(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 2 837 777 A1

(12) EUROPEAN PATENT APPLICATION

(43) Date of publication:
18.02.2015 Bulletin 2015/08

(51) Int Cl.:
F01K 13/02 *(2006.01)* ***F01D 19/02*** *(2006.01)*
G05B 13/04 *(2006.01)*

(21) Application number: **13180644.0**

(22) Date of filing: **16.08.2013**

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA ME

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(54) Method for calculating thermal stress in a thick-walled component of a thermal system

(57) A method for calculating thermal stress in an obstruction-curved and/or thick-walled component through which a medium flows in a thermal system comprises the steps determining a heat transfer coefficient (α) from a measured steam pressure (p), a measured steam temperature (T_s), a steam flow and an inner diameter of the component (d); and determining the thermal stress as a temperature difference between a component surface temperature (T_i) and a mean integral temperature of the component (T_m) based on the measured steam temperature (T_s) and the heat transfer coefficient (α).

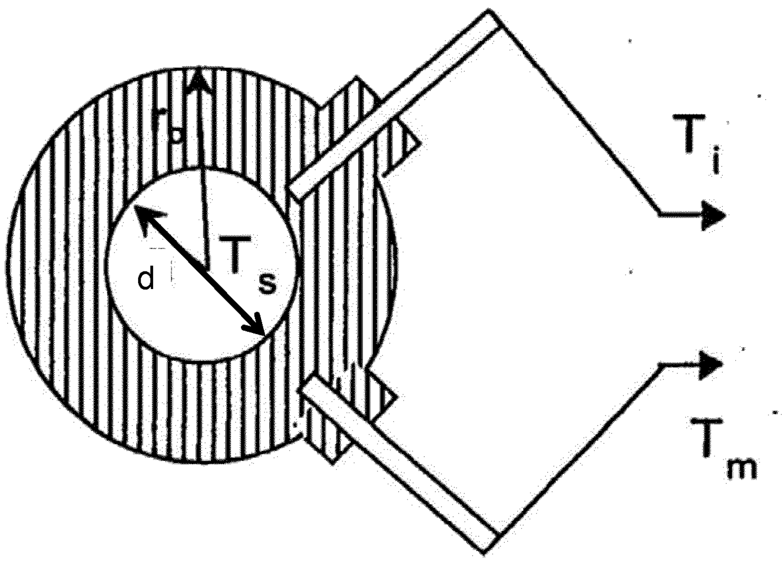

Fig. 1

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Description

[0001] The invention relates to a method for calculating thermal stress in an obstruction-curved and/or thick-walled component through which a medium flows in a thermal system.

[0002] Although the principles described below are applicable to various thermal systems, the example application area concerns the above mentioned components of a steam power plant.

[0003] For obstruction-curved and/or thick-walled components through which a medium flows, thermal stress occurs during operation with high temperature transients or gradients, i.e., during load changes. In a steam power plant, load changes occur particularly during start-ups and shut-downs.

[0004] There are at least two major application areas, where it is of interest to know the thermal stress in the component: life time monitoring and on-line feedback control.

[0005] In life time monitoring, an application is used on-line or off-line, e.g., integrated into a data historian, in order to calculate the lifetime reduction and remaining lifetime as a result of the actual plant operation taking place, typically by calculating the low-cycle fatigue (LCF) using an algorithm for detecting and evaluating stress peaks.

[0006] In on-line feedback control, the aim might be e.g. to achieve load changes and start-ups which take as little time as possible whilst not exceeding a preset limit of the thermal stress.

[0007] An example solution for taking thermal stress into account when controlling load changes in a thermal system is described in EP 1 462 901 A2. Therein, a heat stream density q is determined from a measured temperature difference (T_m - T_i), which is the difference between a mean wall temperature T_m of a thick-walled component and its inner surface temperature T_i, and from geometric and material parameters p of the component through which the medium, i.e. the steam, flows:

$$q = f(p)(T_m - T_i), \text{ with } f(p) \text{ being a function of parameters p.}$$

[0008] From the heat flux density q, a measured temperature of the flowing medium T_s and the surface temperature T_i, a heat transfer coefficient α is determined:

$$\alpha = q/(T_s - T_i)$$

[0009] The calculated heat transfer coefficient α is then used for different purposes.

[0010] The solution described here, on the other hand, is reversely directed towards calculating temperatures occurring in a component, optionally using a calculated heat transfer coefficient α.

[0011] A typical characteristic of the most common class of thick-walled components of a steam power plant is that steam flows through the component on the inside, while it is isolated on the outside, and that it has the basic shape of a cylinder or sphere. These are headers and other vessels collecting and splitting up streams of steam. Another class of thick-walled component comprises turbine rotors, where the outer surface is in contact with the steam and an inner surface is practically non-existent in case of a compact shaft or is the surface of a hollow bore within the turbine. The thick-walled components of a steam power plant are in all relevant cases made of metal, so that the temperatures of the different areas of their walls may also be referred to as metal surface and inner metal temperatures, respectively.

[0012] The following calculation principles apply to both of the above named component classes. The principles are not limited to steam-heated components, but can be expanded to components heated e.g. by flue gas as well.

[0013] At least for components, which can be approximated by the basic shapes "cylinder", "sphere", and "plate", the thermal stress is typically directly characterized by the difference ΔT between the surface temperature and an inner wall temperature of the component, i.e. the difference between the component surface temperature T_i and the mean integral temperature T_m. The latter is the integral of temperature times mass divided by total mass of the component.

[0014] The thermal stress σ is given from the temperature difference ΔT by the well-known formula

$$\sigma = \beta \frac{E}{1-\nu} \Delta T$$

Where β is the factor of differential thermal expansion, E is Young's modulus of elasticity, and ν is Poisson's ratio. Thus, the thermal stress equals the temperature difference multiplied by a more or less constant factor.

[0015] Ideally, the steam temperature, surface temperature, and mean temperature are measured using an arrangement as indicated in Fig. 1. However, one is often confronted with one of the following situations:

- No metal temperature measurements are available.
- Only a measurement of the surface temperature T_i is available.
- T_m is available but deemed unreliable.

[0016] In any of such cases, the information on the metal temperatures needs to be augmented using calculation. Often, these calculations need to be implemented in a distributed control system (DCS), which is known as "soft sensor".

[0017] In order to calculate the temperature difference ($\Delta T = T_m - T_i$), the temperature distribution within the component needs to be known.

[0018] A standard method for calculating the temperature difference is a so called multi-layer algorithm, which is based on dividing the component into a multitude of slices or layers, i.e. finite elements, as indicated by Fig. 2. Since typically one of the basic shapes "cylinder", "sphere", and "plate" is assumed, a one-dimensional division suffices. A typical number of segments is five or more, leading to five or more dynamic states.

[0019] Each of these layers is assigned a temperature state, and between these the temperature flux can be calculated by a one-dimensional Fourier equation. In Fig. 2, the double arrows indicate the heat flux between segments according to the one-dimensional Fourier equation, and the single arrow indicates the heat flux from the medium through the inner surface of the component. The mass m of each segment depends on the inner and outer radius of the respective segment. Fig. 2 also shows the equation for calculating the mean integral temperature T_m.

[0020] Especially for on-line feedback control, but also for lifetime monitoring, it is of interest to simplify the calculation algorithm, as it is time and resource consuming and therefore not suitable for on-line execution. In addition, the algorithm described above may be complicated to be implemented in a data processing unit of an already existing distributed control system (DCS) due to the extensive engineering work involved. If the DCS has limited accuracy, i.e., does not use floating-point arithmetic, it may not be able to execute the algorithm at all.

[0021] Although the DCS problem is mainly related to on-line feedback control, reduced complexity is important for lifetime monitoring and optimization applications as well. It turns out that the multi-layer algorithm gives rise to a stiff ordinary differential equation (ODE) which cannot easily be solved e.g. by a Euler algorithm with longer sampling intervals. This makes it intractable in simple mathematical frameworks.

[0022] Also, the high number of dynamic states may be an obstacle.

[0023] It is therefore an object of the present invention to provide a simplified method for calculating thermal stress in an obstruction-curved and/or thick-walled component through which a medium flows in a thermal system which may be executed on-line during operation of the thermal system.

[0024] The object is achieved by a method as described in independent claim 1.

[0025] The invention and its embodiments will become apparent from the example and its embodiments described below in connection with the appended drawing which illustrates:

Fig. 1 a section through a thick-walled component indicating where the component temperatures and the steam temperature would be measured,

Fig. 2 an illustration of the multi-layered temperature modeling approach,

Fig. 3 an illustration of a PT1 temperature modeling approach,

Fig. 4 a decreasing of the surface temperature during a day,

Fig. 5 the temperature difference obtained with the multi-layered approach and with the PT1 approach for the ramp input of Fig. 4,

Fig. 6 the surface temperature of a real-life data scenario,

Fig. 7 the temperature difference obtained with the multi-layered approach and with the PT1 approach for the real-life input of Fig. 6.

[0026] Fig. 1 shows a section through a thick-walled component indicating where the component surface temperature T_i, the mean integral temperature T_m and the steam temperature T_s would be measured.

[0027] In Fig. 4, the time dependent behavior of the component surface temperature T_i during a day is shown, with a linear decrease in temperature from 500 °C to 400 °C.

[0028] Fig. 5 illustrates, in a solid line, the temperature difference obtained with the multi-layered approach for the ramp input of Fig. 4. The multi-layered approach is based on 10 layers. As can be seen, the temperature difference "DT multilayer" finally reaches about 1 K. The temperature difference decreases with time due to the material properties being functions of temperature.

[0029] A first step to simplifying the calculations of the multi-layered approach is by simplifying the multi-layer model, since the multiple-layer method is in fact too excessive with regard to accuracy. Upon examination of the step response of the mean integral temperature T_m, it is apparent that it is very similar to a first-order system response. Accordingly, it turns out that the mean integral temperature T_m can be approximated by a first-order system:

$$\dot{T}_m = K(T_i - T_m)$$

[0030] This simple type of dynamic system is called "PT1", in particular in German control literature. This term will be used in the sequel in this text as well to denote a first-order system of such structure.

[0031] Fig. 3 demonstrates the structure of the PT1 model as opposed to the multi-layered model in Fig. 2. Only one layer, i.e., one dynamic state is used.
A fundamental property of the temperature difference ($\Delta T = T_m - T_i$) is that, at so-called quasi-stationary conditions, i.e., when the inner temperature T_i describes a ramp with constant gradient, a constant ΔT value is obtained.

[0032] The value of the inverse time constant, K, in the PT1 equation above will now be selected in such a manner that exactly the correct response is obtained at quasi-stationary conditions. The PT1 approximation then has the following characteristics:

- by definition, perfect response of the temperature difference ($\Delta T = T_m - T_i$) at quasi-stationary conditions, i.e. when the surface temperature is described by a ramp and the ΔT has settled;
- perfect maximum value of the temperature difference for a step change;
- perfect response at infinite frequency.

[0033] An exemplary result is shown in Figs. 4 and 5, where a decreasing ramp input leads to a near-constant positive ΔT. The slight decrease in its value with time is due to the metal properties changing as a function of temperature. The PT1 model model, illustrated by the dotted line, has correct response by definition. Fig. 5 also verifies the behavior at quasi-stationary conditions for the multi-layered model, illustrated by the solid line.

[0034] Since the performance of the PT1 approximation is so good in the extreme cases listed above, it is to be expected that it would provide a very close approximation for practical data as well, which indeed is the case as practical experience shows and as is illustrated with Figs. 6 and 7.

[0035] In Fig. 6, the surface temperature of a real-life data scenario, namely a boiler start-up, is depicted. Fig. 7 shows with a solid line the temperature difference "DT multilayer" obtained with the multi-layered approach, and with a dotted line the temperature difference "DT PT1" obtained with the PT1 approach, both for the real-life input of Fig. 5.

[0036] As can be seen, the PT1 approach causes slight conservatism, i.e., over-estimation of the temperature difference, but delivers sufficiently accurate results at considerably reduced processing effort.

[0037] A simple and useful way to determine the factor K, i.e., the inverse of the time constant, in the PT1 approximation is to use a component design datasheet by the component manufacturer. Apart from information on geometry and material, such data sheets often depict the allowable ΔT limits and the corresponding limits of the temperature gradient under quasi-stationary conditions. The negative ratio of the temperature gradient and the corresponding ΔT directly yields the factor K above. This can be shown e.g. by going over to the Laplace domain and noting that the steady-state value of $T_m - T_i$ equals $-r / K$ if $T_i = 1/s \cdot r$ and $s \cdot T_m = K(T_i - T_m)$, where r is the rise speed and s is the Laplace constant.

[0038] A more rigorous way to obtain K is to use component geometry and material data. In the following, the PT1 approximation for a cylinder with heating occurring through the inner wall is given.

[0039] As stated in EP 1 462 901 A2, the heat flux density in this case is given by

$$q = \frac{\lambda(T_i - T_m)}{r_i\left[\left(\frac{r_o^2}{r_o^2 - r_i^2}\right)^2 ln\frac{r_o}{r_i} - \frac{3r_o^2 - r_i^2}{4(r_o^2 - r_i^2)}\right]}$$

[0040] Strictly speaking, this formula applies only to quasi-stationary conditions. However, in accordance with the above, it is used to determine an inverse time constant which will determine the general dynamics of the PT1 approximation.

[0041] By introduction of the auxiliary variable R

$$R := \frac{\lambda}{r_i\left[\left(\frac{r_o^2}{r_o^2 - r_i^2}\right)^2 ln\frac{r_o}{r_i} - \frac{3r_o^2 - r_i^2}{4\left(r_o^2 - r_i^2\right)}\right]}$$

it is obtained:

$$q = R(T_i - T_m)$$

[0042] The heat flux through the inner wall is given by

$$\dot{Q} = qA$$

where the inner surface A is given by

$$A = 2\pi r_i l$$

where l is a (fictitious) component length, which will be eliminated in the sequel.

[0043] The heat flux can be assumed to directly affect the component mean temperature, yielding

$$\dot{T}_m = \frac{\dot{Q}}{c_p \rho V}$$

[0044] Taking into account that the component volume is given by

$$V = \pi\left(r_o^2 - r_i^2\right)l$$

a dynamic equation for the mean temperature is obtained as a function of the inner surface temperature:

$$\dot{T}_m = \frac{2r_i R}{c_p \rho\left(r_o^2 - r_i^2\right)}(T_i - T_m)$$

[0045] The inverse time constant

$$K := \frac{2r_i R}{c_p \rho\left(r_o^2 - r_i^2\right)}$$

contains material properties which may or may not be treated as functions of temperature, according to the degree of accuracy required. Note that - optionally - making K a function of T_m obviously introduces a slight nonlinearity, as can be seen from Fig. 5, where the near-constant temperature difference slowly decreases with time.

[0046] The above formula is useful in cases where the surface temperature T_i is known, or where α can be assumed to be close enough to infinity. Often, however, the actual case in the plant is that no metal temperature measurement is available. Instead, only a steam temperature T_s is available, and T_i and T_m need to be calculated from this using the heat transfer α.

[0047] The influence of α is taken into account by considering that the heat flux density through the inner surface is given by

$$q = \alpha(T_s - T_i)$$

[0048] Substituting this into the equation for q above yields

$$\dot{T}_m = \frac{2r_i R}{c_p\rho(r_o^2 - r_i^2)}(T_s - T_m)/\left(1 + \frac{R}{\alpha}\right)$$

[0049] Subsequently, T_i is obtained as a function of T_s and T_m using above formulas:

$$\dot{T}_i = T_m + 1/\left(1 + \frac{R}{\alpha}\right)(T_s - T_m)$$

[0050] Similar formulas can be obtained e.g. for a sphere or a component heated from the outside. The starting point in each of these cases is the theoretical solution of the heat flux density q, which can be obtained from literature.

[0051] Noticing that a temperature difference is proportional to a certain temperature gradient at quasi-stationary conditions, one might be tempted to directly calculate the temperature difference from a calculated derivative according to

$$\Delta T = -\frac{1}{K}\frac{dT_i}{dt} \sim -\frac{1}{K}\frac{T_i(k) - T_i(k-1)}{t(k) - t(k-1)}$$

or similar. However, such a method will quickly turn out to lead to completely wrong results. This can be demonstrated by taking a step-like temperature change, which will lead to a near-infinite calculated derivative, depending on the sampling interval, and thus a near-infinite ΔT. On the other hand, calculating the derivative using a first-order function, retrieves the PT1 structure, with the correct time constant being determined as discussed above.

[0052] When having only a steam temperature measurement T_s at one's disposal, a convenient way to simplify calculation is to assume α to be infinite, i.e., to set T_i equal to T_s in the PT1 approximation. However, this may yield unnecessarily conservative results, i.e., higher temperature differences than necessary. This is motivated by the following.

[0053] The surface temperature T_i lies between the mean integral temperature T_m and the steam temperature T_s. How close it is to the one or the other depends on the heat transfer coefficient α, i.e., on the heat transfer between steam and metal. For an infinite α value, the surface temperature T_i matches the steam temperature T_s. For lower values of α, the surface temperature T_i is closer to the mean integral temperature T_m, yielding lower temperature differences ($\Delta T = T_m - T_i$). Since α is a function of the steam flow and steam parameters, low α values occur especially during start-ups, where the temperature differences have the most impact anyway.

[0054] Due to these considerations, a calculated value of α may be required. Further, the calculation should be simple enough to be implementable in a data processing unit of a DCS, etc.

[0055] A standard formula for the α value is given for superheated steam under turbulent conditions in Karl Strauß, "Kraftwerkstechnik zur Nutzung fossiler, nuklearer und regenerativer Energiequellen", 5. Auflage. Springer 2006, pp. 200-218:

$$\alpha\,[W / m^2K] = Nu\ \lambda / d = 0.02\ \Phi^{0.8}\ d^{-0.2}\ \lambda^{0.58}\ c_p^{0.42}\eta^{0.38}$$

where

Nu is the Nusseldt number = 0.02 $Re^{0.8}$ $Pr^{0.A2}$ [-],
λ is the heat conductance of the steam [W/mK],
d is the inner diameter of the component [m],
Re is the Reynolds number = Φ d /η[-],
Pr ist die Prandtl number [-],
Φ is the mass flow density [kg/m^{2}s],
η is the dynamic viscosity of the steam [Pa s],
c_p is the specific heat capacity of the steam [kJ/(kg K)].

[0056] This formula is hard to implement, especially in a simple DCS, due to the steam parameters heat conductance λ, heat capacity c_p, and dynamic viscosity η, for which steam tables are required.

[0057] Thus, it is of interest to simplify this equation, or in particular, the term

$$k_{steam} = \lambda^{0.58}\, c_p^{0.42} \eta^{0.38}$$

[0058] The equation for k_{steam} is highly nonlinear and hard to approximate. However, the inventor managed to find the following simple approximation, which appears to be quite good in the ranges for which the equation is valid:

$$k_{steam,approx} =(- (p\,[bar] - 1) / (T_s\,[°C] - 80)^2 + 0.00365) \cdot (1+0.07 / 100 \cdot (500 - T_s\,[°C]))$$

where p [bar] and T_s [°C] are the steam pressure and steam temperature.

[0059] In addition to the above factor k_{steam}, the steam flow [kg/s] and the inner diameter d of the component are needed to calculate the mass flow density Φ and the rest of the equation for α.

[0060] To summarize, what is presented above is a simplified and complete calculation of the temperature difference ($\Delta T=T_m - T_{i\_}i$) and thus the thermal stress in thick-walled components, which can be achieved by combining two different approaches: the simplification of the component temperature distribution calculation, using the PT1 approximation, and the simplification of the α calculation, i.e. of the heat transfer coefficient from medium to metal.

[0061] The α calculation is also of interest in other applications than temperature calculation for thick-walled components, such as for boiler efficiency calculations and in soot-blowing applications, etc.

Claims

1. Method for calculating in a data processing unit thermal stress in an obstruction-curved and/or thick-walled component through which a medium flows in a thermal system, **characterized by** the steps

    • determining a heat transfer coefficient (α) from a measured steam pressure (p), a measured steam temperature (T_s), a steam flow and an inner diameter of the component (d), and
    • determining the thermal stress as a temperature difference between a component surface temperature (T_i) and a mean integral temperature of the component (T_m) based on the measured steam temperature (T_s) and the heat transfer coefficient (α)

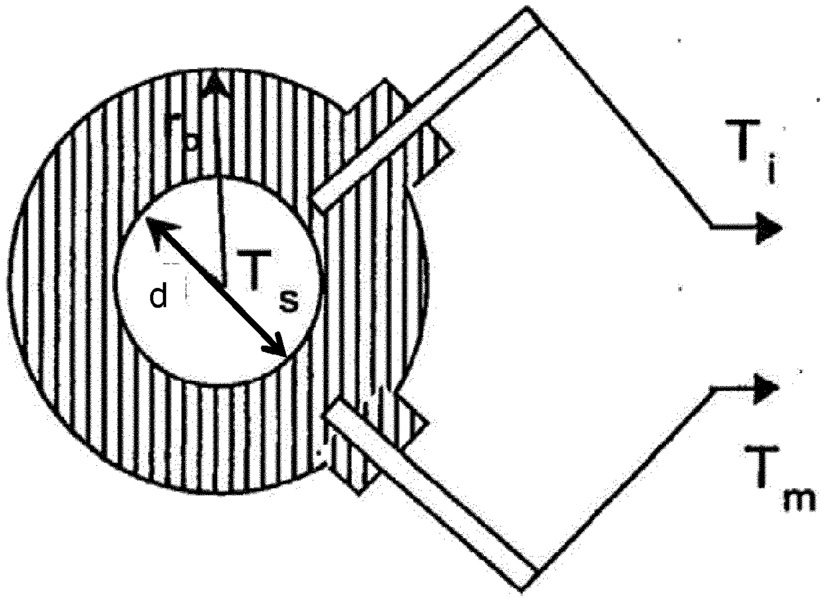

Fig. 1

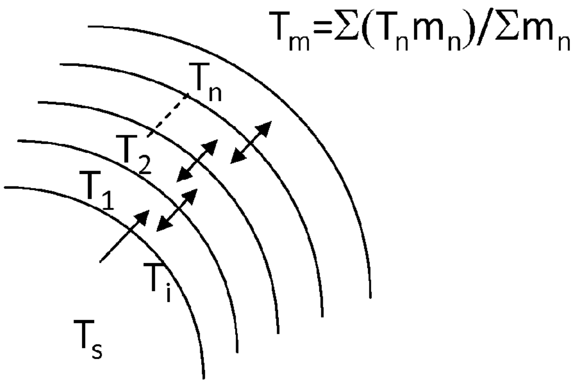

Fig. 2

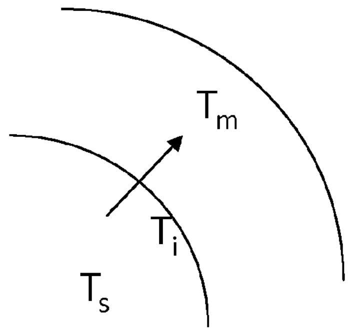

Fig. 3

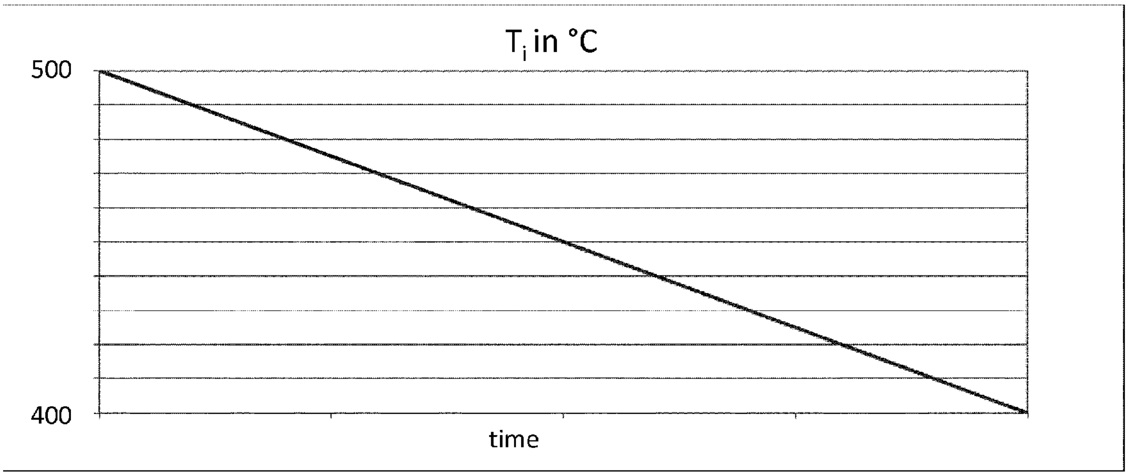

Fig. 4

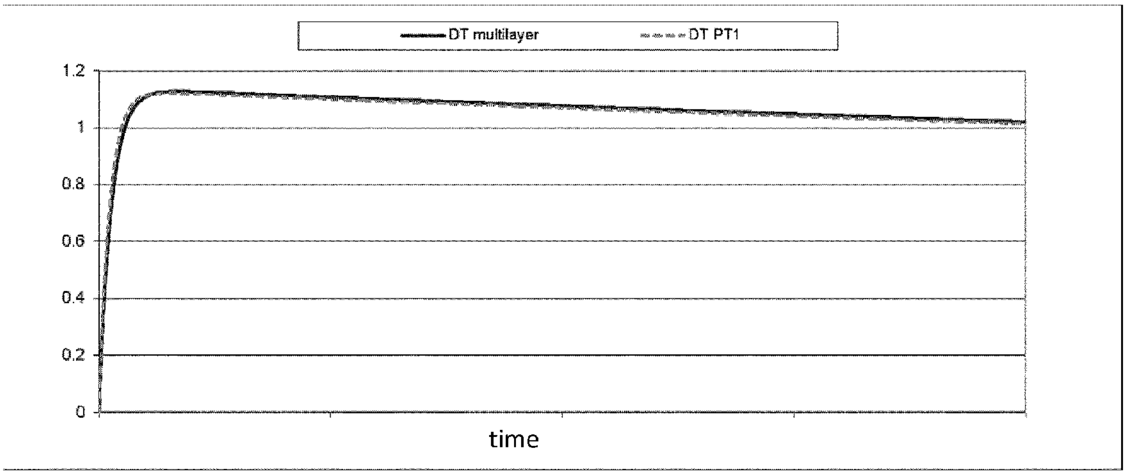

Fig. 5

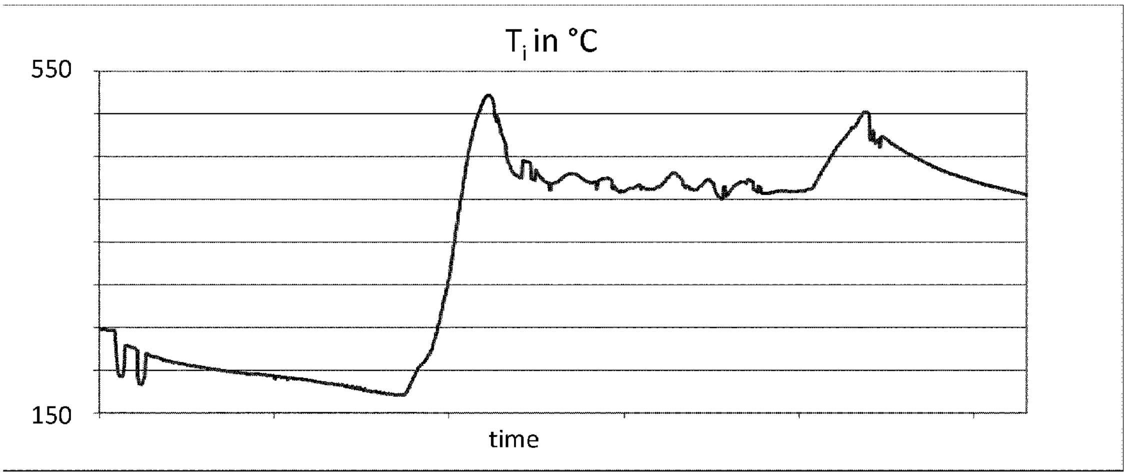

Fig. 6

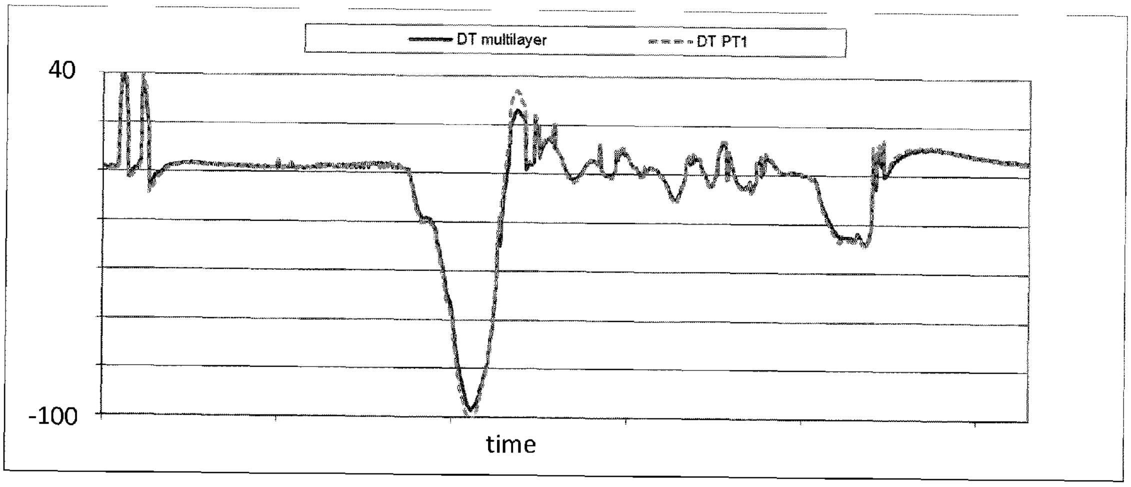

Fig. 7

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EUROPEAN SEARCH REPORT

Application Number
EP 13 18 0644

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
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TECHNICAL FIELDS SEARCHED (IPC)

F22B
G05D
G01N
F01K
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G05B

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
Munich	27 January 2014	De Syllas, Dimitri

1

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