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Studying the Convective Heat Transfer from a Building Model with Infrared Camera Techniques

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Contributed by the Heat Transfer Division of The American Society of Mechanical Engineers for presentation at the Winter Annual Meeting, San Francisco, Calif., December 10-15, 1978. Manuscript received at ASME Headquarters August 14, 1978.

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, UNITED ENGINEERING CENTER, 345 EAST 47th STREET, NEW YORK, N.Y. 10017

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NOMENCLATURE

E	Voltage across foil heaters (volts)
E	Emissivity
F	Radiation shape factor = 1
h	Local convective heat transfer coefficient ($W/m^2 °C$)
$\bar{h}$	Average convective heat transfer coefficient ($W/m^2 °C$)
I	Current into foil heaters (amps)
K	Conductivity of aluminum
$\underline{L}$	Thickness of aluminum (cm)
$\overline{Nu}$	$\bar{h}L/K$ - Average Nusselt number
Re	$\bar{U}L/\nu$ Reynolds number
T_a	Ambient air temperature (°C)
T_r	Reference temperature (°C)
T_s	Model surface temperature (°C)
T_{sg}	Approximation of surface temperature (°C)
u	Wind velocity (m/sec)
$\underline{\nu}$	Kinematic viscosity of air
σ	Stafan-Boltzmann constant

INTRODUCTION

In these days of energy awareness, dwindling natural gas supplies, and high fuel prices, there is increased need for information on the effects of convective heat transfer from buildings for more effective designs and more efficient energy utilization.

There seems to be very limited information in the open literature on the convective heat transfer from buildings. The ASHRAE Handbook of Fundamentals (1) gives the basics of determining the heating load for structures according to the heat conduction of building materials and a convective heat transfer coefficient which was extrapolated from experiments of flow over a flat plate wall of a square duct. Clearly flow over a flat plate is a poor approximation for three-dimensional flow around buildings. Ito (3) reported on a field study of the convective heat transfer of a building in Tokyo using heat meters. The field study results did not agree with estimates based on the convectional ASHRAE relationships for convective heat transfer. Kelnhofer and Thomas (4) conducted wind tunnel experiments on convective heat transfer coefficients of a building model utilizing a square flat-roofed model for shear and uniform flow conditions with varying wind direction, as well as the effects of a single neighboring upstream model on the convective heat transfer coefficients. The model was instrumented with nine small isothermal heaters, one-half inch in diameter, located over one side of the model. Due to a lack of field data and restrictions in geometry, quantitative use of the data could be made but only with caution.

This wind tunnel study attempts to go beyond these earlier tests by using both a variety of model sizes and models entirely heated on all sides, rather than on small areas distributed over a single side.

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This wind tunnel study attempts to go beyond these earlier tests by using both a variety of model sizes and models entirely heated on all sides, rather than on small areas distributed over a single side.

The former situation more closely models the actual situation which occurs on full-size buildings.

The technique used for this study was the prime aspect under investigation. The objectives were to determine if the method would work, to see what refinements could be made, and to determine if the results were an accurate determination of the convective heat transfer coefficients. Thus, this was a preliminary study to see if the infrared-technique combined with a meteorological wind tunnel is a suitable method to investigate the convective heat transfer over various architectural shapes.

METHOD AND TEST APPARATUS

The method involved taking color pictures of the heated building models at varying orientations in an atmospheric boundary layer wind tunnel. Technical problems faced included modeling the flow, preparing the heated models, and operation of the infrared camera to obtain the desired pictures of the various building orientations.

Wind Tunnel and Shear Layer

The atmospheric flow conditions were modeled in the stratification wind tunnel at Colorado State University's Fluid Dynamics and Diffusion Laboratory. The transition section and the heaters were removed from the tunnel (see Fig. 1).

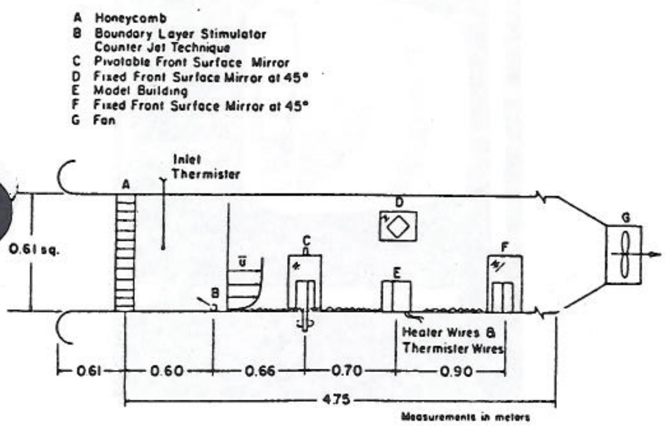

Fig.1 Model building arrangement in boundary-layer wind tunnel.

A short, low-contraction-ratio transition section replaced the section removed. The boundary layer was simulated by a power law velocity profile. The method used to develop this turbulent velocity layer was one of a counter-flow jet technique of H.M. Nagib (1974). Lead shot randomly scattered on the tunnel floor provided upstream surface roughness. A power law profile exponent of $n = 0.18$ was obtained. This power velocity profile was slightly altered when mirrors were placed in the tunnel, as explained later. The final power law profile was given by the exponent $n = 0.185$ (Fig. 2). All the experiments using the heated building models and infrared camera were run with this shear layer profile.

Heated Building Models

The heated building models were designed to meet several constraints. The model walls must supply uniform and constant heat flux on all sides and top. It is preferable if the surface radiation closely approximates grey body radiation behavior. A known temperature reference on the model walls is necessary to calculate the radiosity of the warm surface.

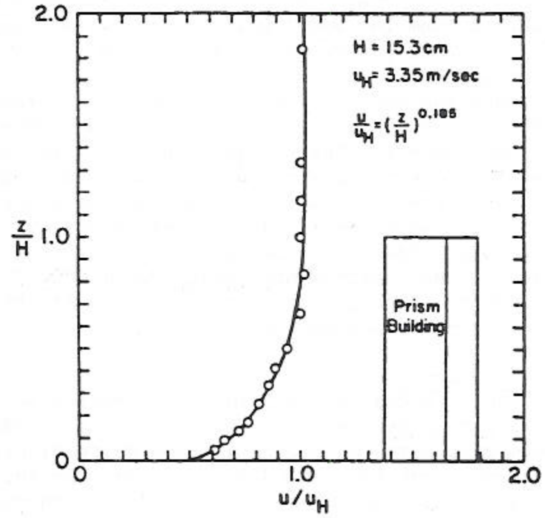

Fig.2 Dimensionless velocity profile (at model site with all mirrors in wind tunnel with back mirror at 45°, conterflow jets at 45° angle, flowrater at 100%, lead shot roughness elements 0.17 inches in diameter).

The models were constructed of 1/16-inch thick aluminum sheets to whose interior sides were attached heaters and thermistors. These wall sections were attached to an interior Plexiglas support with small screws. The heaters were Minco Thermofoil Heaters (Series HK-6040) with an adhesive backing. The standard sizes of the foil heaters were what essentially determined the building model dimensions. The thermistors were Yellow Springs Instrument Company precision thermistors YSI 44004 with a resistance of 2253 ohms at 25°C with an interchangeability of ±0.2°C. The locations of the interior thermistors are given in Fig. 3. The heaters were wired in parallel and the heater and thermistor leads were brought out of the bottom of the model. The model interior was then filled with fiberglass insulation to eliminate any interior convective currents.

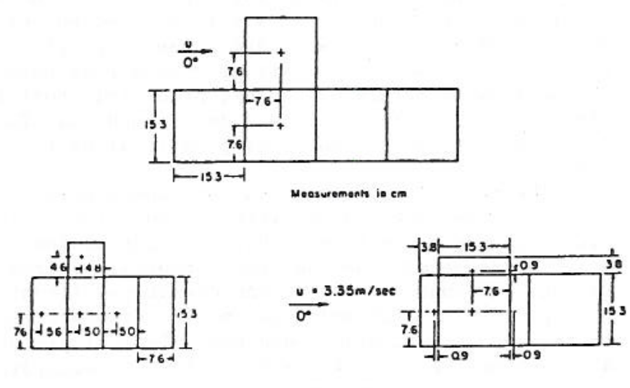

Fig.3 Thermistor locations at 0° orientation of models.

To approximate a grey body temperature source, the model was finished with 3M Company's Nextel Brand optically black coating which uniformly scatters incident light regardless of incident angle. The emissivity of this coating was taken to be approximately ε = 0.95. To reduce the possibility of heat being conducted from the model walls to the wind tunnel floor a rubber gasket material was used for the bottom of the models.

When the models were in operation the heater leads were connected to a Power Stat type variable A-C power supply. The voltages and currents into the models were measured with an amp-probe. The thermistor leads from the models and two ambient temperature thermistors were connected to a Yellow Springs Instrument Company Model 4002 Switchbox and then into a Yellow Springs Instrument Company Model 425C Telethermometer which gave direct indication of the temperatures in degrees Celsius.

Infrared Camera

The infrared camera techniques previously referred to consisted of an AGA Thermovision 680 Camera Unit with the 102B Black and White Display to which was connected the CM700 Color Monitor which displayed ten simultaneous color coded isothermal regions. Placed over the color monitor was a Polaroid camera. Polaroid type 108 color film recorded the ten isothermal regions which were displayed for the variously oriented heated building model walls.

The camera and monitors were placed outside of the wind tunnel throughout the experiment. The pictures were subsequently taken through a thin polyethelene sheet (Saran Wrap) stretched tightly over a movable side panel of the wind tunnel. This film appeared to be completely transparent to infrared radiation. In order to obtain pictures of the varying model positions (top, front, and rear being relative to the fixed wind tunnel) a system of mirrors was placed in the wind tunnel. Mirrors were placed 18 inches above the wind tunnel floor directly over the model site at a permanent 45 deg angle, downstream of the model site at a permanent 45 deg angle, and a mirror on a pivot was placed upstream of the model site (Fig. 1). The upstream mirror was pivoted remotely from a position parallel to the flow so as not to disturb the flow field, to a position of 45 deg to allow pictures to be taken through the mirror. When a picture was taken with the front mirror, the front mirror was quickly placed at a 45 deg angle while the camera was adjusted. After the camera was adjusted, the mirror was replaced in the parallel position for a period of time to compensate for the change in flow characteristics when the mirror was at the 45 deg angle and disturbing the flow. With the camera adjusted, the mirror was rapidly moved into position and a picture was taken. Several velocity profiles were taken with this pivoted mirror parallel to the flow to determine the extent of this position's influence on the flow. It was determined that for this study the pivot would not greatly alter the results.

The models were placed in the tunnel on a turntable and oriented to different positions. The pictures were taken from either the side of the tunnel next to the model (top and side pictures) or from the side of the tunnel upstream and downstream (front and rear pictures). The model was moved through orientations from zero to ninety degrees. Typical pictures are included in Figs. 4-7 and are labeled according to camera location and model orientation.

DATA REDUCTION AND RESULTS

Energy Balance

In the steady state situation the energy supplied to the model by electric heating may be equated to the heat that the model dissipated to its surroundings by

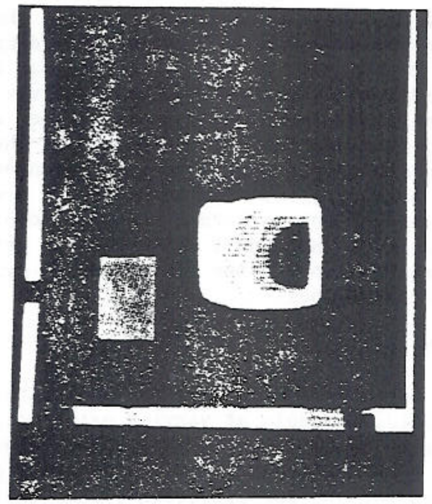

Fig.4 Color isotherms: 15.3 x 15.3 x 15.3 cm building, angle = 22.5°, front face.

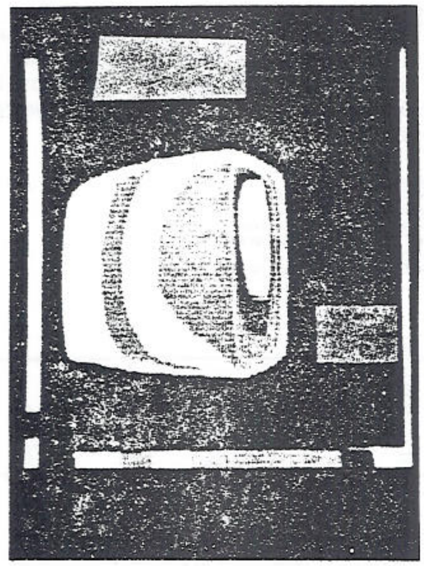

Fig.5 Color isotherms: 15.3 x 15.3 x 15.3 cm building, angle = 22.5°, side face.

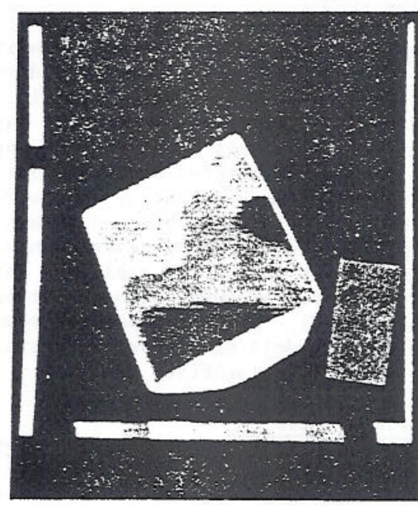

Fig.6 Color isotherms: 15.3 x 15.3 c 15.3 cm building, angle = 22.5°, top face.

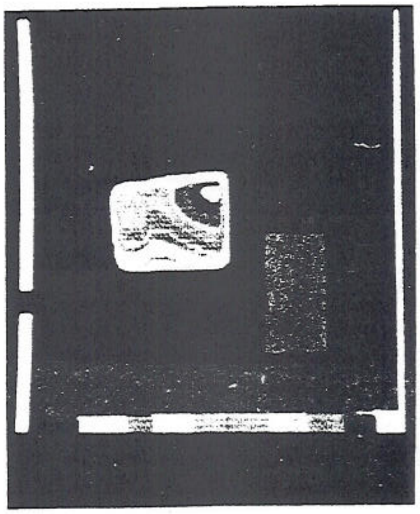

Fig.7 Color isotherms: 15.3 x 15.3 x 15. 3 cm building, angle = 22.5°, back face.

convection and radiation. The energy supplied to the models was the wattage applied by the Power Stat A-C power supply to the foil heaters. There were three modes of heat transfer out of the model. There was conduction of heat out of the power and thermistor leads which ran out of the bottom of the model. There was also conduction of heat from the model edges through the rubber gasket which served as the bottom of the model to the wind tunnel floor. Both these heat losses were taken to be small and neglected. The second route for heat to leave the model was by radiation to the surrounding tunnel walls. This loss was a function of the model emissivity and the fourth power of the model temperature. This loss was found to be approximately twenty percent of the heat losses. The third and last way for heat to be transferred from the model was from convection to the air which flowed past the model. This loss was proportional to the temperature differential between the model walls and the ambient air. Thus, equating the various heat transfer rates, mean energy balance produces the relation

$$EI = h[T_s - T_a] + F \, \varepsilon \, \sigma[T_s^4 - T_a^4]. \qquad (1)$$

The wattage into the model was measured. The infrared camera arrangement and a known reference temperature at a single point within the radiating image gave the values of the model surface temperature over the entire wall. The emissivity and the ambient temperature were also determined. The surface heat transfer coefficient h was the only unknown quantity; hence, rearranging the equation

$$h = \frac{EI - \varepsilon F \sigma [T_s^4 - T_a^4]}{T_s - T_a} \qquad (2)$$

The values of the surface heat transfer coefficient, h, depend heavily on the reference temperature used to calibrate the infrared detector; hence, a calculation was performed to determine if the reference temperature measured on the inside of the model's aluminum walls was indeed an accurate value to use as a surface temperature on the outer surface. The heat flux per unit area is equal to the heat transferred through the aluminum according to Fourier's law. This heat flux per unit area is also equal to the heat transfer at the surface of the model wall from convection and radiation; hence, equating these two terms produces

$$\frac{k}{L}(T_r - T_s) = h(T_s - T_a) + \sigma \, \varepsilon (T_s^4 - T_a^4) \qquad (3)$$

where T_r is the reference temperature at the interior of the aluminum walls. When this relation is solved by trial and error for typical operating conditions the surface temperature was found to agree very favorably with the interior reference temperature being within 99.96 percent.

Surface Temperatures

The surface temperatures were obtained directly from the color pictures taken over the infrared camera monitor. From these pictures, the reference temperatures, and the ambient temperature values of the surface temperature could be determined for each color coded region. At the bottom of each picture the AGA Thermovision camera unit provided a color code from white to black, hottest to coldest regions respectively. These regions are on a scale from 0 to 1.0 in steps of 0.1. Since the camera sensitivity and the scale difference between the reference color, which corresponds to the color over the reference thermistor, and any other color on the scale were recorded at the time of the experiment one can determine the temperature of the colored regions. The product of the relative scale distance between colors and the sensitivity gave the number of isotherm units. Isotherm units may be converted to degrees Centigrade from a calibration curve provided with the AGA Thermovision camera. The final results were regions of known temperature.

Data Reduction Procedure

Heater power dissipation, ambient air temperature, surface emissivity, and local surface temperature have been combined to calculate surface heat transfer coefficient from the equations discussed earlier. Since the AGA Thermosivion presented colored thermograms divided into ten isothermal regions, computed surface heat transfer coefficients were tabulated in terms of their respective color number 1 to 10 ranging from black (cool) to white (hot) surface conditions.

Since the photographs taken directly from the AGA Thermovision display are distorted by camera angle and resolution a series of graphs were prepared which display the surface heat transfer coefficient contours on a series of "unwrapped" views of the models (Figs. 8 to 10). In some cases no surface temperature reference was available; hence no value was assigned to the contour lines. In other cases awkward camera angle precluded a useable photograph.

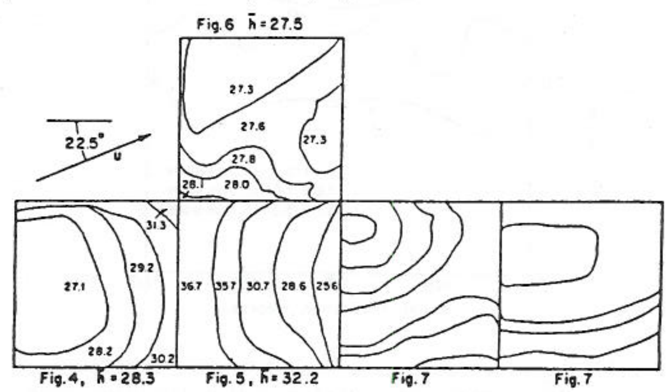

Fig.8 Contours of local surface heat transfer coefficients (15.3 x 15.3 x 15.3 cm model at 22.5° orientation h = W/M^2 - °C).

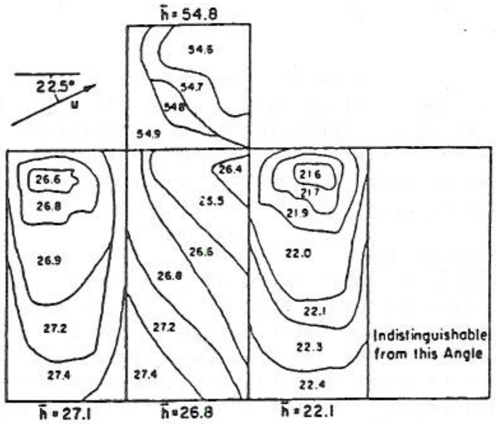

Fig.9 Contours of local surface heat transfer coefficients (7.6 x 7.6 x 15.3 cm model at 22.5° orientation h = W/M^2 - °C).

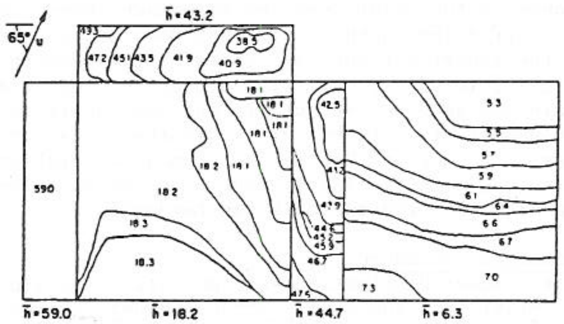

Fig.10 Contours of local surface heat transfer coefficients (3.8 x 15.3 x 15.3 cm model at 65° orientation h = W/M^2 - °C).

The average surface heat transfer coefficient per side, $\overline{h}$, has been calculated by measuring the area between contour lines with a planimeter. Average surface heat transfer coefficient, $\overline{h}$, and midpoint h on each side are plotted as a function of model orientation in Figs. 11 and 12.

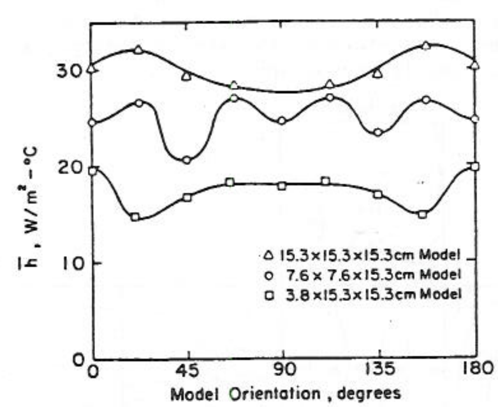

Fig.11 Average convective heat transfer coefficient ($\overline{h}$) (model wall of the heated building model rotated in the wind tunnel 180 deg, extrapolation of some points was made using symmetry of model, at 0° wind is parallel to model wall and at 90° wind is perpendicular to model wall).

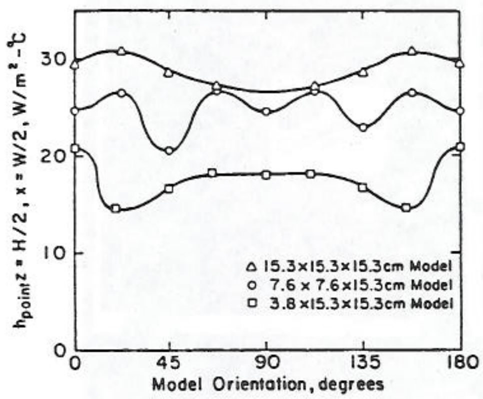

Fig.12 Midpoint convective heat transfer coefficient (model wall of the heated building model rotated in the wind tunnel 180 deg, extrapolation of some points was made using symmetry of model, at 0° wind is parallel to model wall and at 90° wind is perpendicular to model wall).

DISCUSSION OF RESULTS

A wind tunnel investigation was conducted to determine local and average forced convection heat transfer coefficients on the exterior surface of three flat-roofed building models under shear flow conditions as well as varying wind direction. The primary objective of this experiment was to investigate the feasibility of utilizing an infrared camera technique to measure model heat transfer coefficients. As a matter of convenience only a model scale and wind tunnel were utilized which do not provide strict simulation of full-scale conditions. It was found, nonetheless, that the experiment sufficed to delineate the potential and operating problems for the system. In the following sections specific details of the results are discussed in light of the above considerations. A set of recommendations concerning future measurements are contained in Conclusions and Recommendations.

Local Heat Transfer Coefficient Variation

Measured values of local heat transfer coefficient ranged from 5.3 to 59 W/M^2 - °C. This large variation was due in part to the selection of building shapes which included those of slender cross-section resulting in large sheltered regions at some orientations, and thin boundary layers, highly ventilated regions at other orientations.

Flow against an upstream facing wall generally resulted in a low heat transfer coefficient stagnation region directly above wall center. Heat transfer in this case was generally higher at the building edges with maximum values, somewhat unexpectedly, near the ground. Thomas (6) did not observe such a stagnation region in his most equivalent shear flow case; however, measurements made during his uniform approach flow condition suggests such a region. Considering that the power law coefficient for Thomas's profile was 0.3 and the building height to boundary layer height ratio, H_B/δ, was one-third compared to values of 0.18 and 1.0 respectively for this case the variation may be expected. Sharp velocity gradients near the surface for this case result in strong horseshoe-shaped vortex eddies which wrap about the model base and scour the lower surface.

On the side and top faces the highest heat transfer coefficient exists at the upwind edges where the boundary layer is thinnest (see Fig. 8). In those cases where separation occurs the reverse is the case. Low transport rates exist in these recirculating regions (see side wall, Fig. 9). For a symmetric cubical building oriented with one corner upwind, the top and side walls display the expected symmetry. Particularly interesting are the wavy roof patterns which suggest an effect due to dual vortices generated by the delta-wind character of the roof corner.

The downstream faces generally have the lowest values of heat transfer coefficient (Fig. 10) as a result of large slowly recirculating wake eddies. Ito et al. (3) observed a similar front to back variation in the surface heat transfer coefficient. Their full-sized building has a similar geometry to the 3.8 x 15.3 x 15.3 cm model depicted in Fig. 10. Thomas (6) found a much smaller variation in surface heat transfer coefficient from front to back of his model.

A marked discontinuity in surface heat transfer coefficient occurs across the edges between different model faces. Thomas (6) observed a similar result; however, since he employed only nine detectors on each face, the resolution near the edge of each wall is rather poor.

Not all contour lines are identified in Figs. 8 to 10. In some cases a reference temperature was not available, in others there is some question concerning the grey level setting on the AGA Thermovision camera. Some sides do not have any contour lines indicated. These sides were inaccessible for the given camera and mirror arrangement.

The infrared camera method was generally found to give much better resolution over the building surface than the individual heat meters utilized by Thomas (6). The AGA Thermovision system permitted a wide range of surface variation, yet could be adjusted to differeniate small changes in a local area. Heating of all sides of the model building uniformly is more realistic than heating small local surfaces over a given wall. It is to be expected that heat transfer coefficients resulting from a continuous thermal boundary layer will be markedly different than those found from short internal thermal boundary layer regions.

Certain problems appeared during these trial experiments which must be resolved before further experiments are performed. It was difficult to identify the block edges at some orientations on the TV screen. A reflective edge coating may resolve this problem. In some cases a single camera setting could not resolve temperature variations in cases where two sides were viewed simultaneously. Individual shots should be made for each side as necessary.

During experiments for orientations of 0 deg or 90 deg there appeared to be a systematic asymmetry in the contours on the upstream faces. If one considers figures there were higher temperatures and lower convective heat transfer coefficients to the left of the model face center. This shift to the left was caused by the wake of the upstream mirror and its pivoting mechanism. Since the mirror was only 0.6 m upstream of the model a velocity deficit occurs which is reflected in contour appearance. The selection of a longer focal distance TV camera lens may resolve this problem.

Average Heat Transfer Coefficients

Average heat transfer coefficients were calculated for the 15.3 x 15.3 cm face for each of the three models. Its variation as the model is rotated ±90 deg from the approach flow is shown in Figs. 11 and 12. For the cubical building the variation is similar to that reported by Thomas (6). The average heat transfer coefficient is a minimum at a model orientation of 90 deg, but it increases to a maximum for angles turned slightly less than 90 deg, and then it decreases as the face becomes parallel to the flow. This decrease occurs when sheltering separation regions occur on the model face.

Full-scale buildings reside in flows typified by large Reynolds numbers; hence the boundary layers which develop on the building faces are probably turbulent. The local flow developing on the model sides reported herein are most likely laminar. A model Reynolds number based on building height for the tests reported herein was only ~30,000--a value generally considered insufficiently large to produce turbulent boundary layer behavior. Thomas (6) reports values of $\overline{Nu}/Re^{1/2} \approx 0(1)$ for most of his experiments on isolated buildings. Since laminar boundary layer data generally correlates in such a manner he concluded laminar boundary layers persisted in his experiments to Reynolds numbers as large as ~150,000. For the 15.3 x 15.3 x 15.3 cube oriented at 90 deg the average heat transfer coefficient found gave $\overline{Nu}/Re^{1/2} = 0.62$. Thus, these experiments were performed in a similar regime to those of Thomas (6).

It will be possible to increase model Reynolds numbers to values exceeding 300,000 by increasing model characteristic height above 30 cm and utilizing one of the Colorado State University wind tunnels capable of velocities exceeding 15 m/sec. Turbulent flow on reasonable size models could be assured by utilizing rougher model surfaces.

CONCLUSIONS AND RECOMMENDATIONS

Heat Transfer from a Building

The results of the investigation to determine local and average values of forced convection heat transfer coefficients on the square, flat roofed building models, lead to the following conclusions:

1 Local heat transfer coefficients may vary by a factor of three from one face to another of a rectangular building at different building orientations.

2 Local heat transfer coefficients may vary 30 to 40 percent over a single building face depending upon the orientation.

3 Lowest rates of heat transfer occur near flow stagnation regions on forward facing walls, in regions of flow separation on side walls, and over the entire face of leeward walls.

4 Highest rates of heat transfer occur near the edges of forward facing walls and along the upstream edge of the roof and side walls when separation does not occur.

5 Although local values of heat transfer coefficient vary with wall orientation, the average value for the entire face appears to vary by less than five percent with wind direction. Nonetheless accounting for local variations during building climatology calculations may permit energy savings.

Simulation Criteria to Model Exterior Building Climatology

A number of wind engineering simulation principles have been confirmed or reinforced by this study; they are:

1 A shear flow environment as opposed to a uniform approach flow contributed substantially to local heat transfer coefficient variation over the surface of the model building.

2 It will be necessary to operate at sufficiently large Reynolds number to establish turbulent boundary layers along the model surface if full-scale similarity is expected. A value of

Reynolds number greater than 500,000 would be preferred based on laminar turbulent transition experience. Rough wall surfaces are also desirable.

3 In light of 2 above, an investigation should be made into model size and model surface roughness requirements to assure turbulent flow and similarity to full-scale.

4 In winter situations a building may be appreciably warmer than the oncoming flow. Jointly forced and free convection may produce heat transfer rates significantly different from forced convection alone. A series of measurements should be performed where the equivalent Grashop number,

$$Gr = \frac{\rho^2 g\ \beta(T_s - T_\infty)L^3}{\mu}$$

, is developed in the laboratory. Since in turbulent free convection situations $\bar{h} \sim L$ and one expects turbulent forced convection to vary as $\bar{h} \sim L^{2/3}$ such combined influences will be significant on larger buildings.

Laboratory Methodology

As noted previously a primary objective of this study was to evaluate the infrared camera technique as a method to measure heat transfer coefficients in the laboratory. The experience gained during these tests suggests:

1 The infrared thermovision camera technique utilized herein can produce detailed information concerning the variation of surface heat transfer coefficients. There are no major problems concerning thermal range, resolution, accuracy, calibration, or analysis.

2 A number of improvements can be made in the details of data analysis. Thermovision camera output should be recorded on magnetic tape or a computer driven photographic image detection device should be utilized to limit the drudgery of data reduction.

3 An automated system of calibration should be incorporated into the methodology to permit full use of the range and resolution of the infrared camera system without lengthy manual calibration.

4 A longer focal-length infrared camera lens should be employed to permit remote measurements in the larger size wind tunnels necessary for full-scale similarity.

5 A more flexible mirror system should be designed to permit viewing all model sides at face angles of less than 45 deg.

6 Model buildings should be so wired for heating that individual heater power dissipation may be measured and adjusted.

REFERENCES

1 ASHRAE, Handbook of Fundamentals, American Society of Heating, Refrigerating and Air Conditioning Engineers, 1972.

2 Homan, J. P., Heat Transfer, Third Edition, McGraw-Hill Book Company, Inc., New York, 1972.

3 Ito, N., Kimura, K., and Oka, J., A Field Study on the Convective Heat Transfer Coefficient on External Surface of a Building, ASHRAE Transactions, Vol. 78, Part 1, pp. 184-192, 1972.

4 Kelnhofer, W. J. and Thomas, C. J., External Convective Heat Transfer Coefficients on a Building Model, ASME Publication, 76-WA/FE-30, August 12, 1976.

5 Taw-atichat, J., On Modeling of Atmospheric Surface Layers by the Counter-Jet Technique, AIAA Paper No. 74-638, 8th Aerospace Testing Conference, Bethesda, Maryland, July 8-10, 1974.

6 Thomas, C. J., External Heat Transfer Coefficient on Building Models, Dissertation, The Catholic University of American, Washington, D.C., 1974.

METEOROLOGY AND THE HEATING AND VENTILATING OF BUILDINGS:

1993 ASHRAE HANDBOOK, FUNDAMENTALS, I-P Edition, American Society of Heating, Refrigeration and Air Conditioning Engineers, Inc., Atlanta, GA.

Aynsley, R.M., W. Melbourne and B.J. Vickery, 1977, ARCHITECTURAL AERODYNAMICS, Applied Science Publishers, Lt., Essex, UK

Cermak, J.E., M. Poreh, J.A. Peterka, and S.S. Ayad, 1982, WIND-TUNNEL INVESTIGATIONS OF NATURAL VENTILATION, ASCE Winter Annual Meeting, Preprint 82-519, 16 pp.

Cockroft, J.P. and Roberston, 1976, VENTILATION OF AN ENCLOUSRE THROUGH A SINGLE OPENING, Building and Environment, Vol. 11, pp. 29-35.

Dick, J.B., 1950, THE FUNDAMENTALS OF NATURAL VENTILATION OF HOUSES, Institute of Heating and Ventilation Engineers, Jrnl., Vol. 18, pp. 123-134.

Ethridge, D.W. and J.A. Nolan, 1979, VENTILATION MEASUREMENTS AT MODEL SCALE IN A TURBULENT FLOW, Building and Environment, Vol. 14, pp. 43-64.

Meroney, R. N., 1978, STUDYING THE CONVECTIVE HEAT TRANSFER FROM A BUILDING MODEL WITH INFRARED CAMERA TECHNIQUES, 1978 ASME Winter Annual Meeting, December 10-15, 1978, San Francisco, California, May 1978, 28 p., ASME Paper 78-WA/HT-58,

Meroney, R.N., Birdsall, J.B., and Neff, D.E., 1995, WIND-TUNNEL SIMULATION OF INFILTRATION ACROSS PERMEABLE BUILDING ENVELOPES: ENERGY AND AIR POLLUTION EXCHANGE RATES, Proceedings of the 7th International Symposium on Measurement and Modeling of Environmental Flows, San Francisco, CA, November 12-17, 1995

Meroney, R.N., Neff, D.E., and Elbadry, 1995, H., WIND-TUNNEL SIMULATION OF ENVELOPE CONCENTRATIONS FROM RADON EXHAUST VENTILATORS AROUND TYPICAL DOMESTIC BUILDINGS, Proceedings of 9th Int. Conf. on Wind Engineering, New Delhi, India, January 1995, 12 pp.

Meroney, R.N. and Birdsall, J.B., 1995, MODEL SCALE AND NUMERICAL EVAULATION OF TRACER GAS DISTRIBUTION DUE TO WIND FORCED NATURAL VENTILATION, Proceedings of 9th Int. Conf. on Wind Engineering, New Delhi, India, January 1995, 12 pp.

METEOROLOGY AND THE HEATING AND VENTILATING OF BUILDINGS:

1993 ASHRAE HANDBOOK, FUNDAMENTALS, I-P Edition, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA.

Aynsley, R.M., W. Melbourne and B.J. Vickery, 1977, ARCHITECTURAL AERODYNAMICS, Applied Science Publishers, Ltd., Essex, UK.

Cermak, J.E., M. Poreh, J.A. Peterka and S.S. Ayad, 1982, WIND-TUNNEL INVESTIGATIONS OF NATURAL VENTILATION, ASCE Fall Annual Meeting, Preprint 82-??, ?? pp.

Cockroft, J.P. and Robertson, P., 1976, VENTILATION OF AN ENCLOSURE THROUGH A SINGLE OPENING, Building and Environment, Vol. 11, pp. 29-35.

Dick, J.B., 1950, THE FUNDAMENTALS OF NATURAL VENTILATION OF HOUSES, Institute of Heating and Ventilation Engineers Jnl., Vol. 18, pp. 123-134.

Etheridge, D.W. and J.A. Nolan, 1979, VENTILATION MEASUREMENTS AT MODEL SCALE IN A TURBULENT FLOW, Building and Environment, Vol. 14, pp. 53-64.

Meroney, R.N., 1978, STUDYING THE CONVECTIVE HEAT TRANSFER FROM A BUILDING MODEL WITH INFRARED CAMERA TECHNIQUES, 1978 ASME Winter Annual Meeting, December 10-15, 1978, San Francisco, California, Man...1978, 28 p., ASME Paper 78-WA/HT-58.

Meroney, R.N., Birdsall, J.B. and Neff, D.E., 1995, WIND-TUNNEL SIMULATION OF INFILTRATION ACROSS PERMEABLE BUILDING ENVELOPES: ENERGY AND AIR POLLUTION EXCHANGE RATES, Proceedings of the 7th International Symposium on Measurement and Modeling of Environmental Flows, San Francisco, CA, November 12-17, 1995.

Meroney, R.N., Neff, D.E., and Birdsall, 1995 b, WIND-TUNNEL SIMULATION OF ENTRY OF CONCENTRATIONS FROM RADON EXHAUST VENTILATORS AROUND TYPICAL DOMESTIC BUILDINGS, Proceedings of 9th Int. Conf. on Wind Engineering, New Delhi, India, January 1995, 12 pp.

Meroney, R.N. and Birdsall, J.B., 1995, MODEL SCALE AND NUMERICAL EVALUATION OF TRACER GAS DISTRIBUTION DUE TO WIND-FORCED NATURAL VENTILATION, Proceedings of 9th Int. Conf. on Wind Engineering, New Delhi, India, January 1995, 12 pp.