Geometrical shape nozzle and liquid temperature influence on the two-phase free jet characteristics

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L'INFLUENCE DE LA GEOMETRIE DE LA TUYERE ET DE LA TEMPERATURE DU LIQUIDE SUR LES CARACTERISTIQUES DU JET DIPHASIQUE LIBRE

Abstract

The paper traits the behaviour of a two-phase jet outgoing from different nozzle geometries at various liquid temperatures. The tests were realised on an experimental layout from our laboratory. The visualisation of temperature field by a thermal camera are displayed. Based on the experimental results the graphical evolution of the main characteristics of the jet are shown. Due of these tests the influence of Coanda effect related to the nozzle geometry was mentioned. The results of this study are applicable to the fire extinguish with an enhanced efficiency, especially in the closed spaces. The extinguish thermal efficiency was introduced.

Keywords : two-phase jet, Coanda effects, thermal field, fire extinguish thermal efficiency.

Résumé

Le travail traite le comportement du jet diphasique issue des diverses géométries de la tuyère à de différentes températures de l'eau. Des essais ont été effectués sur un banc approprié. La visualisation du champ des températures a été réalisée par une thermo-caméra. Basés sur les résultats issus de l'expérience ont été tracés des diagrammes d'évolution des grandeurs spécifiques du jet. Du aux expériences on a constaté l'application de l'effet de Coanda dans ce cas. Les résultats de l'étude sont nécessaires à l'augmentation de l'efficacité pour supprimer les incendies, en spécial dans les espaces fermés. Une efficience thermique d'éteint a été introduite.

Mots clé : jet diphasique ; effet de Coanda, champ thermique, incendie, efficience thermique d'éteint.

Introduction

The modern systems used for the fire extinguish systems, must have a short time to stop the fire and adequate efficiency. The use of water mist is today an option in aim to realise an economical method and an ecological way for fire extinguish. The most available fluid used is the liquid water. This fluid is not toxic, or pollutant, and has an important heat absorption [1-5]. Droplets transfer processes occurring in the spray surrounding are multifaceted and it is difficult to model. Droplet dispersion cone depends of the atomizer system. The finesse of droplets in aim to have small thermal inertia, must have a diameter less than 20 µm. The dimensions of the pulverized droplets are recommended to be in the range of 10 to 50 µm [2, 5]. It is well known that the time interval for evaporation of liquid reduced for the small droplets. A way to realise this short time is to have an important pressure difference on the nozzle, a hundred bars as example [6]. The authors of this paper proposed a heating of water before the entry in the nozzle. Our experiences were made with the water heated up to 50°C and with the pressure in the range of 4-6 bar [5, 6]. In fact, the vapour generated by the liquid evaporation pushes the other gaseous compounds from the affected space, and the condition to maintain the flame is not fulfilled. Concerning the shape of the jet cone of two-phase fluid outgoing from the nozzle, this is an important parameter in aim to ensure a high volume of it. Consequently, the cone must be more flared in the space. By this way the flame volume captured by jet is important and the water is used efficiently. Totalling these two effects, which take place simultaneously, an efficient cooling of the zone is realisable [5, 7]. Also, by the rapid evaporation of water the gaseous mixture created in the jet has a molar weight of vapour less than the molar weight of the surrounding. Consequently, the mixture rich in vapour rises and form a layer under the ceiling and protect it.

1 – Apparatus arrangement and the nozzle shape characteristics

The paper is focused on the tests made on the below layout. Based on the above considerations, we have proposed an experimental bench producing the mist jet. The tests were made in a semi-open room. The liquid temperature at the discharge head of the tube is modified in aim to put in evidence its influence on the dispersion and its evaporation rate. The liquid feed water pressure is in the range of 2-6 bars and its feed temperature is around 30-55°C. The experimental test layout diagram is shown in the Figure 1 [5, 6]. The tested nozzle diameters are 1.5 and 4 mm.

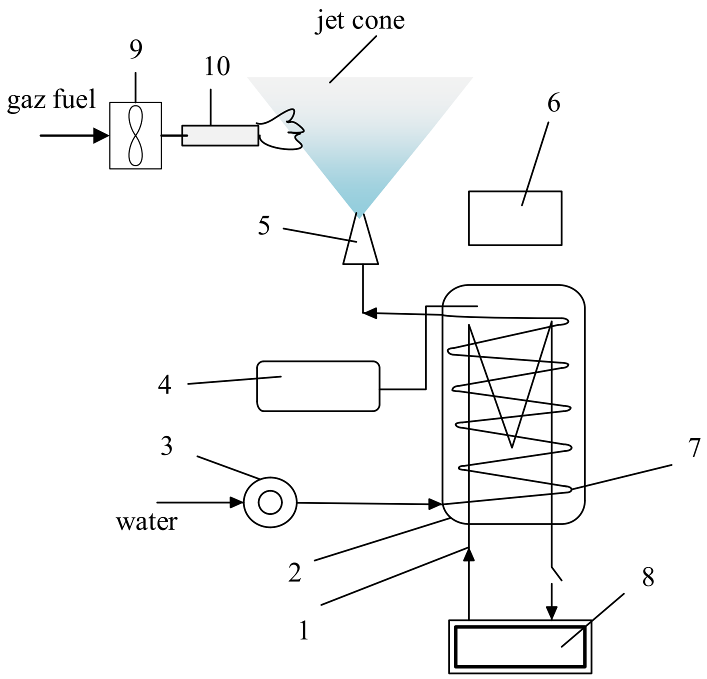


Figure 1: Experimental layout
1 - electrical resistance; 2 - oil tank; 3 - water flowmeter; 4 - electronic thermostat; 5 - water nozzle; 6 - measurement bench; 7 - water heater; 8 - electrical supplier; 9 – fuel gas flowmeter; 10 - gas burner

The cold water passes the mass flowmeter type Coriolis, is driven to the water coil immerged in the oil tank. The oil from the tank is heated by an electrical resistance. The control of the temperature water supplying the jet is ensured by the electronic thermostat and sensor assemblage.

Another important equipment of the test layout is represented by the nozzle assemblage with a swirl chamber. The swirl device, shown in the figure 2, has the function to create the vortex in liquid before that be pushed through the nozzle orifice. On this way a short length of the liquid jet is obtained and the liquid is dispersed in small droplets.

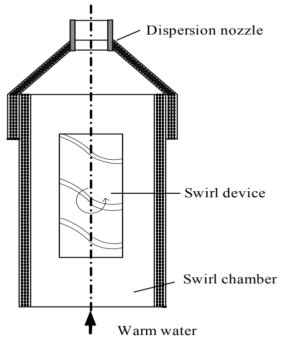


Figure 2: Water dispersion nozzle with swirl device

A key fact of the dispersion nozzle is the shape of the exit section and the angle of this. A qualitative draw is presented on the figure 3, for various exit angles. This form of nozzle was built in aim to use the Coanda effect consequences [8]. By reducing the angle α the jet angle envelope increases too. This enhancement of the cone jet volume is necessary in intention to bring a high space of flame and to use efficiently the water mist. The picture with the nozzles used in test is shown on the figure 4. The nozzle number one has a 1,5 mm orifice diameter and the angle α=35°, and the nozzle number two is attributes to the nozzle with the diameter of 4 mm and the angle α=40°. After the tests we have observed that the most efficient nozzle in the spray development with finest droplets, is realised with the nozzle number 1 of 1,5 mm.

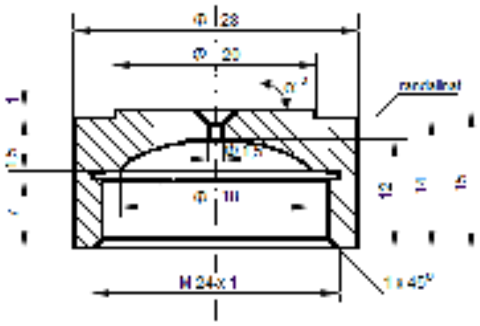
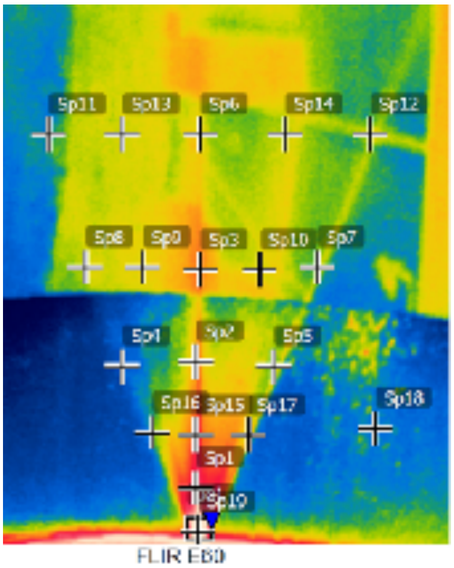


Figure 3. The nozzle shape and details (nozzle 1)

Figure 4. Nozzles shape

On the figure 5 the two-phase jet development for the liquid temperature supply of 52°C is shown. By increasing the inlet liquid temperature, the droplet dimension diminishes. The picture was taken with the infrared camera used for these tests [9].



Measurements		
Bx1	Max	47.0 °C
	Min	24.1 °C
	Average	32.7 °C
Sp1		28.6 °C
Sp2		24.5 °C
Sp3		24.8 °C
Sp4		22.1 °C
Sp5		22.9 °C
Sp6		22.8 °C
Sp7		22.0 °C
Sp8		23.2 °C
Sp9		23.4 °C
Sp10		23.5 °C
Sp11		22.8 °C
Sp12		23.3 °C
Sp13		23.2 °C
Sp14		23.2 °C
Sp15		25.7 °C
Sp16		22.6 °C
Sp17		24.0 °C
Sp18		23.6 °C
Sp19		42.8 °C

Figure 5: The mist jet cone supplied by the liquid at 52°C

a) Infrared image; b) temperature table

The cloudiness, especially at the liquid temperature of 50°C, shows that due to the high evaporation rate the relative humidity around and, in the jet, increases and the saturation conditions are reached. On the other hand, by growing the evaporation rate, the partial vapour pressure in the jet volume rises, and consequently, the oxygen concentration reduces.

The temperature values are indicated on table from figure 5. We observe that on the vertical axes the temperature is greater that on the horizontal axes at the same level. On the figure 6 the axial temperature evolution for the two cases are displayed. On the other hand, the decreasing in temperature for the 52°C temperature at the nozzle exit, at certain height is more accented than for the jet outgoing at 41°C. That means that the temperature decrease rate is more important for the higher exit temperature, so the saturation of the concerned volume in water vapor increases rapidly for the higher exit temperature.

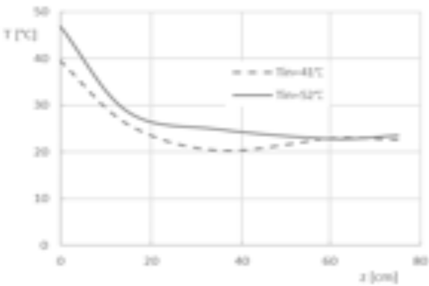


Figure 6 Temperature evolution in the jet axis

2 – Heat balance of the analysed extinguish system

The experimental layout allows us to realise a heat balance of the system. The representative equation of the heat balance is:

$$\dot{V}_{fN}\, H_s = \dot{m}_l \left(h_v - h_l\right) \cong \dot{m}_l\, l_v \qquad (1)$$

where: $\dot{V}_{fN}$ is the fuel gas in $\mathrm{m_N^3/s}$, measured with the gas flowmeter and corrected in function of the gas temperature and pressure according with the manual instruction [10]; H_s is the higher heating value of the fuel, in kJ/m^3_N ; $\dot{m}_l$ represents the water mass flow rate effectively evaporated, in kg/s; h_v – specific enthalpy of the generated vapour in the mixture, kJ/kg; h_l – specific enthalpy of liquids at the nozzle exit, in kJ/kg; l_v is the heat of vaporisation of water, in kJ/kg.

In the above equation we used the higher heating value of fuel, because in contact with the liquid phase the vapour born by the combustion process, in contact with the liquid droplets is considered in liquid phase. This phenomenon is supposed to take place, in the case when the final temperature of gases after the flame extinguish, is practically an environmental temperature before the flame ignition. The mass flowrate of the liquid used for the flame extinguish is calculate from the equation 1, using the thermal properties of water and the fuel flowrate consumption $\dot{V}_N$ [11, 12].

At the exit of the nozzle the mass water flowrate $\dot{m}_w$ is measured with the Coriolis equipment. This mass flowrate differs of the mass flowrate $\dot{m}_l$ calculated with the equation 1, because not all this flowrate is used in the flame extinguish, and only a part of this. We may define the mass efficiency of liquid use for the flame extinguish as a ratio between the mentioned mass flowrates:

$$\varepsilon_{fe} = \frac{\dot{m}_l}{\dot{m}_w} \tag{2}$$

Concerning the heat capacity of the liquid water absorption, the heat flowrate theoretically possible to be captured by the $\dot{m}_w$ flowrate is:

$$\dot{Q}_{t\,abs} = \dot{m}_w \left(h_v - h_l\right) \cong \dot{m}_w\, l_v \tag{3}$$

The real heat flowrate is obtained with the effective use of water:

$$\dot{Q}_{r\,abs} = \dot{m}_l \left(h_v - h_l\right) \cong \dot{m}_l\, l_v \tag{4}$$

The ratio between the two heat flowrate represents the thermal efficiency of liquid use, expressed as:

$$\eta_{te} = \frac{\dot{V}_{fN}\, H_s}{\dot{m}_w \left(h_v - h_l\right)} = \varepsilon_{fe} \tag{2'}$$

where the rel. 1 was used. From the relations 1, 3 and 4 practically the same expression of the mass efficiency of liquid and the thermal efficiency for the flame extinguish results (eq. 2).

Because the liquid is heated a supplementary heat flux is necessary for this scope. The heat flux necessary for this scope is:

$$\dot{Q}_{pr\,l} = \dot{m}_w \left(h_{l\,ex} - h_{l\,in}\right) \tag{5}$$

where $h_{l\,ex}, h_{l\,in}$ are the liquid enthalpies at the exit and at the inlet in the water heater 7 (fig. 1).

We define also the thermal power of flame as:

$$\dot{Q}_{t\,fl} = \frac{\dot{V}_{fN}\, H_s}{60} 10^{-3} \qquad \text{[kW]} \qquad (6)$$

with H_s in kJ/m$_N^3$, and $\dot{V}_{fN}$ in dm^3 $_N$/min. Consequently, the thermal efficiency becomes:

$$\eta_{t\,e} = \frac{\dot{Q}_{t\,fl}}{\dot{m}_w\left(h_v - h_l\right)} = \frac{\dot{V}_{fN}\, H_s\; 10^{-3} / 60}{\dot{m}_w\left(h_v - h_l\right)} \qquad (2")$$

The effective thermal efficiency of liquid use may be defined as:

$$\eta_{t\,eff} = \frac{\dot{V}_{fN}\, H_s}{\dot{m}_w\left(h_v - h_l\right) + Q_{pr\,l}} = \frac{\dot{V}_{fN}\, H_s}{\dot{m}_w\left[\left(h_v - h_l\right) + \left(h_{l\,ex} - h_{l\,in}\right)\right]} \qquad (7)$$

Using our experimental data, some results applying this model are displayed in the table 1.

Heat balance characteristics of the two phase jet and butane flame Table 1

Measured parameters		Calculated quantities			
$\dot{m}_w$ [kg/min]	$\dot{V}_f$ [dm^3/min] *at 20°C, p=750 mm Hg	$\dot{V}_{fN}$ [dm^3$_N$/min]	$\dot{Q}_{t\,fl}$ [kW]	$\dot{m}_l$ [kg/min]	$\eta_{t\,ef} = \varepsilon_{fe}$
0,94	4,5	3,35	7,09	0,170	0,18
1,24	5	3,70	7,88	0,189	0,15
1,483	8,2	6,06	12,92	0,310	0,21
1,61	9,4	6,95	14,82	0,355	0,22
1,85	12,4	9,16	19,55	0,469	0,25

* the flowmeter used was calibrated for 23°C and 760 mmHg, the correction coefficient for butane, according to the apparatus guide, is 0,812 [10].

The used fuel is butane with the higher heating value $H_s = 128\ 000\,\text{kJ/m}_\text{N}^3$ [12]. For the heat absorption we bring the heat of vaporisation of water at triple point l_v= 2501 kJ/kg [11]. This point is a reference one for the moist air at 760 mm Hg. The superheated heat of vapour up to the environmental temperature is negligible.

We note that the thermal efficiency of liquid use is reduced, consequently, an important liquid quantity is unused, and the damage produce increases.

Conclusions

The paper put in evidence the importance of the temperature and the nozzle shape of a extinguish system supplied by an warm liquid having the town layout pressure in the range of 4-5 bar over the atmospheric pressure. So, it is not necessary a supplementary energy for water pump.

The adequate details of the nozzle system is required in aim to improve the fineness of the liquid particles in the jet, and also for the jet flared. This characteristic allows to have a good efficiency in fire extinguish. The Coanda effect is important in the jet flared.

The use of thermal extinguish efficiency was proposed. A high rate of fluid is unused with the important negative consequences. This may represent an important parameter for the system exploitation.

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