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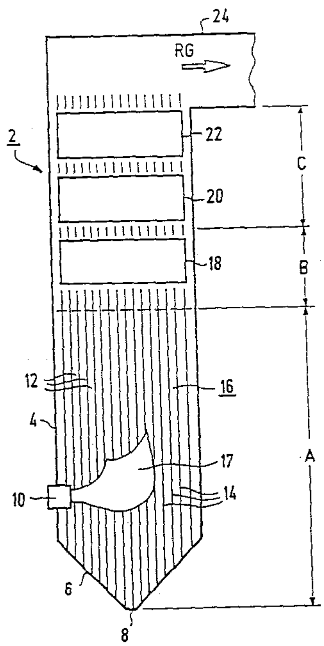
(54) **GENERATEUR DE VAPEUR CONTINU**
(54) **CONTINUOUS STEAM GENERATOR**

$$m = \frac{qi}{C(T_{max} - T_{krit} - \Delta T_w)} \quad (kg/m^2s) \qquad (7)$$



(57) L'invention concerne un générateur de vapeur continu (2) comportant une chambre de combustion (4) dont les tuyaux (12) verticaux ont une face intérieure présentant une structure superficielle (26) et peuvent être traversés de bas en haut par une substance en écoulement (S). Dans ce générateur de vapeur continu (2), une densité de flux massique m particulièrement avantageuse dans les tuyaux (12), pour une charge créant une pression critique (p_{crit}.) dans les tuyaux (12), correspond selon l'invention à la relation (7).

(57) The invention concerns a continuous steam generator (2) having a combustion chamber (4) with vertically extending pipes (12) which have a surface structure (26) on the interior. A flow medium (S) flows upwards through the pipes (12). According to the invention, a particularly advantageous mass flow density m in the pipes (12), at a load at which critical pressure (p_{crit}) prevails therein, corresponds, according to the invention, to the relation (7).

GR 96 P 3052 P

Abstract

Once-through steam generator

In a once-through steam generator (2) having a combustion chamber (4), in which a flow medium (S) is capable of flowing from below upwards through its vertically extending tubes (12) which have a surface structure (26) on their inside, an especially favourable mass flow density $\dot{m}$ in the tubes (12) at that load at which critical pressure (p_{crit}) prevails in the tubes (12) conforms, according to the invention, to the relation:

$$\dot{m} = \frac{q_i}{C(T_{max} - T_{crit} - \Delta T_w)} (kg/m^2 s)$$

FIGURE 3

File this amended text

Description

Once-through steam generator and method for designing a once-through steam generator

The invention relates to a once-through steam generator with a combustion chamber surrounded by a containment wall composed of tubes connected to one another in a gastight manner, a flow medium being capable of flowing from the bottom upward through the vertically extending tubes which have a surface structure on their inside. It relates, further, to a method for designing a once-through steam generator of this type.

A steam generator of this type is known from the paper "Verdampferkonzepte für Benson-Dampferzeuger" ["Evaporator concepts for Benson steam generators"] by J. Franke, W. Köhler and E. Wittchow, published in VGB Kraftwerkstechnik 73 (1993), No. 4, pages 352 to 360. In a once-through steam generator of this type, in contrast to a natural circulation or forced circulation steam generator with only partial evaporation of the water/steam mixture, the heating of evaporator tubes forming the combustion chamber leads to the complete evaporation of the flow medium in the evaporator tubes in a single pass. Whereas, in the natural circulation steam generator, the evaporator tubes are basically arranged vertically, the evaporator tubes of the once-through steam generator may be arranged both vertically and spirally, hence at an inclination.

A once-through steam generator, the combustion chamber walls of which are composed of vertically arranged evaporator tubes, can be produced more cost-effectively than a once-through steam generator having spiral tubing. Furthermore, once-through steam generators with vertical tubing have lower water-side/steam-side pressure losses than those with evaporator tubes which are inclined or are arranged so as to ascend spirally. Furthermore, in contrast to a natural circulation steam generator, a once-through steam generator is not subject to any pressure limitation, so that fresh steam pressures well above the critical pressure of water (P_{crit}= 221 bar), where there is only a slight density difference ....

between the liquid-like and steam-like medium. High fresh-steam pressures are necessary in order to achieve high thermal efficiencies and consequently low CO_2 emissions.

A particular problem, in this case, is to design the combustion-chamber or containing wall of the once-through steam generator with regard to the tube-wall or material temperatures which occur there. In the subcritical pressure range up to about 200 bar, the temperature of the combustion-chamber wall is determined essentially by the value of the water saturation temperature, when wetting of the heating surface in the evaporation zone can be ensured. This is achieved, for example, by the use of internally ribbed tubes. Tubes of this type and their use in steam generators are known, for example, from European Patent Application 0,503,116. These so-called ribbed tubes, that is to say tubes with a ribbed inner surface, have particularly good heat transmission from the inner wall to the flow medium.

In the pressure range of about 200 to 221 bar, the heat transmission from the tube inner wall to the flow medium decreases sharply, so that the flow velocity - the mass flow density usually being used as a measure of this - has to be increased correspondingly, in order to ensure that the tubes are cooled sufficiently. Consequently, in the evaporator tubes of once-through steam generators operated at pressures of approximately 200 bar and above, the mass flow density and therefore the pressure loss due to friction must be selected higher than in once-through steam generators which are operated at pressures of below 200 bar. Particularly in the case of small tube inside diameters, the higher pressure loss due to friction cancels out the advantageous property of vertical tubing but, when there is multiple heating of individual tubes, their throughput also rises. However, since high steam pressures of more than 200 bar are required in order to achieve high thermal efficiencies and therefore low CO_2 emissions, it is necessary, in this pressure range too, to ensure good heat transmission. Consequently, once-through steam generators with a combustion-chamber wall having vertical tubing are conventionally operated with relatively high mass flow densities in the tubes, so as to ensure, in the unfavourable pressure range of about 200 to 221 bar, that there is always sufficiently high heat transmission from the tube wall to the flow medium, that is to say to the water/steam mixture. In this context, the publication "Thermal Engineering" I.E. Semenovker, Vol. 41, No. 8, 1994, pages 655 to 661, specifies a mass flow density at 100% load of about 2000 kg/m^{2}s consistently both for gas-fired and for coal-fired steam generators.

The object on which the invention is based is to specify, for tubes with a containing wall of a once-through steam generator, a design criterion which is suitable in terms of a particularly favourable mass flow density in the tubes.

This object is achieved, according to the invention, in that the steam generator is designed in such a way that the mass flow density $\dot{m}$ in the tubes of the containing walls at that load at which critical pressure p_{crit} prevails in the tubes conforms to the relation:

$$\dot{m} = \frac{q_i}{C(T_{max} - T_{crit} - \Delta T_w)} (kg/m^2 s),$$

in which

q_i (kW/M^2) is the heat flow density on the inside of the tube,

T_{max} (°C) is the maximum permissible material temperature of the tube,

T_{crit} (°C) is the temperature of the flow medium at critical pressure p_{crit},
ΔT_w (K) is the temperature difference between the outer wall and inner wall of the tube, and
$C \geq 7.3 \cdot 10^{-3}$ kWs/kgK is a constant.

The invention proceeds from the consideration that, in the flow-related design of the internally ribbed tubes, two basically contradictory conditions have to be satisfied with regard to the mass flow density. On the one hand, the mean mass flow density in the tubes must be selected as low as possible. This is to ensure that a higher mass flow flows through individual tubes, to which more heat is supplied than to other tubes on account of unavoidable heating differences, than through tubes which have average heating. This natural-circulation characteristic known from the drum-type boiler leads, at the outlet of the evaporator heating surface, to an equalization of the steam temperature and consequently of the tube-wall temperatures.

On the other hand, the mass flow density in the tubes must be selected high enough that reliable cooling of the tube wall is ensured and permissible material temperatures are not exceeded. High local overheating of the tube material and the consequential damage (tube cracks) are thereby avoided. Essential influencing variables for the material temperature are, in addition to the temperature of the flow medium, the external heating of the tube wall and the heat transmission from the inner tube wall to the flow medium (fluid). There is therefore a connection between the internal heat transmission, which is influenced by the mass flow density, and the external heating of the tube wall.

The invention, then, proceeds from the finding that the connection between the internal minimum heat transmission coefficient α_{min} and the mass flow density $\dot{m}$ can be described in permissibly simplified form by the relation:

$$\alpha_{min} = C \cdot \dot{m} \qquad (1)$$

in which

α_{min} (kW/m²K) is the heat transmission coefficient,

$\dot{m}$ (kg/m²s) is the mass flow density in the ribbed tubes, and

C is a constant with the mean value of

$C = 7.3 \cdot 10^{-3}$ kWs/kgK for commercially available tubes. Depending on the structure of the inner surface of the tubes, this constant C can also be selected in the range between $7.3 \cdot 10^{-3}$ kWs/kgK and $12 \cdot 10^{-3}$ kWs/kgK.

The said relation gives an optimum mass flow density in the tubes which both results in a favourable throughflow characteristic (natural-circulation characteristic) and also ensures reliable cooling of the tube wall and consequently adherence to the permissible material temperatures.

A fundamental consideration in deriving the said relation for the mass flow density in the tubes is that, in the case of predetermined external heating of the tube wall - the so-called heat flow density (kW/M²), that is to say the heating per unit area, being used hereafter for this - the material temperature of the tube wall is only slightly, but definitely, below the permissible value. In this case, it is necessary to bear in mind the physical phenomenon that the heat transmission from the inner tube wall to the flow medium is most unfavourable in the critical pressure range of about 200 to 221 bar.

Comprehensive tests show that the highest material stress is obtained when a relatively low mass flow density is combined with the highest occurring heat flow density in the evaporation zone at about 200 to 221 bar. This is the case, for example, in that region of the combustion chamber in which the burners are arranged. If evaporation is subsequently terminated and steam superheating commences, the material stress on the tubes of a combustion-chamber wall decreases again. The reason for this is that, in a conventional burner arrangement and a conventional combustion cycle, the heat flow density also decreases.

It was found, furthermore, that, in other pressure ranges too, no heat transmission problems arise if, when ribbed tubes are used, sufficient cooling of the tube wall is ensured in the said pressure range of 200 to 221 bar. Thus, at low pressures, that is to say of below approximately 200 bar, the internal ribbing of the tubes causes critical boiling to commence only at the end of the evaporation zone, that is to say in a region having a reduced heat flow density. Critical boiling no longer occurs in the supercritical pressure range. Heat transmission, then, is so intensive that sufficient cooling of the tube wall is ensured.

To determine the optimum mass flow density $\dot{m}$ in the tubes of the tube wall, the said optimum mass flow density ensuring an advantageous throughflow characteristic on the one hand and reliable cooling of the tube wall on the other hand, the following procedure can be adopted:

Step 1:

Determination of the heat flow density q_a on the tube outside, based on the thermal calculation of that load at which a pressure of 210 bar prevails in the tubes of the tube wall. This heat flow density determined in this way must be increased by a factor of between 1.1 and 1.5, in order to allow for local irregularities in heat transmission.

Step 2:

Calculation of the maximum permissible material temperature T_{max} at the tube apex on the heated side of the tube wall. If it is assumed that the containing or combustion-chamber wall has a mean temperature which corresponds to the mean value of T_{max} and T_{crit}, the maximum thermal stress is calculated as:

$$\sigma_{max} = \frac{T_{max} - T_{crit}}{2} \beta \cdot E (N/mm^2), \quad (2)$$

with

σ_{max}	maximum thermal stress (N/mm^2)
T_{max}	maximum material temperature (°C)
T_{crit}	temperature of the fluid at the critical point (°C)
β	coefficient of thermal expansion (1/K)
E	modulus of elasticity (N/mm^2)

Since the stresses which are crucial here are thermal stresses, these can be guarded against as secondary stresses according to the ASME Code with triple the value of the permissible stresses σ_{per}. This results in the temperature T_{max} as

$$T_{max} = T_{crit} + \frac{6 \cdot \sigma_{per}}{\beta \cdot E} (°C) \quad (3)$$

The permissible stress can be taken from the particulars supplied by the tube manufacturers.

Step 3:

Conversion of the predetermined heat flow density q_a (related to the outside of the tube wall) to a heat flow density q_i which is related to the inner wall of the tubes:

$$q_i = \frac{K \cdot d_a}{d_i} \cdot q_a \; (kW/m^2) \quad (4)$$

The determination of the heat redistribution factor K is based on temperature field calculations and can be arrived at with sufficient accuracy as follows:

$$K = A \; (d_a^2 \cdot q_a) + B \quad (5)$$

with A = 0.45 and B = 0.625 for $(d_a^2 \cdot q_a) \leq 0.5$ kW and A = 0.25 and B = 0.725 for $(d_a^2 \cdot q_a) > 0.5$ and ≤ 1.1 kW and $A = 0$ and $B = 1$ for $(d_a^2 \cdot q_a) > 1.1$ kW, with

d_a = tube outside diameter (m)
d_i = tube inside diameter (m)
q_a = heat flow density on the outside (kW/m^2)
q_i = heat flow density on the inside (kW/m^2)

Step 4:

Determination of the temperature difference ΔT_w between the tube outer wall and the tube inner wall. The temperature difference ΔT_w is determined by means of the heat conduction equation:

$$\Delta T_w = \left(\frac{1+K}{2}\right) \frac{q_a \cdot d_a}{2\lambda} \cdot \ln \frac{d_a}{d_i} \quad (K) \qquad (6)$$

with λ = thermal conductivity of the tube material (kW/mK).

Step 5:

Determination of the necessary mass flow density $\dot{m}$ according to the relation:

$$\dot{m} = \frac{qi}{C(T_{\max} - T_{crit} - \Delta T_w)} \, (kg/m^2 s) \qquad (7)$$

An exemplary embodiment of the invention is explained in more detail by means of a drawing. In this:

Figure 1 shows a simplified representation of a once-through steam generator with vertically arranged evaporator tubes,

Figure 2 shows an individual evaporator tube in cross-section,

Figure 3 shows, in a graphical representation, curves E, F, G and H for the mass flow density in the case of different geometries of an evaporator tube consisting of the material 13 CrMo 44, and

Figure 4 shows, in a graphical representation, the dependence of the maximum permissible material temperature of 13 CrMo 44 on the permissible stress (N/mm^2).

Parts corresponding to one another are provided with the same reference symbols in all the figures.

Figure 1 shows diagrammatically a once-through steam generator 2 of rectangular cross-section, a vertical gas flue of which is formed from a containing wall 4 which merges at the lower end into a funnel-shaped bottom 6. The bottom 6 comprises a discharge orifice 8, not shown in any more detail, for ash.

A number of burners 10, only one of which can be seen, for a fossil fuel are mounted, in the lower region A of the gas flue, in the containing wall or combustion chamber 4 formed from vertically arranged evaporator tubes 12. In this region A, the vertically arranged evaporator tubes 12 are welded to one another via tube fins or tube webs 14 to form gas-tight combustion-chamber or containing walls. The evaporator tubes 12, through which the flow passes from the bottom upwards when the once-through steam generator 2 is in operation, form an evaporator heating surface 16 in this region A.

When the once-through steam generator 2 is in operation, a flame body 17 occurring during the combustion of a fossil fuel is located in the combustion chamber 4, so that this region A of the once-through steam generator 2 is distinguished by a very high heat flow density. The flame body 17 has a temperature profile which, starting from about the middle of the combustion chamber 4, decreases both upwards and downwards in the vertical direction and in the horizontal direction towards the sides, that is to say towards the corners of the combustion chamber 4. Located above the lower region A of the gas flue is a second flame-distant region B, above which a third upper region C of the gas flue is provided. Convection heating surfaces 18, 20 and 22 are arranged in the regions B and C of the gas flue. Located above the region C of the gas flue is a flue-gas outlet duct 24, via which the flue gas RG generated as a result of the combustion of the fossil fuel leaves the vertical gas flue.

Figure 2 shows an evaporator tube 12 which is provided with ribs 26 on the inside and which, while the once-through steam generator 2 is in operation, is exposed on the outside, within the combustion chamber 4, to heating at the heat flow density q_a and through which the flow medium S flows internally. At the critical point, that is to say at the critical pressure p_{crit} of 221 bar, the temperature of the flow medium or fluid in the tube 12 is designated by T_{crit}. The maximum permissible material temperature T_{max} at the tube apex 28 on the heated side of the tube wall is used for calculating the maximum thermal stress σ_{max}. The inside diameter and outside diameter of the evaporator tube 12 are designated by d_i and d_a respectively. In the case of internally ribbed tubes, it is necessary to use the equivalent inside diameter which allows for the influence of the rib heights and rib valleys. The tube-wall thickness is designated by d_r.

Figure 3 shows, in a system of coordinates, four curves E, F, G and H for different outside diameters d_a (mm) and tube-wall thicknesses d_r (mm). For this purpose, the heat flow density q_a (kW/m^2) on the tube outside is plotted on the abscissa and the preferred or optimum mass flow density $\dot{m}$ (kg/m^{2}s) is plotted on the ordinate. The curve E shows the trend for a tube outside diameter d_a of 30 mm in the case of a tube-wall thickness d_r of 7 mm. The curve F represents the trend for a tube outside diameter d_a of 40 mm in the case of a tube-wall thickness d_r of 7 mm. The curve G shows the trend of the mass flow density $\dot{m}$ in dependence on the heat flow density q_a for a tube 12 having an outside diameter d_a of 30 mm and a tube-wall thickness d_r of 6 mm. The curve H shows the trend of a tube 12 with an outside diameter d_a of 40 mm in the case of a tube-wall thickness d_r of 6 mm. The mass flow densities $\dot{m}$ are calculated for heat flow densities q_a of 250, 300, 350 and 400 kW/m^2 at the critical pressure p_{crit} of the flow medium S for the tube material 13 CrMo 44.

An example of the determination of the optimum mass flow density $\dot{m}$ is shown below. In this case, the following conditions are presupposed:

q_a = 250 kW/m^2; heat flow density on the tube outside at a pressure of 210 bar,

1.4 as raising factor for allowing for local irregularities in the heat transmission to the tubes 12,

d_a = 40 mm tube outside diameter, d_r = 7 mm tube-wall thickness, and tube material: 13 CrMo 44.

It follows from d_a and d_r that: d_i = 26 mm tube inside diameter.

<u>1st Step: Calculating the heat flow density</u>

The heat flow density based on thermal calculation is multiplied by the raising factor. This results in:

$$q_a = 350 \text{ kW/m}^2$$

<u>2nd Step: Determining the maximum permissible material temperature</u>

According to equation (3), this temperature is calculated at T_{crit} = 374°C (temperature of the fluid at critical pressure p_{crit}), with $\beta = 16.3 \cdot 10^{-6}$ (1/K) (coefficient of thermal expansion of 13 CrMo 44), $E = 178 \cdot 10^3$ (N/mm^2) (modulus of elasticity of 13 CrMo 44) and σ_{per} = 68.5 (N/mm^2) (permissible stress of 13 CrMo 44 at the maximum permissible material temperature) as:

$$T_{max} = 515°C.$$

This determination of T_{max}, to be carried out iteratively, shows the dependence of the permissible stress σ_{per} on the material temperature. Figure 4 represents graphically this dependence between the permissible stress σ_{per} on the maximum material temperature T_{max} for the material 13 CrMo 44.

<u>3rd Step: Heat flow density on the tube inside</u>

By means of the equations (4) and (5), there follows for A = 0.25 and B = 0.725 for the heat flow density q_i on the inside of the tubes 12:

$$q_i = 466 \text{ kW/m}^2.$$

<u>4th Step: Determining the temperature difference ΔT_w between the tube outer wall and tube inner wall</u>

According to equation (6), with the thermal conductivity of 13 CrMo 44 of $\lambda = 38.5 \cdot 10^{-3}$ kW/m K:

$$\Delta T_w = 73 \text{ K}.$$

<u>5th Step: Determining the necessary mass flow density</u>

According to equation (7), with $C = 7.3 \cdot 10^{-3}$ kWs/kgK:

$$\dot{m} = 939 \text{ kg/m}^2\text{s}.$$

The optimum mass flow density $\dot{m}$ can thus be determined by means of the available values for the heat flow density q_a on the tube outside and the maximum permissible material temperature T_{max}. This value is represented by broken lines in Figure 3 for the specified conditions. It can be seen that, for the assumed heat flow density q_a of the tube outside of 350 kW/m², optimum mass flow densities $\dot{m}$ of between 740 and 1060 kg/m²s are obtained in the case of tubes 12 having outside diameters d_a of between 30 and 40 mm and wall thicknesses d_r of between 6 and 7 mm.

For the flow-related design of the tubes 12 of the tube wall or containing wall 4, the mass flow density $\dot{m}$ thus determined can still be converted to the conditions prevailing under 100% load. For this purpose, the operating pressure at the inlet of the tubes 12 is calculated at 100%. The abovementioned mass flow densities $\dot{m}$ are subsequently converted in proportion to the operating pressure under 100% load. If, for example, the operating pressure under 100% load is p_B = 270 bar, the mass flow density $\dot{m}$ increases from 740 to 951 kg/m^2 or from 1060 to 1363 kg/m^{2}s.

It may be expedient to allow for uncertainties in the determination of the heat flow density q_a by raising the mass flow density $\dot{m}$ from +15% to +20% in relation to the calculated value.

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List of reference symbols

$\dot{m}$	mass flow density
σ	thermal stress
A, B	region
d_a	outside diameter
d_r	tube-wall thickness
E,F,G,H	curve
p_{crit}	pressure
q_a	heat flow density
q_i	heat flow density
RG	flue gas
S	flow medium
T_{max}	maximum permissible material temperature
2	once-through steam generator
4	containing wall
6	bottom
8	discharge orifice
10	burner
12	evaporator tube
14	tube web
16	evaporator heating surface
17	flame body
18,20,22	convection heating surface
24	flue-gas outlet duct
26	ribs
28	tube apex

Patent Claims

1. Once-through steam generator having a combustion chamber (4) surrounded by a containing wall consisting of tubes (12) connected to one another in a gas-tight manner, a flow medium (S) being capable of flowing from below upwards through the vertically extending tubes (12) which have a surface structure (26) on their inside, characterized in that the mass flow density $\dot{m}$ in the tubes (12) at that load at which critical pressure p_{crit} prevails in the tubes (12) conforms to the relation:

$$\dot{m} = \frac{q_i}{C\,(T_{max} - T_{crit} - \Delta T_w)}\ (kg/m^2s)\ ,$$

in which
q_i (kW/m²) is the heat flow density on the inside of the tube (12), T_{max} (°C) is the maximum permissible material temperature of the tube (12), T_{crit} (°C) is the temperature of the flow medium (S) at critical pressure (p_{crit}), $\Delta\ T_w$ (K) is the temperature difference between the outer wall and inner wall of the tube (12), and $C \geq 7.3 \cdot 10^{-3}$ kWs/kgK is a constant.

2. Once-through steam generator according to Claim 1, characterized in that the heat flow density q_i related to the inner wall conforms to the relation:

$$q_i = \frac{K \cdot d_a}{d_i} \cdot q_a \quad (kW/m^2)$$

with $K = A\ (d_a^2\ .\ q_a) + B$,
in which: $A = 0.45$ and $B = 0.625$ for $(d_a^2\ .\ q_a) \leq 0.5$ kW,
$A = 0.25$ and $B = 0.725$ for $(d_a^2\ .\ q_a) > 0.5$ and ≤ 1.1 kW,
$A = 0$ and $B = 1$ for $(d_a^2\ .\ q_a) > 1.1$ kW, and
~~q_a being the heat flow density on the tube outside (kW/m²) and d_a being the tube outside diameter (m)~~.

q_a being the heat flow density on the tube outside (kW/m^2) and d_a being the tube outside diameter (m).

3. The once-through steam generator as claimed iβn claim 1 or 2, wherein the maximum admissible material temperature T_{max} conforms to the relation:

(°C),

σadm being the admissible thermal stress (N/mm^2), β the coefficient of thermal expansion (1/K) and E the modulus of elasticity (N/mm^2) of the tube material.

4. The once-through steam generator as claimed in one of claims 1 to 3, wherein the temperature difference ΔT_W between the tube outer wall and the tube inner wall conforms to the relation:

(K)

with $K = A\ (d_a^2 . q_a) + B$,
in which A=0.45 and B=0.625 for $(d_a^2 .\ q_a) \leq 0.5$ kW,
A=0.25 and B=0.725 for $(d_a^2 . q_a) > 0.5$ and ≤ 1.1 kW,
A=0 and B=1 for $(d_a^2 . q_a) > 1.1$ kW, and
q_a is the heat flow density on the tube outside (kW/m^2), d_a the tube outside diameter (m), d_1 the tube inside diameter (m) and λ the thermal conductivity of the tube material (kW/mK).

5. The once-through steam generator as claimed in one of claims 1 to 4, wherein, for a tube (12) made from the material 13 CrMo 44, points in a coordinate system which are determined by pairs of values of the heat flow density m (kg/m^2) lie on a curve E defined for a tube outside diameter

d_a of 30 mm and a tube wall thickness d_r of 7 mm and passing through the points determined by the pairs of values:
q_a = 250 kW/m^2, m = 526 kg/m^2s,
q_a = 300 KW/m^2, m = 750 kg/m^2s,
q_a = 350 kW/m^2s, m = 1063 kg/m^2s, and
q_a = 400 kW/m^2, m = 1526 kg/m^2s.

6. The once-through steam generator as claimed in one of claims 1 to 4, wherein, for a tube (12) made from the material 13 Cr Mo 44, points in a coordinate system which are determined by pairs of values of the heat flow density m (kg/m^2) lie on a curve F defined for a tube outside diameter d_a of 40 mm and a tube wall thickness d_c of 7 mm and passing through the points determined by the pairs of values:
is this correct?
q_a = 250 kW/m^2, m = 471 kg/m^2s,
q_a = 300 KW/m^2, m = 670 kg/m^2s,
q_a = 350 kW/m^2s, m = 940 kg/m^2s, and
q_a = 400 kW/m^2, m = 1322 kg/m^2s.

7. The once-through steam generator as claimed in one of claims 1 to 4, wherein, for a tube (12) made from the material 13 Cr Mo 44, points in a coordinate system which are determined by pairs of values of the heat flow density m (kg/m^2) lie on a curve G defined for a tube outside diameter d_a of 30 mm and a tube wall thickness d_r of 6 mm and passing through the points determined by the pairs of values:
q_a = 250 kW/m^2, m = 420 kg/m^2s,
q_a = 300 KW/m^2, m = 576 kg/m^2s,
q_a = 350 kW/m^2s, m = 775 kg/m^2s, and
q_a = 400 kW/m^2, m = 1037 kg/m^2s.

8. The once-through steam generator as claimed in one of claims 1 to 4, wherein, for a tube (12) made from the material 13 Cr Mo 44, points in a

coordinate system which are determined by pairs of values of the heat flow density m (kg/m^2) lie on a curve H defined for a tube outside diameter d_a of 40 mm and a tube wall thickness d_r of 6 mm and passing through the points determined by the pairs of values:

q_a = 250 kW/m^2, m = 399 kg/m^2s,
q_a = 300 KW/m^2, m = 549 kg/m^2s,
q_a = 350 kW/m^2s, m = 737 kg/m^2s, and
q_a = 400 kW/m^2, m = 977 kg/m^2s.

9. A method for designing a once-through steam generator with a combustion chamber (4) surrounded by a containment wall composed of tubes (12) connected to one another in a gastight manner, a flow medium being capable of flowing from the bottom upward through the vertically extending tubes (12) which have a surface structure (26) on their inside, wherein the tubes (12) are selected in such a way that, under the load at which a critical pressure p_{crit} prevails in the tubes (12), a mass flow density m of:

2 which formula ?

flows through said tubes,
q_i (kW/m^2) being the heat flow density on the inside of the tube (12), T_{max} (°C) the maximum admissible material temperature of the tube (12), T_{crit} (°C) the temperature of the flow medium (S) at critical pressure (P_{crit}), ΔT_W(K) the temperature difference between the outer and inner wall of the tube (12), and C $\geq$ $7.3 \cdot 10^{-3}$ kWs/kgK a constant.

10. The method as claimed in claim 9, wherein the tubes are selected in such a way that the heat flow density q_i related to the inner wall conforms to the relation:

(kW/m^2)

with $K = A\ (d_a^2.q_a) + B$, in which: $A=0.45$ and $B=0.625$ for $(d_a^2.q_a) \leq 0.5$ kW, $A=0.25$ and $B=0.725$ for $(d_a^2.q_a) > 0.5$ and ≤ 1.1 kW, $A=0$ and $B=1$ for $(d_a^2.q_a) > 1.1$ kW, and q_a is the heat flow density on the tube outside (kW/m^2) and d_a is the tube outside diameter (m).

11. The method as claimed in claim 9 or 10, wherein the tubes (12) are selected in such a way that the maximum admissible material temperature T_{max} conforms to the relation:

(°C),

σ_{adm} being the admissible thermal stress (N/mm^2), β the coefficient of thermal expansion (1/K) and E the modulus of elasticity (N/mm^2) of the tube material.

12. The method as claimed in one of claims 9 to 11, wherein the tubes (12) are selected in such a way that the temperature difference ΔT_W between the tube outer wall and the tube inner wall conforms to the relation:

(K)

with $K = A\ (d_a^2.q_a) + B$, in which: $A=0.45$ and $B=0.625$ for $(d_a^2.q_a) \leq 0.5$ kW, $A=0.25$ and $B=0.725$ for $(d_a^2.q_a) > 0.5$ and ≤ 1.1 kW, $A=0$ and $B=1$ for $(d_a^2.q_a) > 1.1$ kW, and

q_a is the heat flow density on the tube outside (kW/m^3), d_a the tube outside diameter (m), d_i the tube inside diameter (m) and λ the thermal conductivity of the tube material (kW/mK).

13. The once-through steam generator as claimed in one of claims 9 to 12, wherein the tubes (12) are selected in such a way that, for a tube (12) made from the material 13 CrMo 44, points in a coordinate system which are determined by pairs of values of the heat flow density m (kg/m^2) lie on a curve E defined for a tube outside diameter d_a of 30 mm and a tube wall thickness d_r of 7 mm and passing through the points determined by the pairs of values:

q_a = 250 kW/m^2, m = 526 kg/m^2s,
q_a = 300 KW/m^2, m = 750 kg/m^2s,
q_a = 350 kW/m^2s, m = 1063 kg/m^2s, and
q_a = 400 kW/m^2, m = 1526 kg/m^2s.

14. The method as claimed in one of claims 9 to 12, wherein the tubes (12) are selected in such a way that, for a tube (12) made from the material 13 Cr Mo 44, points in a coordinate system which are determined by pairs of values of the heat flow density m (kg/m^2) lie on a curve F defined for a tube outside diameter d_a of 40 mm and a tube wall thickness d_r of 7 mm and passing through the points determined by the pairs of values:

q_a = 250 kW/m^2, m = 471 kg/m^2s,
q_a = 300 KW/m^2, m = 670 kg/m^2s,
q_a = 350 kW/m^2s, m = 940 kg/m^2s, and
q_a = 400 kW/m^2, m = 1322 kg/m^2s.

15. The method as claimed in one of claims 9 to 12, wherein the tubes (12) are selected in such a way

that, for a tube (12) made from the material 13 Cr Mo 44, points in a coordinate system which are determined by pairs of values of the heat flow density m (kg/m^2) lie on a curve G defined for a tube outside diameter d_a of 30 mm and a tube wall thickness d_r of 6 mm and passing through the points determined by the pairs of values:

q_a = 250 kW/m^2, m = 420 kg/m^2s,
q_a = 300 KW/m^2, m = 576 kg/m^2s,
q_a = 350 kW/m^2s, m = 775 kg/m^2s, and
q_a = 400 kW/m^2, m = 1037 kg/m^2s.

16. The method as claimed in one of claims 9 to 12, wherein the tubes (12) are selected in such a way that, for a tube (12) made from the material 13 Cr Mo 44, points in a coordinate system which are determined by pairs of values of the heat flow density m (kg/m^2) lie on a curve H defined for a tube outside diameter d_a of 40 mm and a tube wall thickness d_r of 6 mm and passing through the points determined by the pairs of values:

q_a = 250 kW/m^2, m = 399 kg/m^2s,
q_a = 300 KW/m^2, m = 549 kg/m^2s,
q_a = 350 kW/m^2s, m = 737 kg/m^2s, and
q_a = 400 kW/m^2, m = 977 kg/m^2s.

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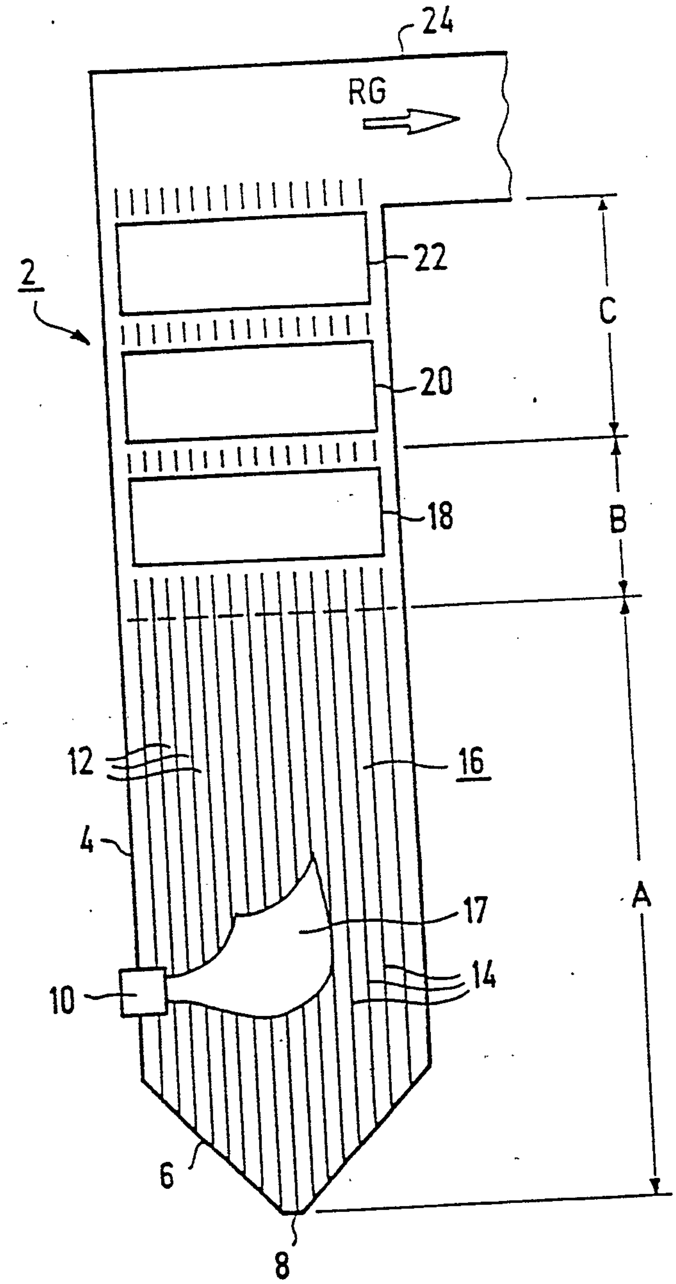

FIG 1

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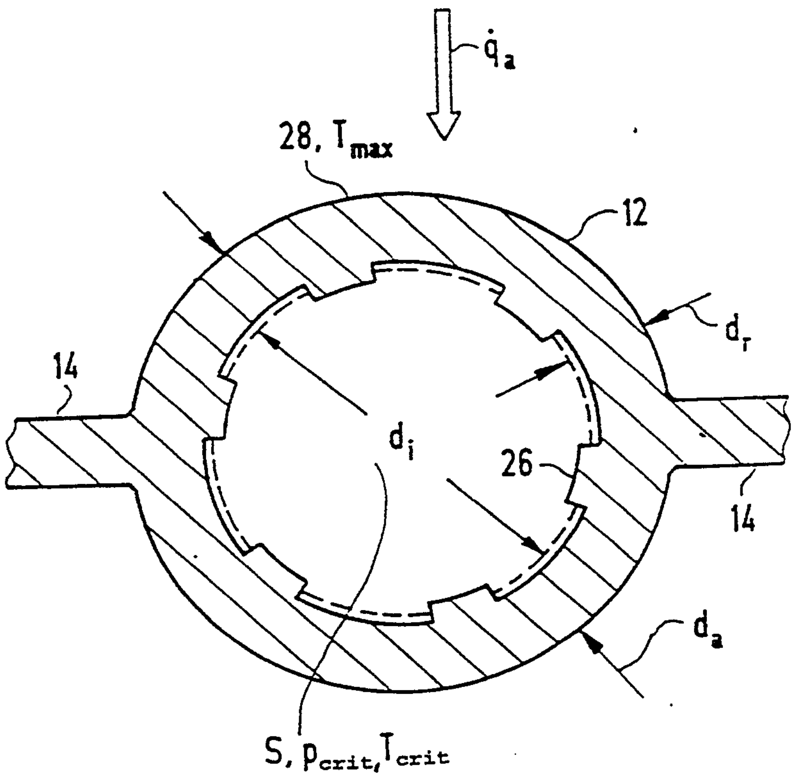

FIG 2

GR 96 P 3052 P

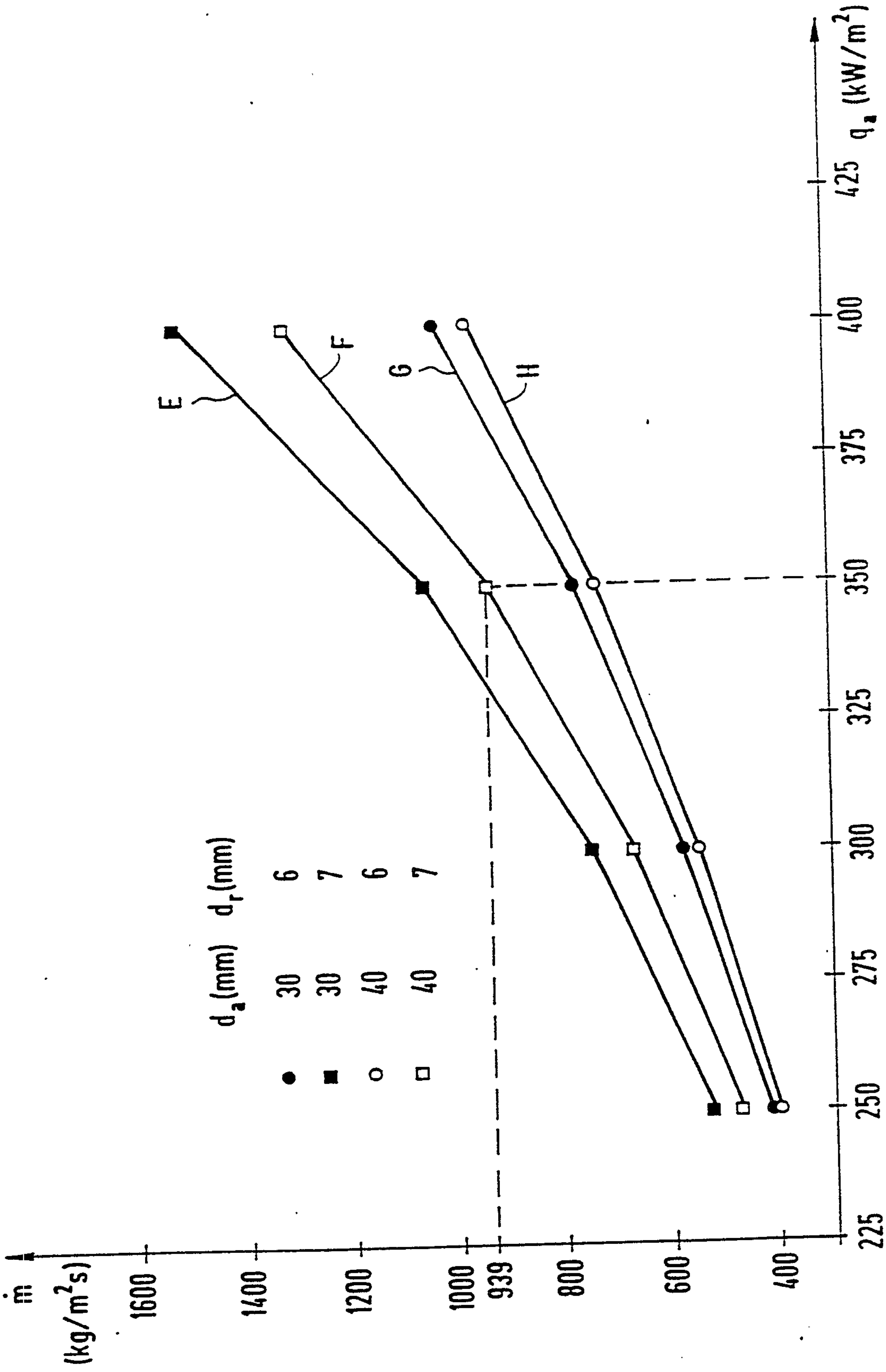

FIG 3

GR 96 P 3052 P

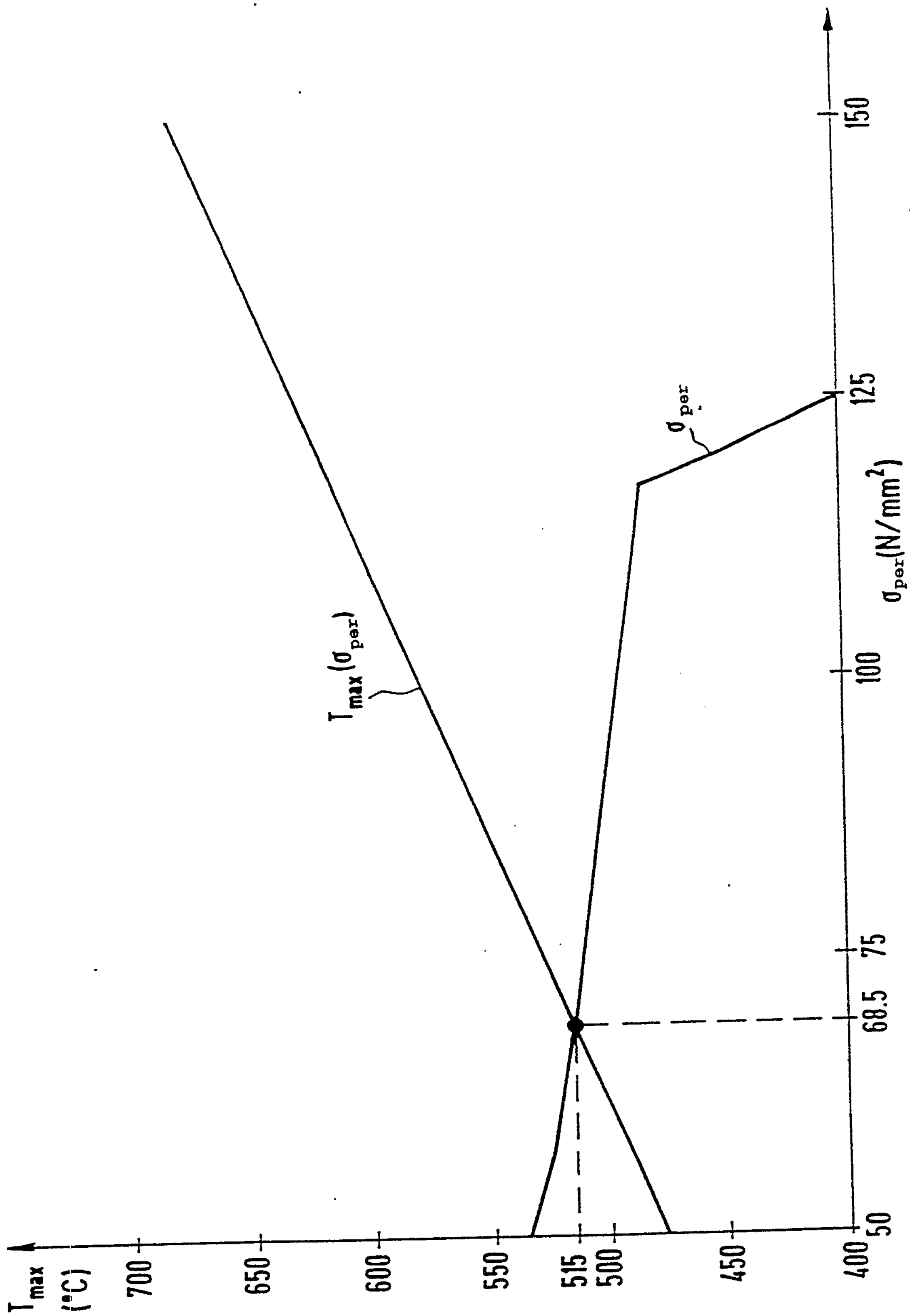

FIG 4

$$m = \frac{q_i}{C\,(T_{max} - T_{krit} - \Delta T_w)} \quad (kg/m^2 s) \qquad (7)$$