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CONVECTION AND CONDUCTIVE HEAT TRANSFER TO THE FLOOR OF TENTS

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DEFENCE RESEARCH ESTABLISHMENT OTTAWA
TECHNICAL NOTE 86-2

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ABSTRACT

Conductive heat transfer to the floor of a single walled tent was examined experimentally and empirically. A heated, Canadian Forces, 5-Man, Arctic Tent was used to measure heat flow rates to the floor, as well as air temperatures and air velocity near the floor. The measured conductive heat flow rates to the floor of the tent were compared with predictions based on empirical correlations from problems of similar yet different geometries. Predicted heat flow rates obtained from stagnation point flow analysis were found to agree with measured heat flow rates to within 25%. The total conductive heat loss to the floor of this single walled tent was found to be approximately 7% of the total heat loss from the tent. Thus, the use of the stagnation point flow correlation would result in an error of approximately 2% in the calculation of the overall heat transfer rate from a single walled tent. The conductive heat flow rate to the floor was found to be approximately 20 W/m^2 and the radial air speed near the floor was found to be less than 0.1 m/s.

RÉSUMÉ

On a procédé à l'emamen théorique et pratique du tranfert de chaleur par conduction au plancher d'une tente à paroi simple. Au moyen d'une tente Arctic des Forces canadiennes chauffée et pouvant contenir 5 personnes, on a mesuré les flux thermiques se rendant au plancher, de même que la température et la vitesse de l'air à proximité du plancher. Les flux thermiques mesurés au plancher de la tente ont été comparés avec les prédictions faites à partir de corrélations empiriques dècoulant de problémes concernant des géométries semblables, quoique différentes. Les flux thermiques que l'on prévoyait obtenir suite à l'analyse du flux au point d'arrêt correspondaient avec les flux thermiques mesurés, dans une marge de moins de 25 pour cent. La perte totale de chaleur par conduction vers le plancher de la tente à paroi simple équivalait à environ 7 pour cent de la perte totale de chaleur de la tente. Ainsi, l'utilisation de la corrélation du flux au point d'arrêt entraînerait un pourcentage d'erreur d'environ 2 pour cent dans le calcul du transfert thermique global d'une tente à paroi simple. On a constaté que le transfert de chaleur par conduction vers le plancher de la tente était d'environ 20 W/m^2 et que la vitesse de la ventilation radiale prés du plancher était inférieure à 0,1 m/s.

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GLOSSARY

g	acceleration due to gravity, $\frac{m}{s^2}$
Gr	Grashof Number = $g\,\frac{\beta \Delta T L^3}{\nu^2}$
h	heat transfer coifficient, $\frac{W}{m^2 K}$
k	thermal conductivity, $\frac{mK}{W}$
L	a characteristic length, m
Nu	Nusselt Number = $\frac{hL}{k}$
Pr	Prandtl Number
q	conductive heat transfer rate per unit area, $\frac{W}{m^2}$
T	temperature, C
U	characteristic air flow rate, m/s
x	a length in the direction of heat transfer, m
β	coefficient of thermal expansion, K^{-1}
ν	Kinematic Viscosity, $\frac{m^2}{s}$

Subscripts

a	air
f	floor

1.0 INTRODUCTION

This study examines the conductive heat transfer to the floor of a single walled, heated tent. Specifically, a Canadian Forces, 5-Man Arcitic Tent was used for the experimental observations. Conductive heat flow at the floor, air temperatures and air velocities near the floor of the tent were measured experimentally. Predictions of the heat flow rate based on correlations between characteristic nondimensional parameters from similar analyses were compared with the measured heat flow rates to determine whether or not a convenient correlation for accurately predicting heat flow rates to tent floors exists in the literature.

This report is part of a larger study, the goals of which are to determine the important heat transfer mechanisms in tents and to provide means of predicting the magnitudes of these heat transfer rates.

It should be noted that this study does not include the heat loss to the floor by means of radiative heat transfer, evaporation/condensation or air infiltration. Radiative heat transfer is a significant fraction of the total heat transfer in many problems. Fortunately it may be analysed separately from, but simultaneously with, conductive heat transfer. Radiant heat transfer will be examined in a subsequent investigation.

Evaporation or condensation of water vapour at a surface can dominate heat transfer mechanisms, a phenomenon observed in a sweating man and in evaporative cooling towers. The experimental portion of the study was performed in mid-summer when the air was dry. It was therefore assumed that there was negligible heat transfer by condensation or evaporation at the floor of the tent.

Air infiltration through the floor may be of importance if the floor is very porous. It is expected that this mechanism would seldom be significant. Air infiltration around the base of the tent could significantly alter the rate of heat transfer to the floor as well as the overall heat transfer from the tent. The tent used in this study was well secured at the ground, and air infiltration could only occur through the fabric of the tent walls.

2.0 BACKGROUND

2.1 CONDUCTION

Conductive heat transfer is a mechanism by which energy is transfered due to the kinetic motion of atoms or molecules. Conductive heat transfer can often be calculated approximately by:

$$q = k(T_a - T_g)/X \tag{1}$$

For finite, non-zero distances, equation 1 is valid if there is no bulk motion in the conducting medium. If bulk motion of the medium occurs, as is often observed in fluids, equation 1 must be replaced by the more general form of the Fourier relationship:

$$q = k(dT/dx) \tag{2}$$

This equation is valid at any point in a conducting medium.

For a heated tent, it is expected that the floor would be cooler than the air above it. If there were no circulation, the heat transfer through the air could be predicted using equation 1. Entrainment by the heater plume and subsidence of cooled air at the tent walls causes air circulation resulting in air movement over the floor. Thus, equation 2 should be used for calculating the heat transfer to the floor.

2.2 CONVECTION

When motion of the conducting medium, subsequently called the fluid, is large enough to cause a significant deviation of the predicted heat transfer rate from equation 1, the heat transfer mechanism is often said to be convective. This is a misnomer as convection refers to the motion of the fluid rather than the transfer of thermal energy. By modifying "heat transfer" by "convective" the reader is being informed that the heat transfer rate is being affected by the motion of the fluid. The effect of convection is to increase the temperature gradient of equation 2 within the fluid. Thus, resulting heat transfer rates can be greater than those predicted from equation 1. The temperature distribution in the fluid is closely connected to the velocity distribution.

Convection can be classified into two forms: free and forced. In free convection, bulk motion of the fluid due to density differences caused by temperature gradients is dominant. In forced convection, bulk motion of the fluid is the result of an impressed pressure gradient. Both free and forced convection may occur simultaneously and may be of equal importance. A measure of the relative importance of the two mechanisms is the ratio of the Grashof number to the square of the Reynolds number [1]. If this ratio is much less than one, then forced convection is dominant over free convection in affecting the heat transfer while if the ratio is much greater than one the opposite is true. If this ratio is approximately 1, both mechansims are important.

Heat transfer from a surface to an adjacent fluid is often calculated by Newton's law of cooling:

$$q = h(T_a - T_g) \tag{3}$$

The heat transfer coefficient, h, is often determined experimentally, especially for turbulent flows and flows with complicated surface geometries. It is sometimes possible to determine the heat transfer coefficient analytically for laminar flow over simple bodies.

Natural convection from a flat plate is either a two or three dimensional process [2,3,4]. Prediction of whether the flow will be two or three dimensional is not currently possible. Thus, empirical correlations from experimental observations are the most convenient means of making predictions for this flow. A full three dimensional analysis of natural convection with heat transfer from a flat plate is possible by numerical techniques, however, it is beyond the scope of this report. The above references contain information on numerical analysis of natural convection from a horizontal flat plate.

The motion of the air at the floor of the tent will depend upon the temperature and pressure gradients within the air. No bulk motion of the air is predicted when there is a stable density gradient (temperatures increasing with height above the floor) and no lateral pressure gradient.

An unstable density gradient with no lateral pressure gradient would cause the air to circulate due to natural convection. A lateral pressure gradient would cause the air to flow along the floor with a small component of vertical velocity. If a temperature gradient also exists, the vertical component of velocity will be increased or decreased depending upon whether the temperature gradient is unstable or stable respectively. Figure 1 shows typical streamlines which would be expected from these flows.

2.2.1 NATURAL CONVECTION CORRELATIONS

Correlations between nondimensional heat transfer coefficients and nondimensional parameters which characterize heat and mass flow conveniently allow practical estimates of heat transfer rates where theoretical analysis are extremely complicated. The rate of heat transfer from a warm, horizontal plate to a cool fluid which is in motion due to natural convection has been found to depend upon several parameters [1]: the plate temperatures; the mean bulk fluid temperatures; a characteristic size of the plate; the thermal conductivity of the fluid; the kinematic viscosity of the fluid; and the coefficient of thermal expansion of the fluid.

The Nusselt number, a nondimensional measure of the heat transfer coefficient, correlates well with a simple function of the Grashof and Prandtl numbers [1]:

$$Nu = c(GrPr)^n \tag{4}$$

Table I lists the empirically determined constants of Equation 4.

TABLE I

Constants c and n of Eq. (4) for Free Convection from a Hot Horizontal Plate at Uniform Temperature to a Cold Fluid Above

Type of Flow	Range of Gr_L Pr	c	n
Laminar	10^5 to 2×10^7	0.54	1/4
Turbulent	2×10^7 to 3×10^{10}	0.14	1/3

The characteristic dimension, L, is taken to be the plate area divided by the plate perimeter [5].

If a tent is unheated, the floor temperature could be warmer or cooler than the air in the tent, depending upon the past history of the heat transfer to the ground under the tent. Ground temperatures can often be significantly different from the ambient air temperatures [6]. Thus, equation 4 may be required to calculate the heat transfer from a warm floor to the cool air above it. For the case of a heated tent, it is reasonable to assume that the air temperature in the tent will be warmer than the floor temperature. Thus, no natural convection from the floor should occur. If the floor temperature is not uniform, however, it may be possible to have some natural convection due to lateral temperature gradients. This mechanism is not expected to result in a substantial heat loss.

2.2.2 FORCED CONVECTION CORRELATIONS

Forced convection over the floor of a tent may result from any of several causes. Air entrainment by a heater plume, air subsidence at the cold walls of the tent, an air-circulating fan, air infiltration by the wind or movement of individuals are potential sources of forced convection. As with free convection, forced convection can occur in either laminar or turbulent flows or a combination of both.

The problem of laminar flow over a flat surface may be solved exactly in some cases. The geometry for this case necessitates the use of numerical analysis to obtain the flow field and temperature distribution. Two examples of exact solutions with similar geometries are included below.

The heat transfer for laminar flow over a flat plate (Figure 2a) of length R can be shown [7] to be given by:

$$q = 0.664\ \rho\ U\ C_p(T_a - T_f)/Pr^{0.3}\ Re^{0.5} \qquad (5)$$

for small U. For stagnation point flow (Figure 2b), the heat transfer can be shown [7] to be:

$$q = 2.058\ \rho\ U\ C_p(T_a - T_f)/Pr\ Re^{0.5} \qquad (6)$$

It should be noted that neither of these examples have exactly the same flow field as the problem at hand, rather they are the closest approximation found in the current literature.

Preliminary results indicated that the air velocity near the floor of the tent in this study was quite low. As the flow was well within the laminar flow regime, no turbulent flow analysis was considered. Additional information on turbulent flow heat transfer may be found in references [1] and [7].

3.0 EXPERIMENTAL INVESTIGATION

A Canadian Forces 5-Man Arctic tent (Figure 3) was used to measure the floor heat transfer rates while the tent was heated. As the tent is approximately symmetrical around the vertical axis, only one sector of the tent was instrumented. The average tent-floor radius was approximately 1.85 m. Air and ground temperatures were measured with an array of thermistors and thermocouples. Heat transfer rates to the ground were measured with heat flow discs.

To separate the convective and radiative components of heat transfer to the floor, aluminum foil was attached to the upper surface of the discs with heat sink compound. The foil had been found to have a reflectivity of approximatley 95% [8]. The radiative component of heat transfer was found to be of a similar magnitude as the conductive heat transfer. Thus, covering the heat flow disks with foil limited the influence of radiant heat transfer to approximately 5%.

Air velocities were measured with several hot-wire anemometers (HWA). The HWA were capable of measuring the air velocities in a single plane in the range of 0 to 3 m/s.

Heat was supplied by two electric, forced air heaters each of which had a nominal heat output of 1300 W and air flow-rate of 0.07 kg/s or approximately 0.06 m^3/s for these experiments. The heaters were supported approximately 5 cm above the floor, pointing upwards at the centre of the tent. The tent as allowed to come to steady state over several days to reduce the possibility of evaporative heat transfer from the ground.

Some of the heat transfer calculations required a freestream temperature and a freestream velocity. The volume weighted mean temperature of the tent [9] was used for the freestream temperature and the radial velocity at a height of fifteen centimeters and radius one meter was used for the freestream velocity.

Fluid properties were evaluated at the average of the floor and the volume weighted mean air temperatures. The volume weighted mean air temperature and the floor temperature were typically 35°C and 26°C respectively. The ambient temperature was typically 15 C. All vents and the door of the tent were sealed for the experiments.

3.1 RESULTS

The air velocity near the floor was found to be mainly radial with a small angular component. Typical radial velocity profiles are shown in Figure 4 for heights 1 and 50 cm. Due to the roughness of the gravel floor, it is doubtful that a fully developed, uniform velocity profile existed at a height of 1 cm. The larger velocities at the greater radii at a height of 1 cm are contrary to what would be expected from a radially converging flow [10]. This may have been caused by the floor roughness which locally accelerated or decelerated the flow. Entrainment by the heater plume may have increased the vertical component of velocity, resulting in a lower radial velocity component as shown in Figure 5.

The vertical component of velocity at a radius of 135 cm was found to vary from zero at a height of 1 cm to approximately 0.1 m/s at a height of 50 cm. It is expected that the vertical component of velocity will vary considerably in magnitude and direction with radius due to entrainment by the plume and subsidence at the tent walls.

The angular component of velocity was found to be approximately 0.01 m/s. It was expected that the angular component of velocity would be zero due to geometric symmetry. This small discrepancy may have been caused by some asymmetry in the heater arrangement or the tent geometry. This low angular velocity was not thought to significantly affect the heat flow at the floor.

Evaluation of the Reynolds number based on the distance from the tent wall indicated that the flow was at the lower end of the laminar regime.

Typical temperature profiles near the floor are shown in Figure 6. The temperature gradient can be seen to be somewhat steeper at the greater radius indicating a higher rate of heat transfer. It was found that the heat transfer rate at a radius of 145 cm was typically 1 to 5 W/m^2 greater than those at a radius of 80 cm. Predicted heat transfer rates based on equation 1 and the vertical temperature measurements were substantially less than the measured heat flow rates. This was presumably due to the bulk motion of the air. The irregularity of the floor made temperature measurement near the floor difficult. Thus, equation 2 could not be approximated closely enough by equation 1 to produce accurate estimates of the heat flow rates.

Table 2 gives the measured heat flow rates as well as the predicted heat flow rates from equation 6. Equation 4 was not applicable to this problem and the predicted heat flow rates from equation 5 were much too low. The measured heat flow rates are the average of four readings, two readings at a radius of 145 cm and two readings taken at 80 cm. Estimates based on equation 6 were found to predict the heat flow to within 25% for most cases. This is not an unreasonable precision considering that this correlation is for a similar problem only and that many correlations are only accurate to approximately 25%.

Included in Table 2 is the percentage of the total heat loss from the tent which is lost by conduction to the floor. The average value was 7% with a maximum of 14%. Errors incurred by using equation 6, though large with respect to the measured heat loss, are small with respect to the global heat loss from the tent.

TABLE 2

Test	Measured Heat Flow (W/m^2)	Eq^n 6 Heat Flow (W/m^2)	% of Total Tent Heat Loss By Conduction To The Tent Floor
1	22.1	26.7	8
2	14.8	23.2	6
3	20.9	19.0	8
4	18.6	24.4	7
5	24.0	22.4	9
6	36.5	27.4	14
7	11.2	12.9	4
8	12.0	14.0	5
9	10.7	13.7	4
10	16.4	19.7	6
11	18.1	21.5	7

4.0 CONCLUSION

The result of this study indicate that the conductive component of heat loss to the floor of a heated tent is approximately 7% of the total heat loss from the tent, with a maximum of 14% having been observed. Equation 6 was found to predict the conductive heat loss to the floor to within 25%. The error incurred by using this equation to determine the overall heat loss from a heated tent was therefore approximately 2%, and was less than 5% in all observed cases.

If no estimate of the air velocity near the floor is available a conductive heat loss of 7% of the total heat loss from the tent could be assumed. If the air velocity above the floor is known in addition to the relevant temperatures, equation 6 could be used.

Conductive heat loss to the floor was found to increase with increasing radius. This was consistent with observed changes in the temperature profile with radius.

The radial component of velocity was found to increase with increasing radius. This may have been due to local disturbances in the flow or due to the entrainment flow from the heater plume.

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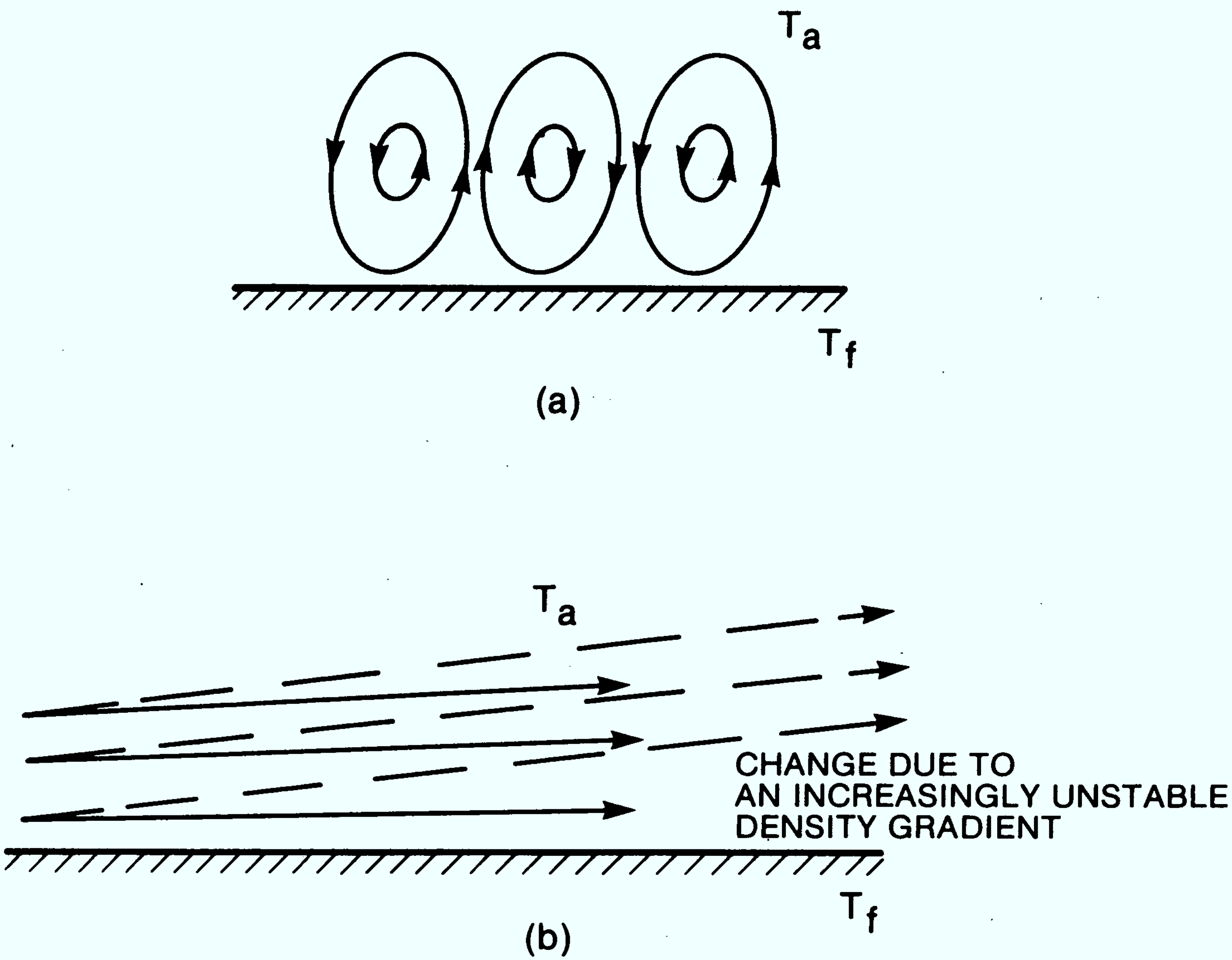

Figure 1: Typical Streamlines Resulting From:
(a) Natural convection from a flat plate;
(b) Forced convection over a flat plate (———— stable density gradient; ------ unstable density gradient).

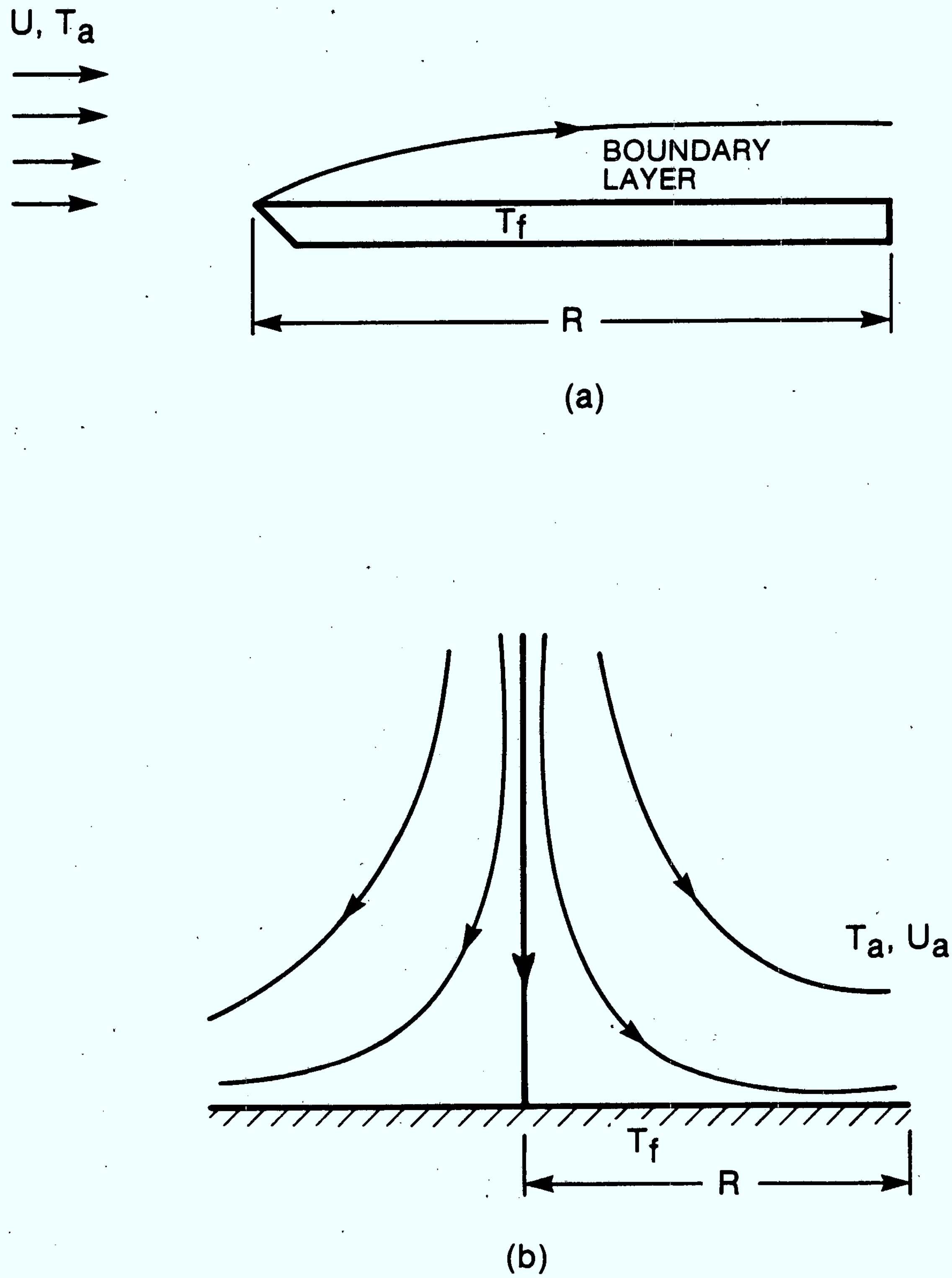

Figure 2: Analysis of:

(a) Flow over a flat plate;
(b) Stagnation point flow.

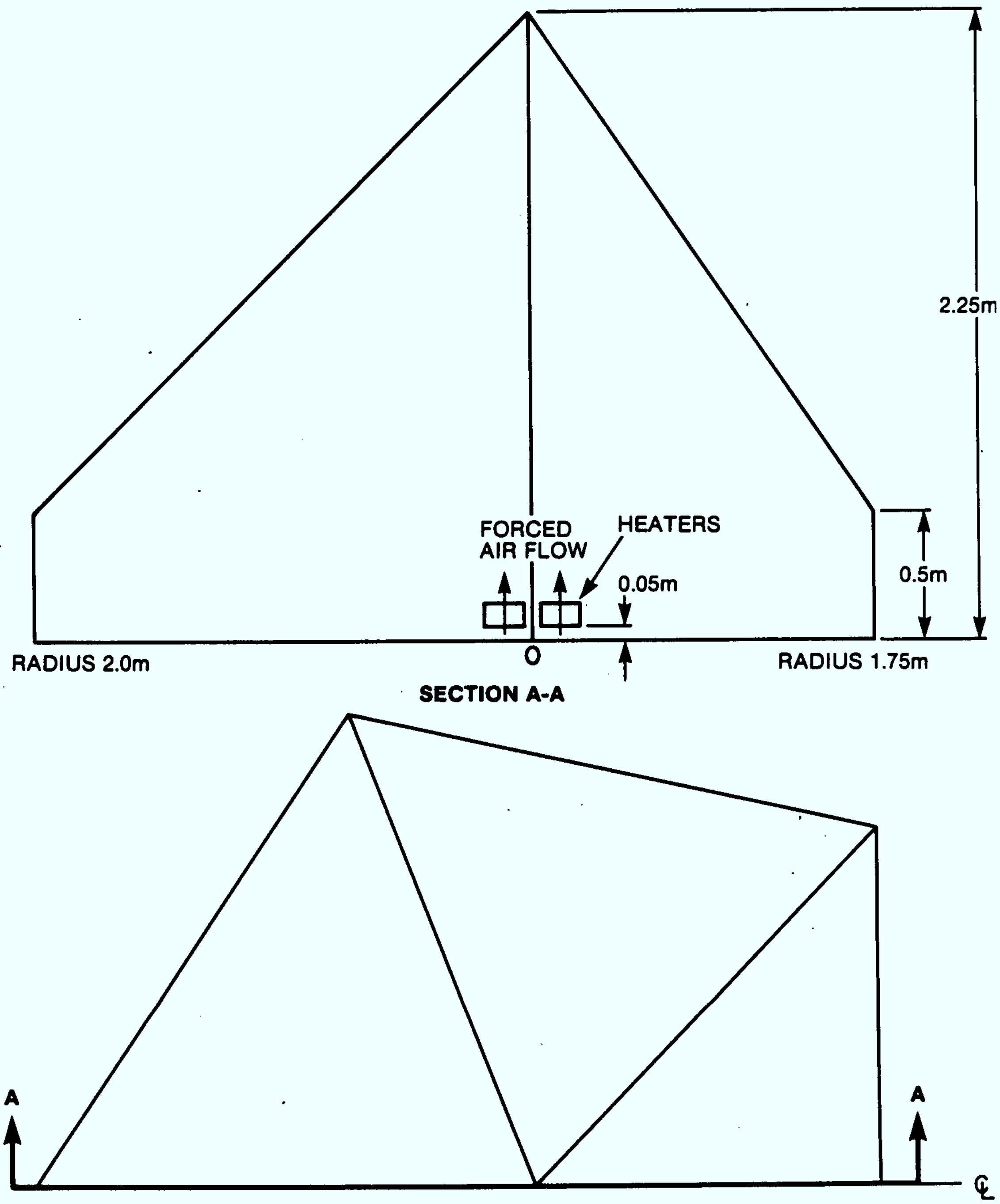

Figure 3: Schematic of the Canadian Forces, 5-Man, Arctic Tent used in the experimental portion of this study.

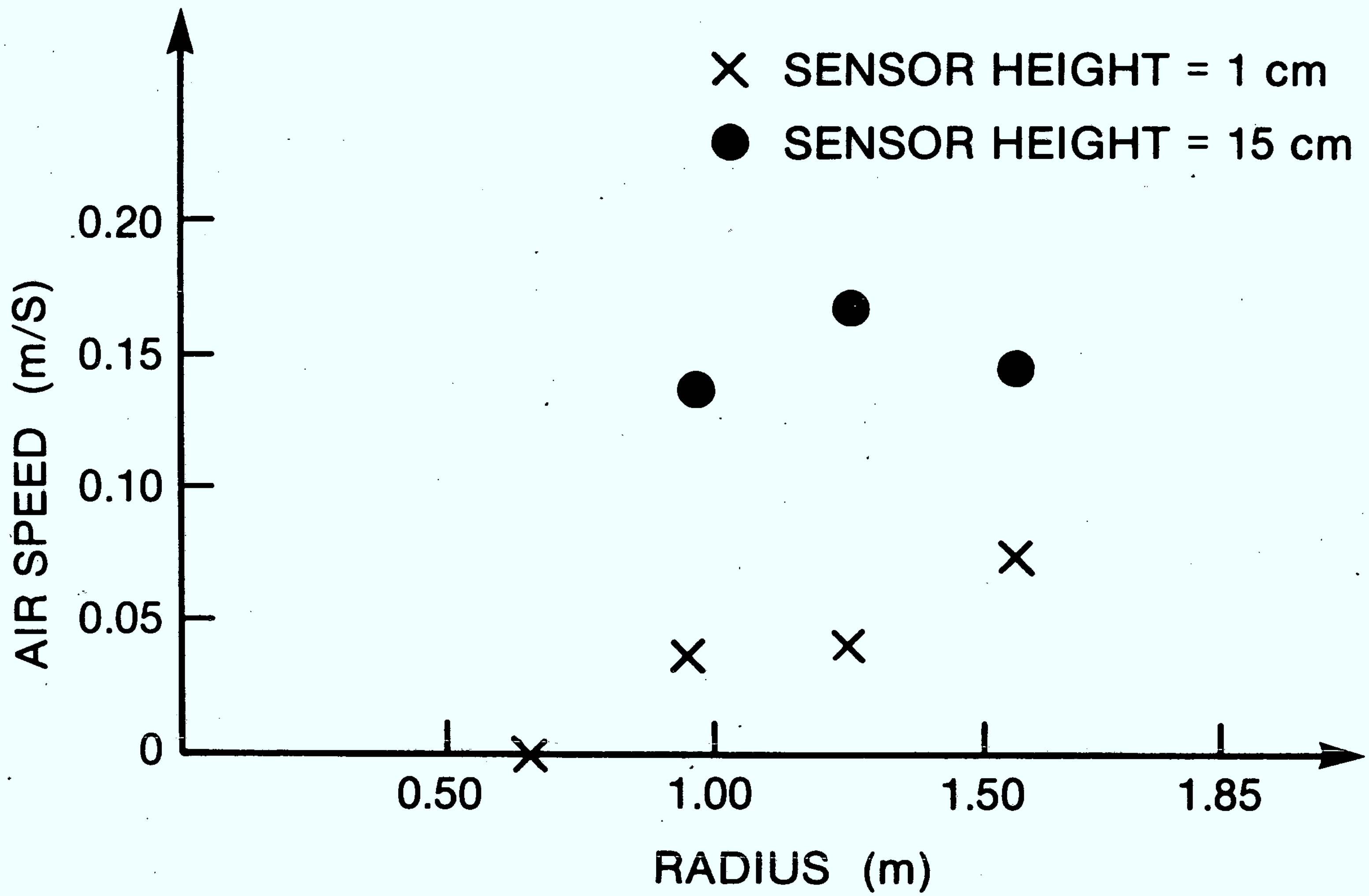

Figure 4: Radial air speed versus radius for the air flow over the tent floor.

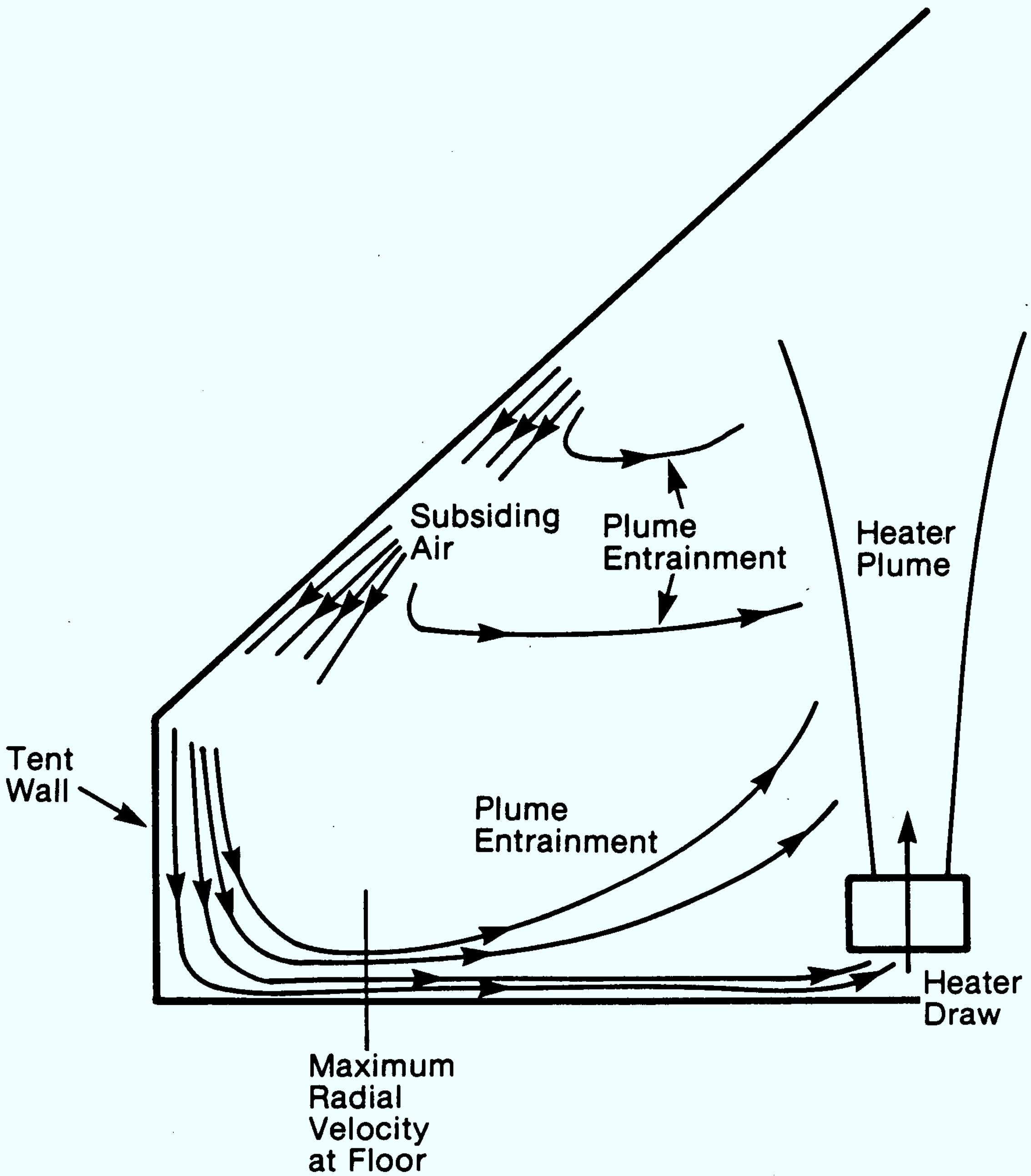

Figure 5: Possible flow patterns for air circulating in a tent due to heater plume entrainment and subsidence at the tent walls.

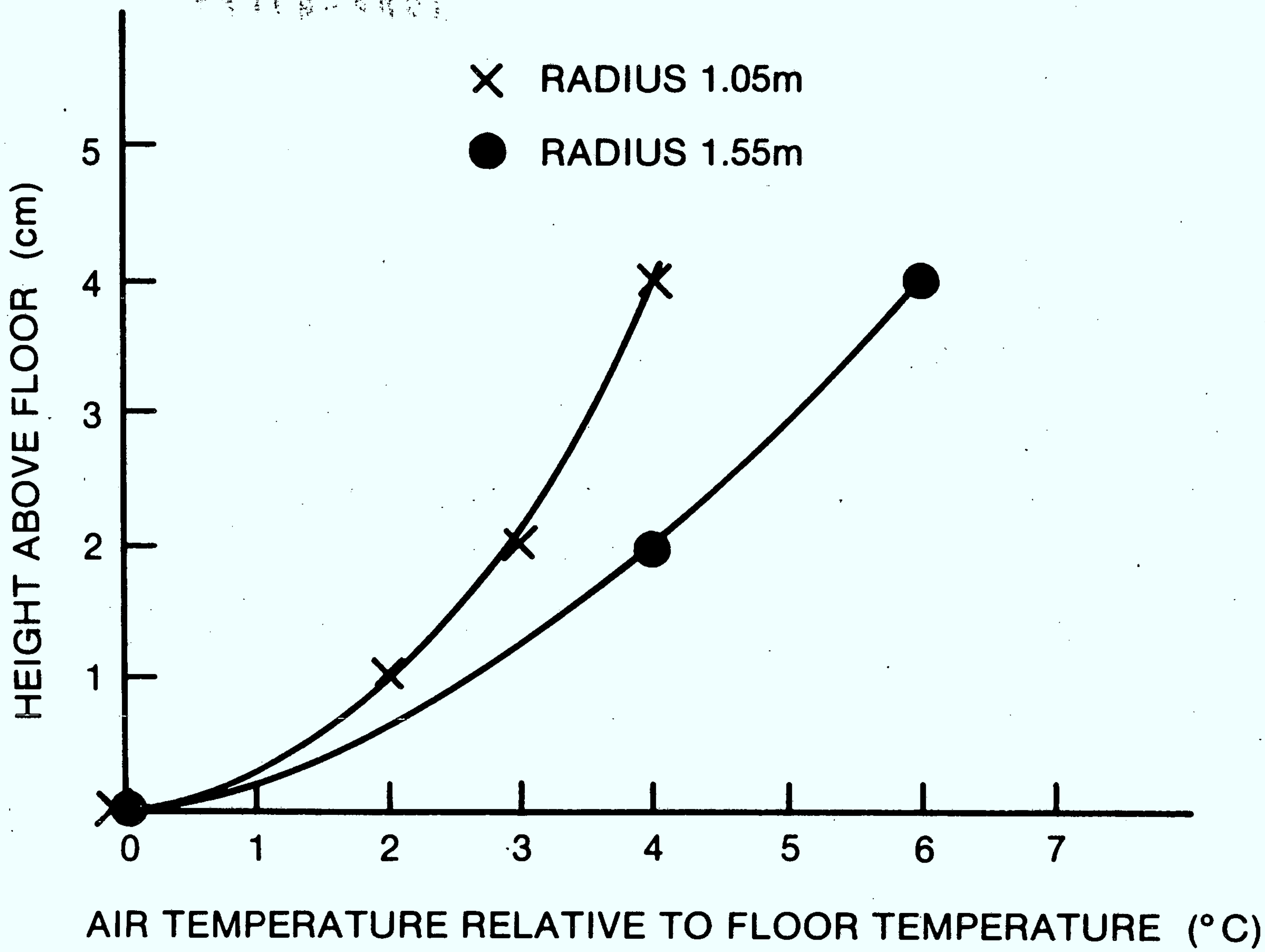

Figure 6: Relative air temperature above the tent floor at two radii.

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13. ABSTRACT

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