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HEAT TRANSFER COEFFICIENT CHARACTERIZATION AT THE SOLAR COLLECTOR WALL-FLUID INTERFACE

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ABSTRACT

In this paper, a numerical study is carried out to characterize the transient local heat transfer coefficient at the fluid-solid wall interface of a solar collector. For that purpose, the considered collector wall geometry is a flat plate with non-negligible thickness which is subjected to a variable solar heat flux. The transient conjugated conduction-convection heat transfer has been taken into account. The heat transfer coefficient is calculated as a function of the plate thickness as well as the position along the plate. A good agreement has been found between the calculated temperatures and other experimental results. The heat transfer coefficient evolutions, as a function of time, have been obtained for various positions along the plate. The results showed that at first, high values of the heat coefficient are reached, and then it decreases and tends to constant values. It has been also noticed that at a fixed value of time, the heat transfer decreases when the position is increased from the beginning of the plate towards its end. The parametric study allowed obtaining a correlation of the transient convective heat transfer coefficient as a function of the steady state coefficient (which depends on the flow velocity and the coordinate of the considered point on the plate), multiplied by a function of time and the plate properties. The results have been used to optimize the heat transfer coefficient measurement technique using the pulse method. The pulse method consists in imposing a heat flux on a wall, and then to calculate, by an inverse method, the heat transfer coefficient from the time evolution of surface temperature (thermo-gram). Measurement of the heat transfer coefficient is based on the introduction into the inverse model of a function that represents the theoretical evolution of this coefficient due to the energy excitation. This function is deduced from the numerical study conducted in this work.

INTRODUCTION

The evaluation of the heat transfer coefficients between a solid and a fluid is necessary for the control and the design of many thermal systems [1] such as for solar collectors. Indeed, such thermal systems are always exposed to variations in time of some thermal parameters (solar heat flux, fluid or solid temperature). Therefore, the heat transfer coefficient at the solid-fluid interface may not be considered constant. At the interface, the conjugate heat transfer problem is considered. Such problem occurs when a regime fluid flow is coupled to the conducting solid wall with a finite thickness [2]. For that case, the effects of heat conduction in the solid wall cannot be neglected. If the conjugate phenomena are neglected, the temperature or the heat flux is assigned as boundary conditions. However, in practice, both temperature and heat flux at the solid–fluid interface are unknown and have to be determined by simultaneous and coupled solutions of the thermo-fluid dynamic equations in the fluid and the energy equation in the solid [3].

NOMENCLATURE

a	[m^2/s]	Thermal diffusivity
C	[J/kgK]	Specific heat
E	[m]	Plate thickness
k	[W/mK]	Thermal conductivity
h	[W/m^{2}K]	Convective heat transfer coefficient
L	[m]	Plate length
P	[N/m^2]	Pressure
T	[K]	Temperature
t	[s]	Time
Pr	[-]	Prandtl number
x, y	[m]	Cartesian coordinates
u, v	[m/s]	Velocity component
Special characters		
μ	[kg/ms]	Dynamic viscosity
ρ	[kg/m^3]	Fluid density
φ	[W/m^2]	Heat flux density
Subscripts		
ext		External
c		Duration of the heat flux pulse
f		Fluid
s		Solid interface (y = 0)
∞		Free stream (at infinity)
0		Initial and final steady-state regimes

For instance, it is well known that, if in the calculation of convective heat transfer coefficients from experimental data the effects of such conjugate heat transfer phenomena are neglected, the corresponding Nusselt numbers are, generally, underestimated [4-6]). The difficulty in the resolution of the coupled problem implies that conjugated problems are usually studied by numerical or approximate methods.

Numerical methods are used especially to solve the complex problems associated with a particular geometry. Recently, numerical techniques have been employed in several studies [7, -10]. The influence of coupled conduction-convection flows in channels has also been studied in [11,12], with periodic variations in the thermal boundary conditions (temperature or heat flux) or flow velocity. It was shown in these studies that characteristic times of heat diffusion in the solid and the fluid are parameters that dominate coupled heat transfers. There have been several researches devoted to the problems of transient heat transfer coefficient calculations in the flat plat for various boundary conditions. Rebay & al. [13] presented a non-destructive procedure for the measurement of the local convective heat transfer coefficient between a plat uniformly heated and an air flow. By an extension of the differential method, Rebay & al. [14] studied the unsteady forced convection heat transfer of a flow over a flat plate with a sudden change of the heat flux density at the surface of the plate without constant pressure along the direction of flow. Forced convection from a microstructure on a flat plate was investigated by Rebay & al. [15]. The axial conduction, usually neglected in boundary layer theory, was considered in [15]. The differential method has been used to reduce the governing partial differential equations to ordinary differential ones, which are solved numerically by the use of a computational code developed by the authors. Mladin & al [16] have modelled the unsteady thermal boundary layer developing along a finite thickness plate under a ramp type variation of temperature on the bottom plate surface. To model the transient heat transfer, two mathematical approaches were used: the integral method based on Karman-Polhausen methodology and the full Navier-Stokes system of equations, numerically solved with the commercial solver FLUENT. Both methods were validated for steady state regime and zero plate thickness, by comparison with solutions commonly reported in the literature. The numerical results revealed that the two methods agree within 5% for the steady state and 2.6% for transient conditions. In the present work, purely numerical method was used to investigate the unsteady laminar forced convection over a flat plate with various thicknesses, subjected to a time-variation of the surface heat flux. Indeed, in practical applications, the variation of the surface heat flux is more observed than the surface temperature (i.e. sun radiations, electric heating or laser radiation…). Moreover, in this study, the case of a sudden positive change of the heat flux from an initial isothermal state has been considered. The main goal is the determination of the transient convective heat transfer coefficient at the fluid-solid interface. The results will help to optimize the technique for measuring the heat transfer coefficient by the impulse method. In this method, a fixed duration heat flux is imposed on a wall, and then the heat transfer coefficient is calculated from the time evolution of surface temperature (thermo-gram) by an inverse method. In the inverse model, a function that represents the theoretical evolution of the heat transfer coefficient due to the energy excitation is introduced. Such function is deduced from the numerical study conducted in this work.

NUMERICAL MODELING

A forced external air flow over a flat plate with (E) thickness and (L) length (Figure 1), initially at thermal equilibrium is considered. Both the fluid and the wall are considered to be initially at the same temperature. At t = 0, the front surface of the plate (y = -E) is suddenly subjected to a time dependent heat flux density φ (t) which is suppressed after a given period of time. For the parallel free flow, velocity and temperature values are respectively U_∞ and T_∞ and thermal properties are assumed to remain constant, such as the fluid hydro-dynamical boundary layer which is not dependent on the temperature field.

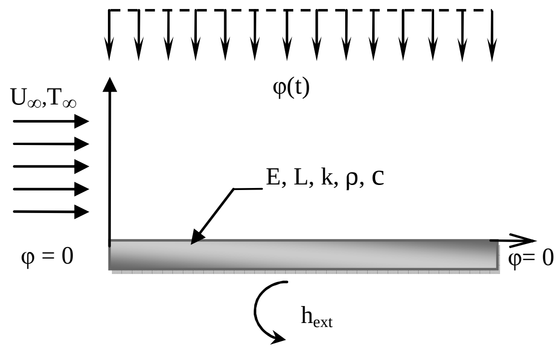


Figure 1 Flat plate with non-negligible thickness and variable imposed heat flux

Under a steady-state flow but transient heat transfer conditions, the governing equations are given by:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{\partial p}{\partial x} + \nu\frac{\partial^2 u}{\partial y^2} \quad (1)$$

The energy equation in the fluid:

$$\frac{\partial T_f}{\partial t} + u\frac{\partial T_f}{\partial x} + v\frac{\partial T_f}{\partial y} = a_f\left(\frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2}\right) \quad (3)$$

The energy equation in the solid:

$$\frac{\partial T_s}{\partial t} = a_s \cdot \left(\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2}\right) \quad (4)$$

The initial and boundary conditions corresponding to the application of the heat flux on the front side of the plate are given by:

At y = 0:

$$-k_s\left(\frac{\partial\, T_s(x,0,t)}{\partial\, y}\right)=\begin{cases} q_s(t)-k_f\left(\dfrac{\partial T_f(x,0,t)}{\partial y}\right) & 0\le t\le t_c \\ -k_f\left(\dfrac{\partial T_f(x,0,t)}{\partial y}\right) & t\ge t_c \end{cases} \quad (5)$$

At y = -E:

$$-k_s\left(\frac{\partial\, T_s(x,\text{-}E,t)}{\partial\, y}\right)=h_{ext}(T_s-T_\infty) \quad (6)$$

At x = 0 and x = L:

$$\left(\frac{\partial\, T_s(0,y,t)}{\partial\, y}\right)=\left(\frac{\partial\, T_s(L,y,t)}{\partial\, y}\right)=0 \quad (7)$$

Conduction-convection coupling is obtained by the equality of both temperatures and heat flux at the fluid-solid interface.
At y = 0:

$$\begin{cases} T_f(x,0,t)=T_s(x,0,t) \\ -k_f\left(\dfrac{\partial T_f(x,0,t)}{\partial y}\right)=-k_s\left(\dfrac{\partial T_s(x,0,t)}{\partial y}\right) \end{cases} \quad (8)$$

NUMERICAL PROCEDURE

The considered governing equations are numerically solved with the commercial solver FLUENT, which is based on the finite volume method. The computational grid has been generated using Gambit Software. The dimensions of the computational fluid domain are 30 cm length and 15 cm high. Various thicknesses have been considered. An unsteady second-order implicit formulation was used with a pressure based solver. For spatial calculation, SIMPLE algorithm for the pressure-velocity coupling and second order upwind for the momentum and energy discretization schema are chosen. A User Defined Function (UDF) was developed in order to introduce the time variation of the heat flux applied to the front face of the plate. A mesh-independence study was conducted to verify that the obtained solution is not influenced by the grid size. For that purpose, four different size grids have been used (Table 1)

Table 1 Different size grids studied

	Grid 1	Grid 2	Grid 3	Grid 4
Solid area	500	1000	1500	2000
fluid area	5000	20000	45000	80000

The temperature variation along the plate interface for the four studied cases is shown in Figure 2. It is clear that, Grids 1 and 4 are not suitable. However, grids 2 and 3 are acceptable, because the difference between the corresponding curves is very small. The same results have been obtained by the two meshes (i.e. grid 2 and grid 3); the smallest one was chosen, namely the Grid 2, which is composed by 20000 elements in the fluid area and 2000 elements in the solid one (Figure 3). All subsequent simulations were performed with the chosen mesh.

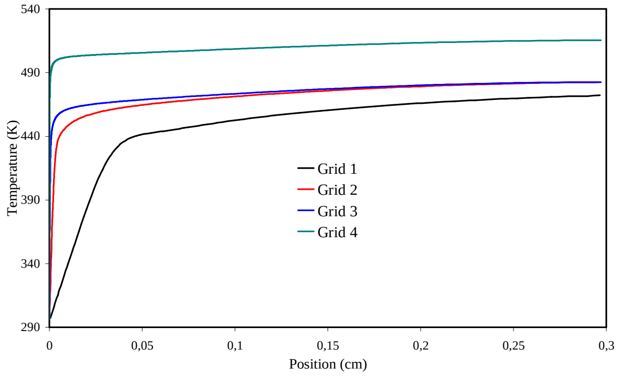


Figure 2 Temperature variations with position along the plate, for the studied grids

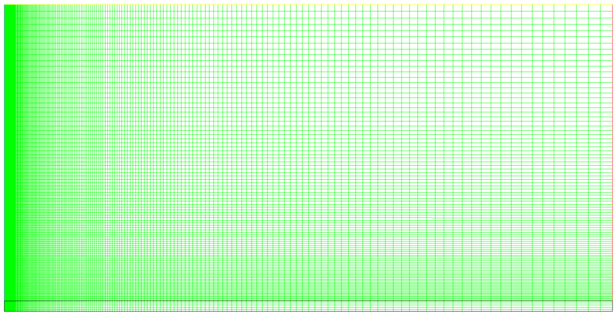


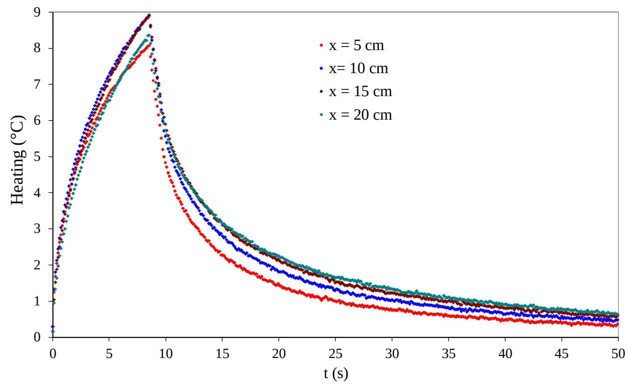
Figure 3 Geometry and mesh by Gambit of the chosen grid

Validation of the Numerical Procedure

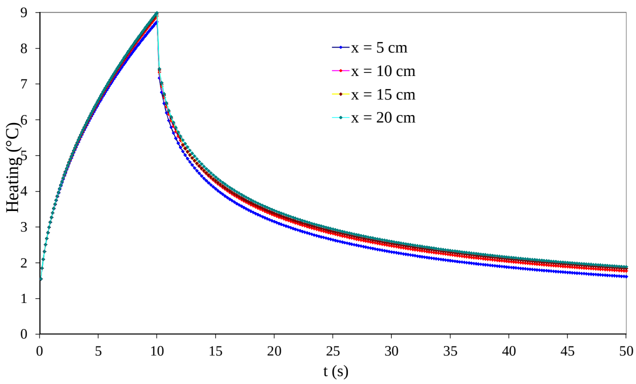
The numerical procedure have been validated by comparing the obtained results for the temperature evolution along the plat interface at different positions, with available experimental data [1]. A good agreement has been obtained. The result is shown in Figure 4.

RESULTS AND DISCUSSIONS

The flat plate is exposed to impulse change in the heat flux applied on its front face. The heat flux is given on the front face, so the temperature profiles start evolving at the same time in both solid plate and fluid. At first, the whole heat flux is penetrating by conduction into the plate; therefore the plate becomes hotter at the interface with the fluid, than in the neitherboring fluid. Due to the temperature gradient, convective heat transfer is taken place between the solid plate and the fluid, which is influencing in its turn the heat flux penetrating in the plate. That why, the interface temperature starts evolution earlier when exposed to heat flux on its front face. Surface temperature reaches a maximum value, and then it decreases. As observed in Figure 5, the maximum value is reached at t = t_c = 10 seconds, at the maximum time of the heat flux impulse.



(a)



(b)

Figure 4 Temperature evolution along the plat interface at different positions: (a) Experiments (b) Numerical calculations

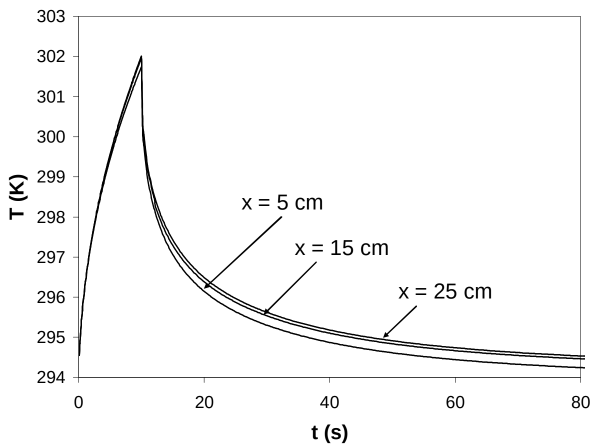
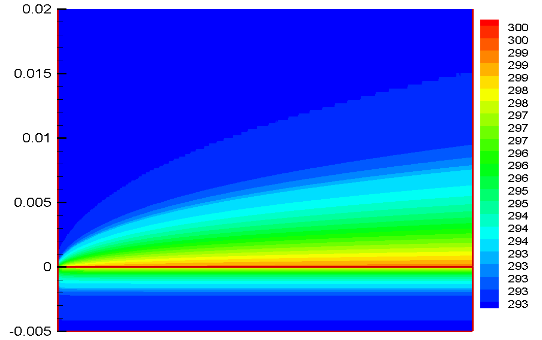
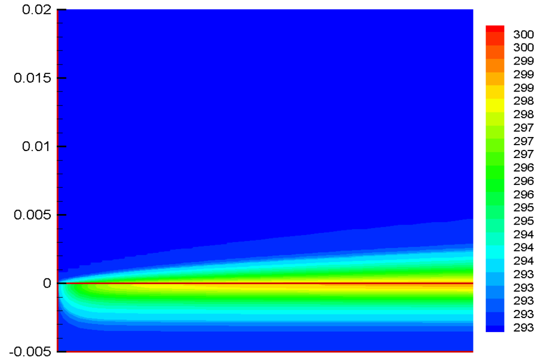


Figure 5 Temperature evolutions in time for a 5 mm thick plate, with a 10s heat flux on the front face of the plate

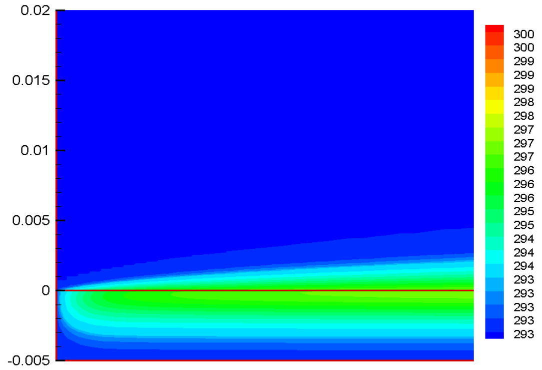
In Figure 6, temperature fields in the plate and in the air are exposed. They correspond to t = 5 s, 9.8 s, 11 s and 100 s respectively. Temperature profiles are evolving at the same time at the solid-fluid interface. A portion of the heat flux is diffusing in the plate and the other one is extracted by the air. Gradually, as the air temperature increases, the heat fluxes increase. It can be clearly seen that the development of thermal boundary layer in the fluid during the heating (Figure 6a and 6b) and the thinning of boundary layer after stopping the application of the heat flux on the interface (Figure 6c).



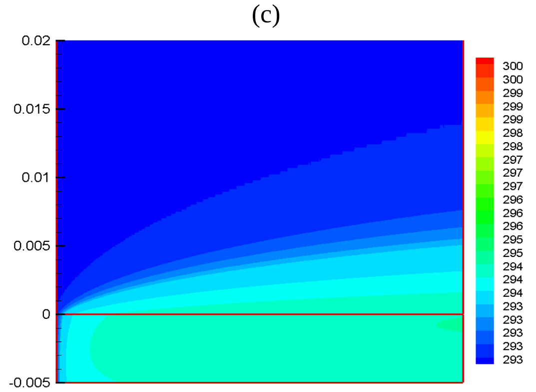
(a)



(b)



(c)



(d)

Figure 6 Temperature fields at t = 5 s (a) t = 9.8 s (b) t = 11 s (c) and t = 100 s (d)

In Figure 7, the convective heat transfer coefficient evolutions versus time are presented for 10 seconds-duration pulse and for four positions along the plate. The heat transfer coefficient reaches very high values at the beginning of the transient process. Subsequently, the heat transfer coefficient decreases, it goes through a minimum value at t = 10 s, and then it increases to reach an asymptotic value corresponding to the value of the steady state regime for each position (x). Because the heat transfer coefficient is defined as the heat flux density divided by the difference between the temperature at the interface and the reference one in the fluid, the heat transfer coefficient takes negative values for a certain period of time after the heat extinction. During this period, the heat flux has a negative value in the fluid because the penetrating portion in the wall is higher than that into the fluid.

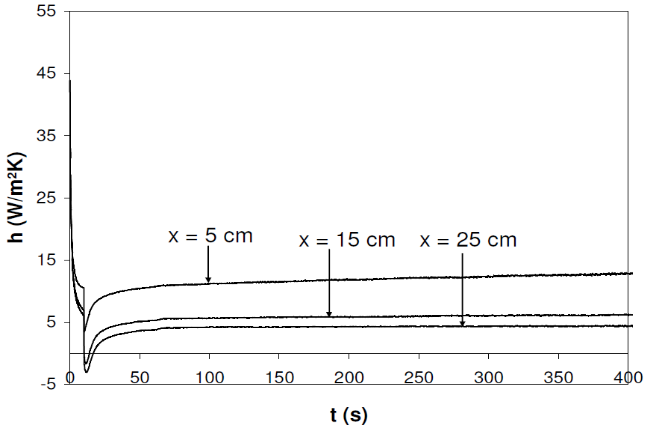


Figure 7 Temporal evolutions of the convective heat transfer coefficient for different positions

The impact of the thickness of the flat plate on the heat transfer coefficient evolution is illustrated in Figure 8 showing the time variation of this coefficient for two different positions (near the inlet and near the exit). For both thicknesses, heat exchange coefficients begin to evolve at the same time, but the minimum value achieved by this coefficient at $t = t_c = 10$ s, for the case of 5 mm thickness plate is significantly less than that corresponding to the plate with 0.5 mm thickness. At all other times and whatever the position (x), the heat transfer coefficient at the interface of the 5 mm thickness plate is always lower than that with 0.5 mm thickness.

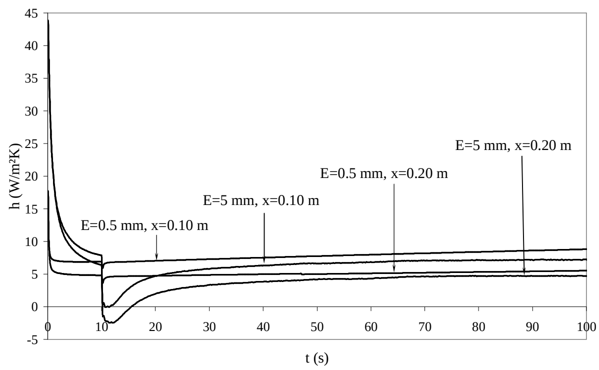


Figure 8 Temporal evolutions of the convective heat transfer coefficient for different thickness plates

HEAT TRANSFER COEFFICIENT CORRELATION

From this numerical study, the transient heat transfer coefficient can be correlated as a function of position (x) and time (t). The convective heat transfer coefficient is assumed to be represented as a product of a constant coefficient h_0 (heat transfer coefficient in steady state) which is multiplied by a temporal and space function f(t,x), which represents the evolution due to the application of the time-dependent heat flux required by the use of the pulsed photothermal method. Thus, the heat transfer coefficient can be expressed as [1]:

$$h(x,t) = h_0(x).f(t,x) \tag{9}$$

With h_0 is the convection coefficient of the initial and final steady-state regimes, which is given by:

$$h_0(x) = f(U_\infty, \mathrm{Pr}, x^{-1/2}) \tag{10}$$

And

$$f(t,x) = f(t).C(x) \tag{11}$$

The function f(x,t) has been deduced from the resolution of the conjugated conduction-convection problem in transient regime.

$$C(x) = 0{,}025x^2 - 1{,}0545x + 16{,}934 \tag{12}$$

For 0 < t < 10 s:

$$f(t) = t^{-0.86} \tag{13}$$

For 10 s < t < ∞

$$f(t) = a_4t^4 + a_3t^3 + a_2t^2 + a_1t + a_0 \tag{14}$$

The constants 'a_i' depend on the geometrical and thermo-physical properties of the plate.

CONCLUSION

In this paper, numerical results have been obtained for the conjugate convection-conduction problem with a time variation in the heat flux density which is applied on a front face of a flat plate. The Fluent numerical code has been used for that purpose. The highly unsteady behaviour of the surface temperature and the convective heat transfer coefficient were clearly exhibited. Temporal evolutions of these parameters were found to be strongly dependent on the boundary conditions. The available experimental results obtained by infrared thermography on the fluid-wall interface of the plate were used to validate the numerical procedure. It has been shown that the heat transfer coefficient depends on the position where it is calculated and the thickness of the plate. It has also been shown that the heat transfer coefficient reaches very high values at the beginning of the transient process, and then decreases, it goes through a minimum value and then increases to reach an asymptotic value corresponding to the value of the steady state regime for each x. The convective heat transfer coefficient is assumed to be represented as a product of a heat transfer coefficient corresponding to the initial and final steady-state regimes multiplied by a temporal and space function. That function represents the evolution due to the application of the time-dependent heat flux required by the use of the pulsed photothermal method.

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