

The Effect of In-Duct Sprinkler Operation on Exhaust of Fire Gases from Clean Room Wet Benches

Tak-Sang Chan and Hong-Zeng Yu
Factory Mutual Research Corporation
Norwood, MA 02062 USA

ABSTRACT

Experiments were conducted to study the effect of sprinkler operation on the exhaust of fire gases from the wet bench for a fire occurring in a clean room wet bench. In this study, the pressure loss across the sprinkler spray in a horizontally-oriented duct connected to the wet bench was measured and analyzed to address the following operation variables: orientation of sprinkler water discharge relative to exhaust flow, duct size, sprinkler model and orifice diameter, water discharge pressure, gas temperature and exhaust flow rate. The pressure loss increases with water discharge pressure and exhaust rate. As the sprinkler orifice size increases, the spray-induced pressure loss increases for the same water discharge pressure, but decreases for the same water discharge rate. Furthermore, discharging water against the exhaust flow induces considerably greater pressure loss than discharging water perpendicularly to the flow. The pressure loss data for sprinklers discharging perpendicularly into the exhaust flow can be correlated with the above operation variables. The correlation suggests that the spray-induced pressure loss is mainly due to the drag resistance of the water drops generated in the duct.

KEYWORDS: Wet bench fires, in-duct sprinkler spray, flow resistance

INTRODUCTION

The process hazards associated with the fabrication of semiconductor devices in clean rooms lie in the heavy use of toxic, highly corrosive and flammable gases and liquids¹. Adding to the high risk of fire loss is the use of plastic wet benches. Because process equipment is expensive and the product in process is extremely susceptible to fire smoke, there is a great potential for substantial loss resulting from non-thermal damage even though the fire is contained in a very small area. Therefore, elaborate air handling systems are deployed in clean room facilities to properly exhaust harmful fumes and smoke.

There is evidence that sprinkler operation in exhaust ducts connected to wet benches during a fire can completely obstruct flow in the ducts and thus back up the fire gases to contaminate the clean area^{2,3}. However, only limited information is available for the exhaust rate reduction caused by a single sprinkler spray in a specific exhaust arrangement. It is difficult to extend this information to other exhaust systems, operation conditions, and multiple sprinkler operations in the duct. To be able to evaluate the reduction of exhaust flow rate for different exhaust systems under different sprinkler operation conditions, it is believed that the flow resistance induced by single sprinkler sprays should be quantified for different operation conditions just like other components in the exhaust system. Once the information is available, the overall exhaust flow can be calculated based on the overall flow resistance in the system, for both single and multiple sprinkler operations. Although pressure losses in ducts due to water films, suspended water drops and solid particles have been investigated in numerous studies^{4,5,6,7}, it is difficult to apply these results to our current problem, since little is known about the two-phase flows resulting from sprinkler sprays in ducts. As a result, experiments were conducted to measure the spray-induced pressure losses for different exhaust flow orientations (horizontal, upward and downward) for a range of exhaust conditions pertaining to normal wet bench operations. In this paper, the pressure drops across single sprays in a horizontal duct are presented for sprinklers discharging water perpendicularly to and against the exhaust flow.

EXPERIMENTAL SETUP AND PROCEDURES

Figure 1 schematically shows the experimental setup, which consists of a fire enclosure, a burner, ductwork and a blower. The fire enclosure measured 0.91 m x 0.91 m x 1.82 m long and was made of galvanized steel. One end of the enclosure was open and the other end connected to the exhaust ductwork. Inside the enclosure, a heptane spray burner consisting of four furnace nozzles was used as a fire source to provide different inlet gas temperatures in the exhaust duct.

The galvanized steel ductwork was divided into three sections, identified as G1, G2 and G3 in Figure 1. Ducts G1 and G3 were designed for volumetric flow rate measurements; duct G2 was provided for sprinkler operation. The duct diameter in sections G1 and G3 was 0.25 m for all the experiments. Two sizes of duct, 0.25 m and 0.51 m, were used in section G2 so that the effect of duct size could be examined.

The volumetric flow rate upstream of the sprinkler was measured with an orifice plate located ten duct diameters downstream of the fire enclosure. Pressure taps were placed one diameter upstream and one-half diameter downstream of the orifice and the pressure differential between the two taps was monitored using a pressure transducer. The volumetric flow rate downstream of the sprinkler was measured using an averaging pitot tube (model 302, Mid-West Instrument) installed in duct G3. The averaging pitot tube was installed seven diameters downstream of the 90° elbow as shown in Figure 1. A centrifugal blower was connected via a tee at the end of the duct. The exhaust flow rate from the enclosure could be controlled by the blast gate installed in the tee shown in Figure 1.

Sprinklers employed for the experiments included a 1/2-in pendent sprinkler with a discharge K-factor of $7.9 \text{ dm}^3/\text{min}/(\text{kPa})^{1/2}$, and a 3/8-in pendent sprinkler with a K-factor of $4.3 \text{ dm}^3/\text{min}/(\text{kPa})^{1/2}$. The sprinklers were installed in duct section G2. Two discharge

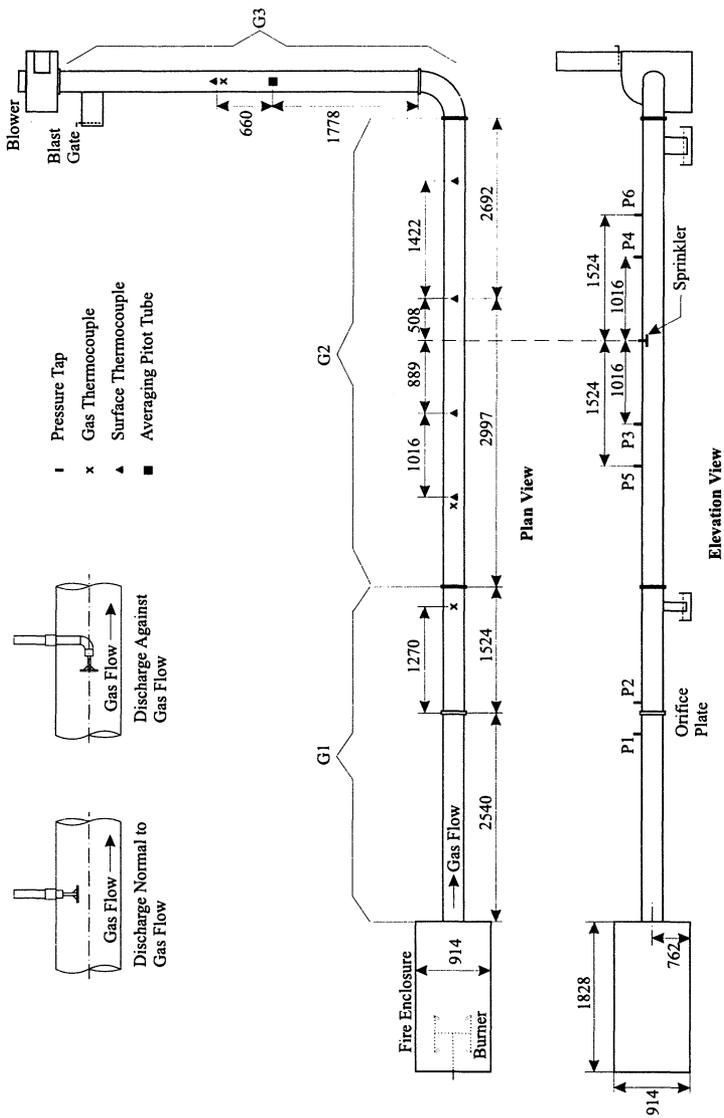


Figure 1. Experimental setup for the 25-cm diameter duct. A 51-cm duct was also used for section G2 to examine the effect of duct size. All dimensions shown in the drawing are in millimeters.

orientations were employed for the measurements: discharging water perpendicularly to the exhaust flow and discharging against the flow. The positions of the sprinklers inside the duct for the two orientations are shown in the upper left corner of Figure 1. For the normal discharge orientation, the sprinkler deflector was parallel to the flow direction. For the against-flow orientation, the sprinkler was placed at the center of the duct with its deflector normal to the flow direction. A pressure transducer was used to monitor the water discharge pressure.

All thermocouples were constructed from 30-gage, chromel-alumel, inconel sheathed wires. Thermocouples marked with a cross in Figure 1 were used to monitor the gas temperature inside the duct. A single thermocouple probe was placed in the center of the duct at each location, except at location A where two additional thermocouples were placed, one 3.8 cm from the top and the other 3.8 cm from the bottom of the duct. In addition to the gas temperature measurements, the exterior surface temperatures of the duct were also monitored. The locations of the surface thermocouples are identified in Figure 1 by the triangle symbols.

Pressure taps were provided at locations P3, P4, P5 and P6 for monitoring the pressure drops across the sprinkler. The pressure differentials between locations P3 and P4 and locations P5 and P6 were monitored with pressure transducers.

Two water drains were provided in the ductwork to prevent water from accumulating inside the duct during sprinkler operation. One of the drains, which had a diameter of 10.2 cm, was placed upstream of the sprinkler at the end of duct G1. The second drain was 20 cm in diameter and was located downstream of the sprinkler near the 90° elbow. The outlets of the drains were submerged in water to prevent exhaust flow leakage.

Before each measurement, the initial volumetric exhaust flow rate was adjusted to the designated value. The blower was then turned off and the computer started collecting pre-ignition data for thirty seconds. At the end of the thirty seconds, the blower was turned on and allowed to run for one minute. The heptane spray burner was then ignited. Thirty seconds after the ignition water was discharged through the sprinkler at a pre-selected discharge pressure. The test continued for a number of increasing water discharge pressures. The test time for each discharge pressure was typically five minutes. At the end of the highest discharge pressure, water to the sprinkler and heptane fuel to the nozzles were shut off and a test series was thus completed. Data signals were acquired at a rate of 1 scan per second.

EXPERIMENTAL RESULTS

When an in-duct sprinkler discharges water perpendicularly to or against the exhaust gas flow, additional pressure loss is incurred. The objective of the experiments was to provide such pressure loss data necessary for the assessment of the flow capacity of an exhaust system during sprinkler operation. In the absence of a fire, the increase in pressure loss caused by the operation of an in-duct sprinkler, ΔP_{sp} , can be defined by

$$\Delta P_{sp} \equiv \Delta P_w - \Delta P_o \quad (1)$$

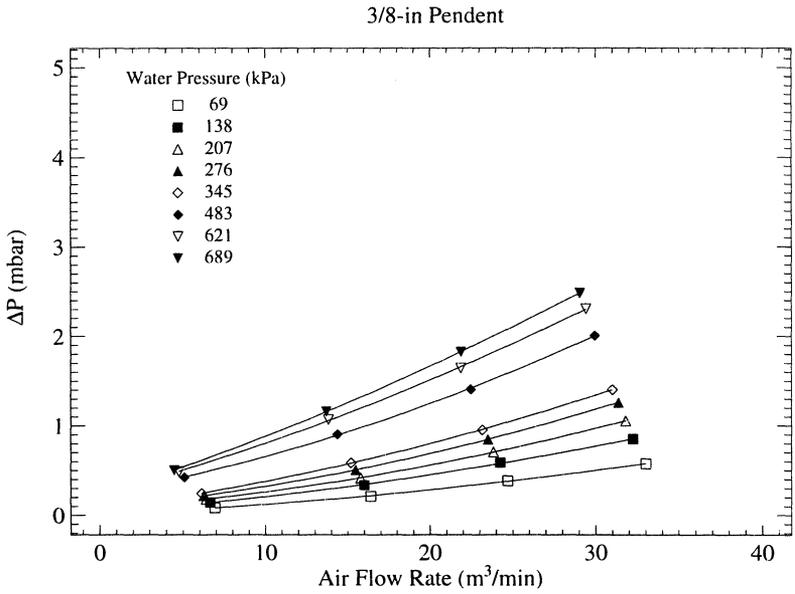
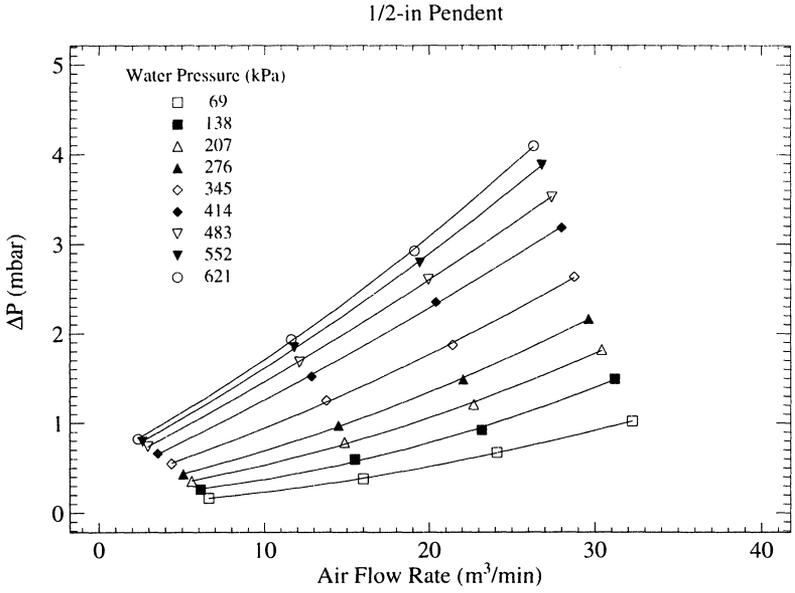


Figure 2. Spray-induced pressure losses for the 1/2-in and 3/8-in sprinklers with water discharge direction normal to the airflow direction. Duct diameter = 25 cm, air temperature = 20 °C.

where ΔP_w is the pressure drop across the sprinkler spray, and ΔP_0 the pressure drop when the sprinkler is not operated.

The pressure losses ΔP_{sp} for the 1/2-in and 3/8-in sprinklers are presented in Figure 2 for the case when the sprinkler water discharge is perpendicular to the duct axis. The results presented correspond to the pressure drop measurements made between taps P5 and P6, which agree well with the results obtained from the measurements made between taps P3 and P4. Each curve in the figure corresponds to a constant water discharge pressure. For a sprinkler operating in the duct with no fire in the enclosure, the flow resistance is seen to increase with the water discharge pressure and the volumetric flow rate in the duct. For a given water discharge pressure, the 1/2-in sprinkler causes higher flow resistance than the 3/8-in model. However, by examining the pressure loss data for the same water flow rate, the resistance of the 1/2-in sprinkler is found to be less than that of the 3/8-in sprinkler. This can be seen in Figure 2, for instance, by comparing the pressure loss data of the 1/2-in sprinkler at 207 kPa with those of the 3/8-in sprinkler at 690 kPa. Both the sprinklers provide the same water discharge rate of 113 dm³ at the above corresponding water pressures, since the water discharge rate is the product of K-factor and the square root of water pressure.

In Figure 3, the pressure losses for the 1/2-in sprinkler discharging water in the against-flow orientation are compared with those of the normal discharge orientation. It is seen that, as the water pressure is increased, the pressure losses for the against-flow orientation becomes considerable greater. This is because the relative velocity between the water drops and the exhaust flow are higher in the against-flow orientation, resulting in larger drag resistance of the water drops.

DATA ANALYSIS

It is assumed that the spray-induced pressure loss is due only to the drag resistance of the water drops, and the interference between water drops is negligible. It is also assumed that the total drag of water drops of non-uniform sizes can be estimated by the total drag of uniform-sized drops with a characteristic diameter d . From dimensional analysis, ΔP_{sp} is related to the drag of a drop by the following functional relationship:

$$K_{sp} \equiv \frac{\Delta P_{sp}}{\frac{1}{2} \rho U^2} = f \left(\frac{n F_p}{\frac{1}{2} \rho U^2 A} \right) \quad (2)$$

where U is the exhaust flow velocity and ρ is gas density, and K_{sp} is defined as the sprinkler spray-induced pressure loss coefficient. The total number of water drops in the measurement volume is denoted by n . F_p is the drag force per drop and A is the area of the duct cross section. The drag force of a drop can be expressed:

$$F_p = \frac{1}{2} \rho U_r^2 C_D A_d \quad (3)$$

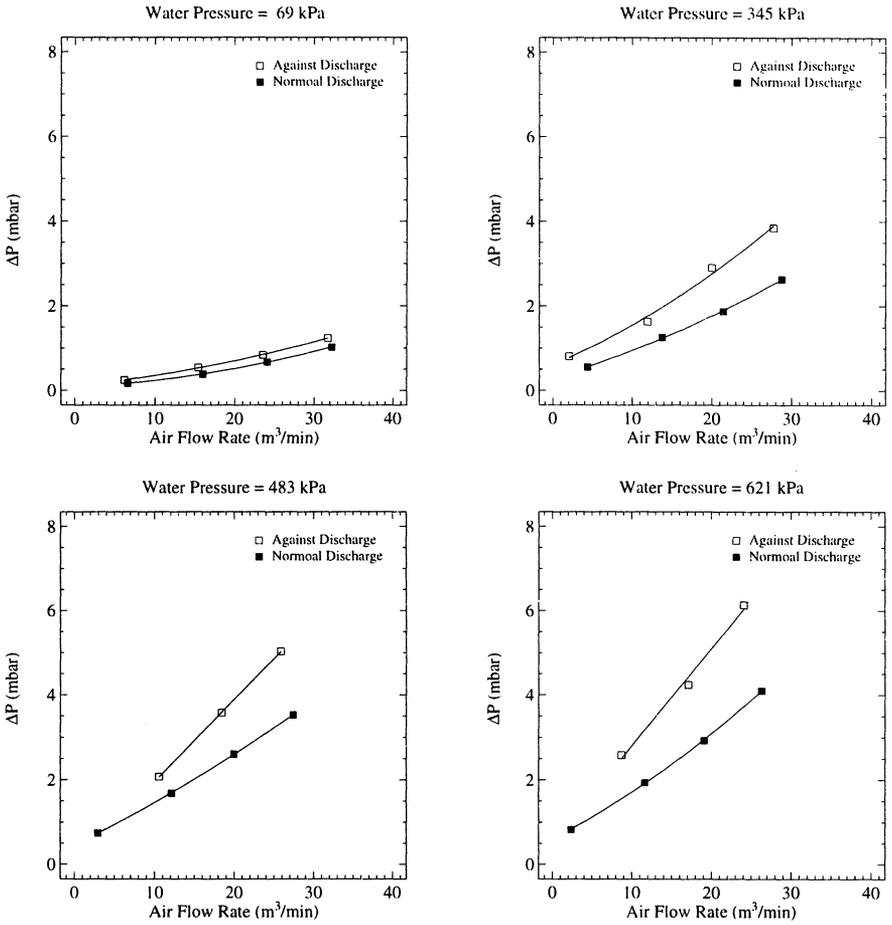


Figure 3. Comparisons of spray-induced pressure losses by the 1/2-in sprinkler between two discharge orientations: against the flow and normal to the flow. Duct diameter = 25 cm, air temperature = 20 °C.

where C_D is the drag coefficient, A_d is the frontal area of the drop and U_r is the relative velocity between drop and the exhaust flow. The drag coefficient of a sphere depends on the droplet Reynolds number, $Re = \rho U_r d / \mu$ (d is the droplet diameter and μ is gas viscosity). For the range of Reynolds numbers expected for drops in sprinkler sprays C_D can be approximated: ⁽⁸⁾

$$C_D \propto Re^{-1/2} \quad (4)$$

According to Eqs. (3) and (4) the drag force per drop, F_p , can be written:

$$F_p \propto \frac{\pi(\rho\mu)^{1/2}}{8} (U_r d)^{3/2} \quad (5)$$

The total number of suspended water drops, n , is approximated by:

$$n \propto \left(\frac{\rho_w \dot{Q}_w}{m_d} \right) \left(\frac{l}{u_w} \right) \quad (6)$$

where \dot{Q}_w is the volumetric flow rate of water discharged by the sprinkler, ρ_w is the water density and m_d is the mass of a water drop, i.e. $(\pi/6)\rho_w d^3$. The term in the first bracket of Eq. (6) corresponds to the number generation rate of water drops. The second term is a time scale expressed with a length scale and the average water discharge velocity, $u_w = 4\dot{Q}_w / \pi D_{or}^2$ (D_{or} being the diameter of the sprinkler orifice). For the case where the water is discharged normally to the exhaust flow, the length scale, l , may be taken as the distance from the sprinkler deflector to the opposite side of the duct wall, which is equal to $D - e$, the diameter of the duct minus the insertion depth of the deflector.

Substituting Eqs. (5) and (6), together with the definitions for m_d , u_w , l and A , into Eq. (2) and setting $U_r = U$ for the perpendicular discharge orientation, the following equation is obtained:

$$K_{sp} = f \left\{ \frac{3}{2} \left(\frac{\mu}{\rho U D} \right)^{1/2} \left(\frac{D-e}{D} \right) \left(\frac{D_{or}}{D} \right)^{1/2} \left(\frac{D_{or}}{d} \right)^{3/2} \right\} \quad (7)$$

The approximation of U_r with U is reasonable because the injection velocity of the droplets in the flow direction would be much smaller than U when the water was discharged in the direction perpendicular to the flow.

For a given sprinkler, the average drop size can be expressed explicitly by the following relation: ⁽⁹⁾

$$\frac{d}{D_{or}} \propto We^{-1/3} \quad (8)$$

where We is the Weber number equal to $\rho_w u_w^2 D_{or} / \sigma$. Substituting Eq. (8) into Eq. (7) for the ratio D_{or}/d and discarding the leading constant, we have:

$$K_{sp} = f(\lambda)$$

$$\lambda = \left(\frac{We}{Re_D} \right)^{1/2} \left(\frac{D-e}{D} \right) \left(\frac{D_{or}}{D} \right)^{1/2} \quad (9)$$

where Re_D is the Reynolds number based on the flow velocity and the duct diameter.

Figure 4 plots the pressure loss coefficient K_{sp} as a function of λ for the 1/2-in and 3/8-in sprinklers. The data for the 0.51-m duct experiments are also included in the figure. As seen in the figure, the experimental data are correlated well using the functional relationship presented in Eq. (9).

Experiments were also conducted with the presence of a fire in the enclosure to investigate the effect of gas temperature on the sprinkler spray-induced flow resistance. For elevated temperature conditions, Eq. (1) was modified to obtain the spray-induced pressure loss between P5 and P6.

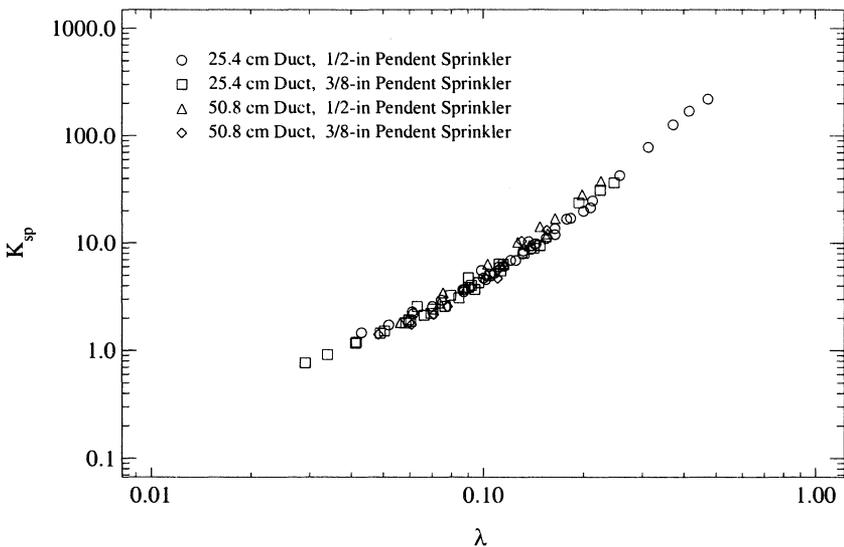


Figure 4. Correlation of the spray-induced pressure loss coefficient for the 1/2-in and 3/8-in. sprinklers discharging water perpendicularly to the exhaust flow. Air temperature = 20 °C.

$$\Delta P_{sp} = \Delta P_w - \Delta P_o + \rho_5 U_5^2 - \rho_6 U_6^2 \quad (10)$$

where subscripts 5 and 6 denote the two pressure tap locations upstream and downstream of the spray. The first two terms on the right hand are defined the same as in Eq. (1). The last two terms account for the effect of spray cooling on the momentum of the gas flow.

The upstream momentum at P5 was calculated from the flow rate measurements with the orifice meter and the temperature measurements at location A (Fig. 1). Analysis of the flow rate data

measured downstream by the averaging pitot tube showed that the mass flow rates at the location of the averaging pitot tube were close to those upstream of the sprinkler, although some water vapor was expected to be generated by heat. The maximum amount of water vapor in the bulk flow at location P6 can be estimated by assuming that the reduction of the gas temperature from P5 to P6 is mainly caused by water evaporation. With T_5 , T_6 and the mass flow rate measured by the orifice plate, the maximum mass fraction of the water vapor downstream of the sprinkler at P6 was estimated to be less than 15% for all the experiments conducted at elevated temperatures. As a result, the downstream velocity U_6 can be approximated by $U_5\rho_5/\rho_6$. Using the ideal gas law, the last two terms in Eq. (10) can be substituted by $\rho_5 U_5^2(1 - T_6/T_5)$. For the present test conditions, the maximum value of $\rho_5 U_5^2(1 - T_6/T_5)$ was equivalent to a static pressure of 4 mm water.

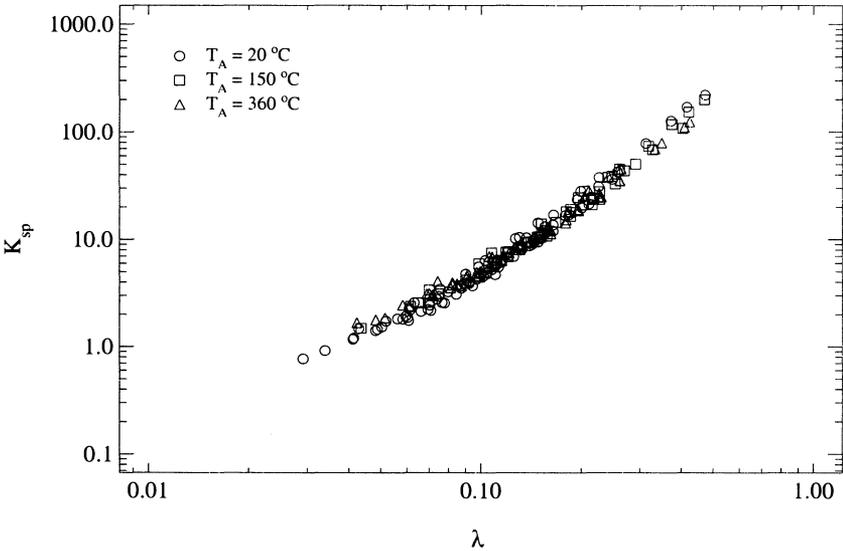


Figure 5. Correlation of the spray-induced pressure loss coefficient at different temperatures for the 1/2-in and 3/8-in sprinklers discharging water perpendicularly to the gas flow direction.

The spray-induced pressure loss coefficient K_{sp} at elevated gas temperatures are presented in Figure 5 for the 1/2-in and 3/8-in sprinklers operated in the normal discharge orientation. For comparison, the corresponding cold flow data in Figure 4 are also included in the figure. In the figure, K_{sp} was derived from the pressure drop data in accordance with Eq. (10) normalized by the downstream dynamic pressure. The product of the density and velocity in Re_D was evaluated based on the mass flow rates measured by the orifice plate, while the viscosity was evaluated at the room temperature. It is seen that the pressure losses at elevated temperatures are also correlated well with Eq. (9).

The gas temperature at location 6 associated with the pressure loss data of Figure 5 can also be correlated with the dimensionless group λ , as shown in Figure 6. In the figure, θ

corresponds to a dimensionless temperature ratio, $(T_6 - T_w) / (T_5 - T_w)$, where T_w is the temperature of the water before discharge.

CONCLUSIONS

The pressure loss due to sprinkler operation in a wet bench exhaust duct increases with the water discharge pressure and gas flow rate in the duct. For the same water discharge pressure, larger pressure losses are generated by sprinklers with larger orifices. However, for the same water discharge pressure, the pressure losses caused by sprays discharged from sprinklers with larger orifices are less than those of sprinklers with smaller orifices. Furthermore, the pressure losses for the case where the sprinklers discharge water against the exhaust flow become considerably larger than for the case where the sprinklers discharge in the direction perpendicular to the flow.

For sprinklers discharging water in the duct perpendicularly to the exhaust flow, the pressure loss data are satisfactorily correlated. The correlation includes the effects of duct diameter, gas flow velocity and temperature, sprinkler orifice diameter and sprinkler discharge pressure. The correlation indicates that the pressure loss is mainly due to the drag resistance of the water drops generated inside the duct.

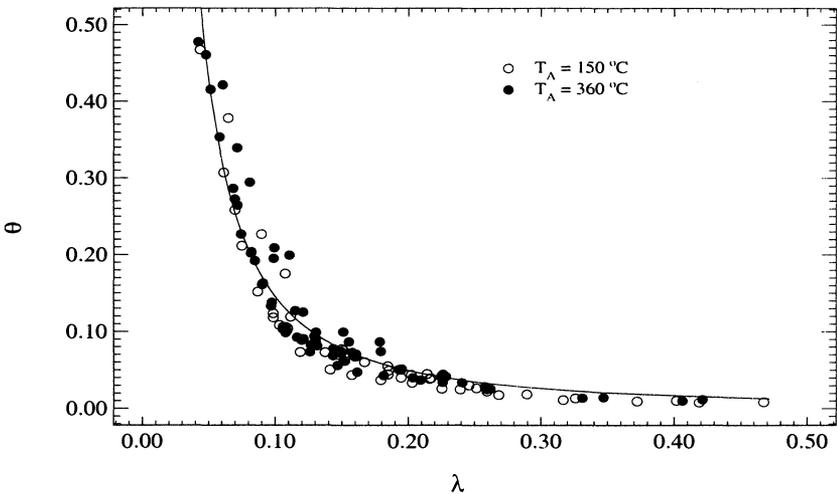


Figure 6. Correlation of the temperature data for the 1/2-in. and 3/8-in. sprinklers discharging water perpendicularly to the exhaust flow.

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