EXPERIMENTAL STUDY OF ENCLOSURE FIRES WITH HORIZONTAL VENTS

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INTRODUCTION

A very important flow and transport circumstance that arises in enclosure fires is that of heat and mass transfer across a horizontal vent Such vents exist in enclosed regions such as rooms and energy storage and ventilation systems. It is important to understand the basic nature of the transport processes which arise due to density and pressure differences across such vents. These two mechanisms give rise to very complicated flow patterns, depending on the relative magnitude of each.

Buoyancy induced flows generated by fires in enclosures have received considerable attention in the literature (Quintiere, 1977, Jaluria, 1980, Jaluria and Cooper, 1989). However, not much work has been done on flows through openings or vents such as those between containment areas in nuclear power systems, connecting rooms in buildings and between decks in ships. Vertical vents have been studied in recent years because of their importance in several practical problems, such as room fires and electronic and energy systems (Gebhart et al., 1988 and Abib and Jaluria, 1988). However, the work done on the flow through horizontal vents, such as the one shown in figure 1, is very limited. The flow rate for this case can be estimated, as done for vertical vents, by using Bernoulli's equation (Emmons, 1988). This model breaks down when there exists density, $\Delta\rho$, and pressure differences, Δp, across the vent (Cooper, 1990). Some experimental work has been done on the buoyancy driven flow through horizontal vents for the special circumstance of zero pressure difference (Brown, 1962 and Epstein, 1988). However, there have been few studies with non-zero pressure differences.

This paper represents the second phase of an on-going experimental study on the flow of air across a horizontal vent in an enclosed region for arbitrary values of the governing variables $\Delta\rho$, Δp and L/D, where L and D are the length and diameter of the vent respectively. The objective is to accurately quantify some of the results obtained from an earlier visualization study (Jaluria et al., 1993).

EXPERIMENTAL ARRANGEMENT

The experimental set-up consists of four sub-systems. These are the glass tank, the temperature measurement and regulation system, the pressure measurement and regulation system and the data acquisition system. The details of this set-up are given in Jaluria et al. (1993). Basically, the interior of the glass tank measures 0.41m x 0.41m x 0.46m, and all the interior edges and corners were sealed to ensure an air-tight enclosure.

To measure the temperature within this tank, type-T thermocouples were mounted vertically along one wall. On the top of this wall, a pressure tap was also placed. Due to the low pressures of interest, mean flows within the enclosure rendered accurate pressure readings difficult. Therefore, a diffuser for the inlet as well as a buffer for the pressure tap were added. This is shown in figure 2. An ultra-precise pressure transducer with an accuracy of less than 0.1 mPa was employed for the pressure measurements.

To control the temperature within the double-panel glass enclosure, a temperature controller was used to operate finned strip heaters. In order to control the pressure, a stagnation chamber was used to pressurize the glass enclosure. This chamber had an inside dimension of 0.15m x 0.15m x 0.15m. Its pressure was raised by a compressor, and controlled by a pressure regulator. A flow meter was connected upstream of this chamber to measure the mean flow rate, $\dot{m}$. To achieve uniform conditions within the tank, a layer of diffusive material was added. This consists of copper nettings which were layered uniformly to provide flow diffusion and thermal capacitance.

After verifying the enclosure to be air-tight, sufficient time was allowed for the entire system to reach steady state. Once steady state was achieved, the internal temperatures were measured before and after the experiments to ensure uniform temperature. Typically the measurements showed a temperature difference between the top and bottom of the glass tank to be less than 5 % of the average value. Most importantly, the top region near the exit duct attained temperature uniformity of better than 1 %. Also the ambient temperature and pressures were recorded before and after each experiment. During the experiment, the outputs from the thermocouples and pressure transducer were recorded through a data acquisition system.

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RESULTS AND DISCUSSIONS

Figure 3 shows one of the flow regimes observed in the earlier visualization study. This sequence is for flows through a vent with an L/D ratio of 4.0 and nonzero pressure and temperature differences. It clearly shows the oscillatory nature in the flow. This behavior was found to exist for pressures ranging from zero up to flooding. Attempts were made to correlate the frequency of the oscillation with the pressure difference across the vent. The intention was to test whether at the purging pressure, either the magnitude or the frequency of the oscillation would be significantly different. However, no such effect was found. Instead, the flooding pressure, Δp_c, was correlated (correlation coefficient > 0.98) against the temperature difference using a second order polynomial. This can be seen in figure 4. Attempts are also presently underway to track the oscillation in temperatures.

Although the oscillatory nature of the flow made the determination of the mean pressure very difficult, the correlation of the net flow rate against pressure differences across the vent was more successful. The data were fitted with a polynomial where the correlation coefficient was greater than 0.97. A typical fit is shown in figure 5a. Results were obtained for vents with four L/D ratios (0.0144, 1, 2, 4) and two temperature differences (ΔT= 20 K and 40K). As shown in figures 5b-d, the basic trends resemble that of a Bernoulli relation, where the slight deviation from is due to existing viscous and buoyancy forces. Also, the correlation showed that at low pressures, the mass flow rate was independent of temperature difference or the length of the vent. However, at a higher pressure, the flow rate increases with decreasing vent length and temperature differences. This is intuitively consistent since a shorter vent would present less frictional force while a smaller temperature would present less opposing buoyancy force. Efforts are presently underway to investigate the effect of different vent diameters.

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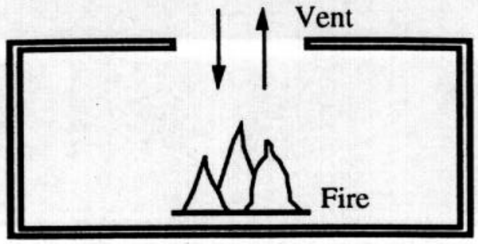


Figure 1: Schematic of a typical enclosure with a horizontal vent.

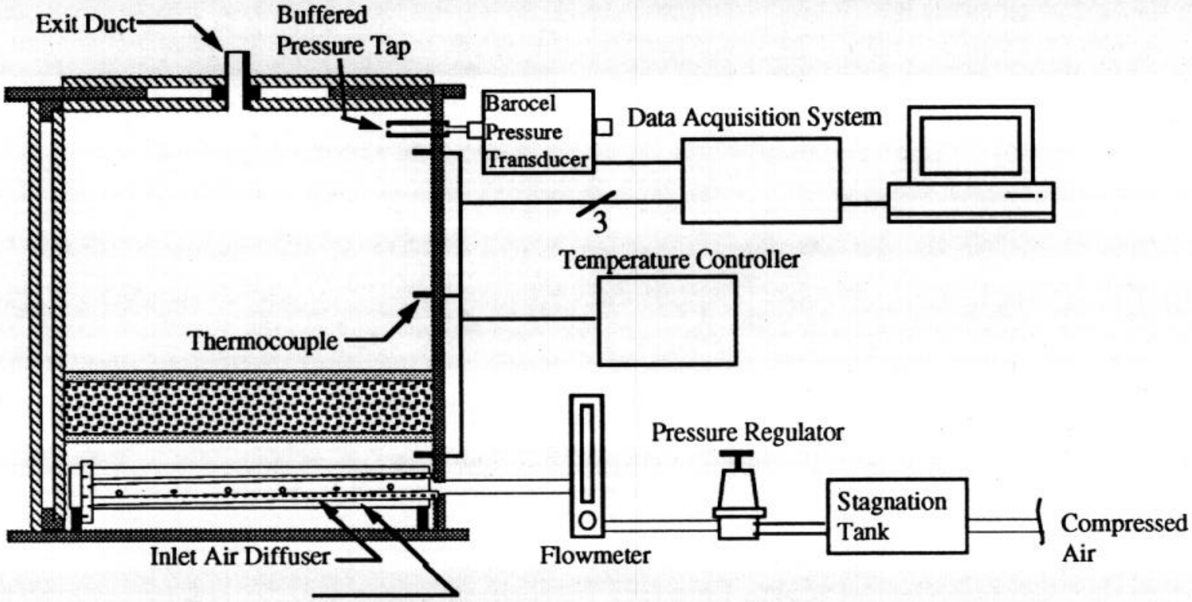


Figure 2: Schematic showing the subsystems of the experimental set-up.

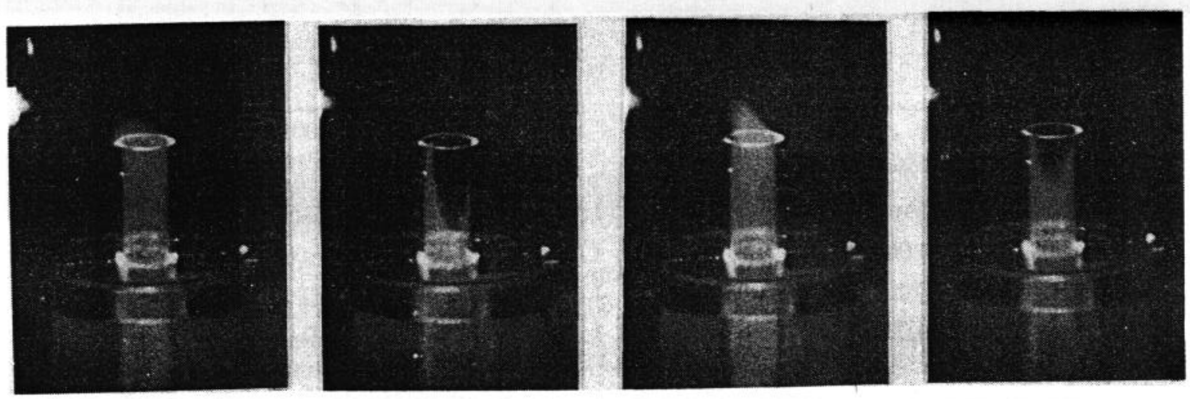


Figure 3: Photographs of an oscillatory flow in a vent of L/D=4.0.

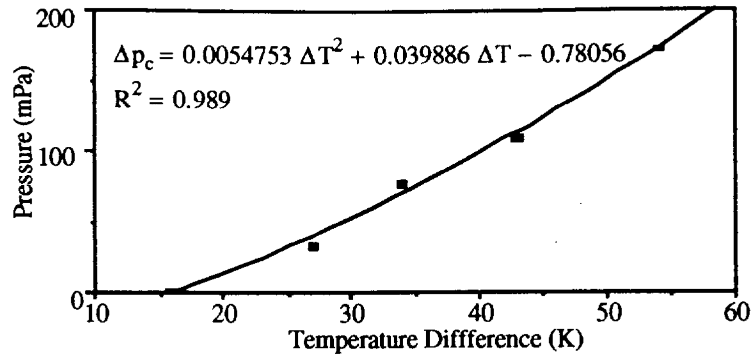
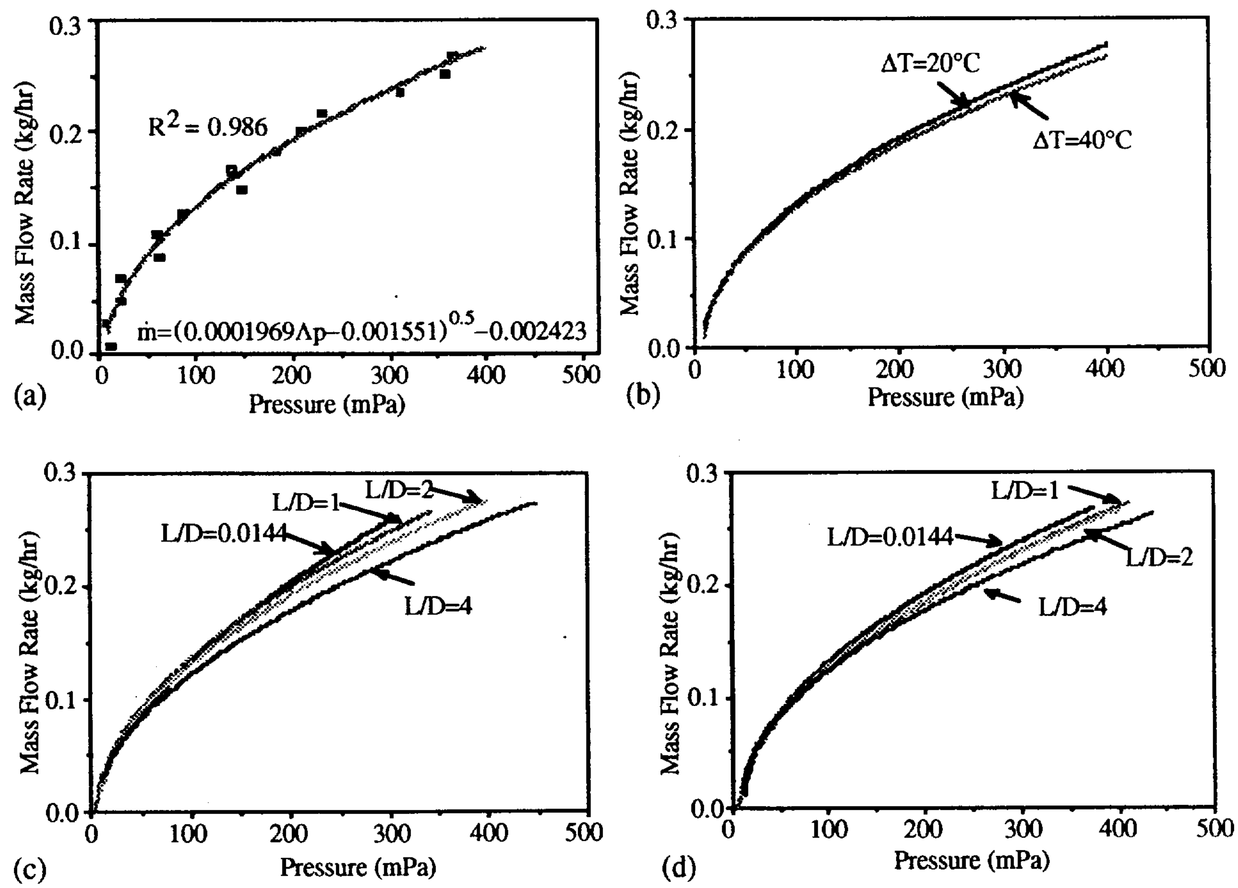


Figure 4: Plot of the purging pressure as a function of temperature differences for a vent with a L/D ratio of 0.0144.



L/D=0.0144: $\dot{m}=(0.0003078\Delta p+0.004068)^{0.5}-0.053$

L/D=1: $\dot{m}=(0.0002224\Delta p-0.0008244)^{0.5}-0.009375$

L/D=2: $\dot{m}=(0.0001969\Delta p-0.001551)^{0.5}-0.002423$

L/D=4: $\dot{m}=(0.0001803\Delta p-0.00042214)^{0.5}-0.01020484$

L/D=0.0144: $\dot{m}=(0.0001973\Delta p+0.002417)^{0.5}+0.0002564$

L/D=1: $\dot{m}=(0.0002032\Delta p-0.0007603)^{0.5}-0.015313$

L/D=2: $\dot{m}=(0.0001781\Delta p-0.001715)^{0.5}+0.001835$

L/D=4: $\dot{m}=(0.0001535\Delta p-0.00147538)^{0.5}+0.00662617$

Figure 5: Plots showing (a) the typical curve fitting for the present data, (b) the effect of different temperatures on the net mass flowrate, and the effect of different L/D ratios for a temperature difference of (c) 20K and (d) 40K.