Experimental investigation on thermal insulation performance of air interlayer under an impinging jet at high temperature

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Abstract. Air interlayers are widely used in building envelope to reduce heat transfer between indoor and outdoor environments. However, it is less common to apply to industrial processes in confined spaces, especially when the heating source is at high temperature with high velocity. In this paper, a special structure with air interlayer between two plates for thermal insulation was derived from an industrial manufacturing application. The effects of the interlayer on heat transfer was investigated under an impinging jet at high temperature. An experimental mock-up was designed to produce high temperature (~ 500 °C) impinging gas jet with relative stable high velocity (~56 m/s) at the nozzle exit. The impingement surface of the test structure was covered with a 2mm thick stainless-steel plate at a distance of 5mm, which formed a horizontal air insulation layer. Benchmark test was also conducted after the removal of the stainless-steel layer. In each case, a standard procedure within a duration of 30 min was applied to minimize the variations of testing conditions. The temperatures at both inner and outer surfaces of the structure, as well as the temperatures from inside and outside the air interlayer, were measured by thermocouples. In addition, an analytical model to account for the effects of air insulation was proposed to compared with experiments. The results showed that, the air interlayer can reduce the highest temperatures for about 48 °C. The temperature difference at the end of the experiment was about 30 °C. The analytical model was in good agreement with the experimental data and the maximum difference was ~6.5 °C.

1. Introduction

When a vehicle, such as train, equipped with a high-power gas turbine runs slowly or stands by in a ferryboat, tunnel or some confined space, its exhaust discharges at a high temperature and a high speed. The exhaust jet not only causes the ambient air temperature to rise sharply, which affects the safety of personnel, but also has a strong impact on the walls and some facilities on the top of the space. At the same time, this also causes the exhaust fan of the entire space to withstand higher temperatures, and once the cooling facilities are incomplete or damaged, it will cause significant losses.

Ye et al. took a spray cooling method for high temperature exhaust gas jet using nozzle array in confined space [1]. This method successfully reduces the ambient temperature in the confined space. In some cases, when there is no cooling facility, the space enclosure needs to be insulated to prevent the high temperature impinging jet from damaging the equipment. In fact, the insulating material, such as polycrystalline fiber or aluminum silicate fiber, is effective without damage. However, the powerful kinetic energy of the exhaust impinging jet often destroys the laying structure of the insulating material. Engineers must take some steps to prevent this from happening.

It is well known that air interlayers are widely used in the insulation of building envelopes, such as application to perforated bricks [2, 3]. We considered using stainless steel plates to build an air interlayer...
on the surface of the enclosure to deal with this situation. In this paper, we will study the thermal insulation properties of unsealed air interlayers under impinging jet conditions.

2. Theoretical analysis of the horizontal air interlayer
As shown in the figure 1, the horizontal cavity with thin air interlayer is heated from below. Assume that the upper and lower surfaces are large flat plates with temperatures $T_c$ (K) and $T_h$ (K), respectively, and in this case, the airflow characteristics are determined based on the Grashof number ($Gr$) or Rayleigh numbers ($Ra$):

$$Gr = \frac{g \beta (T_h - T_c) L^3}{\nu^2}$$  \hspace{1cm} (1)

$$Ra = Gr \times Pr$$  \hspace{1cm} (2)

where $g$ is gravitational acceleration (m/s²) and $\beta$ is volumetric thermal expansion coefficient (K⁻¹). $L$ is length and $\nu$ is kinetic viscosity (m/s²). $Pr$ is Prandtl number. For $Gr \leq 2430$ [4] or $Ra \leq 1708$ [5], buoyancy forces cannot overcome the resistance imposed by viscous forces and there is no advection within the cavity. Therefore heat transfer from the bottom to the top surface occurs by conduction, for thinner air interlayers, in addition to heat conduction, also by radiation.

![Figure 1. Schematic of the horizontal air interlayer.](image)

The heat flux $q$ across the air interlayer is expressed as

$$q = \frac{k}{L} (T_h - T_c) + \varepsilon \times \sigma \times (T_h^4 - T_c^4)$$  \hspace{1cm} (3)

The heat flux $q$ is heat transfer rate (W/m²). The parameter $k$ is a transport property known as the thermal conductivity of air (W/m·K). $\varepsilon$ is a radiative property of the surface termed the emissivity and $\sigma$ is the Stefan-Boltzmann constant ($\sigma = 5.67 \times 10^{-8}$ W/m²·K). All properties of the air are evaluated at the average temperature, $T_m = (T_h + T_c)/2$.

3. Method
3.1. Experimental mockup
Figure 2 gives a schematic of the experimental mockup. The mockup included a full-premixed gas burner, a high pressure centrifugal fan, a combustion mixing chamber with the outer diameter $D=600$ mm and an impinging jet nozzle with internal diameter of 150 mm. The nozzle impinged obliquely at an angle $\alpha$, between the nozzle centre and the normal direction of the impingement surface, as shown in figure 3(a). The jet came from the designed nozzle and hot gas was supplied by gas burner and centrifugal fan. A pressure measurement device was installed on the gas pipe to monitor the gas dynamic pressure. A thermocouple thermometer was also installed on the gas pipe to ensure that jet exit temperature could be maintained at 500℃. Some device parameters of experimental mockup are summarized in the table 1. The air insulation layer was installed at the bottom of the test piece, as shown in figure 4(b), and the configuration of the test piece is consistent with the actual engineering application.

| Device          | Type            | Total pressure [Pa] | Speed [rpm] | Air volume [m³/h] | Power [kW] | Function |
|-----------------|-----------------|--------------------|-------------|-------------------|------------|----------|
| Gas burner      | full premix     | -                  | -           | -                 | 100–300    |          |
| Centrifugal fan | 4-72-4A         | 2014 ~ 1320        | 2900        | 4072 ~ 7419       | 5.5        | supply   |
| Centrifugal fan | 4-72-4.5A       | 2554 ~ 1673        | 2900        | 5712 ~ 10562      | 7.5        | exhaust  |
3.2. Experimental procedure

In order to ensure consistency and repeatability of test conditions for all cases, a series of standard procedures were established as follows:

- Check every part of the test equipment. If it is normal, install the test object on the equipment;
- Arrange the measuring points on the surface of the test object;
- Turn on the Fluke Recorder and the digital pressure gauge;
- Turn on the exhaust fan, and adjust the variable voltage and variable frequency (VVVF) facility of the exhaust fan to 40 Hz, and at this time the air flow rate at the inlet of the test equipment is about 1.9 m/s;
- Turn on the supply fan, and adjust its VVVF facility to 8 Hz;
- Turn on the full premix gas burner;
- Record the time when the burner is successfully ignited;
- Adjust the VVVF facility of the supply fan and the burner controller to make the temperature and the dynamic pressure meet the test requirements (velocity about 56 m/s and temperature up to 500°C);
- Turn off the burner after 30 minutes;
- Wait for the whole system to cool to room temperature.
4. Results and discussion

During the experiment, the temperature at nozzle exit was controlled at 500±20℃, the velocity of gas at nozzle exit was controlled between 55m/s and 58m/s. All the time of the testing lasted for half an hour. The sectional airflow rate at the inlet of the test equipment is 1.9±0.1m/s. The bottom of the air insulation layer is 2mm thick stainless steel plate, and the top is 7mm thick EH36 steel plate.

Case 1 is to remove the air insulation layer, and case 2 is to have an air insulation layer. In figure 4(a), the red line represents the temperature rise curve of the case 1, and the blue line represents the temperature rise curve of the case 2. At the end of the test procedure of 1800 seconds, the maximum temperature of Case 2 is 185°C, and the maximum temperature of Case 1 is 212°C. The maximum temperature difference during heating is 48 °C, which occurs in the heating period of 516s to 572s, shown in figure 4(b). Before 516s, the temperature difference between case 1 and case 2 increases rapidly, and the temperature difference gradually decreases after 572s. We calculated the slash slope between the temperature difference ($\Delta T$) and the time interval ($\Delta t$) of the experimental data, and the time interval is set to 20s. The results are shown in figure 4(c). Before 540s, the temperature change rates of case 1 is more than case 2. In the range of 540s to 560, the temperature change rates are the same, and then the $\Delta T/\Delta t$ value of the case 1 becomes smaller and gradually lower than the case 2. All the analysis shows that the 5mm thickness air insulation layer can delay heat transfer under high temperature impinging jet conditions, which can reduce the thickness of other insulating materials used at the same time and save space.

Based on the experimental data, we performed a theoretical calculation on the simplified model with air insulation layer, as shown in figure 3(c). According to the conservation of energy, there are the following equations (4) and (5):

\[\text{(4)}\]
\[\text{(5)}\]
(4) \[ h_g \cdot A_1 \cdot (T_b - T_h) = \frac{\sigma \cdot (T_b^4 - T_h^4)}{\varepsilon_1 \cdot A_1 + \frac{1 - \varepsilon_1}{A_1}} + \frac{k \cdot A_2}{L} (T_h - T_b) + \rho \cdot C_p \cdot A_2 \frac{dT_h}{dt} \]

(5) \[ h \cdot A_2 \cdot (T_h - T_b) = h_g \cdot A_1 \cdot (T_h - T_b) + q_f + \sigma \cdot \varepsilon_2 \cdot A_2 \cdot (T_b^4 - T_{inf}^4) \]

where \( h_g \) and \( h \) are convective heat transfer coefficient of the impinging gas jet and ambient air, respectively. \( T_h \), \( T_b \) and \( T_{inf} \) are the temperature of bottom surface, top surface and the ambient, respectively. \( \varepsilon_1 \) and \( \varepsilon_2 \) are the emissivity of the bottom and top surface, respectively. \( A_1 \) and \( A_2 \) are the surface area of the bottom and top surface, respectively. The parameter \( k \) is the conductive coefficient of the air interlayer. \( L \) is the thickness of the air interlayer. The \( \rho \) is the density of the stainless steel. The parameter \( \varepsilon_p \) is the specific heat at constant pressure of the stainless steel. The parameter \( t \) is time. The steady state of heat transfer is the condition for the equation (5) to be established.

In equation (4), \( h_g \) is very complicated. For a round nozzle, Martin [6] has recommended the following correlation to solve the convective heat transfer problem of the impinging jet:

\[ \frac{Nu}{Pr^{0.42}} = \frac{D}{r} \left[ 1 - \frac{1.1 \frac{D}{r}}{1 + 0.1 \left( \frac{H}{D} - 6 \right) \frac{D}{r}} \right] \times 2 \frac{Re^{0.5}}{200} \left[ 1 + \frac{Re^{0.05}}{200} \right]^{0.5} \]  

(6)

The ranges of validation of equation (6) are 2000 ≤ Re ≤ 400,000, 2≤H/D≤12 and 2.5≤r/D≤7.5. Here \( r \) is the radial distance from the stagnation point. \( D \) is the diameter of the nozzle and \( H \) is the distance from the nozzle to the impingement surface. \( Re \) is Reynolds number. Average Nusselt number \( Nu \) in equation (6) is determined by

\[ Nu = \frac{h_g \cdot D}{k_g} \]  

(7)

where \( k_g \) is the thermal conductivity of the impinging jet. Therefore, we can get the correlation of \( h_g \) as follows

\[ h_g = \frac{k_g}{r} \left[ 1 - \frac{1.1 \frac{D}{r}}{1 + 0.1 \left( \frac{H}{D} - 6 \right) \frac{D}{r}} \right] \times 2 \frac{Re^{0.5}}{200} \left[ 1 + \frac{Re^{0.05}}{200} \right]^{0.5} \times Pr^{0.42} \]  

(8)

In equation (5), \( q_f \) is the fin heat transfer rate. The straight fin of uniform cross section model can be used to evaluate \( q_f \), and the calculation method can be found in related papers or manuals and will not be listed here.

The two equations were solved using the Interact Heat Transfer 4.0 program, which is downloaded from the website at www.wiley.com. The initial condition parameter values of the equation from (4) to (8), such as temperature, flow rate, incident angle, constant of physical property, etc., are consistent with the test condition parameters. As a result shown in figure 4(d), the temperature rise rate was deviated. For the highest temperature, the calculated value is 178.5 °C, which is only 6.5 °C deviation compared to the test value of 185 °C. If the high temperature surface is at the bottom, equations (4) and (5) can be used to predict horizontal thin air insulation effects in design.

5. Conclusion

The paper briefly analyses the heat transfer mechanism of the horizontal thin air insulation layer (bottom heating), and uses the IHT4.0 program to calculate the back surface temperature of the model under the experimental conditions. Computational models can be used to predict the effect of horizontal air insulation.
An experimental mockup was constructed and a heat transfer experiment of high-temperature high-speed gas impinging jet was completed. Two experimental cases were successfully carried out, and the temperature of the back surface of the test piece in the case of air-insulated layer and without air insulation layer was measured. The highest temperature, temperature difference, and heating rate were compared between the two cases. The air insulation layer delays heat transfer and saves space.

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