INfiltration of AIR INTO COLD STORES

Q.T. PHAM
D.W. OLIVER
Meat Industry Research Institute of New Zealand (Inc.),
Hamilton (New Zealand)

1. INTRODUCTION

Infiltration of warm air is a major factor in the design, operation and performance of cold stores. It can constitute more than half the total heat load, it is usually the main source of frost forming on coils, and it causes temperature and humidity fluctuations that can badly affect product quality.

Tamm /1/ derived the following theoretical equation to predict the air interchange rate, \( Q \) (m\(^3\)/s), through a door between a warm area and a cold area:

\[
Q = \frac{2wh}{3} \sqrt{\frac{2gh(1-s)}{(1+s^{1/3})^{3}}} 
\]  

(1)

where \( w \) is the door width (m), \( h \) the door height (m), \( g \) the gravitational acceleration (m/s\(^2\)) and \( s \) the ratio of warm air density to cold air density. He also gave expressions for the air velocity profile. Longdill and Wyborn /2/ verified these equations for a 1.2 m wide by 1.6 m high doorway in a 177 m\(^3\) test room and also reported on the performances of various types of air curtains. However, no test results on full-scale cold stores under realistic operating conditions have been published.

Such tests are now possible owing to recent developments in tracer-gas methods. In these methods a gas is injected into a room, and provided that it is well mixed with the air, the rate at which its concentration, \( c \), decreases with time can be related to the rate of air change, \( r \), by:

\[
\ln \left( \frac{c}{c_0} \right) = -rt
\]  

(2)

where \( c_0 \) is the initial concentration and \( t \) is the time after injection.

This paper reports some results from both tracer tests and direct velocity measurements on cold store doors.

2. AIR VELOCITY ACROSS COLD STORE DOORS

To verify Tamm's equations, air velocities across cold-store doors were measured with an "Airflow Development" vane anemometer precalibrated in a wind tunnel. For each door, vertical velocity profiles were measured at three to five sections and the measurements repeated one to four times. Six doors were tested, with door sizes ranging from \( w \times h = 1.08 \times 1.98 \) m to \( 3.0 \times 3.6 \) m, room volumes from 177 to 37 000 m\(^3\), room heights from 3.8 to 21.2 m and temperature differences from 19.5 to 34°C.

A typical vertical air velocity profile is shown in Figure 1, together with Tamm's prediction. Measured values are lower than predicted and the lower half of the measured profile is more flattened than the theoretical; i.e., velocity does not increase as the floor is approached, indicating that frictional effects become important near the floor. This, and the presence of obstructions (product stacks) in the store, may explain the discrepancy between theory and measurement.

Table 1 summarizes the test conditions and results for total air interchange. Figure 2 shows that Tamm's equation consistently overpredicts the total air interchange (calculated by averaging the inflow and the outflow) by 46%. Thus, the right-hand-side of eq. (1) should be multiplied by an empirical factor 0.68 ± 0.04 (factor ± s.d.). We will call the resulting expression Tamm's modified equation.
Fig. 1 - Air velocity profile for a 3m-high door.
(Inside temperature -16.5°C, outside temperature 13.5°C)

Table 1. Air interchanges across cold store doors

<table>
<thead>
<tr>
<th>Store height, m</th>
<th>Door width, m</th>
<th>Door height, m</th>
<th>Inside temperature, °C</th>
<th>Outside temperature, °C</th>
<th>Measured interchange, m^3/s</th>
<th>Interchange from eq. (1), m^3/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.4</td>
<td>3.05</td>
<td>2.75</td>
<td>-16.5</td>
<td>13.5</td>
<td>3.22</td>
<td>4.91</td>
</tr>
<tr>
<td>8.4</td>
<td>3.05</td>
<td>2.75</td>
<td>-16.5</td>
<td>13.5</td>
<td>3.24</td>
<td>4.91</td>
</tr>
<tr>
<td>5.1</td>
<td>2.70</td>
<td>2.90</td>
<td>-18.0</td>
<td>15.0</td>
<td>3.41</td>
<td>4.86</td>
</tr>
<tr>
<td>6.5</td>
<td>3.00</td>
<td>3.60</td>
<td>-17.0</td>
<td>7.0</td>
<td>4.33</td>
<td>6.40</td>
</tr>
<tr>
<td>21.2</td>
<td>1.77</td>
<td>2.13</td>
<td>-20.0</td>
<td>14.0</td>
<td>1.22</td>
<td>2.04</td>
</tr>
<tr>
<td>3.8</td>
<td>1.08</td>
<td>1.98</td>
<td>-4.5</td>
<td>15.0</td>
<td>0.63</td>
<td>0.83</td>
</tr>
</tbody>
</table>

Fig. 2 - Measured vs predicted air interchanges across open cold-store doors.
3. CONTROLLED TRACER TESTS

**Experimental:** Sulfur hexafluoride tracer was used in the tracer tests. (Details of the sample collection and analysis procedures are given in another paper /3/.) In each test the tracer gas was released into the store, all the doors were closed and the recirculation fans started, enough time was given the tracer to disperse evenly (15 - 60 min depending on the store), and air samples were taken at the four corners of the room. The door being tested was then opened for a specified period, closed again, and 15-60 minutes later four more samples were taken. The total air change was then calculated according to eq. (2). Table 2 summarizes the ranges of test conditions used.

<table>
<thead>
<tr>
<th>Store description</th>
<th>A, B, C, D: conventional palletized stores with gross volumes 4000, 5600, 11 000 and 38 000 m³ respectively; E: high-rise (21.2 m) store with racking, 12 300 m³.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Door situation</td>
<td>enclosed (by a loading area) or exposed.</td>
</tr>
<tr>
<td>Door protection:</td>
<td>none, air curtain (horizontal or vertical), plastic strip curtain, air and plastic strip curtains.</td>
</tr>
<tr>
<td>Forklift traffic:</td>
<td>none or one pass/minute.</td>
</tr>
<tr>
<td>Internal circulation fan:</td>
<td>on or off.</td>
</tr>
</tbody>
</table>

**Effect of internal partitioning:** An unexpected feature of the tracer-test results is that in most cases the rate of air change was influenced by the duration of door opening (Fig. 3). The rate of air change agrees with anemometer-measured values at short opening durations, but becomes less for longer durations. The explanation is that eq. (2) relies on the assumption that the air in the room is perfectly mixed. In practice, the presence of product stacks tends to divide the room into several zones, so that some of the air entering the building goes back out without thorough mixing with air in the rest of the room. To interpret quantitatively what happens, Pham /4/ constructed theoretical models based on two interconnected well-mixed regions. He was able to correlate the results in Fig. 3 by the model of Fig. 4, with α = 0.5 and f = 0.1.

![Graph](image)

**Fig. 3 - Air change vs door opening duration** (store A, 2.7m-wide x 2.9m-high door).

**Fig. 4 - Simplified two-region model** (V: total volume, v: total interchange rate).
When the mixing of the entering and leaving air streams near the door is the predominant factor, Pham's /4/ model predicts that the rate of air change no longer depends on door opening duration, but is reduced by a constant factor \( t/(1 + t) \). This was found to be the case for a high-rise store (E in Table 2) which had a single door, 2.1 m wide x 1.8 m high. The air interchange measured at the door with an anemometer was 0.50 air change/hour (ac/h); however, tracer tests give 0.26 ac/h, a reduction factor of 0.52 (corresponding to \( t = 1.1 \)), irrespective of door opening duration. This was caused by product being stacked just inside the door, partially isolating it from the rest of the room.

Because of the internal-partitioning effect, the infiltration rate due to two unprotected doors that were open simultaneously was much less than the sum of infiltrations through each door on its own. Differences ranged from 13% when doors were on opposite walls to 44% when doors were right next to each other. Similarly, infiltration was not proportional to door width but increased more slowly.

In practical situations doors are usually opened intermittently. We did some test on doors that were alternately open and closed at one-minute intervals to simulate practical working conditions. The ratio of measured/predicted infiltrations was 0.54 \( \pm \) 0.15 (prediction was based on the modified Tamm equation), showing that under realistic working conditions the rate of air change was nearly halved by the internal partitioning effect.

**Effectiveness of protective devices:** The effectiveness of a door protective device (air curtain or plastic strip curtain) is defined as the fractional decrease in airflow across the door when the device is fitted. For obvious experimental reasons it is difficult to measure airflows directly (with an anemometer) when such devices are present, and tracer tests must be used. However, one may wonder whether internal partitioning effects will distort the results as shown in the last section. Fortunately, air curtains and plastic strip curtains reduce the infiltration rate to quite small fractions of what it was, as will be seen presently, and so Pham's model /4/ predicts that internal-partitioning effects will become insignificant when protective devices are fitted (as the internal mixing rate becomes much larger than the external interchange).

Table 3 lists our results for the efficiencies of door protective devices measured on stores A, B and C, with airflows through unprotected doors based on Tamm's modified equation. The air curtains had been adjusted as recommended by Longdill and Wyborn /2/. Their efficiencies agree with results obtained by the same authors.

<table>
<thead>
<tr>
<th>Curtain type</th>
<th>Slot width, mm</th>
<th>Jet velocity, m/s</th>
<th>Efficiency (mean ( \pm ) s.d.)</th>
<th>Efficiency from /2/</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air curtain, vertical, non-recirculating</td>
<td>115</td>
<td>10.0</td>
<td>79 ( \pm ) 3</td>
<td>78</td>
</tr>
<tr>
<td>Air curtain, horizontal recirculating</td>
<td>60</td>
<td>6.8</td>
<td>76 ( \pm ) 3</td>
<td>78</td>
</tr>
<tr>
<td>Plastic strip curtain</td>
<td>-</td>
<td>-</td>
<td>93 ( \pm ) 1</td>
<td>-</td>
</tr>
<tr>
<td>Horizontal air curtain + plastic strip</td>
<td>60</td>
<td>6.8</td>
<td>91 ( \pm ) 1</td>
<td>-</td>
</tr>
</tbody>
</table>

**Effect of forklift traffic:** Tests were carried out with a forklift plus loaded pallets moving in and out of 3m-wide x 3.6m-high doors at one pass per minute. With unprotected doors the infiltration rate decreased by 21% \( \pm \) 10% due to blockage of the doorway. With air and/or plastic curtains, infiltration generally tended to increase, although the data are very scattered (increases of -19% to +105% were recorded) due to the small magnitude of the infiltrations being measured.

**Effect of internal circulation fans:** In some installations the circulation fans for the forced draft cooling system are turned off when doors are opened. We found that in general this caused a slight reduction in the air change rate (-9% \( \pm \) 6% s.d.). This is in agreement with the two-region model of Figure 4, since turning off the fan would reduce the internal
recirculation factor \( f \), and hence slow down the effective air change of the whole system \( /4/ \).

However, for one door infiltration rate consistently increased when the fans were turned off, by 35% when the door had no protection and by 77% when a vertical air curtain was operating. A probable explanation of this effect is that this door was under the end of the circulation fan's delivery duct. The jet of air from this duct would have run down the end wall and promoted turbulent mixing near the door in a way similar to an air curtain. This would lead to a decrease \( \left( 1/(1+f) \right) \) factor in infiltration, as mentioned earlier. The temperature difference between the entering and leaving air streams decreased from 28°C to 22°C when the recirculation fans were turned on, which confirmed that there was increased mixing between the two air streams.

**Infiltration under normal work conditions:** An example will illustrate how the results presented in this paper can be applied. Store D of Table 1 (gross volume 38 000 m³, net air volume 33 000 m³) was tested under actual working conditions as follows: tracer gas was released into the store and samples of air were collected with an automatic sampler \( /3/ \) every 90 minutes for 2 weeks. The total time doors were open was recorded every day using microswitches connected to electric clocks. The doors were unprotected, 2.75 m-wide x 3.05 m-tall. Inside and outside temperatures were -17°C and +16°C, respectively.

![Graph](image)

**Fig. 5 - Air change vs door-opening time under normal work conditions**

(Store D)

The total air change over each day, \( y \), was then plotted against total door opening time per day, \( x \), in Figure 5. Linear regression yielded the following equation:

\[
y = 0.19x + 0.11
\]

(3)

with a correlation coefficient of 0.974.

Tamm's modified equation predicts that each open door should cause an infiltration of 3.3 m³/s, or 0.36 ac/h. However, results given earlier in this paper show that an internal-partitioning effect will cause a reduction of 46% ± 15% (for intermittently opened doors). Thus, the predicted slope for eq. (3) is 0.42 ac/h x 0.54 = 0.20 ± 0.05 ac/h, which is in agreement with the measured value of 0.19 ac/h.

4. **DISCUSSION AND CONCLUSION**

Tamm's equation overpredicts air infiltration because it neglects frictional effects, and an empirical factor 0.68 ± 0.04 (s.d.) should be applied to it for practical cold store conditions.

Internal partitioning tends to further reduce the rate of air change. Its importance depends on how the doors are operated and how products are stacked, but a reduction factor of about 0.53 has been observed for unprotected doors in a variety of situations. It is
expected that this reduction factor does not apply to doors protected by efficient air curtains or plastic strip curtains.

Forklift traffic reduces air infiltration through unprotected doors but increases it when air curtains and/or plastic strip curtains are used. Air curtain efficiencies are about 75-80%, and plastic-strip-curtain efficiencies about 93%, but forklift traffic will reduce this figure slightly.

These results can be used to predict the rate of air change under normal working conditions if door opening times and internal partitioning factors are known. However, whether the refrigeration load is directly proportional to the rate of air change is another question. It depends on whether and how the entering air is cooled before it re-emerges from the room. If cooling is done purely by turbulent mixing with cold air, then heat and tracer gas would be exchanged between the air inside and outside the store at exactly the same rate. Thus, the rate of decay of tracer gas will be directly proportional to the rate of heat loss by infiltration. This would apply to store E since most of the mixing took place just inside the door. On the other hand, if the entering warm air is quickly brought into contact with the cooling coils or other cold surfaces, (for example when doors are located directly under air return ducts), then tracer test results will underestimate the actual heat load. In any case Tamm's modified equation will give an upper limit to the infiltration heat load. Thus, to take advantage of internal partitioning effects, air return ducts should not be positioned directly above doors as is often done, and turbulent mixing should be promoted near the doors.

This paper points out the inadequacy of using a "standard air change rate" to calculate heat loads in designing a cold store. The relevant factors are how large the doors are, how they are protected and how heavily they are used. We are collecting further data on these parameters.

REFERENCES


ENTRES D'AIR PAR LES PORTES DES CHAMBRES FROIDES

RESUME : Les échanges d'air, à travers les portes ouvertes de chambres froides, ont été mesurés à la fois avec un anémomètre et par une méthode avec traceur. Les résultats avec anémomètre montrent qu'on doit appliquer un facteur de 0,68 à l'équation de Tamm. Un autre facteur a été évalué (environ 47 % de réduction); il concerne le mélange imparfait de l'air. Les rideaux d'air réduisent l'infiltration à environ 75-80 % et les rideaux plastiquest à environ 93 %. Le trafic des chariots et les ventilateurs de brassage interne influencent les échanges d'air. La relation entre ces échanges et la consommation d'énergie est souvent complexe et dépend de la manière dont l'air entrant est refroidi.