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Visualization of Transient Natural Convection Heat Transfer from a Vertical Rectangular Fin

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***Abstract*:** Experiments on thermal visualization of transient natural convection from short vertical rectangular fins were conducted using the technique of laser holographic interferometry. A sequence of infinite-fringe interferograms recorded for the heating regime of an aluminum fin demonstrates the effect of fin base heating on local convection coefficients and reveals alternating and oscillatory buoyancy-driven flows similar to those over the top surface of heated horizontal plates. The effect of fin base heating results in greater uniformity of the local heat transfer coefficient along the fin. It also significantly reduces the steady-state heat transfer coefficients of short vertical fins compared to their transient values. Hence, the use of steady-state solutions for the design of short vertical fins operating in transient conditions may not introduce as much error as was previously thought.

***Keywords*:** transient natural convection, laser holographic interferometry, vertical rectangular fins, thermal visualization, interferograms.

***Nomenclature*:**

dT/dy	temperature gradient of air at the fin wall, K/m
h_x	local heat transfer coefficient, W/m^2·K
k	thermal conductivity of air, W/m·K
L	fin length or height, m
n	refractive index of air
n_o	refractive index of unheated ambient air
N	fringe order number
t	elapsed time from start of fin heating, s
T	air temperature, K
T_S	fin surface temperature, K
T_∞	freestream temperature, K
W	fin width in direction of laser beam, m
x	distance along fin measured from fin base, m
y	distance from fin surface, m
$\Delta\phi$	optical pathlength difference, m
λ	wavelength of laser beam, nm

1. Introduction

The dissipation of heat using extended surfaces in natural convection is often relied upon for economical, noiseless and maintenance-free cooling of electronic equipment. Finned surfaces are especially attractive for these applications since they effectively increase the heat transfer rate if properly designed. Although in many cases, fins operate under transient conditions, they are usually designed for the worst possible conditions at steady-state operation, which results in larger than necessary fin weight.

Although the use of fins is widespread, the literature is relatively sparse for natural convection heat transfer from vertical fins with a horizontal base. Starner and McManus (1963) performed an experimental investigation of steady-state natural convection from rectangular fin arrays. They found that for short fins, the horizontal base orientation performs better than inclined orientations. Harahap and McManus (1967) reported that for a long fin, the mass flow rate of fresh air having to move a long distance along the fin and getting heated at the same time, will cause a single chimney flow. This single chimney flow is favorable to high rates of heat transfer from horizontal fin systems. They also found that for shorter fins, sliding chimney strip flow occurred due to insufficient inflow of air. Leong et al. (1994) applied the technique of holographic interferometry to the visualization of temperature fields surrounding a vertical plate, vertical fin and fin array under steady-state natural convection. Qualitative observations from interferograms of a vertical fin and fin array show the effect of boundary layer interaction between adjacent fins and also the effect of a chimney flow reported by other investigators. Leong and Kooi (1996) using the same technique of holographic interferometry demonstrated the effect of base heating on steady-state natural convection from vertical aluminum and mild steel fins of rectangular profile. The local heat transfer coefficients were found to vary along the fin with maximum values of 22 – 48 % of the fin height measured from the base.

Numerous theoretical studies were performed on transient heat transfer from longitudinal, spine-shaped and annular fins. (Chapman, 1959; Suryanarayana, 1975; Aziz and Na, 1980; Chang et al., 1982; Assis and Kalman, 1993). The literature is relatively sparse in experimental work relating to the transient behavior of fins. Sobhan et al. (1989) performed an experimental study of transient natural convection from horizontal fin arrays. Steady-state results were compared with those of a single vertical fin of 70-mm height and a vertical flat plate. A maximum value for the local heat transfer coefficient at about 25 – 30 % of the fin height from the base was obtained for steady-state natural convection from a single fin. A recent study by Assis et al. (1994) made use of the liquid crystal technique to visualize the temperature distribution along the fin. However, there were no results on the thermal visualization of the air temperature field surrounding the fin.

Following up on recent studies on steady-state natural convection from vertical fins using the non-intrusive optical technique of holographic interferometry by the author and his co-workers, this paper presents the results of thermal visualization studies for transient natural convection from vertical rectangular fins with a horizontal base. The objective is to obtain a better understanding of the effect of fin base heating on heat transfer from short vertical fins. The variation of local heat transfer coefficient with fin height from the onset of fin base heating to steady-state operation will also be presented.

2. Experimental Set-up and Procedures

To visualize the temperature field and to obtain the variation of local heat transfer coefficient along the fins, a laser holographic interferometry system was set up. The optical arrangement of the system is shown in Fig. 1. An Omnichrome air-cooled 130 mW Argon ion laser was used as the coherent light source. This laser was operated at 514 nm wavelength in the experiments due to the better fringe visibility obtained compared to that obtained by using the laser at 488 nm. A reference-to-object beam ratio of 3:1 was chosen as it was found to produce the best fringe contrast. The collimated beams of 50-mm diameter were allowed to impinge on a Newport HC-301 thermoplastic plate of size 30 mm by 30 mm placed onto an HC-300 thermoplastic recorder. A video camera was used to record the real-time interferograms for a specified duration of transient heat transfer. The entire optical system sits on a pneumatically supported honeycomb table.

The fin assembly used in the experiments is shown in Fig. 2. The fins were made from aluminum 6061 and mild steel SS 41 with thermal conductivity values of 186 W/m·K and 56.7 W/m·K, respectively. Each fin has a thickness of 3 mm and a height of 20 mm. The fin base has a length of 60 mm, width of 100 mm and thickness of 5 mm which is sufficient to ensure a uniform base temperature. The width is long enough to ensure that the air surrounding the fin can be considered as a two-dimensional phase object. The fins were all machined out of integral metal blocks to eliminate contact resistance between the fins and their fin bases.

A 60 W mica heater seated in a groove machined on a Bakelite board provided heating to the fin base. The Bakelite board reduces heat loss from the back of the fin base. A 0.58 kVA transformer was used to vary the electrical power to the fin heater. A wattmeter with a precision of 1 W was used to display the power consumption of the fin heater.

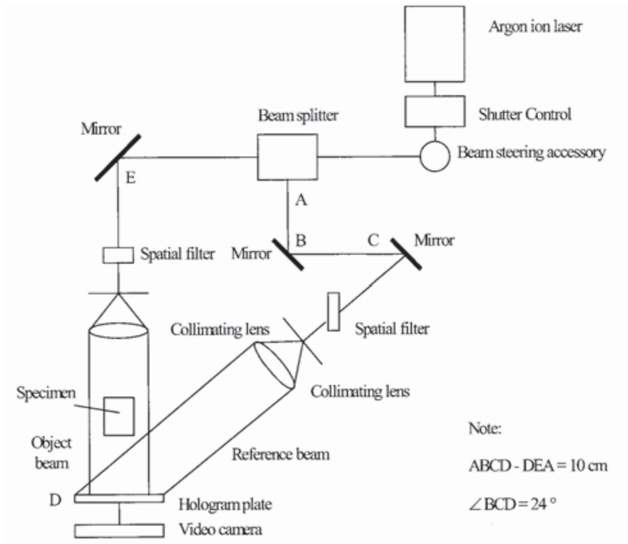


Fig. 1. Optical set-up.

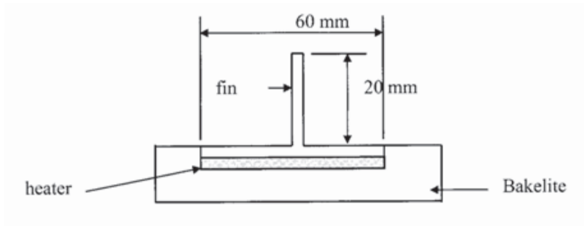


Fig. 2. Fin assembly.

The entire fin assembly was mounted on a stand placed in the path of the object beam. The experiments were conducted in an enclosed dark room under still air conditions. The temperature of the air in the room was measured throughout the duration of the experiment and found to be constant at 28°C. The real-time holographic interferometry method was used. Its advantages compared to Mach-Zehnder interferometry are described in detail by Leong et al. (1994). Initially, a holographic exposure of the unheated air surrounding the fin was made. Electrical power to the heater was switched on and real-time interference fringes were produced by the superposition of the object beam with the reconstructed initial hologram achieved by illuminating the hologram plate with an identical and coherent reference beam. Infinite-fringe interferograms produced on the hologram plates were recorded by a video camera for the entire duration of a 3-hour VHS videotape. By playing back the video recording, frames in which a new constructive (or bright) fringe was formed were digitized using Multimedia Video Producer software.

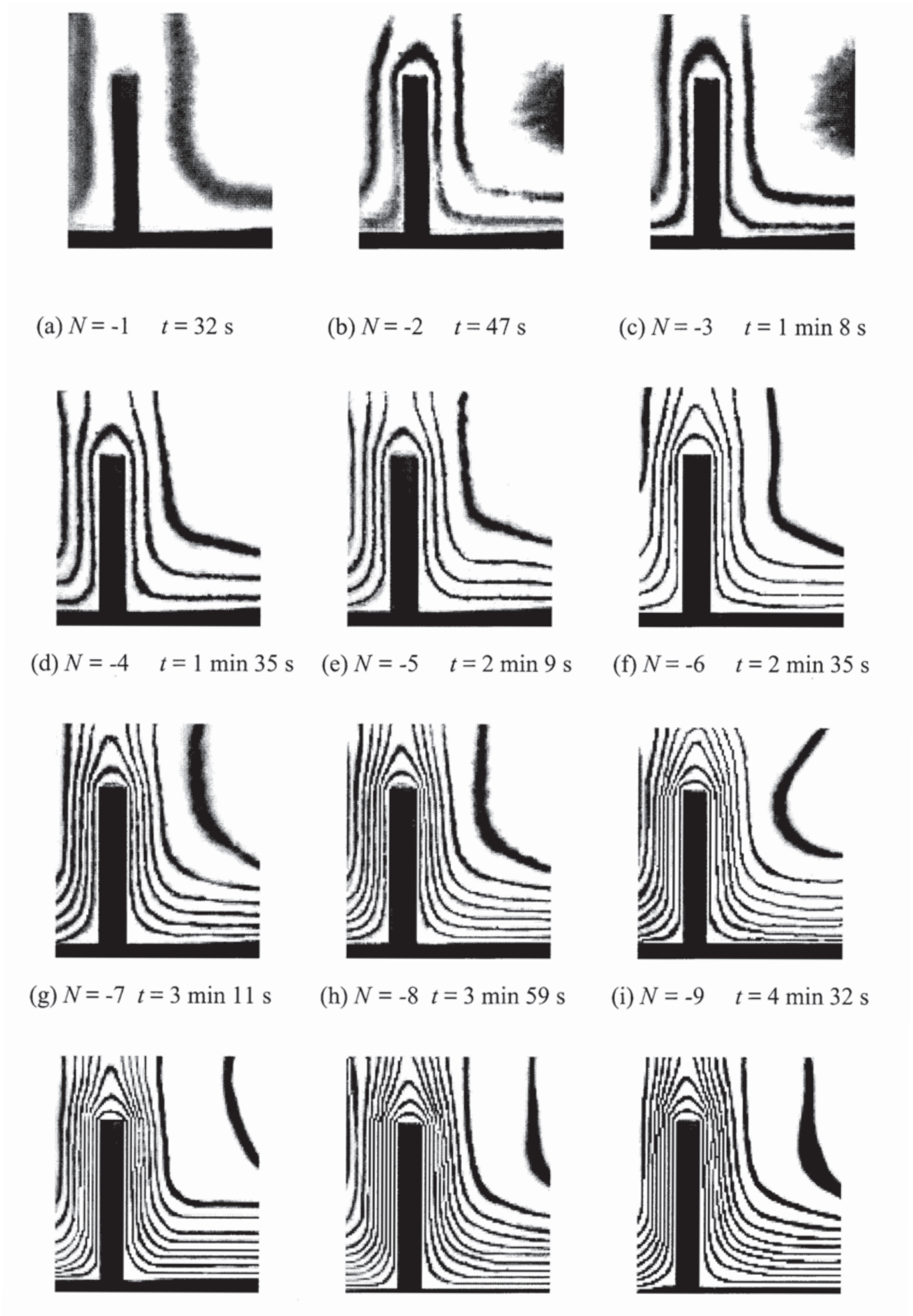
3. Fringe Interpretation and Data Reduction

In order to extract quantitative information of the temperature field from the interferograms, the infinite fringes have to be interpreted or related to the absolute temperature of the air. The interferogram pathlength difference of a two-dimensional phase object can be described by

$$\Delta\phi\ (x, y) = [n\ (x, y) - n_0]\ W = N\lambda \tag{1}$$

The refractive index of air at $\lambda = 514$ nm as a function of the absolute temperature was calculated from the following expression by Radulovic (1977):

$$n - 1 = \frac{0.294036 \times 10^{-3}}{1 + 0.369203 \times 10^{-2}\, T} \tag{2}$$



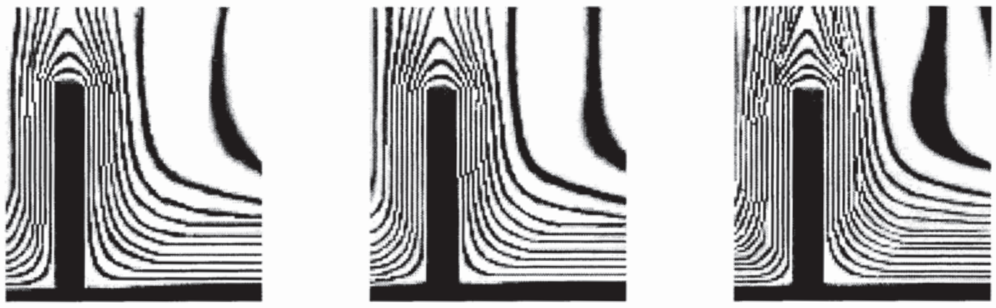
(a) $N = -1$ $t = 32$ s (b) $N = -2$ $t = 47$ s (c) $N = -3$ $t = 1$ min 8 s

(d) $N = -4$ $t = 1$ min 35 s (e) $N = -5$ $t = 2$ min 9 s (f) $N = -6$ $t = 2$ min 35 s

(g) $N = -7$ $t = 3$ min 11 s (h) $N = -8$ $t = 3$ min 59 s (i) $N = -9$ $t = 4$ min 32 s

(j) $N = -10$ $t = 5$ min 32 s (k) $N = -11$ $t = 6$ min 22 s (l) $N = -12$ $t = 8$ min 4 s

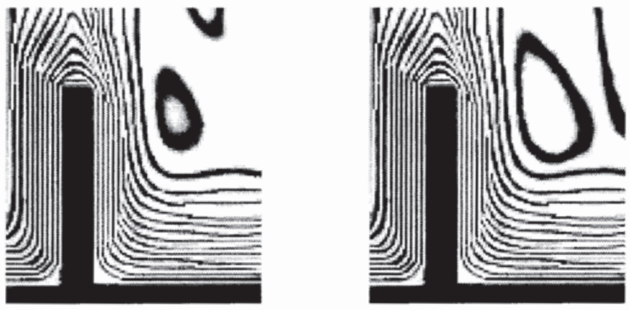
Fig. 3. Interferograms showing transient performance of an aluminum fin.



(m) N = -13 t = 9 min 20 s (n) N = -14 t = 11 min 53 s (o) N = -15 t = 14 min 10 s

(p) N = -16 t = 18 min 24 s (q) N = -17 t = 27 min 30 s (r) N = -18 t = 37 min 48 s

(s) N = -19 t = 48 min 36 s (t) N = -20 t = 1 h 3 min 59 s (u) N = -21 t = 1 h 24 min 38 s



(v) N = -22 t = 1 h 55 min 46 s (w) N = -23 t = 2 h 45 min 42 s

Equation (2) is based on the Gladstone-Dale relations with wavelength dependence calculated according to Meggers and Peters (1918). From Equation (2), the refractive index of the unheated air, n_o can be calculated if the ambient temperature is known.

By combining Equations (1) and (2), the air temperature at any position y measured from the fin surface can be determined as

$$T = \frac{1}{0.00369203}\left(\frac{0.294036 \times 10^{-3}}{N\lambda / W + n_o - 1} - 1\right) \tag{3}$$

The distance of the center of each fringe to the fin wall was measured from the digitized fringe images using GLOBAL-LAB Image software. A fourth-order polynomial approximation for the temperature profile in the thermal laminar boundary layer as recommended by Hauf and Grigull (1970) was assumed. It has the form

$$T = ay^4 + by^3 + cy^2 + dy + e \tag{4}$$

where a, b, c, d, and e are constant coefficients.

Curve fitting to the experimental data was performed using SigmaPlot software based on the Marquardt-Levenberg algorithm which minimizes the sum of squares of the differences between the dependent variable values in the equation model and the observed values. Simple differentiation of Equation (4) yields the temperature gradient dT/dy at the fin wall allowing the local heat transfer coefficient h_x to be calculated from

$$h_x = \frac{-k(dT/dy)_{y=0}}{T_s - T_\infty} \tag{5}$$

4. Results and Discussion

4.1 Infinite-fringe Interferograms

Figure 3 shows a sequence of digitized infinite-fringe interferograms of transient natural convection from an aluminum fin. The bright and dark fringes correspond to isotherms in air. Each image corresponds to the formation of a new bright fringe. For a three-hour duration from the start of fin heating, a total of 23 such fringes were formed. The labels beneath each image show the order number N of the latest bright fringe and the elapsed time t from the start of heating.

At the start of the experiment, the formation of the fringes was relatively fast. The time interval between the appearance of fringes $N = -1$ and $N = -2$ is only 15 s. The time interval between successive fringes increases with time. The elapsed time between the appearance of fringes $N = -22$ and $N = -23$ is almost 50 minutes. For early times, the fringes are stable. However, at later stages of the experiment corresponding to high fin base temperatures, the fringes become unstable. From Figure 3 *r*, it can be seen that sufficiently far from the fin, the flow is driven by descending and ascending parcels of fluid. Conservation of mass dictates that cold fluid descending from the ambient is replaced by warm fluid ascending from the fin base. The flow patterns are similar to that occurring above a heated horizontal plate (Incropera and DeWitt, 1996) in which the same pattern of buoyancy-driven flows will be expected to occur in the span-wise direction of the fin and the flow is actually three-dimensional in nature for plates of short spans. The presence of these buoyancy-driven flows can result in slight thinning of the vertical fin boundary layer (Fig. 3 *r* and *s*). A careful examination of Fig. 3 *r*, *s* and *t* reveals the unstable and oscillatory nature of the buoyancy-driven flows during the transient period. Unfortunately, due to the limitation of the printed page, this phenomenon is not as clearly seen as compared to watching a video playback of the fringe development.

The effect of fin base heating is evident from the thick boundary layer above the horizontal fin base even for early times (Fig. 3 *a* - *c*). For an elapsed time of 1 min 8 s (Fig. 3 *c*), the horizontal boundary layer thickness is about 25 % of the fin height from the base. Because of the short height of the fin, the horizontal boundary layer can extend to the entire fin height at long elapsed time approaching the steady-state condition (Fig. 3 *t*). The effect of fin base heating results in value of zero for the convection coefficient at the fin base, and low convection coefficients throughout the entire fin.

A plume of heated air can also been seen rising from the fin tip. Interaction between the heated air at the tip and the buoyancy induced flow along the fin wall resulted in a slight inward bending of the isotherms near the fin tip. Above the fin tip, the isotherms are spaced further apart compared to elsewhere along the fin indicating that fin tip heat loss is small compared to heat loss along the fin.

4.2 Local Heat Transfer Coefficients

Figure 4 shows the development of the local heat transfer coefficient along the fin height for an aluminum fin at various times t from the onset of heating. There is an increase in heat transfer coefficients with time from 5 to 25 min., which is due to the increased temperature difference between the fin and the ambient fluid. This phenomenon can also be deduced from the increased number of fringes and decreased fringe spacing as shown in Fig. 3. The increase in heat transfer coefficient with heating time was also observed by Sobhan et al. (1989) in his experiments on transient natural convection from fin arrays. There is a slight increase in the local heat transfer coefficient h_x with fin height for $t = 5$ min. For longer times of 15 and 25 min., the variation along the fin height is more significant because of increased horizontal boundary layer thickness and interaction with the vertical fin boundary layer. For an elapsed time of 35 min., the horizontal boundary layer extends almost to the fin tip resulting in a decrease in local heat transfer coefficients. The relatively high values of h_x near the fin tip is due to the combined effects of thinning of the boundary layer by the buoyancy-driven flows described in the previous section and the interaction with the rising plume from the fin tip. At steady-state conditions, which are reached after 3 hours from the start of heating, the local heat transfer coefficients have values between 6 – 8 $W/m^2 \cdot K$ which are about the same magnitudes as those corresponding to $t = 5$ min. The variation of heat transfer coefficient with fin height is also not as significant compared with the results attained for aluminum fins of 40 mm height (Leong and Kooi, 1996). The base temperature attained for steady-state condition is 538 K. The lower values of steady-state heat transfer coefficients compared to its transient state is characteristic of short vertical fins and is a direct result of fin base heating. The effect of the horizontal boundary layer is not significant for long fins such as those investigated by Sobhan et al. (1989). The above observations are also true for mild steel fin as shown in Fig. 5. The fin base temperatures at steady state is 379 K, which result in lower values of the heat transfer coefficient, compared to the aluminum fin. Because of the short fin height, the larger variation in the mild steel fin temperature compared to that of the aluminum fin (Sobhan, et al., 1989; Leong and Kooi, 1996) has negligible effect on the variation of the heat transfer coefficient with fin height.

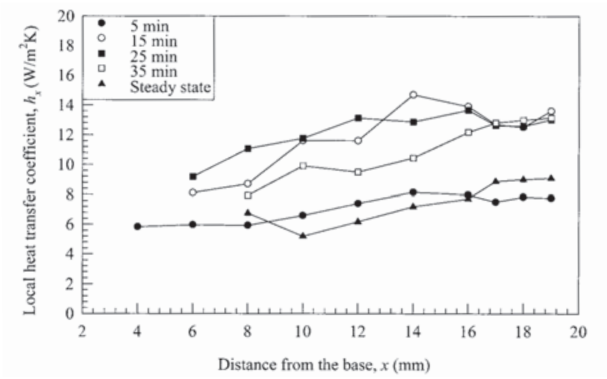


Fig. 4. Local heat transfer coefficients against the fin height at various times for an aluminum fin of height 20 mm.

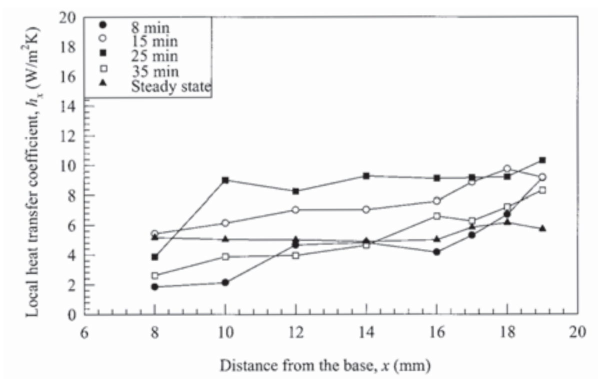


Fig. 5. Local heat transfer coefficients against the fin height at various times for a mild steel fin of height 20 mm.

5. Conclusions

The use of holographic interferometry enables the dynamic and changing temperature field of transient heating of fins to be observed. In the case of vertical fins with a horizontal base, alternating and oscillatory buoyancy-driven flows similar to those over the top surface of heated horizontal plates are present. Coupled with the presence of a horizontal boundary layer which extends almost to the entire fin height in the case of short fins, the variation of local heat transfer coefficient along the fin is negligible compared to longer fins. The assumption of uniform heat transfer coefficient commonly employed in fin analysis is therefore applicable to short vertical fins. The effect of fin base heating also significantly reduces the steady-state heat transfer coefficients of short vertical fins compared to the transient values. Hence, the use of steady-state solutions for the design of short vertical fins operating in transient conditions may not introduce as much error as was previously thought.

Acknowledgments

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