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Gravity Driven Counterflow Through an Open Door in a Sealed Room

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Flow measurements using tracer gas techniques were made on the exterior doorway of a test house for indoor–outdoor temperature differences of 0.5–45 K. The time for door opening and closing was constant at 3.75 s, and fully open hold time varied from 0.5 s to 120 s. Predictions of a variable density steady flow model were in good agreement with the measurements when adjustments were made for the time-varying size of the opening and for the effect of cross-stream mixing between the incoming and outgoing air streams. The flow rate is shown to be governed by an effective density very close to the average of inflow and outflow densities, and the control condition at the doorway is fixed by the jet-like behavior of the inflow stream. Dependence of cross-stream mixing on interfacial stability caused the orifice and coefficient to increase from 0.4 to 0.6 as temperature difference increased. This varying orifice coefficient is well represented by the combination of a discharge coefficient for streamline contraction combined with a mixing coefficient which accounts for mixing between the inflow and outflow.

NOMENCLATURE

C_d discharge coefficient
C_m Q_m/Q, interfacial mixing coefficient for counterflowing streams
Fr $\overline{U^i}/(g'H/2)^{0.5}$, densiometric Froude number
Gr $g'H^3/\nu^2$, densiometric Grashof number
g gravitational acceleration, m s^{-2}
g' $g\Delta\rho/\rho_a$, effective gravitational acceleration, m s^{-2}
H door height, m
K door orifice coefficient including viscous effects and cross-stream mixing
P absolute pressure at height z above the interfacial streamline, Pa
Q^i ideal inviscid inflow or outflow at ρ_a in the absence of streamline contraction, flow separation, viscous losses and cross-stream mixing, m^3 s^{-1}
Q gross flow rate including all viscous effects except cross-stream mixing, m^3 s^{-1}
Q_m effective cross-stream mixing rate, m^3 s^{-1}
Q_n net flow rate, m^3 s^{-1}
Re_H U_nH/ν, net flow Reynolds number
T_i indoor temperature, K
T_o outdoor temperature, K
T_a temperature that would produce the average density ρ_a, K
$T(z)$ temperature on centerline of doorway, K
t time after start of opening, s
t_c closing time ($\theta = 90°$ to $\theta = 0°$), s
t_h fully open time (total time that $\theta \geqslant 90°$ is constant), s
t_o opening time ($\theta = 0°$ to $\theta = 90°$), s
$\overline{U^i}$ $Q^i/(WH/2)$, average ideal inviscid outflow or inflow velocity, m s^{-1}
U_i^i local inviscid outflow velocity of indoor air at distance z from interfacial streamline, m s^{-1}
U_o^i local inviscid inflow velocity of outdoor air at distance z from interfacial streamline, m s^{-1}
U_n $Q_n/(WH/2)$, net inflow or outflow average velocity, m s^{-1}
V net total volume exchanged, m^3
V_c net exchange volume during closing time, m^3
V_h net exchange volume during fully open time, m^3
V_o net exchange volume during opening time, m^3
W door opening width, m
W_e time dependent effective width of doorway during opening and closing times, m
z height above the interfacial streamline, m
ΔT indoor–outdoor temperature difference, K
$\Delta\rho$ absolute value of indoor–outdoor density difference, kg m^{-3}
ρ_i indoor density, kg m^{-3}
ρ_o outdoor density, kg m^{-3}
ρ_a average of indoor and outdoor density, kg m^{-3}
ρ_e reference density, kg m^{-3}
θ angular position of door from doorway plane, degrees
ν kinematic air viscosity at temperature T_a, m^2 s^{-1}

Subscripts
o outdoor air conditions in the inflowing stream
i indoor air conditions in the outflowing stream
n net flow after re-entrainment by cross-stream interfacial mixing

Superscripts
i ideal inviscid flow with no streamline contraction, flow separation, or cross-stream interfacial mixing.

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INTRODUCTION

AIR FLOW through open doorways influences air infiltration heat loss, room air circulation patterns and the distribution of indoor air contaminants between rooms in a building. Estimates of doorway flow rates are required for smoke control in fires, tracking indoor air pollutants and the design of clean rooms, hospital operating theatres, paint booths and restaurant kitchens. This flow may be caused by indoor–outdoor temperature differences, occupant motion, wind, door motion, and supply or extraction of air from the interior space by mechanical ventilation. For exterior doorways in residential buildings, flow induced by the indoor–outdoor temperature difference is usually the most important

exchange mechanism, and will be the focus of our investigation.

In this study, a model for buoyancy dominated flow is developed and tested for an exterior doorway using measurements in a test house at the Alberta Home Heating Research Facility. We will show that the effect of the counterflowing streams of warm and cold air on the orifice coefficient can be accounted for by dealing separately with the contributions of streamline contraction and cross-stream mixing. During the opening and closing cycle the swinging door will be approximated as a quasi-steady flow through an orifice by varying width, and the pumping action neglected.

INVISCID FLOW

The counterflowing streams of incoming cold air and outgoing warm air through an open door can be idealized as a pair of inviscid flows driven by pressure forces produced by density differences. Inflow and outflow rates are coupled by the requirement that the total volume inflow to a sealed room from all sources must equal the outflow. While we consider only the case of a heated room, the results for a cooled room would be similar, with flow directions reversed in Figs 1 and 2.

Figure 1 shows a schematic of the velocity and temperature profiles on the vertical centerline at the doorway. The cross-stream mixing in the counterflowing shear layer has two important effects on the air exchange rate. Its most important effect is to change the temperature (or tracer gas concentration) profiles in the two streams. Mixing also reduces the speed of both streams as momentum is conserved during the constant pressure mixing

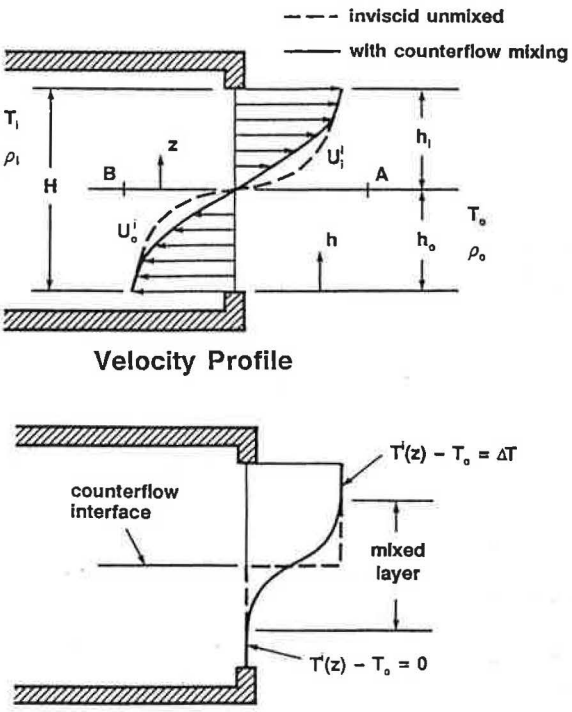

Fig. 1. Inviscid and actual velocity and temperature profiles in the doorway of a sealed room.

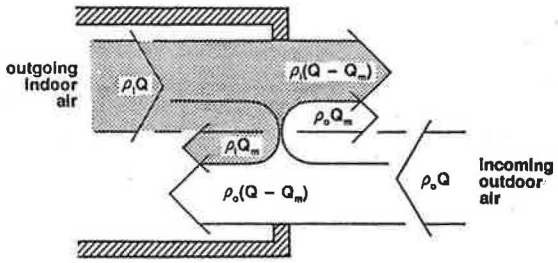

Fig. 2. Reentrainment by cross-stream interfacial mixing between counterflowing streams.

process. Figure 1 shows these effects, both of which act to reduce the net air exchange rate. To deal with streamline contraction, kinetic energy losses and cross-stream mixing, an orifice coefficient will be measured, and applied to the theoretical ideal inviscid flow. To determine this inviscid flow rate we assume:

- There is no mixing or heat transfer between the incoming flow with density ρ_o, temperature T_o, and the outgoing flow with ρ_i and T_i
- Air infiltration from other leakage sites in the room is negligible, and there is no mechanical supply or exhaust ventilation
- The flow is steady, and with no supply or exhaust ventilation the indoor and outdoor pressures are in equilibrium along the interfacial streamline, so that $P_A = P_B$ in Fig. 1
- Streamlines are straight and parallel at the door orifice, with only hydrostatic pressure differences acting across them. These pressures are assumed to be in equilibrium with the outside pressure in the outgoing flow, and with the inside pressure in the incoming flow.

This last assumption, that incoming air is immediately in pressure equilibrium with indoor air, is reasonable if the room is wider than the door, so that undisturbed indoor air lies on each side of the inflowing stream, causing the incoming air to enter the room as a jet.

It is important to note that approximating the counterflowing streams as emerging jets will be valid only when the room or entryway is significantly wider than the doorway. For the case of a sliding door that completely spans the width of a hallway, a lock-exchange flow would occur, with hydrostatic pressure in the inflow set by its own density ρ_o rather than the density ρ_i of the room air. As described by Benjamin [1], the inflow rate in a lock exchange is controlled by a momentum balance between both streams rather than just the incoming stream. Our experiments show than an emerging jet approximation is appropriate for both an open room and a narrow entryway, and only the jet model will be developed here.

For the incoming flow, all streamlines originate in a stagnant outdoor reservoir at pressure P_A, so the Bernoulli equation for an inviscid streamline in the inflow of outdoor air passing through the door at elevation z is:

$$\frac{P(z)}{\rho_o} + \frac{U_o^{i^2}}{2} + gz = \frac{P_A}{\rho_o}, \tag{1}$$

where P_A/ρ_o is the Bernoulli constant of all inflow stream-

lines. The pressure $P(z)$ in the incoming jet is set by the hydrostatic variation inside the room:

$$P(z) = P_B - \rho_i g z. \tag{2}$$

Using the assumption of indoor–outdoor pressure equilibrium $P_A = P_B$, (1) and (2) may be combined to find the inviscid ideal velocity profile:

$$U_o^i = \left[2g\frac{\Delta\rho}{\rho_o}z\right]^{0.5}, \tag{3}$$

where $\Delta\rho = |\rho_o - \rho_i|$. Integrating this velocity profile over inflow height h_o to find the volume flow:

$$Q_o^i = \frac{2W}{3}h_o^{1.5}\left[\frac{2g\Delta\rho}{\rho_o}\right]^{0.5}, \tag{4}$$

which is identical to the inviscid flow over a ventilated rectangular weir. The inviscid outflow rate Q_i^i and velocity U_i^i are derived in the same way with ρ_o and ρ_i interchanged:

$$U_i^i = \left[2g\frac{\Delta\rho}{\rho_i}z\right]^{0.5}, \tag{5}$$

$$Q_i^i = \frac{2W}{3}h_i^{1.5}\left[\frac{2g\Delta\rho}{\rho_i}\right]^{0.5}. \tag{6}$$

The velocity profiles are not symmetric, but differ only by the constant factor $(\rho_i/\rho_o)^{0.5}$.

In the absence of mechanical ventilation and other openings, the constant room volume forces the volume inflow and outflow rates to be equal. For a sealed room filling with density driven exchange, it is volume and not mass flow rates that are equal in the inflow and outflow streams. The density driven inflow displaces the air in the room without mixing with it, and pressure equilibrium forces an equal volume of outflow to occur. This lack of mixing between the inflowing outdoor air and the indoor air remaining in the room is a reasonable assumption for temperature differences larger than a few degrees, and for no mechanical mixing by fans in the room. Equating (4) and (6) determines the ratio of inflow and outflow heights in Fig. 1:

$$\frac{h_o}{h_i} = \left[\frac{\rho_o}{\rho_i}\right]^{1/3}. \tag{7}$$

For a total door height of $H = h_i + h_o$, (7) can be written as:

$$h_i = \frac{H}{1+(\rho_o/\rho_i)^{1/3}}. \tag{8}$$

Using (8) in (6) gives the inviscid inflow or outflow $Q^i = Q_i^i = Q_o^i$ as:

$$Q^i = \frac{W}{3}\left[g\frac{\Delta\rho}{\rho_e}H^3\right]^{0.5}, \tag{9}$$

where the effective density ρ_e is:

$$\rho_e = \rho_i\frac{(1+(\rho_o/\rho_i)^{1/3})^3}{8}. \tag{10}$$

Replacing this effective density ρ_e with the average $\rho_a = (\rho_o+\rho_i)/2$ causes an error of only 0.1% in Q^i at a 40 K indoor–outdoor temperature difference, and ρ_a will be used to approximate ρ_e in the rest of our analysis.

This result justifies the arbitrary assumption by Brown and Solvason [2], Shaw [3], Shaw and Whyte [4] and others that a constant density ρ_a may be used to normalize $g\Delta\rho/\rho$. Their analyses also set the dividing streamline at the door mid-height $h_i = H/2$, rather than allowing it to be determined by the constant room volume condition that leads to (8). Equation (8) predicts $h_i = H/2.05$ for a 40 K temperature difference confirming that $h_i \approx H/2$ is a reasonable assumption.

VISCOSITY AND INTERFACIAL MIXING

Discharge coefficient

The ideal inviscid flow rate Q^i does not account for streamline contraction, flow separation or viscous losses through the orifice. These factors are included in the discharge coefficient $C_d = Q/Q^i$, so that the actual inflow or outflow rate $Q = Q_i = Q_o$ is, with no cross-stream interfacial mixing:

$$Q = C_d\frac{W}{3}(g'H^3)^{0.5}, \tag{11}$$

where the density difference in (9) is now expressed as the effective acceleration of gravity $g' = g\Delta\rho/\rho_a$. Shaw and Whyte [4] obtain the same result. The discharge coefficient for a sharp-edged orifice or weir is about $C_d \approx 0.60$, with streamline contraction as the dominant factor.

In practice, the actual inflow and outflow will not have the symmetry shown in Figs 1 and 2. The floor of the room will keep the incoming flow horizontal, while the warm outflow will rapidly bend over and rise as a buoyant plume. A window, with no floor to keep the streamlines horizontal may have a discharge coefficient that is significantly different than a doorway. Here, we neglect asymmetry and include this effect in the discharge coefficient.

Mixing coefficient

While the discharge coefficient gives the gross inflow or outflow rate, it does not tell us what fraction of the inflowing air is indoor air that has been carried outside by Q_i, re-entrained by Q_o and brought back inside, as shown schematically in Fig. 2. It is this re-entrainment by interfacial mixing that is the distinguishing feature of the counterflowing streams through a doorway, and that produces the gradual variation in temperature through the mixing layer shown in Fig. 1.

Mixing between incoming and outgoing streams also produces a reduction in the velocity of the incoming air by a transfer of outgoing momentum to the counterflowing incoming stream. The same effect occurs in the outgoing stream, and it is this cross-stream momentum exchange that produces most of the difference between the inviscid U^i and actual U velocity profiles shown in Fig. 1.

The net flow rate Q_n which results after the combined effects of re-entrainment and momentum transfer may be expressed in terms of an effective cross-stream mixing flow rate Q_m, so that $Q_n = (Q - Q_m)$, as shown in Fig. 2.

Defining an interfacial mixing coefficient $C_m = Q_m/Q$, the net flow rate must be:

$$Q_n = (1 - C_m)Q. \tag{12}$$

Using this, the doorway orifice coefficient K is:

$$K = C_d(1 - C_m), \tag{13}$$

and, (7) becomes:

$$Q_n = \frac{K}{3} W(g'H^3)^{0.5}. \tag{14}$$

The lower limit of $C_m = 0$ occurs when there is no interfacial mixing, at the two extremes of very large and very small densiometric Grashof number $Gr = g'H^3/\nu^2$. When the Grashof number is small, mixing is suppressed by viscous dissipation and the counterflows are laminar. At large values of the Grashof number, density differences suppress the turbulence generated by counterflow shear, and C_m is again forced to zero.

The largest values of C_m will occur when cross-stream mixing is the greatest, in a range of Grashof number high enough to allow complete transition to turbulence, but low enough to allow turbulence production by counterflow velocity shear to overcome suppression of turbulence by the stable density gradient between the streams. Under these fully turbulent conditions, mixing will be complete if the streams are in contact over a sufficient distance in the flow direction, and both streams will be mixed to the average density ρ_a. It is easy to show (Fig. 2) that a mixing flow $Q_m = Q/2$ is required to force each stream from ρ_o or ρ_i to the average ρ_a. This sets the upper limit at $C_m = 0.5$ for complete cross-stream mixing.

FLOW CONTROL BY THE DOORWAY

The inviscid equations for inflow and outflow rates are identical to open channel flow over a ventilated sharp edged weir, with the free liquid surface replaced by the counterflow interface. In the same way that a weir provides the control condition which sets the flow rate in an open channel, or a sonic orifice produces choked flow of a compressible fluid, a doorway acts to govern the flow rate into a sealed room. Because flow in the doorway is choked, there will be no effect of room geometry on flow rate until the inflow moves across the floor, strikes an interior wall, and is reflected back to the doorway. The speed of travel of this cold air current has been predicted and measured by Kiel and Wilson [5] and Lane-Serff and Linden [6]. Their results can be used to estimate the time for which the quasi-steady solution in (14) will apply. In most practical situations, the door opening/closing cycle is less than the reflection return time.

For free surface flows the doorway control condition sets a critical value for the inviscid densiometric Froude number:

$$Fr = \frac{\overline{U^i}}{\left(g'\dfrac{H}{2}\right)^{0.5}}, \tag{15}$$

where $\overline{U^i} = Q^i/(WH/2)$ is the average inviscid velocity of the uncontracted gross flow from (9). Using (9) in this definition, the inviscid Froude number is always equal to its critical value of $Fr = 2^{1.5}/3 = 0.94$ in the emerging jet from a doorway.

When the doorway orifice is about the same width as the room, it may no longer control the flow. For the case of a long corridor with a full width door, the flow behaves like a lock exchange, with symmetric gravity currents moving in opposite directions when the gate separating the two fluids is suddenly removed. Because corridor counterflow does not have quiescent air on either side, the hydrostatic pressure variation in the incoming air is set by its own density ρ_o rather than the indoor static pressure, which is governed by ρ_i. As a result, the inviscid inflow and outflow velocities in a corridor are uniform, rather than having the parabolic profile of (3) in an emerging jet from a doorway. For lock exchange Benjamin [1] found that $Fr = (1/2)^{0.5} = 0.71$. The lower Froude number of 0.71 for flow along a corridor compared to 0.94 for weir flow through a doorway orifice implies that doorway flow rates should be larger. However, streamline contraction causes a door orifice to have a discharge coefficient of about $C_d \simeq 0.6$ while channel flow through a full-width door in a corridor has $C_d \simeq 1.0$, making the net flow rate somewhat smaller for the door orifice control condition. In the present study, room widths of two and seven times the door width were tested to examine the control condition.

EXCHANGE DURING AN OPENING AND CLOSING CYCLE

Equation (14) may now be used to answer the practical question of how much air is exchanged during a door opening and closing cycle. A typical door cycle can be divided into three periods: an opening period during which the door moves from the closed position to its 90° open position, a fully open period during which the door is open 90° (or more) and finally, a closing period as the door moves from 90° open to the closed position. The total air exchange that would occur during the fully open period of duration t_h, can be determined using (14) with $V_h = Q_n t_h$.

The passage of an occupant through an exterior door is often so rapid that the door is in the fully open position for only an instant. In this case, a significant portion of the total buoyancy driven exchange occurs while the door is opening and closing. An estimate of this buoyancy-driven exchange during opening and closing may be made by assuming quasi-steady flow. Integrating (14) for the period when the door is moving from a closed position to a 90° open position:

$$V_o = \frac{K}{3}(g'H^3)^{0.5} \int_0^{t_o} W_e \, dt. \tag{16}$$

For a swinging door, the effective orifice width W_e is $W \sin\theta$ where θ is the angular position of the door measured from the plane of the doorway. Assuming that the orifice coefficient is constant, despite the changing orifice shape, and that the door velocity is constant, the total buoyancy-driven exchange volume from (16) is $V_o = 2Q_n t_o/\pi$. For identical opening and closing motions, the exchange during the opening, fully open hold and closing cycle is:

$$V = Q_n(t_h + 4t_o/\pi). \quad (17)$$

The factor of $4/\pi$ in (17) results from the assumption of constant angular swing velocity. A constant linear velocity sliding door would have a factor of 0.5 rather than $2/\pi$.

The added exchange caused by the pumping action of the swinging door has been measured by Kiel and Wilson [7] using the data from the present study, combined with 1 : 20 scale model simulations using salt–water mixtures. These experiments showed that the additional pumped air volume was linearly proportional to swing speed, and inversely proportional to density difference $\Delta\rho/\rho$.

EXPERIMENTAL STUDY

Field test facility

Air flow measurements were made using a test house at the Alberta Home Heating Research Facility, described in [8]. This test house is a detached single level wood frame structure with floor dimensions of 6.5 × 7.1 m and an interior wall height of 2.4 m, as shown in Fig. 3. The stairway to the full basement, and all floor openings were sealed to isolate the level in which the door was located. The interior of the house was a single large room with an interior volume of 106.7 m^3 (after deducting the volume of the investigator and the measuring equipment). A computer-controlled motor driven door opener was used to govern the motion of the 2.06 m high and 0.91 m wide exterior door which was located near one end of a 6.5 m long wall. The swing times t_o and t_c to open or close the door were maintained constant at 3.75 s. The fully open hold time t_h was varied from 0.5 to 120 s to vary the total volume exchanged.

The effect of occupant passage through the doorway was not investigated. A moving occupant may increase exchange by sucking outside air in his or her wake, or reduce the net exchange by blocking the doorway flow area, or by increasing cross-stream mixing by wake turbulence. Shaw [9] found that a person walking through an open door induced an exchange volume that varied from 0.29 m^3 for a fast walk to 0.087 m^3 for a slow walk. Tamura [10] measured orifice coefficients with a person standing in an open doorway. This reduced the orifice coefficient for unidirectional flow by about 12 to 15%, equivalent to a flow area reduction of about 0.25 m^2 caused by the blockage.

Short walls and doorframes, shown in Fig. 3, were located inside the house to allow various entryway configurations. For this study, two interior configurations were used. The full-house geometry was formed by leaving both the interior doorways open, so that incoming flow had minimal obstruction. In the hall geometry the interior doorway on the side was sealed and an interior partition constructed which channeled the flow through the 0.94 m wide interior door into a corridor 7.11 m long and 1.26 m wide that narrowed to 1.02 m.

The exterior doorway was sheltered from wind by the wall of an identical adjacent house 2.5 m away. Plastic sheets, shown in Fig. 3, were used as windbreaks to seal the 2.5 m wide gaps at the end of the passage, to form a stagnant outside reservoir with a plan area of 2.5 × 7.2 m, open at the top. In spite of these precautions, measurements will show that wind was able to exert a significant influence on the doorway flow by changing the turbulence level in this sheltered reservoir of outdoor air.

Measurement techniques

Air exchange through the exterior doorway was measured using sulphur hexafluoride as a tracer gas. With the door closed, and the indoor temperature at about 295 K, tracer gas was added to the room to bring the concentration level of SF_6 to about 5 ppm using mixing fans to produce a uniform concentration within the single large room. The tracer gas concentration was measured using a computer-controlled infrared absorption gas analyser. The decay rate of this concentration was monitored for several minutes to determine the natural air infiltration rate through the envelope of the structure.

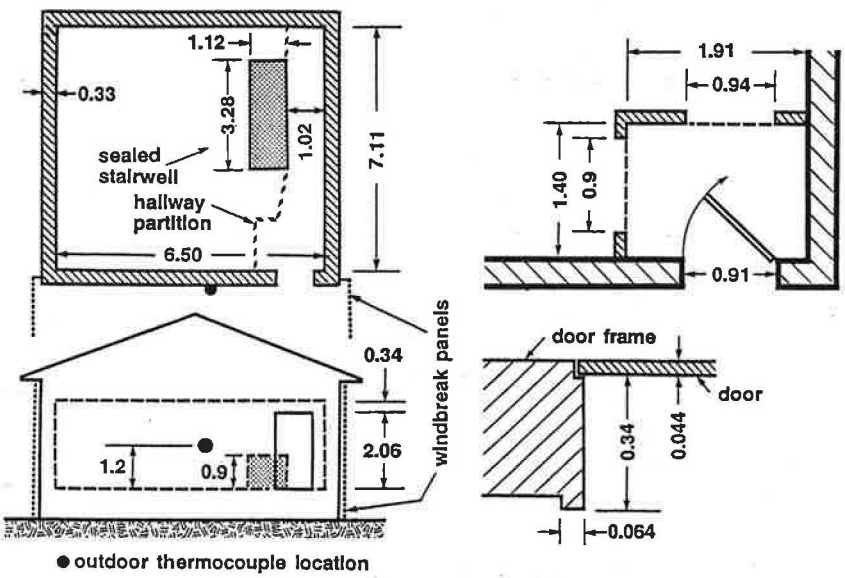

Fig. 3. Dimensions of the test house (m).

Then, the mixing fans were turned off to allow the indoor turbulence to decay, after which the outside door was opened at a constant swing speed, held open for time interval t_h and then closed with the same swing speed. After the door was closed, the fans were again turned on and the internal air mixed for a few minutes to produce a uniform concentration. This concentration was monitored for several minutes to give a second value for the natural infiltration rate.

Using the known internal volume of the single room house, the change in concentration during a door opening cycle, and the natural infiltration rate, the net volume of air exchanged was computed. At a typical test with an indoor–outdoor temperature difference of 30 K, about 10% of the house volume was exchanged during a 20 s opening–closing cycle time. During this period, natural infiltration through a leakage area of about 0.01 m^2 distributed over the building envelope was small compared to flow through the 1.9 m^2 doorway area. Corrections were made to the door flow rate to account for this infiltration.

Steady flow occurred shortly after the door reached its fully open position, allowing the steady state flow rate through the opening to be estimated from the slope of the volume exchanged versus open time curve. The period of steady state flow terminated when the incoming flow of cold outdoor air struck the walls of the room and transmitted a reflected wave back to the opening, altering the flow rate. For all the tests reported here, the space into which the density current flowed was large enough and times short enough that the effect of wave reflection could be neglected.

Temperature profiles were measured on the vertical centerline of the doorway opening to observe the effect of cross-stream mixing. A single vertical row of copper-constantan thermocouples spaced 0.1 m apart were sampled rapidly during the air exchange period, and then time averaged to obtain the temperature distribution across the counterflowing streams.

MEASURED EXCHANGE FLOWS

Orifice coefficient

Typical tracer gas exchanges are shown in Fig. 4. For each set of tests, the net exchange volume was correlated with opening time to determine the steady state flow rate. The good linear fit shown in Fig. 4 is typical of most of the experimental results, and confirms that the fully open flow was steady.

The orifice coefficient was computed by rearranging (14) to solve for K. These orifice coefficients are shown in Fig. 5 for two differing geometries, the full house interior and the long hallway connected to an entryway shown in Fig. 3. Figure 5 includes only measurements where the wind speed was less than 10 km h^{-1}, where mixing between the inflowing and outflowing streams should be dominated by flow-generated turbulence. The measured orifice coefficient decreases from about $K = 0.6$ at large temperature differences to about $K = 0.4$ at small temperature differences. The best fit linear regression is:

$$K = 0.400 + 0.0045\,\Delta T. \tag{18}$$

Data obtained by Fritzsche and Lilienblum [11] from tests on a 2.5 m high and 1.8 m wide cold room door are also shown in Fig. 5. Their orifice coefficient was based on net flow rate determined from spatially integrated temperature and velocity measurements, and was correlated by:

$$K = 0.476 + 0.0043\,\Delta T. \tag{19}$$

Fritzsche and Lilienblum's cold room orifice coefficients [11] have the same slope with ΔT, but define an upper bound to our data. This disagreement may be caused by differences in ambient turbulence levels in the two cases. Fritzsche and Lilienblum's experiments were conducted in an indoor laboratory, where there was little ambient turbulence. In the present study, high levels of outdoor turbulence appear to increase interfacial mixing, and are the probable cause of our lower K values.

By measuring the velocity profile on the vertical mid-plane of the doorway, van der Maas *et al.* [12] found a velocity coefficient of 0.75 for stratified room air with a temperature difference ΔT of 1.5°C at the floor and 3.6°C at the top of a 1.4 m high doorway. Their theoretical flow rate included the effect of this linear temperature stratification. Because they measured only velocity, their flow coefficient corresponds to the ratio of mixed to unmixed average center plane velocities (Fig. 1). Making the crude assumption of uniform mixing across the entire outflow, it is easy to show by a momentum balance that this velocity coefficient is equal to $(1-2C_m)^{0.5}$, so that $C_m = 0.22$ for their flow. For a discharge coefficient of

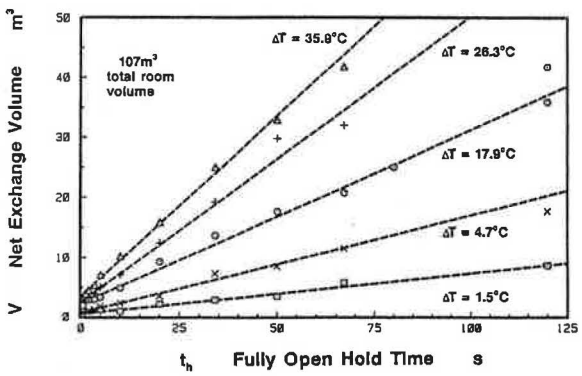

Fig. 4. Total net exchange volumes for door swing times $t_o = t_c = 3.75$ s.

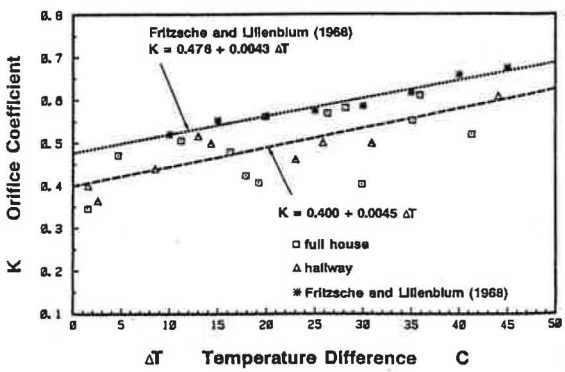

Fig. 5. Measured orifice coefficient for full-house and hallway configurations for wind speed less than 20 km h^{-1}.

0.60 this corresponds to $K = 0.47$, compared to $K = 0.41$ from (18) for the present study, and $K = 0.49$ from Fritzsche and Lilienblum [11] using (19).

Riffat [13] attempted to measure the orifice coefficient of a doorway connecting two zones within a house by using two simultaneous tracer gas decays. The measurements show K decreasing from 0.60 at $\Delta T = 0.5$ K to 0.26 at $\Delta T = 17$ K. This is opposite to the trend observed by other investigators, and in the present study, which predicts K increasing from 0.40 to 0.48, and suggests that errors caused by other gravity driven leakage paths may have been dominant in his study.

Cross-stream mixing and temperature profiles

An important question is whether the observed variation in orifice coefficient is due to a variation in discharge coefficient, mixing coefficient or both. One way to answer this question is to measure the actual velocity profile, integrate, and determine only the discharge coefficient. Shaw [9] examined gross flow rates in this manner for ΔT from 0 to 10 K and found that the discharge coefficient was nearly constant at 0.62 for ΔT values between 3 and 10 K. If $\Delta T < 3$ K mechanical ventilation caused errors which increased Shaw's measured orifice coefficients.

Another method of determining C_d is to conduct experiments at low Grashof and Reynolds numbers in order to suppress mixing. Data from salt–water scale model experiments conducted by Kiel and Wilson [7] show that under low Reynolds number conditions the discharge coefficient is constant at 0.60 over a range of density ratios equivalent to air temperature differences of 1–45 K. Negligible cross-stream mixing was observed in these experiments, further supporting the idea that cross-stream interfacial mixing is responsible for decreasing the full scale orifice coefficient at small ΔT. To further confirm this hypothesis, temperature profiles from the air exchange tests were examined.

Figure 6 shows three normalized temperature profiles measured on the vertical centerline of the opening. Each profile was determined from a 10 s time average after the door was fully open and the flow was steady. The profile associated with the largest temperature difference of 28.8°C shows that there is almost no entrainment of the outflowing air into the inflowing stream. In contrast, the upper portion of this temperature profile indicates significant entrainment of outdoor air into the outflowing stream. This skewed temperature profile can be explained by the fact that the counter-flow streams inside the house are in contact over a large interfacial area, while outside the house the buoyant outflow rises rapidly upon exit from the door, limiting the exterior interfacial contact area. This result emphasises the importance of the downstream flow conditions on cross stream mixing. In contrast, the coefficient of discharge $C_d = 0.6$ agrees well with values for unidirectional orifice flow, indicating that streamline contraction is relatively insensitive to downstream conditions.

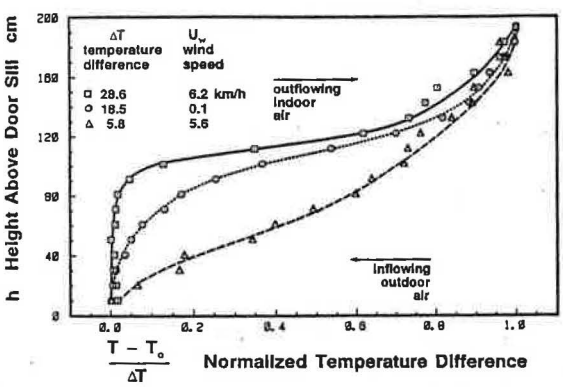

Fig. 6. Temperature profiles on the doorway vertical centerlines.

It is apparent from temperature profiles at small ΔT that a significant amount of warm indoor air was entrained by the inflowing outdoor air. In contrast, less interfacial mixing occurred inside the house, as indicated by the shift toward a normalised temperature difference of 1.0 in the upper region. This is probably due to increased mixing outside the door, producing modified inflow velocity and temperature profiles with a more stable interface. At these small temperature differences the stabilising effect of the buoyancy difference is less, increasing the ability of wind-generated turbulence to promote cross-stream mixing outside the house.

The mixing coefficient can vary from $C_m = 0$, when no cross-stream mixing occurs, to $C_m = 0.5$ when complete mixing produces a uniform density ρ_a across the doorway. At this upper limit, the temperature of each stream is the average of indoor and outdoor temperatures and the outgoing stream is half indoor air and half outdoor air. For a discharge coefficient of $C_d = 0.60$ the lower bound on orifice coefficient is $K = 0.30$. The measured values of K in Fig. 5 indicate a lower limit of $K \simeq 0.40$ for small ΔT, which implies $C_m \simeq 0.33$. The upper limit of $C_m = 0.5$ is unlikely to be observed in practical situations because it requires complete mixing over the entire door height.

Reynolds number correlation for orifice coefficient

Direct correlation of K with ΔT in (18) is simple, and allows the net flow rate Q_n to be computed directly from (14) without iteration. However, (18) is limited to a single fluid, ambient air, and to a range of temperature differences from about 1 to 45 K. To extend these measurements to other fluids and density differences, K was correlated as a function of net flow Reynolds number:

$$Re_H = \frac{U_n H}{\nu}, \tag{20}$$

where $U_n = Q_n/(0.5\ WH)$ is the average net inflow or outflow velocity, including the effects of streamline contraction, orifice flow separation and interfacial mixing between the counterflowing streams. The empirical equation:

$$\frac{K-0.3}{0.6-K} = 3.0 \times 10^{-5} Re_H, \tag{21}$$

used in Fig. 7 produces the correct asymptotic values of $C_d = 0.6$, $C_m = 0.5$ at low Reynolds number and $C_d = 0.6$, $C_m = 0$ at high Reynolds number.

The correlation in (21) should be used with caution below a lower limit of $Re_H < 10^4$, set by the experiments. It is expected that at some critical Reynolds number less

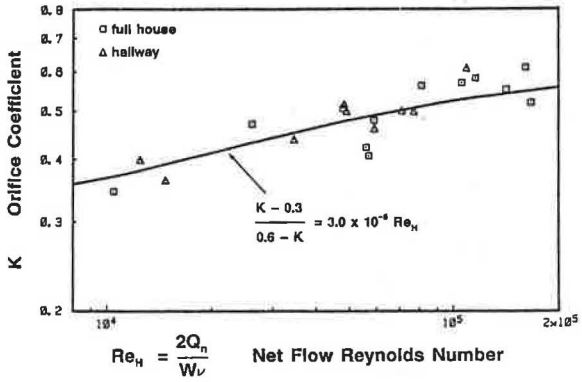

Fig. 7. Variation of orifice coefficients with Reynolds number at wind speeds less than 10 km h^{-1}.

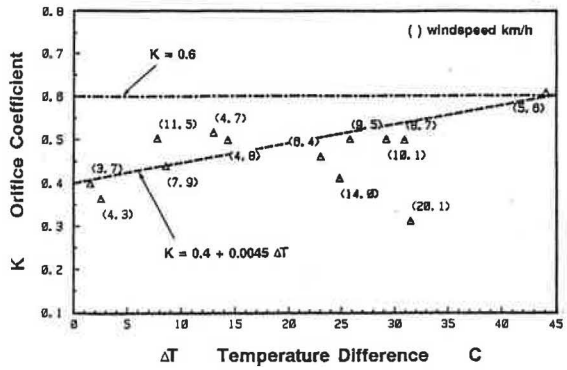

Fig. 9. Effect of wind speed on hallway orifice coefficient.

than 10^4 viscous dissipation will force the counterflow to become laminar, causing K to suddenly increase from near 0.3 to 0.6 as the mixing coefficient changes from its fully mixed lower limit of $C_m = 0.5$ to a laminar value of $C_m = 0$.

The correlation of orifice coefficient with Re_H as given in (21) implies that cross-stream mixing rate and mixed layer thickness vary linearly with opening height. Because only one door height was tested in this study it is not possible to confirm this hypothesis. Caution should be exercised when applying these correlations to openings which are much larger or much smaller than a conventional residential doorway.

Effect of interior partitions

The orifice coefficients in Fig. 7 were found to be the same for both the long hallway and full house configurations. This result might be expected, because the 0.94 m wide exterior doorway provides the flow control condition in both cases. What is more important, is that insensitivity of flow rate to interior room geometry implies that the cross-stream mixing coefficient C_m is the same for an open room and a long hall. This suggests that flow conditions immediately inside and outside the doorway are the major determining factors in interfacial mixing, a result that greatly simplifies the task of predicting steady flow rates through doorways.

Influence of outdoor turbulence

The measured orifice coefficients in both configurations are influenced by windspeed, as shown by Figs 8 and 9. At small ΔT, windspeeds of 20 km h^{-1} measured 10 m above ground produced flow rates through a sheltered doorway that were about 25% smaller than calm conditions. For both the full house (Fig. 8) and the hallway (Fig. 9) configurations the data scatter increases with wind speed, particularly for small temperature differences. While changing wind direction may have obscured some of the trend, windspeeds above 10 km h^{-1} appear to reduce K at a fixed ΔT. The influence of wind speed was surprising, partly because care had been taken to shelter the doorway with windbreaks, but mostly because increasing windspeeds tended to produce a decrease in the net doorway flow rate. At first this decrease was puzzling, until we realized that for a house with a sheltered outside opening, the most important effect of wind is to increase the intensity of outdoor turbulence, which will increase interfacial mixing between incoming and outgoing airstreams. This mixing lowers the net flow rate by increasing C_m, which reduces the orifice coefficient K in (13).

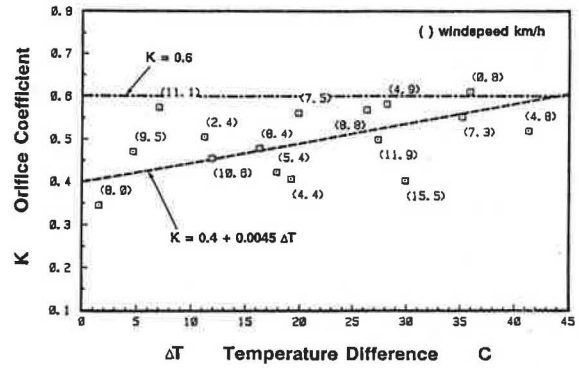

Fig. 8. Effect of wind speed on full house orifice coefficient.

Door swing pumping

Having determined the variation in the overall orifice coefficient, attention may now be directed at the air exchange that occurs while the door is swinging open and closed. Setting the fully open time $t_h = 0$ in (17) yields a quasi-steady flow estimate of the buoyancy driven component of the exchange volume during opening and closing. Figure 10 shows that this quasi-steady approximation accurately predicts total exchange volumes when ΔT exceeds about 4 K. At small values of ΔT the pumping effect of the swinging door makes a significant con-

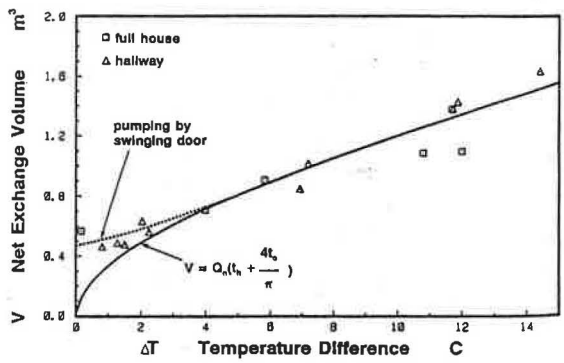

Fig. 10. Effect of door swing cycle on the quasi-steady flow approximation for short hold times $t_h = 0.55$, $t_o = t_c = 3.75$ s.

tribution to the total exchange. Kiel and Wilson [7] found that the volume pumped by the opening and closing cycle at $\Delta T = 0$ is about 0.9 m^3, for $t_o = t_c = 3.75$ s, about 30% of the volume swept by the opening and closing door. For $\Delta T = 1$ K, this pumped volume is equivalent to the steady flow through the fully open door for a hold time of about 15 s.

CONCLUSIONS

Measurements of flow rates and temperature profiles in density driven counterflow in an open door, combined with a simple quasi-steady inviscid flow model lead to the following conclusions.

(1) The measured orifice coefficients in this study, and the velocity profiles of Fritzsche and Lilienblum [11] support the assumption that the flow control condition at the doorway is set by jet-like behaviour of the inflowing stream, with hydrostatic pressure set by room air density rather than the density of the inflowing outdoor air. This jet-like behaviour produces a critical Froude number of 0.94, rather than the value of 0.71 predicted for counterflowing streams in a corridor with a full width door. Measured orifice coefficients showed that the assumption of jet-like inflow behaviour was valid for an entryway hall that was only twice as wide as the doorway.

(2) The theoretical inviscid flow model showed that a large density difference between incoming and outgoing streams has a negligible effect on the unmixed flowrate if the average density is used to normalize buoyancy forces.

(3) The study identified separate discharge and cross-stream mixing coefficients which combine to form the orifice coefficient K for counterflowing streams. Measurements suggest that the discharge coefficient C_d remains relatively constant at the usual unidirectional flow value of 0.6. The mixing coefficient C_m ranged from zero to 0.33 depending on the degree of cross-stream mixing, which decreases with increasing density difference between the streams. These results are consistent with the allowable range $0 \leqslant C_m \leqslant 0.5$ set by the unmixed and completely mixed limits. The incomplete mixing observed at small ΔT may be caused by the short contact length of the counterflow interfaces.

(4) Measured orifice coefficients K were found to be the same for a long hallway as for a large open room. Because the hallway allows a large interfacial area for cross-stream mixing, while the open room has only a small mixing region near the doorway, this result led us to conclude that most cross-stream mixing caused by interfacial shear occurs near the doorway opening.

(5) Buoyancy driven air exchange that occurs while the door is opening and closing can be accurately estimated by assuming quasi-steady flow at each position of the swinging door, and integrating the steady flow equation with time-varying door orifice width. Buoyancy driven exchange completely dominates door swing pumping for temperature differences larger than 4 K, at the 3.75 s door swing time.

(6) Net flow rates decreased significantly at higher wind speeds. This unexpected result appeared to be caused by an increase in wind induced ambient turbulence in the sheltered reservoir outside the house. This higher level of turbulence caused increased cross-stream mixing which entrained some of the outgoing air back into the incoming stream, reducing the net volume exchanged.

The present study provides a set of practical equations for estimating the flow through open doors. Further measurements are needed to determine whether the mixing coefficient should scale with the entire door height, or be limited to a thin interfacial mixing layer between the counterflowing streams. In addition, the observation that cross-stream mixing can be significantly altered by ambient turbulence suggests that experimental studies and theoretical models should attempt to account for the amount of room and outdoor turbulence in determining counter-flow orifice coefficients.

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