devices through which flue gases from said boiler pass after heating said steam and heat carrier for heating of materials for ancillary services to the boiler by the flue gases whereby the flue gases are cooled to a lower temperature than the lowest temperature of said heat carrier during its circulation.

2. A steam circuit as set forth in claim 1 wherein each of said intermediate superheaters comprises steam tubes and means for conveying said heat carrier along the outside of said tubes in counter-current flow to the flow of the steam.

3. A steam circuit as set forth in claim 2 wherein said intermediate superheaters comprise said steam tubes each surrounded by a concentric tube through which said heat carrier flows in a counter-current direction.

4. A steam circuit as set forth in claim 2 comprising guide walls in each said intermediate superheater located to guide said heat carrier in a path which crosses said superheater tubes but which is overall in counter-current to the steam flow within the tubes.

5. A steam circuit as set forth in claim 1 where said furnace incorporates burners operating under pulsating combustion, said burners being provided with coloing jackets through which said heat carrier is circulated.

6. A steam circuit as set forth in claim 5 wherein the heat carrier on its return to the boiler first passes through said cooling jackets.

7. A steam circuit as set forth in claim 5 wherein the heat carrier in its passage through the boiler last passes through said cooling jackets.

8. In a steam power plant including a steam boiler and at least one multi-stage steam turbine, a pipe system for steam and water, a superheater at said turbine for superheating steam from said boiler, reheaters arranged between the stages of said turbine, a pipe system for a



liquid heat carrier, said boiler including a combustion chamber lined with one set of tubes in which the steam for said multi-stage turbine is generated and another set of tubes in which said heating medium is heated, said sets of tubes being arranged in two systems built into the boiler of which the one tube system disposed where the flue gases in the combustion chamber are cooler serves for generating steam for operation of said turbine and the other tube system disposed adjacent the high combustion temperature portion of the combustion chamber contains said liquid heat carrier comprising a non-evaporating liquid which flows through said pipe system, passes through said superheater and reheaters and serves for heating the steam in said superheater and reheaters before and between the turbine stages.

9. A steam power plant according to claim 8, in which each said steam reheaters disposed in front of each turbine stage has straight tubes for conducting the steam around which the heat carrier is conducted in counter-current or transverse current.

10. A steam power plant according to claim 8, and including a preheater between said boiler and said superheater and, in which the liquid heat carrier is heated to the required hot steam temperature and the pipe system therefore is divided between the superheaters and the reheaters arranged in parallelism, and the parallel portions of said pipe system are recombined into a single pipe to heat the steam in said preheater.

11. The power plant of claim 8, in which the heat carrier is selected from the group consisting of a molten salt, lead, lead bismuth mixture, and lead manganese mixture.


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