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THE APPLICATION OF THERMAL IMAGING TECHNIQUES TO DETERMINE HEAT TRANSFER COEFFICIENTS FOR SLOW THERMAL TRANSIENTS

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The Application of Thermal Imaging Techniques to Determine Heat Transfer Coefficients for Slow Thermal Transients

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Abstract

This paper discusses the application of thermal imaging techniques to measure heat transfer coefficients on rotating machinery where, in general, the boundary conditions neccessary for the conventional analysis methods associated with such techniques are impractical. Both an infra-red thermal imager and wide-band thermochromic liquid crystals have been employed to measure surface temperature histories, and Duhamel's method has been used to compute the heat flux history. Duhamel's method, which involves the numerical solution of the time integral of the measured surface temperature, is appropriate for *slow thermal transients* for which there are no analytical solutions to Fourier's equation.

The measurement technique has been applied to an experiment in which a jet of hot air, characterised by a time varying temperature, impinged onto an initially cold rotating disc. The transient surface temperature of the disc was recorded using thermal imaging methods. Initial experimental results demonstrate that accurate measurements of heat transfer coefficients can be obtained.

Introduction

Thermochromic liquid crystals (TLC) are used widely in turbomachinery research [e. g. 1-4]. When they are employed to measure heat transfer coefficients, a semi-infinite surface is suddenly exposed to a step change in fluid temperature. Usually the initial temperature of the surface and the fluid temperature are known, and the surface temperature is measured at a known time after the step change. The heat transfer coefficient can then be computed from an exact solution of Fourier's equation (or the one-dimensional transient conduction equation as it is also known, see for example [5,6]). Most experimentalists use narrow-band TLC, which changes colour from red to green to blue over a temperature range of about 1 C. A single colour (yellow for example) can be used to determine the surface temperature to an accuracy of about ± 0.1 C.

In some experiments it may not be possible to create a step change in fluid temperature owing to, for example, the thermal inertia of pipework upstream of the test section. This case is referred to here as the *slow thermal transient.* Whereas the step-change transient requires only one measurement of the surface temperature, the slow thermal transient requires the history of the surface temperature to be known. Futhermore, an exact solution to Fourier's equation is not available under these circumstances and a numerical integration is required. Thin film resistance gauges have been used in conjunction with such a method to obtain heat flux histories [7].

Liquid crystals can be used to obtain surface temperature histories if wide-band TLC, which typically exhibit changes in colour over a range of 10 C, are used. Such crystals are generally used in conjunction with the hue technique and temperatures can be obtained to a much higher degree of accuracy than those obtainable with a single colour. Details of the hue technique are given in references [4,8,9]. Knowing the variation of surface temperature with time over the 10 C range allows an evaluation of the heat flux from a numerical solution of Fourier's equation using the measured temperature as a boundary condition.

An alternative method of acquiring surface temperature histories is to use an infra-red (IR) thermal imager. The IR imager has the advantage that the surface temperature can be obtained over bandwidths much greater than 10 C, thereby providing a more complete boundary condition for Fourier's equation. The disadvantages are that the IR equipment is significantly more expensive than that associated with TLC measurements and special windows are required to view the test piece, as glass and acrylic plastics are virtually opaque in the IR wavelengths.

Both the IR and TLC techniques have the advantage that remote sensing is used and intrusive instrumentation, such as thermocouples, are unnecessary. This is particularly useful for temperature measurements in rotating systems where thermocouples or thin film instrumentation require sliprings to transfer the measured voltages from the rotating to the stationary reference frames. In addition, embedded instrumentaion can create thermal disturbances which can have a significant effect on the measured temperatures. It should be noted that the temperature-hue calibration of TLC are unaffected by centripetal accelerations up to 16,000 g [10].

This paper demonstrates how the heat transfer coefficient can be determined experimentally from slow thermal transients using thermal imaging techniques. The method is applied to experiments involving jet impingement on rotating discs. There are plans to apply the method to future transient experiments involving rotating discs which simulate gas-turbine aero-engine components. Generally, step-changes in fluid temperature are impractical in experiments conducted with rotating-disc apparatus. Further discussion of this work is available in a laboratory report [11].

Theoretical Method

The quantity of interest in the experiments discussed here is the convective heat transfer coefficient (h), which is the constant of proportionality between the local surface heat flux (q_s) and the difference between the surface temperature (T_s) and convection driving temperature of the gas (T_{aw}, or *adiabatic wall temperature*) at that point:

$$q = h(T_s - T_{aw}) \ . \tag{1}$$

The adiabatic wall temperature is chosen since it is desirable to define, and measure, a heat transfer coefficient which is independent of the temperature boundary conditions and a function of the aerodynamic character of the flowfield alone. This is not strictly possible as there are well known and predictable differences between an isothermal wall situation and a constant heat flux boundary condition. However, for small gradients in temperature and constant fluid properties it has been found that a wall temperature variation has only a minor influence on local convective heat transfer [12]. For situations where compressibility effects are negligible and only one fluid is present, this adiabatic wall temperature is the local mainstream fluid temperature.

Consider the case of a semi-infinite substrate with a uniform initial temperature T_i at time $t = 0$. The substrate has known conductivity k, density ρ, and specific heat c. For the one-dimensional case, Fourier's conduction equation becomes,

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} , \tag{2}$$

where T is the temperature at depth x normal to the surface, and $\alpha = k/\rho c$ is the thermal diffusivity.

Consider further that the test section is suddenly exposed to hot or cold flow (the so-called case of a step-change in thermal boundary conditions) and the transient response of the test surface, as indicated by the TLC colour display or IR camera, is observed. The surface temperature is governed by Fourier's equation and the solution is well known:

$$\frac{T_s - T_i}{T_{aw} - T_i} = 1 - \exp\left[\frac{h^2 t}{\rho c k}\right] \operatorname{erfc}\left[\frac{h\sqrt{t}}{\sqrt{\rho c k}}\right] . \tag{3}$$

Here erfc is the complementary error function [5,6].

For situations where T_{aw} is known (incompressible, two-temperature problem), h can be determined from equation (3) by measuring the time required for the surface to reach a prescribed value as indicated by the TLC or IR sensor. This method is extended to cases where T_{aw} is unknown (three-temperature problem, e. g. situations with a secondary fluid) by noting that both h and T_{aw} can be obtained from simultaneous solution of two equations in the form of equation (3).

There is no analytical solution to Fourier's equation for cases where the fluid temperature varies significantly over the time during which the experiment is conducted. However, a solution may be obtained by numerical integration of the measured surface temperature using Duhamel's method.

Details of Duhamel's method are available elsewhere [5,6] and the details are not discussed here. The heat flux at the surface, q_s, can be shown to given by the following integral [11]:

$$q_s = \frac{\partial T}{\partial x}_{x=0} = \frac{1}{\sqrt{\pi \alpha}} \left[\frac{T_i - T_s}{\sqrt{t}} + \int_0^t \frac{T_i - T_s}{2(t-\tau)^{3/2}} \mathrm{d}\tau\right] \tag{4}$$

Knowing the variation of surface temperature with time, the above equation can be used to calculate the instantaneous heat flux, and the heat transfer coefficient where

$$h = \frac{q_s}{T_s - T_{ref}} \tag{5}$$

and T_{ref} is a suitable fluid reference temperature. Only if $T_{ref} = T_{aw}$, the adiabatic wall temperature, will h be invarient with time and with the changing temperature field. Since the mainstream fluid temperature changes with time, T_{aw} will also be time dependent together with q_s and T_s.

Experimental Apparatus

A schematic diagram of the experimental apparatus is shown in figure 1. Under normal operation, heated air suddenly impinges onto the centre of a rotating polycarbonate disc. The disc surface is coated with TLC and the temperature rise is monitored using either an infra-red (IR) imager or a CCD video camera. The working fluid is compressed air which flows from a reservoir through a British Standard orifice meter and in-line electric heater to a three-way ball-type diverter valve. The direction of the flow through the diverter valve is controlled by manually turning a handle through 90 degrees.

The diverter valve, which is thermally insulated, is used to bypass the heated flow away from the disc until both the air and the pipework reach the desired temperature. The diverted air passes through a control valve, which can be adjusted to equalise the flow resistance between the short section of pipe leading to the disc and the bypass circuit. When the resistance to flow is equal, and the insulated valve has reached a steady temperature, the flow rate and temperature will remain unchanged when the air is re-routed to impinge onto the disc. The temperatures of the impinging jet and the bypass air were measured by total-temperature probes.

This arrangement can be used to create a step change in heat transfer on the rotating disc. A slow thermal transient can be achieved by deliberately creating a difference in resistance through the two flow paths or alternatively the flow can be diverted before the pipework in the two circuits has reached a steady state. The speed of the slow transient can be controlled by varying the initial condition at which the diverter valve is operated.

Thermochromic Liquid Crystal

The polycarbonate disc, which is 225 mm diameter and 6 mm thick, is painted black to provide both an optimum background for the TLC and known emissivity for the IR imager. The TLC is microencapsulated and commercially available. When applied as a thin layer the coating is essentially clear and displays colour of decreasing wavelengths (red, green then blue) with increasing temperature once a threshold temperature has been reached. Beyond the active bandwidth, the TLC becomes clear again. A range of nominal threshold temperatures are available, as well as a range of bandwidths. In the work discussed here, three wide-band crystals were employed, active over the ranges 25-35 C, 35-45 C and 45-55 C (Hallcrest R25C10W, R35C10W and R45C10W respectively).

The light reflected from the TLC coating is monitored using a Panasonic WV-CL700 colour video camera fitted with a 300 mm lens, and the signals were recorded on videotape by a Panasonic AG-7355 cassette recorder. The video frames were then analysed using a Silicon Graphics Indigo R3000 Workstation fitted with a video board, and the conversion of Red-Green-Blue (RGB) to Hue-Saturation-Intensity (HSI) was performed using software from the Silicon Graphics Image Vision Processing Library.

The CCD camera is positioned to gain a near-normal view of the surface of the rotating disc. The video recorder acquires data at a rate of 25 frames per second during the transient. The polycarbonate disc has a steel hub with an outer radius of 65 mm, and the surface of the annulus is coated with the three wide-band crystals. The TLC with the highest threshold is sprayed onto the inner band of the annulus, the central band is coated with the TLC active over the intermediate temperatures, and the TLC on the outer band has the lowest threshold temperature.

Infra-Red Thermal Imager

An Agema 870C IR imager is used to complement the TLC measurements, and the IR scanner converts the electromagnetic radiation from the test surface into an electronic video signal. The scanner, dedicated PC and CASTE software were purchased as a complete package. Image data files are transferred from the PC to a Silicon Graphics workstation for post processing.

The IR imager is positioned to access a view normal to the disc surface and the PC is programmed to gather a sequence of images during the transient experiment. The maximum data-acquisition rate is seven frames per second.

Calibration

The calibration of the IR imager and the TLC is shown in figure 2 where an electrically-heated copper block is employed to create an isothermal surface. The copper is painted black and coated with TLC, and the thermal response of the surface, as indicated by the TLC colour display and the IR imager, is compared with measurements made using a thermocouple. The uncertainty in the thermocouple reading was estimated to be 0.1 C in the range 20 to 70 C. With an emission setting of 0.96, the IR imager was found to agree within 0.1 C of the thermocouple reading over most of the range 47 C to 58 C, the maximum error being 0.3 C. By averaging the hue over a large number of pixels (> 104), the 95% confidence for the TLC readings was better than 0.005 C. Using a single pixel, the "2 σ uncertainty" for the TLC was around 0.4 C. Further details about the calibration are available [10]. A typical TLC calibration result is shown in figure 3.

The copper block provides an isothermal surface for which, in theory, only one value of hue appears in the video image. The video hardware and software can bias the signal when a wide range of hue values appears in the image [10]. Ideally, an in situ calibration of the TLC should be carried out on the rig being used, but this is not always practicable.

Determination of heat transfer coefficient for a rotating disc and slow transient

Air temperature measurements

Figure 4 illustrates air-temperature histories for both a step-change and a slow transient. The temperatures in this figure were obtained by the total temperature probes in the bypass circuit and in the impinging jet, as shown in figure 1, and in both tests the flow is diverted in about one second. For the step-change, the temperature of the air impinging onto the centre of the rotating disc is invariant with time while the temperature in the by-pass circuit decreases slowly once the flow is diverted. For the slow transient, the temperature of the impinging air increases exponentially during the course of the experiment. The time constant of the transient can be shortened or lengthened by reducing or increasing, respectively, the differences in temperature between the two routes before the diverter valve is operated.

Surface temperature

The surface temperature distribution of the disc, rotating at 250 rev/min, is plotted as a function of disc radius in figure 5, and the associated air temperature for this test is the slow transient illustrated in figure 4. There is a series of curves in this figure, each curve recorded in sequence throughout the slow transient using the infra-red imager. In all, 100 sequences were stored into computer memory at a rate of 6.5 readings per second, and only 11 of these sequences are shown in the figure. The curve at the lowest temperature (the initial temperature of the disc before the flow was diverted) is the temperature defined (arbitrarily) at $t = 1$ second. The other curves show the surface temperature distribution at $t = 1.61$, 2.38, 3.92, 5.46, 7.0, 8.54, 10.1, 11.6, 13.2, 14.7 and 15.5 seconds.

It can be seen from figure 5 that, as expected, there is a decrease in surface temperature with radius and an increase with time. The sequence of curves is similar and monotonic apart from the small annular bands around 130 mm and 175 mm. These two bands correspond to regions of the polycarbonate disc which were not painted black. The temperatures shown in figure 5 are based on an emissivity of 0.96 (i.e. the emissivity of the black paint under clear liquid crystals) and are not accurate within the two unpainted bands.

The surface temperature history, measured by the IR imager, at one radius of the disc (135 mm) is shown in figure 6.

Computation of q_s and h

The surface temperature history in figure 6 was used to calculate the heat flux by Duhamel's method. The integration is carried out numerically and the results are shown in figure 7, where the sign of the heat flux has been (arbitrarily) chosen to be negative. There is an initial steep rise in the magnitude of the computed heat flux, lasting approximately 1.5 seconds, after which the magnitude of heat flux decreases with time as the difference in the temperatures of the air and the surface decreases.

The instantaneous heat transfer coefficient, h, at this radius can be obtained by dividing the heat flux, q_s, by the difference between the rotating disc surface temperature, T_s, and air temperature, T_f, from figure 4. Since both q_s and $T_s - T_f$ are known functions of time, the heat transfer coefficient can be computed as a function of time. Only if h is based on $T_s - T_{aw}$ will h be invariant with time, as discussed previously.

The value of T_{aw} which yields a constant heat transfer coefficient has been calculated and is plotted as a function of time in figure 8. This heat transfer coefficient was approximately 370 $Wm^{-2}K^{-1}$. The adiabatic wall temperature is seen to approach an asymptotic value near 41 C. The air temperature measured in the impinging jet approached a temperature near 67 C, which is significantly greater than T_{aw}. The entrainment of cold air into the boundary layer on the rotating disc will obviously cause a reduction in the driving temperature for heat transfer, which serves to explain why T_{aw} is much lower than the air temperature.

Comments

Future work will be to compare the heat transfer coefficient obtained using the slow thermal transient with that measured from a step-change experiment under identical flow conditions. Since the heat transfer coefficient, to first order, depends only upon the aerodynamic character of the flowfield, these two heat transfer coefficients should be identical. This work will be undertaken using a stationary target plate for simplicity.

The variation of surface temperature measured throughout the slow transient presented here was only 3 C. Measurements over a greater temperature range will increase the accuracy of the calculations of q_s and T_{aw}. If measurements were recorded over a longer time, this temperature variation would have increased, though care must be taken to insure that the elapsed time and corresponding depth of heat penetration into the polycarbonate disc are limited if the assumption of a semi-infinite slab is to be valid. Alternatively, a higher inlet air temperature would create a greater rise in T_s.

The surface temperature was measured by employing the IR imager, but the measurements could have been made using the TLC and CCD camera. The active range of the wide-band TLC is 10 C, which is three times the range used in the slow-transient tests described here.

It is appropriate to mention that a distinct advantage of the thermal imaging techniques discussed here is that all (visible) surfaces are thermally active, and hence global measurements of heat transfer are possible. The data presented above corresponds to a radius of 135 mm but information is available at any visible radius of the disc.

Conclusions

It has been demonstrated that Duhamel's method can be used in conjunction with IR or TLC measurements to calculate the heat transfer coefficients for slow-transient tests in semi-infinite solids.

The method was applied to an experiment in which a jet of hot air impinged on the centre of an initially cold rotating disc made from polycarbonate. The heat transfer coefficient was computed at one radius using Duhamel's method applied to smoothed surface temperatures, the latter being measured by an IR imager. From the computations of heat flux, a time-varying adiabatic surface temperature was calculated which kept the heat transfer coefficient constant.

Having demonstrated the accuracy and simplicity of this method, it will be applied to future transient experiments in which step-changes are impracticable, as is often the case in rotating-disc rigs. In principle, either wideband TLC (using the hue technique) or IR measurements can be used with this method to give good spatial resolution of h, the choice of measurement depending on the experimental constraints.

References

[1] T. V. Jones and S. A. Hippensteele, 1988, *High-Resolution Heat-Transfer-Coefficient Maps Applicable to Compound-Curve Surfaces Using Liquid Crystals in a Transient Wind Tunnel,* NASA Technical Memorandum 89855

[2] J. W. Baughn, 1995, *Liquid Crystal Methods for Studying Turbulent Heat Transfer,* International Journal of Heat and Fluid Flow, **16**, 365–375

[3] Kim, Y. W., Downs J. P., Soechting, F. O., Abdel-Messeh, W., Steuber, G. D. and Tanrikut, S., 1995, *A Summary of the Cooled Turbine Blade Tip Heat Transfer and Film Effectiveness Investigations Performed by Dr DE Metzger,* Journal of Turbomachinery, (**117**), pp 1–11

[4] P. T. Ireland, Z. Wang and T. V. Jones, 1995, *Liquid Crystal Heat Transfer Measurements,* von Karman Institute for Fluid Dynamics, Lecture Series

[5] G. E. Myers, 1971, *Analytical Methods in Conduction Heat Transfer,* McGraw Hill, New York

[6] P. J. Schneider, 1955, *Conduction Heat Transfer,* Addison-Wesley, Reading, Mass.

[7] D. L. Schultz and T. V. Jones, 1973, *Heat Transfer Measurements in Short-Duration Hypersonic Facilities,* AGARDOGRAPH 165

[8] C. Camci, K. Kim and S. A. Hippensteele, 1992, *A New Hue Capturing Technique for the Quantitative Interpretation of Liquid-Crystal Images Used in Convective Heat Transfer Studies,* Journal of Turbomachinery, **114**, 765–775

[9] M. Wilson, B. J. Syson and J. M. Owen, 1993, *Image Processing Techniques Applied to Wide-Band Thermochromic Liquid Crystals,* Eurotherm 32: Heat Transfer in Single Phase Flows: Oxford, July 1993

[10] B. J. Syson, R. Pilbrow and J. M. Owen, 1996, *Effect of Rotation on Thermochromic Liquid Crystal,* International Journal of Heat and Fluid Flow, **17**: 491–499

[11] G. D. Lock, J. M. Owen and B. J. Syson, 1996, *Determination of Heat Transfer Coefficients for Slow Thermal Transients,* Report No. 27/96, School of Mechanical Engineering, University of Bath

[12] Eckert, E. R. G., Hartnett, J. P. and Birkebak, R., 1957, *Simplified Equations for Calculating Local and Total Heat Flux to Nonisothermal Surfaces,* Journal of Aeronautical Sciences, (**24**), No. 7, pp 549–550.

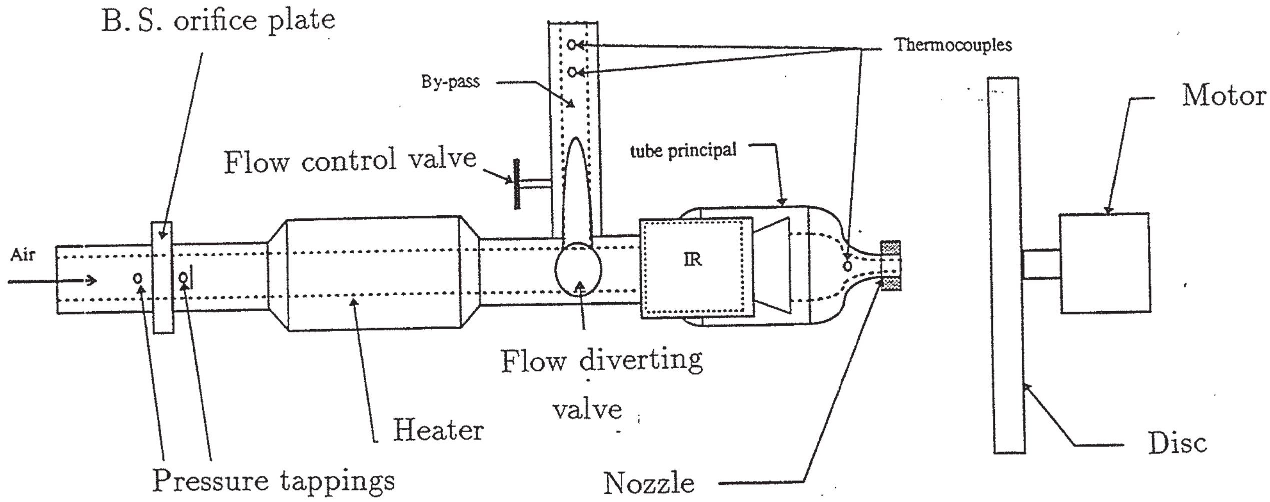

Figure 1: Experimental apparatus

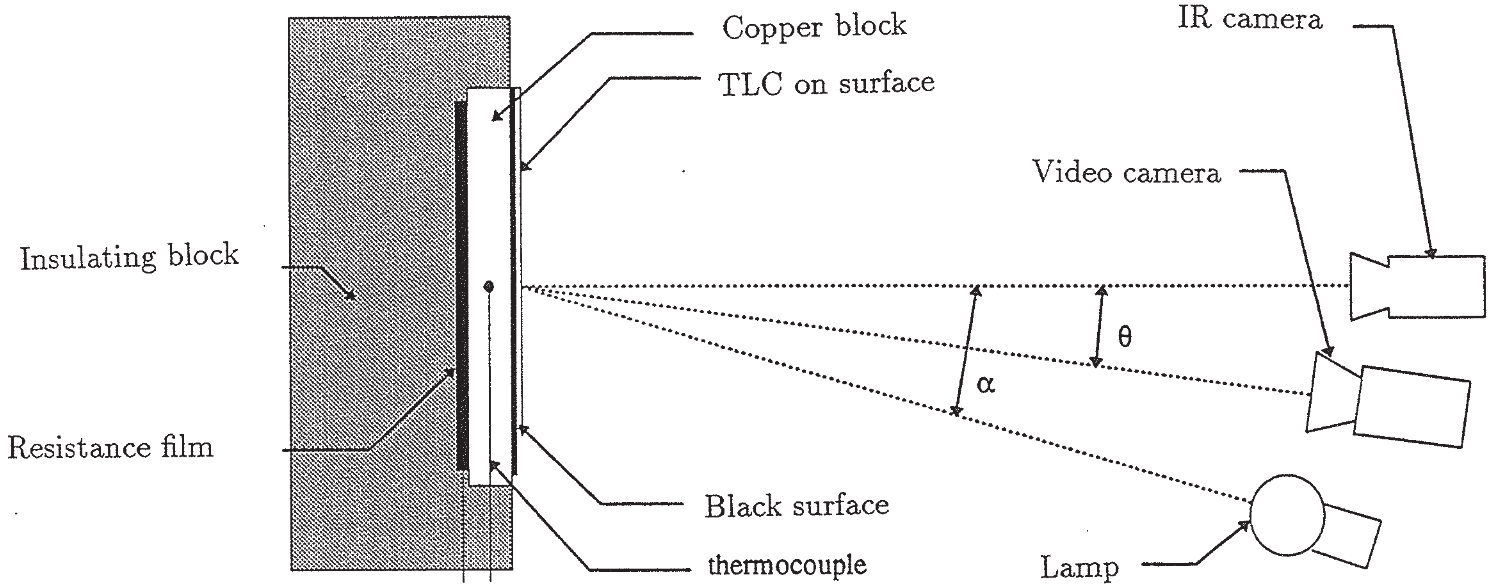

Figure 2: Calibration apparatus

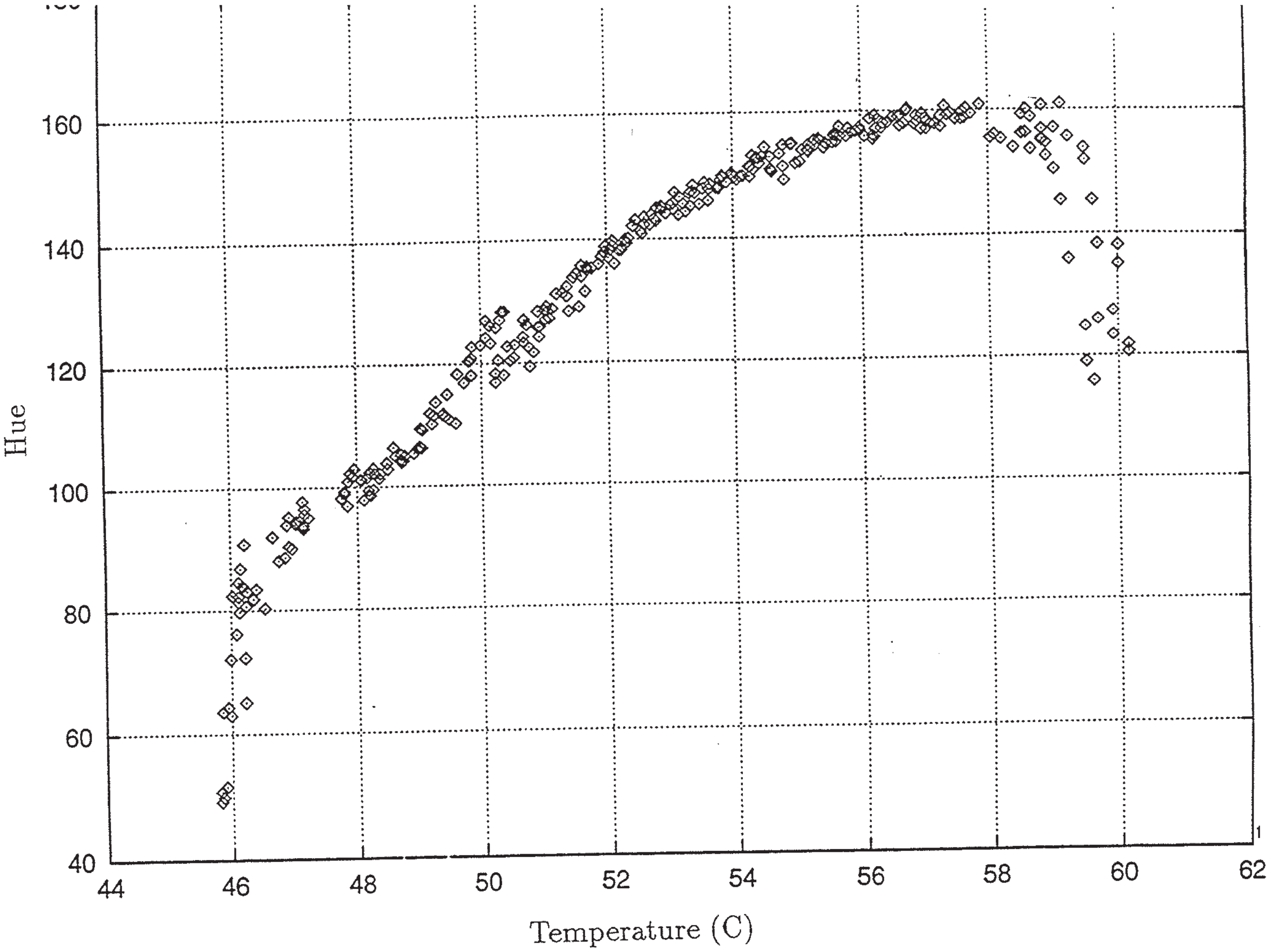

Figure 3: TLC calibration results

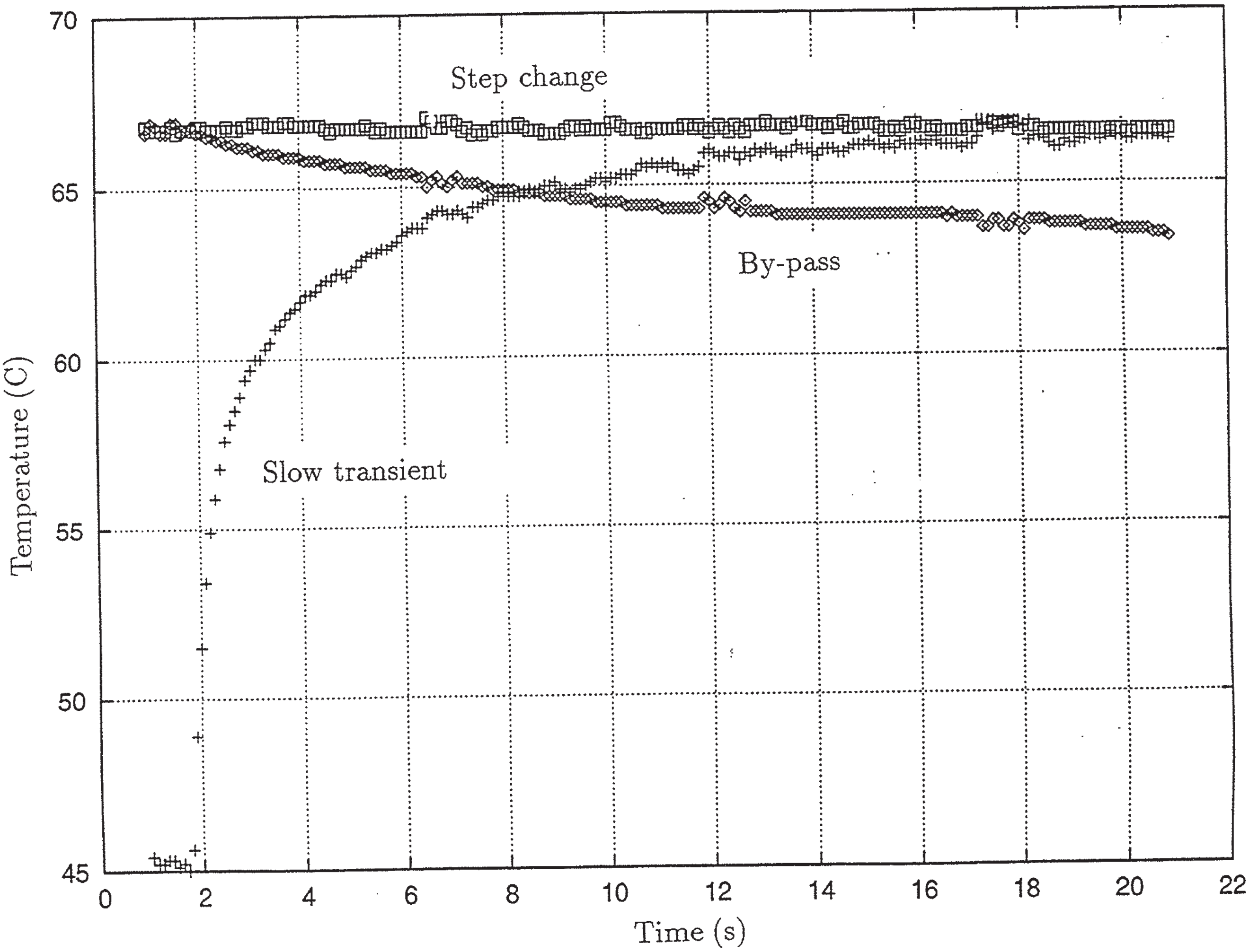

Figure 4: Inlet air temperature histories

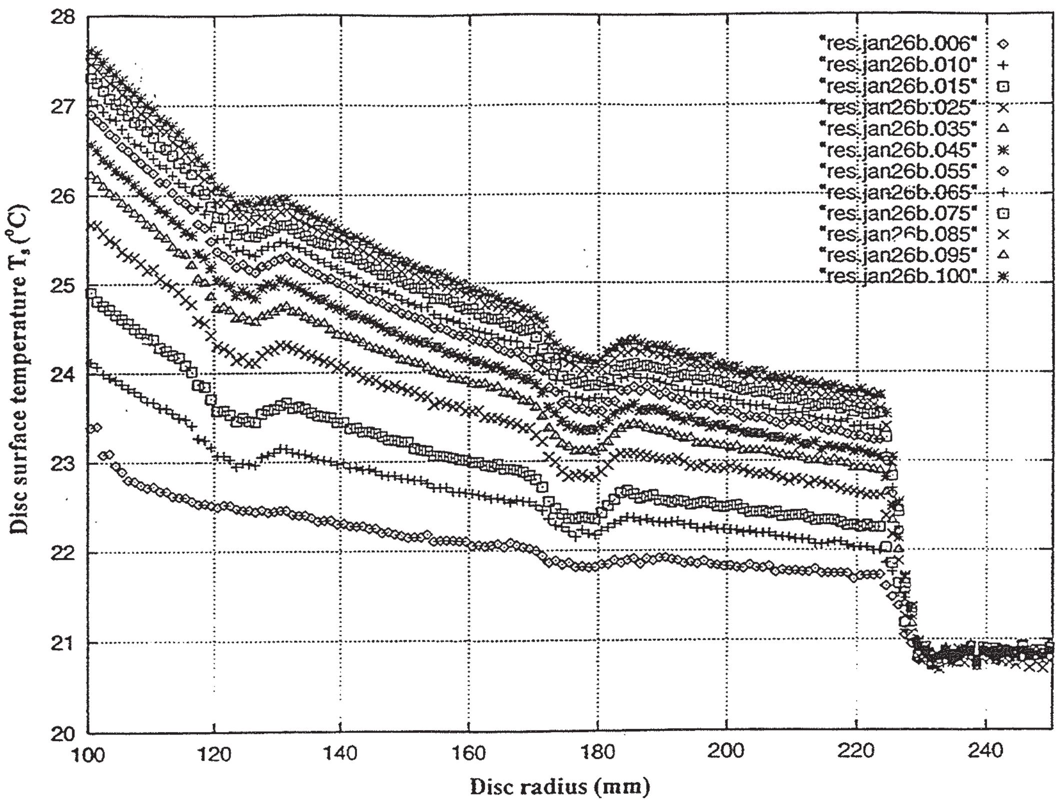

Figure 5: Measured variation of T_s with radius at various times during the slow transient

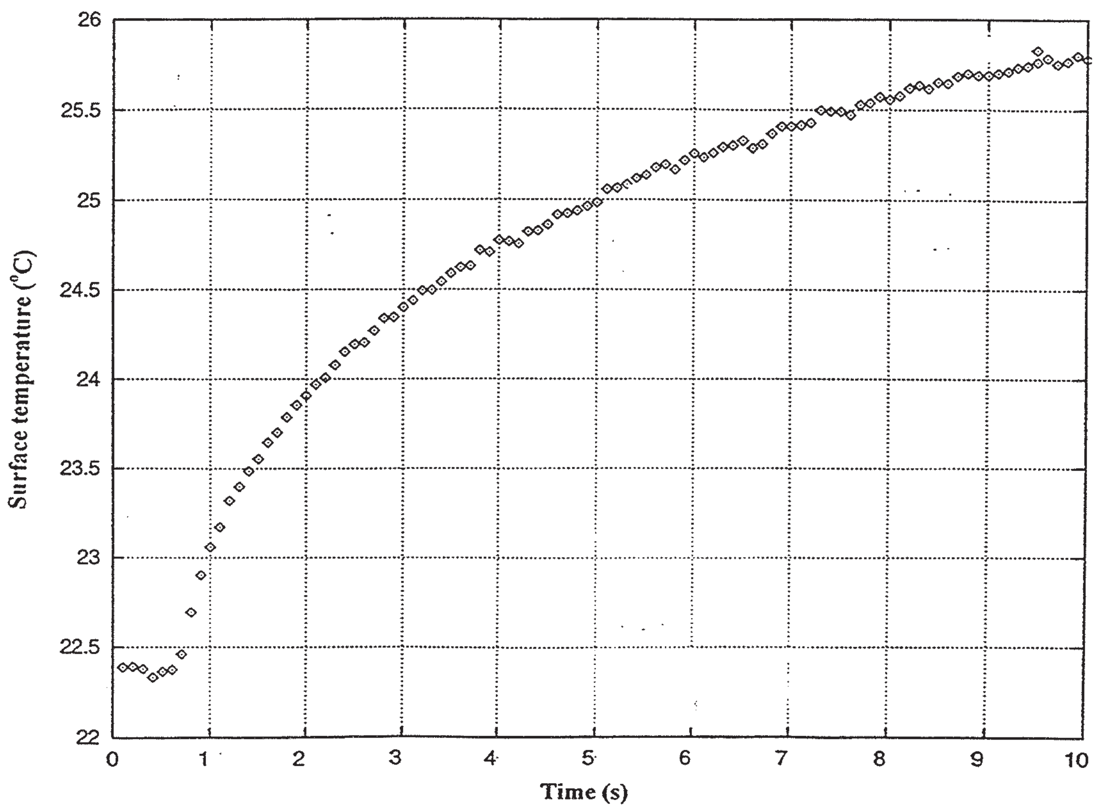

Figure 6: Measured variation of T_s with time at a radius of 135 mm for the slow thermal transient

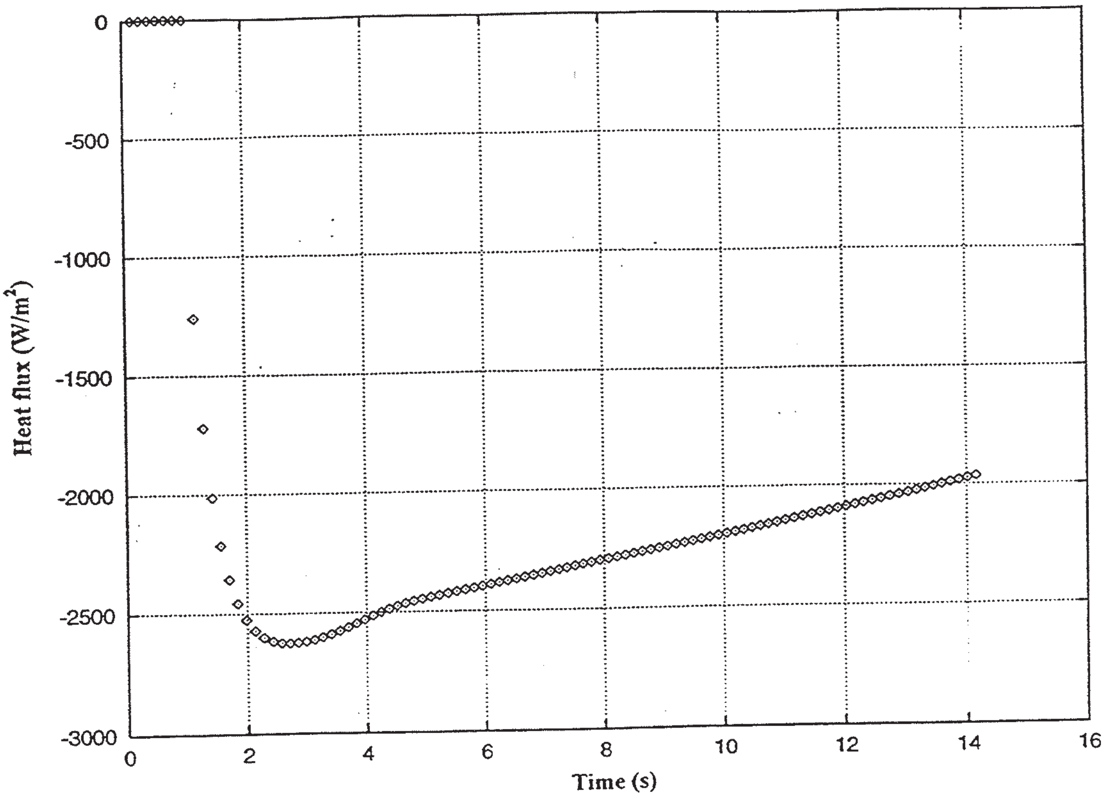

Figure 7: Computed variation of q_s with time at a radius of 135 mm for the slow thermal transient

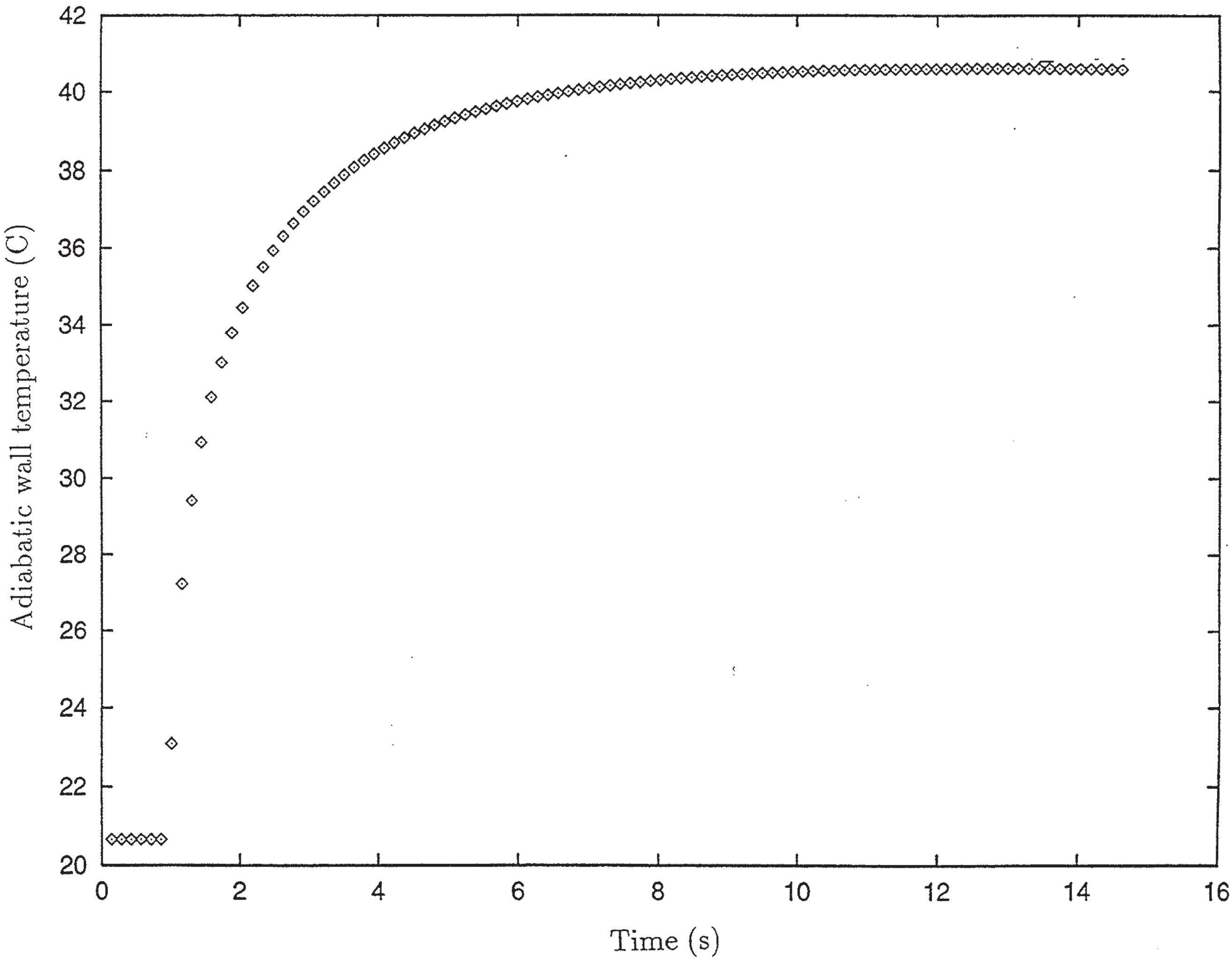

Figure 8: Computed variation of T_{aw} with time for a constant heat transfer coefficient throughout the slow thermal transient