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Apparatus and techniques for study of unsteady thermal/fluid-induced loads on electronic components

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Jiang, Jiaqin, M.S.

San Jose State University, 1993

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**APPARATUS AND TECHNIQUES FOR STUDY OF
UNSTEADY THERMAL/FLUID-INDUCED LOADS
ON ELECTRONIC COMPONENTS**

A Thesis

Presented to

The Faculty of the Department of Mechanical Engineering

San Jose State University

In Partial Fulfillment

of the Requirements for the Degree

Master of Science

By

Jiaqin Jiang

May, 1993

APPROVED FOR THE DEPARTMENT
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ABSTRACT

APPARATUS AND TECHNIQUES FOR STUDY OF UNSTEADY THERMAL/FLUID-INDUCED LOADS ON ELECTRONIC COMPONENTS

by Jiaqin Jiang

This thesis focused on the development of apparatus and techniques for study of unsteady thermal/fluid-induced loads on a hot, wall-mounted resistor as a typical electronic component. Much work was done on the design and assembly of experimental apparatus used for this study. The time-dependence of the flow, the temperature-dependence of flow properties were all considered in this study. The study was accomplished by conducting flow visualization in the low speed wind tunnel which housed a resistor model. The results showed that this wind tunnel can be used for the testing over the flow speed and temperature range that was typical of electronic components in a forced convection environment. The flow pattern was successfully visualized using the smoke-wire technique, and videotape and pictures were taken to record the information. It was concluded that for the flow speed and temperature range studied, the overall flow pattern did not change significantly with heating.

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NOMENCLATURE

d	mesh diameter;
D	diameter of the cylinder;
f	vortex shedding frequency;
g	acceleration of gravity;
Gr	Grashof number, $Gr = \frac{g\beta(T_w - T_\infty)D^3}{\nu^2}$;
K	coefficient of pressure loss for grid, honeycomb and screen;
L	length of cylinder;
L/D	aspect ratio of cylinder;
M	mesh length for grid and screen;
q	dynamic pressure of the air flow, $\rho \frac{U_\infty^2}{2}$;
Q	flow rate;
Re_d	Reynolds number for mesh or wire, $Re_d = \frac{U_\infty d}{\nu}$;
Re_{d1}	characteristic Reynolds number for screen, $Re_{d1} = \frac{U_\infty d}{(1-S)\nu}$;
Re	Reynolds number for cylinder, $Re = \frac{UD}{\nu}$;
Ri	Richardson number, $Ri = \frac{Gr}{Re^2}$;
S	solidity for grid or screen, $S = 1-\beta$;

NOMENCLATURE (Continued)

St	Strouhal number, $St = \frac{fD}{U_\infty}$;
T_w	temperature of cylinder surface;
T_∞	temperature of free stream;
U_l	characteristic velocity at the center of screen cells, $U_l = \frac{U_\infty}{1-S}$;
U_∞	free stream velocity.

Greek symbols

β	perforated ratio for grid or screen, or thermal coefficient of expansion;
ν	kinematic viscosity;
ρ	air density;
Δp	pressure loss;
ΔT	temperature difference, $\Delta T = T_w - T_\infty$;
Σ	summation.

I. INTRODUCTION

1. BACKGROUND

Flow-induced loads have received much attention over the past few decades. As a matter of fact, significant flow-induced loads act on all kind of bluff bodies in which the flow separates from a large section of the body surface. Fluid flow around bluff bodies can cause destructive consequences as well as useful ones. Most heat exchanger interior components are bluff bodies, whose primary purpose is not to gain lift or minimize drag like aircraft components, but rather to contain flow and provide heat transfer surface. These structures are therefore not aerodynamically optimized. Flow-induced loads are often neglected, or considered as a secondary design parameter, at least until a failure occurs. The present study examines flow-induced loads on electronic components.

One reason that flow-induced loads are becoming more important is that designers are using materials to their limits, causing structures to become progressively lighter and more flexible [Kraus and Bar-Cohen 1983]. This is the trend in design of electronic components, e.g., resistors, capacitors, or integrated circuits mounted on printed wiring boards (PWBs). As the electronic components are lightened to reduce cost, flow-induced loads around them may eventually lead to failure of the components or their electrical connections. As a result, the present study aims to provide better understanding of the flow-induced loads around electronic components.

Electronic components are characterized by their small, discrete geometry and, of course, large heat dissipation. Within a forced convection environment, they are further characterized by low-speed airflow (in the range of 0.5 *m/s* to 5 *m/s*) and complex flow patterns due to vortex shedding from component edges. Due to the three-dimensional nature of most electronic components, flow separation behind them often leads to instability of the flow itself. A further point worth stressing is that most components are wall-mounted [Incropera 1988]. The combination of the above means that the flow patterns around different components depend on many parameters (e.g., component shape and dimensions, heating, air speed, etc.)

Different approaches have been developed for the study of flow-induced loads in general: numerical solutions and test model experiments. While some numerical solutions for the flow field have been achieved using digital computers, those solutions have generally been limited to either simple shapes away from a wall at low Reynolds number (i.e., $Re < 1000$) where the flow is assumed laminar, or to solutions where viscosity has been neglected. Moreover, few studies have considered the effect of the heating of the body. Although these numerical solutions provide benchmarks for real flow behavior, they cannot yield information of quantitative relevance to engineering applications. As Sarpkaya [1992] summarized in his recent review paper, separated flows in general are time-dependent (i.e., unsteady) flows and provide fertile territory for fundamental research in fluid dynamics. For most practical cases, unsteady effects may alter the turbulent flow, and since fluid properties vary the problem becomes even more complex. Under these conditions, *the equation of conservation of mass, the Navier-Stokes equations and the energy equation*, are fully coupled [White 1990],

making the problem extremely difficult to solve numerically. Therefore, an experimental approach seems more realistic, suitable, and inexpensive.

An experimental study of flow-induced loads requires special apparatus to accommodate the necessary measurements. The development of such a facility, designed specifically for this study, became a primary objective of the present work. To provide specific design goals, parameters for the flow and models to be used are identified next.

2. TEST PARAMETERS FOR THIS STUDY

The present study is concerned with one of the typical electronic components -- a circular cylinder-shaped resistor subjected to forced-air cooling. The parameters for this type of electronic component in general can be summarized below:

- ♦ Air speed in forced convection mode around electronic components ranges from 0.5 m/s to 5 m/s .
- ♦ The geometric length ranges from about 2 mm to 20 mm for most electronic components.
- ♦ All electronic components should be treated as heat sources *per se*, which usually operate at temperatures about $80 \text{ }^{\circ}\text{C}$ higher than ambient air:
 $T_w = 100 \text{ }^{\circ}\text{C}$, $T_{\infty} = 20 \text{ }^{\circ}\text{C}$.
- ♦ The heat flux removable by forced air can vary from 0.1 to 0.5 W/cm^2 , with a maximum value as high as 2 W/cm^2 [Kraus and Bar-Cohen 1983].

A circular cylinder model was chosen to simulate a discrete resistor mounted on a PWB for the present study (Figure I-1). Its parameters are summarized below in Table I-1, in which fluid properties were evaluated at a temperature of 20 °C.

Table I-1 Parameters for cylinder model

D (mm)	L/D	U_{∞} (m/s)	ΔT (°C)	Re	Gr	Ri
5	2~5	0.5~5	80	167~ 1667	1500	0.00054~ 0.054

Figure I-1 Parameters for the test model and test section
(Scale distorted to show model geometry)

3. REVIEW OF PREVIOUS STUDIES

The flow pattern behind an infinitely long unheated cylinder located away from the wall has been known since Theodore von Kármán [1911] published his classic paper illuminating the phenomena now known as the "Kármán vortex street". The parameters shown above in the Table fall into the Reynolds number range in which the boundary layers undergo laminar separation and the Strouhal number of the vortex shedding, $St = fD / U_\infty$, remains at a constant value of 0.20. For our study, we expect the influence of cylinder heating and wall proximity to affect the flow pattern significantly as compared to that observed by Kármán for an infinitely long, unheated cylinder (i.e., assuming $T_w = T_\infty$).

A great amount of literature has reported the sensitivity of vortex shedding at low Reynolds numbers to small external perturbations. At present, the reasons for wake sensitivity are not fully understood. A relevant study by Merkin [1977] reported numerically the mixed convection boundary layer on a circular cylinder in a stream flowing vertically upwards by assuming $Gr \gg 1$ and $Re \gg 1$. It was found that heating the cylinder delays separation and can suppress it completely, if the cylinder is warmed enough (when $Ri = Gr/Re^2 \cong 1.76 \sim 1.78$). Jain and Lohar [1979] also numerically investigated the unsteady mixed convection from a isothermal horizontal circular cylinder by solving the coupled principle equations for two cases [1) $Re = 100$, $Gr = 8 \times 10^4$ and 2) $Re = 200$, $Gr = 4 \times 10^5$], with surrounding fluid also flowing in the upward direction. Variation of heating and drag with time was obtained, and the total drag was found to increase due to the presence of the free convection. Another recent interesting discovery is due to Strykowski and Sreenivasan [1990]. They conducted an experiment leading to a

novel method of suppression of low-Reynolds-number ($Re = 46 \sim 120$) vortex shedding, by suitably placing in the wake of the vortex shedding cylinder a much smaller 'control' cylinder. The heating in air of the control cylinder (up to 400°C) with direct current was found to dramatically widen the region of suppression. Leung *et al.* [1992] made measurements of the mean pressure distributions and Strouhal numbers on partially grooved cylinders with different groove geometries over a Reynolds number range of 2.0×10^4 to 1.3×10^5 . A progressive shift of the flow regime to a lower Reynolds number was observed with higher groove subtended angles and 75 degrees was found for optimum drag reduction. The effects of surface roughness were found to promote boundary-layer transition and significantly affected the subsequent flow development at Reynolds numbers higher than the subcritical regime. Those observations as well as many previous works verified the fact that global changes in the wake can be produced through only local modification (i.e., geometry, perturbation, etc.)

It is interesting to note, at this point, that only few of above mentioned studies provided the aspect ratio L/D for the cylinder(s) used in their study: Strykowski and Sreenivasan, $L/D = 10 \sim 60$; Leung *et al.*, $L/D = 4.3$. All the other studies had implicitly assumed $L/D = \infty$. As a result, the influence of finite aspect ratio is still unknown.

4. OBJECTIVE OF THIS STUDY

The objective of the present study is to develop the necessary experimental apparatus and techniques, and use them to examine the effect of heating on the flow around a wall-mounted, finite-length cylinder in a forced convection

environment. From flow visualization, the study will infer the nature of any flow-induced loads which may be anticipated and how these loads might depend on heating.

II. EXPERIMENTAL APPARATUS

1. THE WIND TUNNEL

1.1 OVERVIEW

Recently with the increasingly important role of electronic packaging in the electronic and computer industry, the wind tunnel has become a critical tool for electronic packaging experiments involving thermal/fluids testing. Presently there are at least 8 major companies in the United States using the wind tunnel as a tool for heat transfer or cooling analysis. With the present trend, it is predicted that wind tunnels will continue to be needed for 10 or more years. The present experiment was conducted in a wind tunnel especially designed for this project.

The wind tunnel was an open circuit or nonreturn tunnel, where the surrounding room acts as an air return. It was a low speed, small scale tunnel as opposed to those developed for sophisticated aerodynamic experiments. Put briefly, the wind tunnel was designed to provide high-quality, controllable flow for the test.

1.2 DESIGN CONSIDERATIONS

1.2.1 Test Section Requirements

In this particular design in which a flow visualization experiment was performed, it was necessary that the test section should provide visual access in order for the flow pattern to be 'seen' by photographic equipment. This was achieved by making one side of the test section of plexiglas.

1.2.2 Flow Quality Issues

The flow quality in a wind tunnel refers to the various characteristics used to quantify the quality of the air flow. The following criteria are conventionally used to characterize good flow quality.

Table II-1 Flow quality criteria

Uniformity of free stream:	$\pm 0.5\%$
Turbulence level:	0.25
Angularity/swirl:	0.5%
Temperature control:	0.2 °C
Temperature uniformity:	0.1 °C

This wind tunnel, with the above strict flow quality goals, used three kinds of flow conditioning manipulators: grid, honeycomb and screen. The following describes the theory used and the design process followed to select flow manipulators.

1.3 DESIGN OF THE WIND TUNNEL COMPONENTS

A schematic of the wind tunnel is shown in Figure II-1. The tunnel was of open circuit design and consisted of a blower, a transition adapter, a settling chamber, and the test section. The various tunnel components are discussed in the following sections.

Figure II-1 Schematic of the low speed wind tunnel

1.3.1 Blower

The tunnel was driven by a Buffalo, type BL Limited Load Single Inlet centrifugal fan, supplying a volume flow rate of 8490 *cfm* (ft^3/min) at 1/2 *in w. g.* SP (Static Pressure) with a maximum fan speed of 1370 *rpm* (in light of the maximum power supply of 5 *horsepower*). The exit duct from the fan was 24×17 *in*². This gave a maximum flow speed of 15.24 *m/s* in the outlet. This blower was chosen simply because of its ready availability and suitability for the required flow and pressure rise.

The rpm of the fan were controlled by means of a variable speed transmission.

Figure II-2 shows the measured fan rpm at 9 different gear settings available from the fan for the experiment.

Figure II-2 Fan rpm at different gear settings

1.3.2 Transition Adapter Duct

The available outlet cross-sectional area of the fan was $24 \times 17 \text{ in}^2$. A transition adapter was therefore designed to connect the blower with a follow-on settling duct having a cross sectional area of $20 \times 20 \text{ in}^2$. Four steel plates were welded together to construct this duct with a length of 18 in . It was almost a constant area duct (since the area ratio was only $\frac{24 \times 17}{20 \times 20} = 1.02$.)

1.3.3 Grid

At the blower outlet a layer of grid was inserted. The benefit of this arrangement was that it rapidly reduced the swirl and velocity shear from the fan exit. This upstream grid gave modest onset angularity for the downstream honeycomb. The grid used here was made of woven wire mesh.

The parameters of the grid used for $U_{\infty} = 5 \text{ m/s}$ are given in Figure II-3, quoted from Roach [1988].

Table II-2 Dimension of the grid (square-mesh array)

d (mm)	β	K	Re_d
0.914	0.615	0.69	305

Figure II-3 Grid in square-mesh array shape

1.3.4 Settling Duct and Pressure Drops

Turbulence in the test section was further reduced by the installation of honeycomb and two screens. Those flow conditioning parts are separated by the settling duct with a cross sectional area of $20 \times 20 \text{ in}^2$ and length of 48 in . By doing so, enough distance between the honeycomb-screen combination and the second screen is assured.

Honeycomb was installed at the end of the adapter duct. Just downstream of honeycomb the first screen was installed adjacently. Loehrke and Nagib [1972] reported that an important benefit can be derived from locating a fine-scale flow control device, such as a damping screen here, near the exit plane of a honeycomb

section. A honeycomb-screen combination produces a much lower exit turbulence level than a honeycomb alone since the large-scale jets exiting from the honeycomb cells are broken up into smaller scale eddies. Both screens and honeycombs reduce lateral and axial turbulence, due to an energy exchange as the turbulence tends toward isentropy downstream.

The flow is expected to become extremely steady and uniform. Unfortunately this amount of flow manipulation is obtained at the expense of a sizable pressure drop through the settling duct. Honeycomb has a comparatively small pressure drop; screens have a relatively large pressure drop in the flow direction, which reduces the higher velocities more than the lower, and thus promotes a more uniform axial velocity.

The pressure loss coefficient, K , is defined as the pressure loss Δp across the flow conditioning manipulator (i.e., grids, screens or honeycombs) divided by the mean flow dynamic pressure q :

$$K = \frac{\Delta p}{q} = \frac{\Delta p}{\rho \frac{U_{\infty}^2}{2}}$$

1.3.5 Honeycomb

As shown in Figure II-1, a honeycomb was installed at the end of the adapter duct. According to Rae and Pope [1984], the pressure loss for honeycomb with hexagonal-section cells is $K = 0.20$, with a length/diameter = 6.0 and equal tube areas (see Figure II-4).

Figure II-4 Honeycomb shape -- hexagonal section

1.3.6 Screens

DeVahl [1964] reported that the resistance coefficient for screens can be given as

$$K = K_0 + \frac{55.2}{Re_d}$$

where

$$K_0 = \left(\frac{1 - 0.95\beta}{0.95\beta} \right)^2$$

$$\beta = \frac{\text{Open area}}{\text{Total area}} = \left(1 - \frac{d}{M} \right)^2$$

d = wire diameter

M = mesh length

It is noted that the above equation also applies to the grids, which were actually considered as coarser screens. The loss coefficient $K = 0.69$ for the grid used was computed this way.

Derbunovich *et al.* [1981] recommended optimum wire screens for control of turbulence in wind tunnels. They found that the overall turbulence after the screen is a function of solidity $S (= 1 - \beta)$. The conditions at which transcritical flow

occurs past the screen were found to be dependent on the Reynolds number based on the local velocity U_l at the centers of the cells of the screen, rather than on the free-stream velocity U_∞ as is usually done.

Since the flow rate is constant throughout the screen, the relationship follows

$$U_l = \frac{U_\infty}{1-S} ,$$

so that the characteristic Reynolds number becomes

$$Re_{d1} = \frac{U_\infty d}{(1-S)\nu} .$$

Transcritical flow past the screen occurs at $Re_{d1} \geq Re_{cr} \cong 175$.

On the basis of their research, the following parameters were chosen from one of their experimental samples ($U_\infty = 5 \text{ m/s}$).

Table II-3 Dimensions of the screens

M (mm)	$Mesh$ (in ⁻¹)	d (mm)	β	K	Re_{d1}
1.4	18.1	0.31	0.61	1.06	169

A problem with screens is their well-known tendency to accumulate dust. The dust accumulation rate is usually nonuniform, resulting in constant change of the screen's porosity and pressure drop, which in turn changes the velocity, and flow angularity distribution in the test section in an arbitrary way with time. A filter

box was installed at the inlet of the blower to avoid this problem. This is explained in Section 1.5.

1.4 The Test Section

The test section of the wind tunnel consisted of 4 flat plates assembled to form a cross sectional area of $20 \times 20 \text{ in}^2$ (or 2.78 ft^2), with a length of 48 in . One of the side walls was made of transparent plexiglas, which made it possible to conduct flow visualization.

An enlarged-scale resistor model was mounted on the bottom surface inside the test section because it was readily available and also made flow visualization easier to see. The model was a commercially available cylindrical metal heater with a configuration of $D = 10 \text{ mm}$, $L = 40 \text{ mm}$. Compared to a full scale of $D = 5 \text{ mm}$, $L \cong 20 \text{ mm}$, the model was scaled up 2 times. To keep the Reynolds number matched, the upper bound air speed of 10 m/s typical of forced convection around electronic components should also be scaled down to 5 m/s , which meant a maximum flow rate of only 2734 cfm was needed from the fan.

1.5 The Filter Box

As mentioned above, the screen requires clean air to maintain its good performance. Air must pass through a dust/particle filter before entering the fan inlet. A filter box was constructed to provide an air-tight enclosure for a total of 6 filter elements. The filter box was made of plywood panels. The schematic view of the filter box is shown here in Figure II-5.

Figure II-5 Schematic view of the filter box with key dimensions

Room air enters into the filter box in three directions, and two elements in series are installed in each direction. The $24 \times 24 \text{ in}^2$ filter elements were commercially available, and were 2 inch and 12 inch thick, respectively. The dimensions for the filter elements are also shown here. Note that the last row gives the pressure drop at clean or loaded conditions.

Table II-4 Dimensions of the filter elements

	Pre-filter	Inner-filter
Nominal size (in)	24×24×2	24×24×12
Actual size (in)	$23\frac{3}{8} \times 23\frac{3}{8} \times 1\frac{3}{4}$	23.38×23.38×11.5
Rated flow Q (cfm)	2000	2000
Pressure drop Δp (in w. g.)	0.24/0.9	0.65/1.5

1.6 SYSTEM RESISTANCE OF THE WIND TUNNEL

Overall, the wind tunnel can be considered as a system consisting of filter box, blower, grid, transition duct, honeycomb-screen, settling chamber, second screen, and test section. Pressure drop occurred across the test section and various flow conditioning elements (including the filter box), while the fan gave necessary pressure rise to balance the pressure drop within the system. Also there was a pressure drop due to the filter box before the air was drawn into the blower. The pressure variation is illustrated in Figure II-6.

Figure II-6 Schematic view of the pressure variation
through the wind tunnel

Calculation of the pressure drop across various components, based on the maximum flowrate of 2734 *cfm* at $U_{\infty} = 5 \text{ m/s}$ follows.

1.6.1 Filtration

The filter was a sieve to prevent undesired particle from flowing through so that relatively "clean" air could be achieved. Based on the designed layout, the flow rate for each of the three filter elements was

$$q = \frac{Q}{3} = \frac{2734cfm}{3} = 911.333cfm$$

The pressure drop with the filters was considered to be linear with flowrate.

Referring to the parameters for the filter elements, the pressure drop across the filter box was

$$\Delta p_{filter} = 0.9in.water \times \frac{911.333cfm}{2000cfm} + 1.5in.water \times \frac{911.333cfm}{2000cfm} = 1.094in.water$$

with all filters in loaded conditions.

1.6.2 Grid

The loss factor was $K = 0.69$, thus the pressure drop across the grid was

$$\begin{aligned}\Delta p_{grid} &= K \times \frac{\rho}{2} \times \left(\frac{Q}{A}\right)^2 \\ &= 0.69 \times \frac{0.0023slug / ft^3}{2} \times \left(\frac{2734 ft^3 / min \times 1min / 60s}{2.83 ft^2}\right)^2 \\ &= 0.206lbf / ft^2 \\ &= 0.040in.water\end{aligned}$$

1.6.3 Honeycomb

For honeycomb with $K = 0.2$, the pressure drop was

$$\Delta p_{honeycomb} = K \times \frac{\rho}{2} \times \left(\frac{Q}{A}\right)^2$$

$$\begin{aligned}
&= 0.2 \times \frac{0.0023 \text{ slug / ft}^3}{2} \times \left(\frac{2734 \text{ ft}^3 / \text{min} \times 1 \text{ min} / 60 \text{ s}}{2.78 \text{ ft}^2} \right)^2 \\
&= 0.062 \text{ lbf / ft}^2 \\
&= 0.012 \text{ in. water}
\end{aligned}$$

1.6.4 Screens

The loss factor for each screen was $K = 1.06$; The pressure drop was therefore

$$\begin{aligned}
\Delta p_{\text{screen}} &= 2 \times K \times \frac{\rho}{2} \times \left(\frac{Q}{A} \right)^2 \\
&= 2 \times 1.06 \times \frac{0.0023 \text{ slug / ft}^3}{2} \times \left(\frac{2734 \text{ ft}^3 / \text{min} \times 1 \text{ min} / 60 \text{ s}}{2.78 \text{ ft}^2} \right)^2 \\
&= 0.655 \text{ lbf / ft}^2 \\
&= 0.126 \text{ in. water}
\end{aligned}$$

The total pressure drop was thus

$$\begin{aligned}
\Delta p &= \sum_{i=1}^4 \Delta p_i \\
&= \Delta p_{\text{filter}} + \Delta p_{\text{grid}} + \Delta p_{\text{honeycomb}} + \Delta p_{\text{screen}} \\
&= (1.094 + 0.040 + 0.012 + 0.126) \text{ in. water} \\
&= 1.272 \text{ in. water}
\end{aligned}$$

The following is the a tabular summary of spreadsheet analysis for pressure drops across each part at various flowrates.

Table II-5 Summary of pressure drops across wind tunnel components

Q (cfm)	Filter	Grid	Honeycomb	Screens	Total (in. w.g.)
0	0	0	0	0	0
500	0.2000	0.0029	0.0004	0.0137	0.2170
1000	0.4000	0.0077	0.0016	0.0316	0.4408
1500	0.6000	0.0144	0.0036	0.0536	0.6716
2000	0.8000	0.0231	0.0064	0.0799	0.9094
2500	1.0000	0.0338	0.0099	0.1103	1.1540
3000	1.2000	0.0463	0.0143	0.1449	1.4055

The total pressure drop Δp was thus obtained as a function of flow rate Q . These data are plotted in Figure II-7 along with the fan performance curves.

Figure II-7 Wind tunnel system resistance curve and fan performance curves

2. TESTING INSTRUMENTS

2.1 OVERVIEW

To perform flow visualization in the test section of the wind tunnel using the smoke-wire technique [Westphal 1982], several preparations were done beforehand. Optical access to the test section is needed for lighting the smoke sheet and photographing the result. The plexiglas surface was designed to allow the access of necessary lighting and to allow the camera to 'see' the smoke.

The conduct of the experiment seemed more successful at night when the lab was darkened. Since the smoke must be seen through the plexiglas, which would act as a mirror and thus made it impossible to photograph the smoke inside the duct, care was thus taken to properly position the two high-intensity spot lights (300W and 500W) to eliminate glare from reflections. Two of the interior surfaces of the test section (back and bottom) were blackened by covering them with black plastic sheets.

A schematic of the setup of the instruments around the test section is shown in Figure II-8 below. The camera was put on the tripod so that sharp pictures were assured.

Figure II-8 Schematic of flow visualization experiment setup

2.2 WIRE

Nichrome wire ($d = 0.004 \text{ in}$) was used because of its high resistance to breakage at extremely high temperatures. Theoretically, wire diameter must be small enough to avoid producing an unstable smoke streakline due to Reynolds number effects. Westphal [1982] pointed out that for Reynolds number based on diameter

(Re_d) less than about 20, the wire wake remains straight and stable. As long as $Re_d < 60$, coherent streaklines could be obtained. For our 0.004 *in* wire, this implies a limiting flow velocity below 10 *m/s*. The wire diameter should always be as large as possible to give a long smoke burn and less chance of breakage.

Oils were coated on the wire to generate smoke upon joule heating. It was generally expected to produce dense white smoke and have proper surface tension to give an appropriate streakline spacing. In this study, fog fluid was used. Coating of the wire was done very carefully, especially with high flow speed, because thick coatings increase the effective wire diameter (producing an unstable wire wake) and require higher wire temperature to vaporize the oil. Thumb and forefinger using were actually found to be quite reliable and simple to repeatedly brush the wire and wipe off excess oil.

The optimum DC voltage applied to the wire was found to be about 32 V. It heated the wire and thus vaporized the oil to generate smoke. A variable voltage supply was used to find this by trial and error and later on to adjust to slightly different conditions during the observation. Smoke duration is typically around 0.5 second at 4 *m/s*, while a much longer duration is possible for lower speeds (more than 1 sec at 2 *m/s*).

2.3 MANOMETER

In order to keep track of the flow speed for different running cases, Dwyer® manometer (microtector®) was used to measure the velocity of the air in the test section. According to Figure II-8, the pitot tube was clamped at the edge of the

test section end so that it was positioned in the center of the test section. The dynamic pressure measured by the pitot probe was used to calculate the flow speed.

2.4 THERMOCOUPLE

The purpose of using the thermocouple was to know the temperature of the resistor when it was heated. Since the thermocouple was a contact type, its end was simply clamped to the resistor.

2.5 CAMERA AND FILM

The camera was a Pentax with 55 mm lens. Kodak Tmax 400 professional black and white film was used to record the smoke; Fuji HG1600 color film was also tried.

2.6 VIDEO

A VHS video recorder was also used to record the moving smoke, which had 1/500 and 1/1000 shutter speed settings. It was found better motion pictures were taken with shutter off. The video recorder was also put on the tripod for better results.

III. RESULTS AND CONCLUSIONS

1. CONTROL OF PHOTOGRAPHIC TECHNIQUES

The smoke-wire technique described before was proved viable for airflow speed ranging from 2 *m/s* to 5 *m/s* in this study. Due to the short duration of smoke, video taping did not prove too useful (the videotape that recorded the smoke pattern was kept by the thesis advisor). Photographs seemed to provide much better resolution and as much information. Success in obtaining the photographs is attributed to the following items:

- a. Careful arrangement and positioning of the high-intensity spot lights to provide optimum lighting, while eliminating reflections from the surfaces.
- b. Good control of time delay between switching on the smoke generation circuit and triggering the camera.
- c. Proper photographic techniques and film processing.

2. RUNNING CASES OF THE EXPERIMENT

Flow visualization was performed for several different cases. As explained in Chapter I, this study was mainly concerned with the effect of heating on the flow pattern around a wall-mounted, finite-length cylinder in a forced convection environment. During the experiment, the resistor was gradually heated up from room temperature to 100 °C. Pictures were actually taken when the resistor temperature reached 60 °C, 80 °C and 100 °C in order to carefully examine any effect that may occur due to temperature variations. The same procedures were performed four times for various air flow speeds of 2~5 *m/s*. The table below

showed the matrix describing all cases performed for this study, with appropriate Re and Gr numbers.

Table III-1 Matrix of running cases of flow visualization

	$U_{\infty} = 2 \text{ m/s}$ ($Re = 667$)	$U_{\infty} = 3 \text{ m/s}$ ($Re = 1000$)	$U_{\infty} = 4 \text{ m/s}$ ($Re = 1334$)	$U_{\infty} = 5 \text{ m/s}$ ($Re = 1667$)
$\Delta T = 40 \text{ }^{\circ}\text{C}$	$Gr = 750$	$Gr = 750$	$Gr = 750$	$Gr = 750$
$\Delta T = 60 \text{ }^{\circ}\text{C}$	$Gr = 1125$	$Gr = 1125$	$Gr = 1125$	$Gr = 1125$
$\Delta T = 80 \text{ }^{\circ}\text{C}$	$Gr = 1500$	$Gr = 1500$	$Gr = 1500$	$Gr = 1500$

3. ANALYSIS OF RESULTS

For all cases, the smoke generated by the electric current in the Nichrome wire can be clearly seen. As seen in the two pictures in Figure III-1 and Figure III-2, the flow around the hot, wall-mounted resistor showed different pattern from the well-known, classic Kármán Vortex street. For $U_{\infty} = 2 \text{ m/s}$ ($Re = 667$), the separated flow was easily seen. As air speed gradually increased to 5 m/s , the flow near the resistor got more and more turbulent.

Besides the effect from the air speed increase, the effect of the resistor temperature increase, which was the main objective of the study, was examined. The resistor was heated from room temperature to as high as $100 \text{ }^{\circ}\text{C}$. To the naked eye, the flow pattern showed no significant difference from the unheated case. The turbulent pattern of the flow behind the resistor did not change much as the Grashof number increased to as high as 1500.

Figure III-1. Picture showing the flow pattern with resistor at
 $T_w = 80\text{ }^{\circ}\text{C}$ and $U_{\infty} = 2\text{ m/s}$)

Figure III-2 Picture showing the flow pattern with resistor at
 $T_w = 100\text{ }^{\circ}\text{C}$ and $U_{\infty} = 2\text{ m/s}$)

4. CONCLUSIONS AND RECOMMENDATIONS

Much effort was put into the design and setup of the wind tunnel, and the control of photographic techniques. All work was done in the Fluids Mechanics Laboratory in the School of Engineering. The low speed wind tunnel developed for this study was found to be suitable for thermal/fluids testing. The smoke-wire technique was successfully used to conduct the desired flow visualization.

The smoke was very clearly seen during the experiment. Complete comparisons for the resistor at different thermal and flow conditions were thus not difficult to make with simply the naked eye. Photographic techniques turned out to be so critical that pictures were taken several times to wind up with the best quality pictures.

The result obtained from the present study showed that temperature variation of the resistor as a typical electronic component did not appear to play as significant a role as mean flow speed in altering the flow pattern. Thus, it was the flow speed instead of the heating that controlled the flow pattern behind the resistor.

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