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Investigating convective heat transfer with an iron and a hairdryer

Manuel I González and Jesús H Lucio

Departamento de Física, Universidad de Burgos, Avda. Cantabria E-09006 Burgos, Spain

E-mail: miglez@ubu.es

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Abstract

A simple experimental set-up to study free and forced convection in undergraduate physics laboratories is presented. The flat plate of a domestic iron has been chosen as the hot surface, and a hairdryer is used to generate an air stream around the plate. Several experiments are proposed and typical numerical results are reported. An analysis and discussion of the results can be useful even for students at the most elementary levels; for higher levels, comparisons between the measured heat transfer coefficients and the well-established empirical correlations are included, showing good agreement.

(Some figures in this article are in colour only in the electronic version)

1. Introduction

Heat transfer is a class of physical phenomena whose interest in everyday life cannot be ignored. The machinery we use, the clothes we wear and the homes we live in must usually be designed or chosen in order to optimize heat transfer between them and their surroundings. Unequal attention is commonly paid to the three classical heat transfer mechanisms (conduction, convection and radiation) in first-year undergraduate textbooks [1, 2]: conduction is the most widely explained, with the statement of the basic Fourier law and the definition of quantitative parameters such as thermal conductivity and thermal resistance; radiation often includes the Stefan law, at the very least, and the definition of emissivity; finally, convection is usually reduced to a very short qualitative outline.

The complexity of the laws that govern convective heat transfer is the main reason for reducing the depth of its treatment, despite the fact that convection is the predominant mechanism in thermal exchange involving electronic devices, the human body, buildings, etc. In this sense, the existing gap between the very elementary physics textbooks and the more specialized engineering literature should act as an incentive for including quantitative topics on convection in the laboratory. We consider this to be especially true in the context of physics courses intended for engineering students.

This is the aim of the low-cost experimental set-up described in this work. Two very common domestic electrical appliances, namely an iron and a hairdryer, have been chosen as the sources of convection with the help of which several experiments concerning natural and forced convection are performed. Although experimental precautions are less than those involved in the research literature [3, 4], the reported results allow us to reach a number of qualitative and quantitative conclusions that can be adapted to students at various levels. With the non-elementary levels in mind, comparisons are drawn between the results and some commonly used experimental correlations. On the other hand, the experimental data processing also helps students to exercise their basic knowledge of radiative heat transfer.

2. Theoretical outline

The rate of convective heat transfer between a surface and a surrounding fluid is given by

$$q_c = \bar{h}_c A(T - T_\infty), \tag{1}$$

where q_c is expressed in W, A is the surface area, T is its temperature and T_∞ is the fluid temperature far from the surface. h_c is the convective heat transfer coefficient (W m^{-2} K^{-1}), the local value of which may change from one point of the surface to another. In practical applications, however, the average heat transfer is usually of more interest. It is denoted by the bar and defined precisely by (1). As a first approximation, $\bar{h}_c$ is independent of the dissipated q_c. However, as will be seen below, deviations from this ideal behaviour occur.

Natural or free convection occurs when the fluid around the surface moves only due to differences in density caused by thermal gradients. When a fan or a pump causes a relative motion between the fluid and the surface, the process is called forced convection. On the other hand, flow around the surface can be either laminar (ordered) or turbulent (chaotic). Generally speaking, the values of h_c in forced convection are higher than those under similar conditions of free convection. Similarly, turbulent heat transfer is usually faster than laminar heat transfer.

The equations governing free and forced convection seldom permit an analytical solution (especially when the flow is turbulent) except for a number of simple cases. Instead, the use of experimental correlations is a common practice in engineering. The general form of these correlations can be expressed as [5]

$$\bar{h}_c = F(\text{dimensionless parameters, geometric shape, boundary conditions}). \tag{2}$$

These allow us to estimate the heat transfer coefficient for a particular problem on the basis of previously measured experimental data.

Under conditions of natural convection, the most common dimensionless parameters are the Prandtl (Pr) and Grashof (Gr) numbers. The former is defined by

$$\mathrm{Pr} = \frac{\nu}{\alpha}, \tag{3}$$

where ν is the kinematic viscosity of the fluid and α is its thermal diffusivity. The Grashof number is

$$\mathrm{Gr}_L = \frac{g\beta(T - T_\infty)L^3}{\nu^2}, \tag{4}$$

where g is the gravity, β is the thermal expansion coefficient of the fluid and L is some characteristic length of the surface.

The convective heat transfer coefficient itself is usually offered in the correlations (2) through a new dimensionless quantity: the Nusselt number, whose definition is

$$\mathrm{Nu}_L = \frac{\bar{h}_c L}{k}, \tag{5}$$

where k is the thermal conductivity of the fluid.

Under conditions of forced convection, it is customary to use the Reynolds number:

$$\mathrm{Re}_L = \frac{U_\infty L}{\nu}. \tag{6}$$

Here, U_∞ is the non-perturbed velocity of the fluid that moves around the surface.

In the above-mentioned dimensionless parameters, the fluid properties should be evaluated at the so-called film temperature, defined as the average of T and T_∞. The exception is β, which must be evaluated at T_∞ [5].

Physically, the Prandtl number measures the relative importance of momentum and energy transport in the fluid. The Grashof number represents the ratio of the buoyancy forces to the viscous forces, and the Reynolds number represents the ratio of the inertial to viscous forces. The transition between laminar and turbulent flow is governed by the Reynolds number in forced convection, whereas in natural convection the $\mathrm{Pr} \cdot \mathrm{Gr}_L$ product plays a similar role [6].

On the other hand, when the fluid is a gas, as will be in our case, the emission of radiation cannot be ignored. According to the well-known Stefan–Boltzmann law, the net radiative steady-state heat transfer between a surface at temperature T and an enclosure at temperature T_∞ is given by

$$q_r = \epsilon\sigma A\left(T^4 - T_\infty^4\right), \tag{7}$$

where σ is the Stefan–Boltzmann constant and ϵ is the surface emissivity. The latter expression can be rewritten in the same way as in (1) to define a radiative heat transfer coefficient in the following way:

$$h_r = \frac{\epsilon\sigma A\left(T^4 - T_\infty^4\right)}{T - T_\infty}. \tag{8}$$

This expression will be used to discriminate the thermal contributions of radiation in the experimental records from those whose origin is convective.

3. The experimental set-up

Figure 1 shows the proposed experimental set-up. The surface under study is the flat plate of a domestic iron, the area of which is $A = 195\ \mathrm{cm}^2$. The electric power supplied to the iron was controlled with the help of a variac transformer and was measured with a watt meter. Typical powers supplied in the experiments were always much lower than the nominal power of the iron, in order to keep the internal thermostat disabled. On the other hand, a domestic hairdryer was used to carry out the experiments in forced convection.

A conventional thermometer and a K-thermocouple surface probe were chosen to measure ambient and plate temperatures, respectively. Wind speeds were measured with a rotary vane anemometer. Typical accuracies were 0.5 W for the watt meter, 3% for the anemometer and a few tenths of a degree for the thermometers.

The iron and the hairdryer were clamped onto a vertical mast. The clamps allowed the orientation of the iron to be easily modified and the distance between the hairdryer and the iron to be adjusted, so as to control the speed of the air stream. The speed was also controlled by means of the built-in fan velocity switch. Heating of the air stream was always disabled.

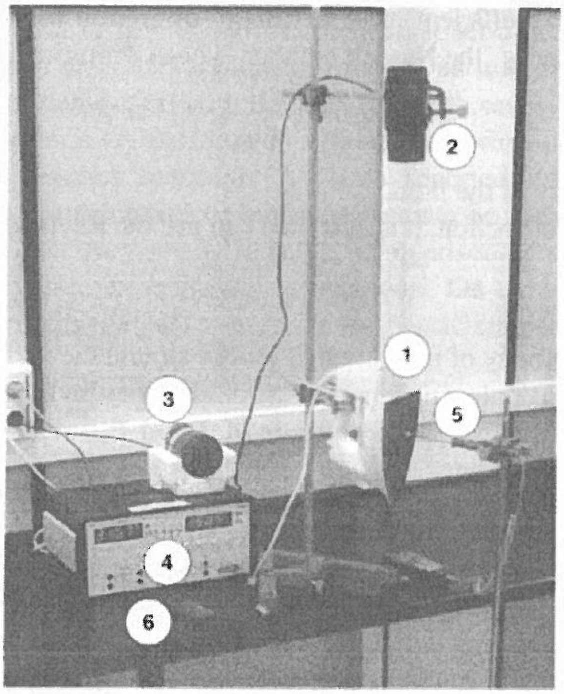

Figure 1. Scheme of the experimental set-up. (1) Iron, (2) hairdryer, (3) variac, (4) watt meter, (5) temperature probe, (6) anemometer.

Two preliminary tests were conducted with the above-described set-up. The first one has to do with the boundary conditions in (2). Two main classes of experimental correlations are available in the literature: those applicable in the case of uniform heat flux and those valid under conditions of uniform surface temperature. Although our experimental set-up cannot be used to assess the former hypothesis, the distribution of temperatures in the hot plate can be easily obtained in order to check the latter. To do so, a free convection experiment was carried out, with the hot plate facing upwards and an electric supply of 41 W. The plate was divided into 25 portions of approximately equal area and the surface temperature in the centre of each was measured when the steady state was reached. The results were used to draw up the contour plot shown in figure 2. The minimum and the maximum temperatures were 94.2 and 104.7 °C, respectively; the average computed temperature was 100.5 °C. As may be seen, the differences in temperature from one region to another are moderate; the hypothesis of uniform temperature, corresponding to neglecting the ±5 °C variations, will result in an error of 10–12% in the results below, which is acceptable within the scope of the present paper. Moreover, the results in figure 2 helped us to decide where the surface probe should be placed in subsequent experiments, in order to ensure (at least to some extent) that the recorded temperature was the mean plate temperature.

The second test is a distance-air stream velocity calibration, which was carried out with the anemometer for distances ranging from 20 to 100 cm. The results are depicted in figure 3. The two graphs correspond to the two available fan velocities. The calibration curves used in the subsequent forced convection experiments are also displayed.

4. Natural convection experiments: horizontal plate

The first experiment to be reported consists of measuring the heat transfer coefficient for the upward-facing hot plate, subjected to free convection. Electric powers q_e ranging from

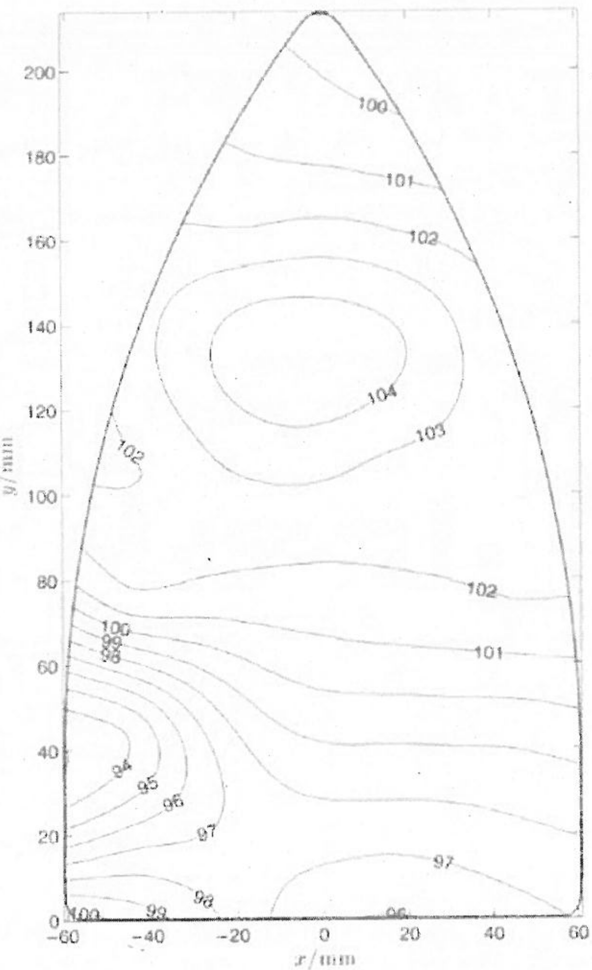

Figure 2. Contour plot for the temperature field in the hot plate (in °C). The conditions are natural convection, hot plate facing upwards and 41 W supplied power.

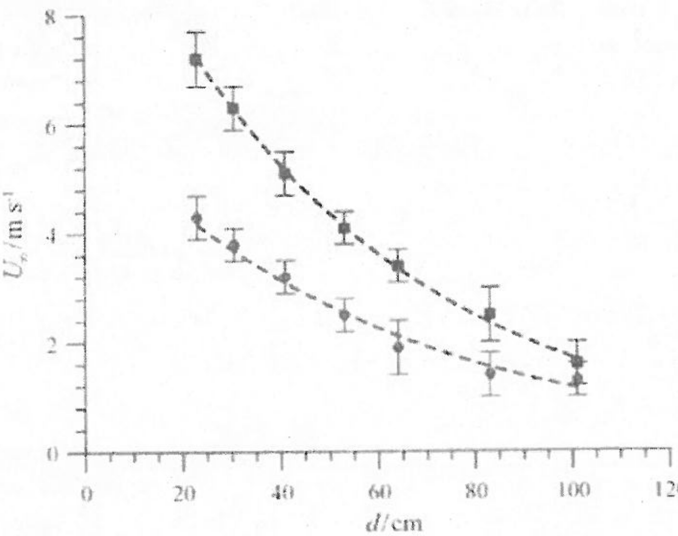

Figure 3. The air stream velocity as a function of the distance to the hairdryer mouth. Lower data set: fan velocity I, upper data set: fan velocity II. Dashed lines: exponential calibration curves, $U_\infty^{(I)}(d) = 6.1 \times 10^{-0.017 \cdot d}$, $U_\infty^{(II)}(d) = 11.2 \times 10^{-0.019 \cdot d}$.

20 W to 50 W were supplied to the iron. For every value of q_e and when the steady state was attained, plate and ambient temperatures were recorded. T ranged from 65 to 115 °C, whereas T_∞ was about 20 °C.

The recorded data were insufficient to work out the heat transfer coefficients, since we cannot ensure that every Watt supplied to the iron will be dissipated from its plate. Instead, we may assume that a fraction f of q_e corresponds to the plate, and the remainder is dissipated through the iron shell:

$$q_T^{(\text{plate})} = f\, q_e, \tag{9}$$

where the subscript T stands for 'Total' and includes combined radiation and convection. The value of f is unknown for the moment; as a credible hypothesis we will assume that f remains constant as q_e increases (this, of course, cannot be ensured), and will be handled as a parameter. To find it, we can use the following correlation [6], which is valid for flat hot surfaces facing upwards and laminar flow:

$$\text{Nu}_L = 0.54\,(\text{Pr}\cdot\text{Gr}_L)^{1/4}, \tag{10}$$

where the characteristic length L is the surface area divided by its perimeter. Other correlations apply for the turbulent flow regime ($\text{Pr}\cdot\text{Gr}_L > 2\times10^7$), but these are not of direct interest here, as the product $\text{Pr}\cdot\text{Gr}_L$ is about 2×10^5 in our experiments.

The proposed data-processing technique consists of selecting any trial value for f, then calculating, for every value of q_e, the global (convection + radiation) heat transfer in the following manner:

$$\bar{h}_T = \frac{q_T^{(\text{plate})}}{A(T - T_\infty)}, \tag{11}$$

and then subtracting the radiative heat transfer coefficient (8) from $\bar{h}_T$. The emissivity was set to $\epsilon = 0.9$, as the hot plate was covered with a dark grey paint [7]. If these calculations are performed on a conventional worksheet, it is very easy to modify the initial value of f until the best agreement is achieved between the set of computed heat transfer coefficients and the correlation (10). This procedure results in an optimum $f = 0.70$ (see the upper data set in figure 4), which means that 70% of the supplied power is dissipated from the plate, whereas the remaining 30% is dissipated through the iron shell. Although the correlation (10) has been used as a part of the fitting procedure, the goodness of the fit can be considered at least as a partial validation of the correlation itself.

Incidentally, the computed radiative heat transfer coefficients range between 6.5 and 8.2 W m^{-2} K^{-1}, and are similar in magnitude to the convective ones, meaning that both mechanisms are of near equal relevance under our experimental conditions. This is also true for the other natural convection experiments reported below.

Similar measurements have been performed with the iron plate facing downwards. In this case, a suitable correlation is [6]

$$\text{Nu}_L = 0.27\,(\text{Pr}\cdot\text{Gr}_L)^{1/4}\quad (\text{laminar flow, } \text{Pr}\cdot\text{Gr}_L < 3\times10^{10}). \tag{12}$$

A comparison between the coefficients in (10) and (12) suggests that the heat transfer coefficients must be lower in the latter case. This fact can be easily interpreted even by the students of more elementary levels: when the plate faces downwards, its own presence acts as a barrier for ascent of the air layers heated in contact with it. This implies that the plate temperature must be higher in this case and, in fact, the measured values ranged from 70 to 125 °C for the same supplied powers as in the case when the plate pointed up. The students could become surprised with the fact that a considerable drop of the heat transfer coefficient results only in slightly higher plate temperatures. To explain this, one must take into account

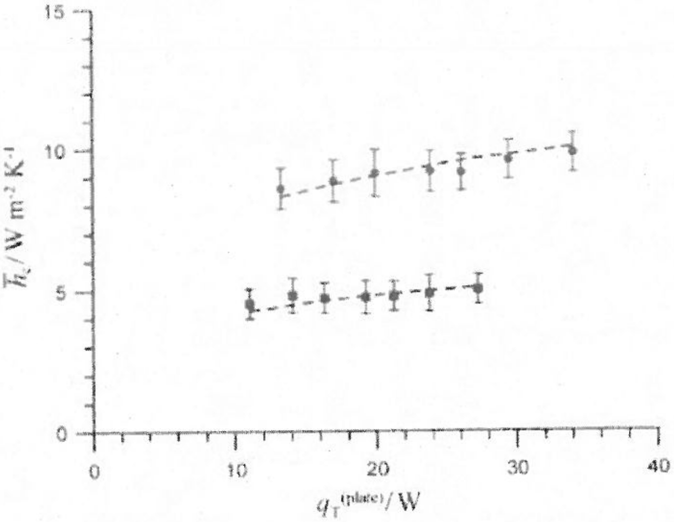

Figure 4. Convective heat transfer coefficient for the iron plate. Symbols: experimental values, dashed lines: fitted correlations. Upper data set: plate facing upward and $f = 70\%$ fit, lower data set: plate facing downward and $f = 55\%$ fit.

that the new position of the iron greatly favours the dissipation of heat through its (upward facing) shell and restricts dissipation from the plate. In other words, the percentage of heat dissipated by the plate must be much lower when the plate faces downwards. Indeed the best agreement between our new data and (12) is found when $f = 0.55$, which means that nearly half the heat dissipation corresponds to the shell.

As can be seen from figure 4 the convective heat transfer coefficient is a slightly increasing function of both dissipated power and plate temperature, although the advanced heat transfer textbooks do not offer a simple physical explanation of this fact. Nevertheless, error bars in figure 4 can be also interpreted as compatible with the hypothesis of constant $\bar{h}_c$. As we see, $\bar{h}_c \approx 9$ W m^{-2} K^{-1} for the plate facing upwards and $\bar{h}_c \approx 4.6$ W m^{-2} K^{-1} for the plate facing downwards. The order of magnitude of both values is consistent with typical natural convection coefficients reported for air and moderate temperature differences [6]. In this context, specifying a constant heat transfer coefficient could be enough for elementary level students, and is also a common practice in engineering. For instance, technical data sheets for heatsinks in electronics are usually characterized by a nominal thermal resistance[1] (an indirect way of expressing $\bar{h}$) which is considered as a constant within the range of operating conditions to which the heatsink will probably be subjected.

5. Natural convection: vertical plate

The same kind of test as performed in the previous section was carried out with the hot plate in a vertical position (see figure 1). The correlation to be used has been extracted from [5]. It is slightly more complicated than that previously used, and includes a coefficient that depends on the exact shape of the surface. The details of the calculation may well be of no interest to the students and lead to

$$\mathrm{Nu}_L \approx 0.55\,(\mathrm{Pr}\cdot\mathrm{Gr}_L)^{1/4}. \qquad (13)$$

[1] See, for example, a catalogue of heatsinks in http://www.aavidthermalloy.com/products/standard/index.shtml.

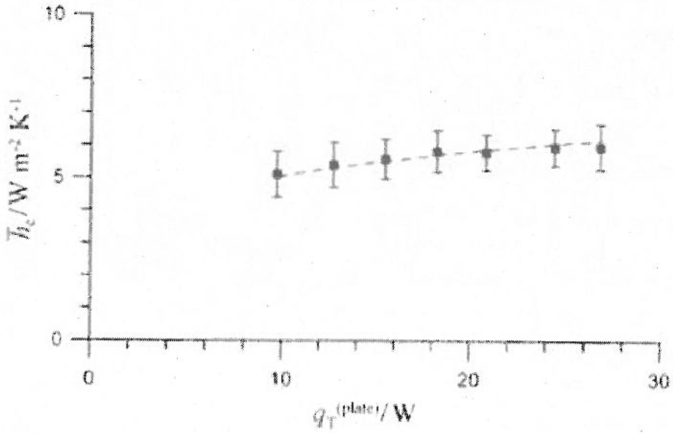

Figure 5. Convective heat transfer coefficient for the iron plate in a vertical position. Symbols: experimental values, dashed line: best fit proportional to $(T - T_\infty)^{1/4}$.

where a purely laminar flow has been assumed again and L is now the maximum vertical dimension of the plate ($L = 21$ cm).

Figure 5 shows the obtained transmission coefficients. Their average value is $\bar{h}_c \approx 5.9$ W m^{-2} K^{-1}, which is lower (higher) than that obtained when the hot plate is facing upwards (downwards), as expected. As in the previous section, the fraction f has been estimated with the help of (13). The value obtained is $f = 0.54$.

The students may be surprised at such low values of the fraction f as they tend to consider the iron shell as a perfect thermal insulator. However, the teacher's role is to help them realize that the shell area is much larger (more than four times larger in the iron we used) than the plate area. In addition, the surface temperature in the vicinity of the plate is nearly as high as in the plate itself. The combination of these factors explains the large amount of heat that is dissipated from the shell. In this sense, one might suspect that when the plate was facing upwards the relatively high value of f could be considered as an exception instead of the general rule.

To avoid repetition, a comparison between the results in figure 5 and the correlation (13) is not shown. Instead, a slightly different approach to the data could be of interest. Combining equations (3)–(5) with (13), we obtain

$$\bar{h}_c = C\frac{k}{(\nu\alpha)^{1/4}}(T - T_\infty)^{1/4}, \tag{14}$$

where C is a constant. For common gases at atmospheric pressure, the quantity $k/(\nu\alpha)^{1/4}$ is nearly constant for a wide range of film temperatures. For example $k/(\nu\alpha)^{1/4} \approx 5.8$ (SI units) for air. This implies that the heat transfer coefficient must be roughly proportional to $(T - T_\infty)^{1/4}$, a fact that has been previously proved by Spuller and Cobb in an undergraduate laboratory experiment [8]. Indeed, the data in figure 5 clearly show this trend. The same dependence with the temperature difference can also be verified for the data in figure 4.

6. Forced convection tests

Several experiments may be performed by directing the air stream thrown out by the hairdryer towards the iron plate. The two elements were arranged as shown in figure 1, with the plate in a vertical position and parallel to the air stream. The straight edge of the plate was chosen as the leading edge.

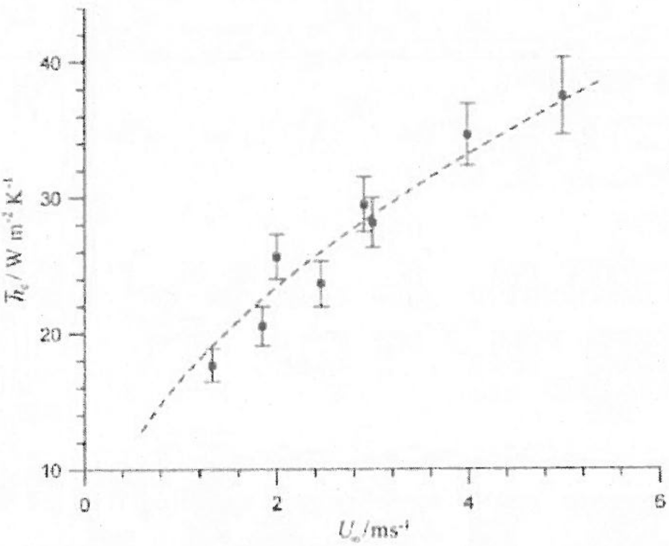

Figure 6. Symbols: forced convection heat transfer coefficients versus air stream velocity under conditions of constant supplied power, assuming $f = 0.54$. Dashed line: best $U_\infty^{1/2}$ fit.

The classical boundary-layer theory [9] for forced convection around a flat plate establishes that the flow regime is laminar near the leading edge, and turbulent at a longer distance, with an intermediate transition zone. This zone occurs at a distance x from the leading edge for which $\mathrm{Re}_x \approx 3 \times 10^5$–$5 \times 10^5$. Under the experimental conditions reported below, the Reynolds number ranges between 2×10^4 and 7×10^4 at the farthest points on the plate. Consequently, a fully laminar flow will be assumed. Under such conditions, the (exact) correlation for a rectangular plate is

$$\mathrm{Nu}_L = 0.664\,\mathrm{Pr}^{1/3}\,\mathrm{Re}_L^{1/2}. \qquad (15)$$

As our plate is not rectangular in shape, the maximum plate length L should be replaced with another (unknown) characteristic length L_{ch} to apply equation (15). Nevertheless, it could be of more interest to investigate its functional dependence with the air stream velocity. Combining equations (5), (6) and (15), we get

$$\bar{h}_c = 0.664\,\mathrm{Pr}^{1/3}\,\frac{k}{\nu^{1/2}}\,\frac{1}{L_{ch}^{1/2}}\,U_\infty^{1/2}. \qquad (16)$$

The Prandtl number for air is nearly constant within the range of temperatures under consideration (0.71 at 30 °C, 0.70 at 80 °C) [6]. The same is true for $k/\nu^{1/2}$ (6.62 SI units at 30 °C, 6.59 at 80 °C), because viscosity and thermal conductivity in a gas are governed by similar microscopic mechanisms. Then we can expect $\bar{h}_c$ to be roughly proportional to $U_\infty^{1/2}$.

To check this result, a first experiment was designed in which the supplied power was fixed ($q_e = 47.5$ W) and the distance between the plate and the hairdryer was varied between 40 and 90 cm, i.e. U_∞ ranged from 1.3 to 5.0 m s^{-1}. Shorter distances and higher velocities were avoided, to keep air velocities around the plate uniform to within 10–15%. The measured steady-state plate temperatures ranged from 55 to 76 °C, which clearly shows that forced convection heat transfer coefficients are larger than those in natural convection.

Figure 6 shows the computed values, together with their best $U_\infty^{1/2}$ fit. In spite of the experimental conditions being somewhat rudimentary, the goodness of the fit can be considered as acceptable; therefore, our measurements are compatible with the expected $U_\infty^{1/2}$

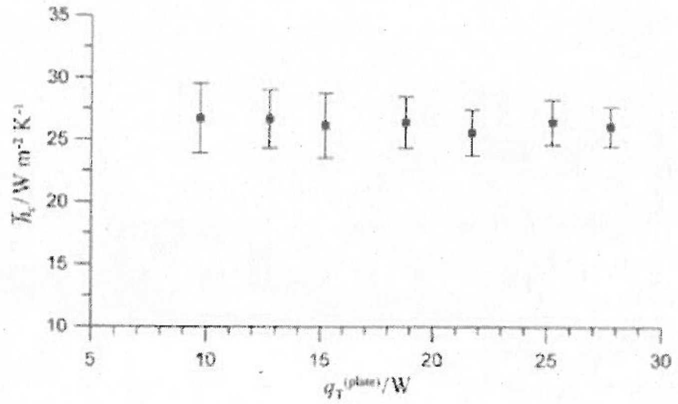

Figure 7. Symbols: forced convection heat transfer coefficients versus dissipated power under conditions of constant air velocity, assuming $f = 0.54$.

proportionality. It must be noted that the coefficients shown in figure 6 have been calculated assuming the same f as in the previous section; nevertheless, the goodness of the fit is essentially the same if other reasonable values are assumed.

The comments made below equation (16) also suggest that, for a fixed fluid velocity, the heat transfer coefficient approaches its ideal constant behaviour to a greater extent than in the previous tests. To verify this fact experimentally, the air stream velocity was fixed at 2.5 m s^{-1} and the supplied power was varied between 15 and 52 W. Plate temperatures ranged from 37 to 67 °C. Computed heat transfer coefficients, assuming $f = 0.54$, are displayed in figure 7, showing a nearly constant $\bar{h}_c \approx 26.3$ W m^{-2} K^{-1}. The same trend is observed assuming other values of f.

7. Conclusions

The proposed low-cost experimental set-up can be useful for revealing some qualitative and quantitative features of free and forced convective heat transfer. A number of experiments can be conducted and their results can be easily adapted to students at various levels. Calculations also include radiative heat transfer, since the plate heat emission is mixed.

The free convection experiments described here were used to check the most popular correlations in the laminar flow case: in particular three plate orientations were considered: upward, downward and in a vertical position. The correlations also suggest the fraction of supplied electrical power which is dissipated through the plate itself: about 70% with the plate facing upwards and about 55% in the other two positions. Arguments have been put forward to explain these apparently low values. On the other hand, it has been shown that the average heat transfer coefficient is roughly proportional to the fourth root of the temperature difference between the plate and its surroundings.

Regarding forced convection, several features concerning heat transfer coefficients have been verified: (i) their values are clearly higher than those under natural convection conditions, (ii) they are approximately proportional to the square root of the air velocity and (iii) they are nearly independent of the dissipated power for a fixed air velocity.

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