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Design and construction of a hydrofluidization system. Study of the heat transfer on a stationary sphere

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ABSTRACT

In chilling and/or freezing of foods by hydrofluidization (HF), the liquid refrigerant is pumped through orifices or nozzles, creating agitating jets. The objectives of this work were to design and build a HF system of pilot scale and to study the heat transfer on a stationary copper sphere impinged by a liquid jet generated by the HF system. Parameters from the literature were taken into account for the HF system construction. Measurements of surface heat transfer coefficients in the HF system were carried out using a solution of NaCl–water as refrigerant. Copper spheres of different diameters (D) and co-linearly placed in front of a round jet generated by a 3 mm (d) diameter orifice were used. Re, Pr, the orifice-sphere distance (H/d), and the curvature of the sphere surface (d/D) were considered in a correlation for Nusselt number for the HF system. The ranges studied were: $15{,}000 \leqslant Re \leqslant 115{,}000$; $9.2 \leqslant Pr \leqslant 11.5$; $1.66 \leqslant H/d \leqslant 16.66$; and $0.15 \leqslant d/D \leqslant 0.30$. The HF system constructed is very versatile and it may be easily modified for experimental studies related to chilling and/or freezing of foods by hydrofluidization. Nusselt number range was 70–400, it increased when Re and Pr increased, it had a maximum for $3 < H/d < 10$, and it decreased when d/D increased.

1. Introduction

The modification of the conventional processes of food preservation to gain efficiency and minimize equipment size is a recognized world trend. Chilled and frozen foods represent 68% of the economic weight of the activities related to cold applications (Guilpart, 2007). For this reason, one of the objectives of the researchers in this area is to develop refrigeration systems at the lowest cost, with a small size and that allows obtaining high velocities of cooling.

Immersion chilling and freezing processes (ICF) involve direct contact of the food product with a low freezing point liquid medium maintained at a low temperature. Generally, those refrigerants are aqueous solutions of solutes such as NaCl, $CaCl_2$, KCl, ethanol, glucose, etc. A simultaneous heat and mass transfer takes place during immersion. The movement of water from the food towards the immersion solution and the movement of the solute in the opposite direction are the main differences with the conventional freezing method (Lucas and Raoult-Wack, 1998).

Immersion freezing shows as main advantages not only those associated with a fast freezing method, but also it allows saving considerable amounts of energy (Chourot et al., 2003), the technology is environmentally friendly, and new delicious products can easily be formulated including appropriate antioxidants, flavorings and macronutrients to extend the shelf-life of the products and to improve their nutritional value and sensory properties (Fikiin, 2003). The major drawbacks are the degradation of the immersion solution and the uncontrolled uptake of solutes by the foodstuffs (Lucas and Raoult-Wack, 1998).

A recently developed method of freezing called hydrofluidization (HF) (Fikiin and Fikiin, 1998) essentially uses a circulating system that pumps the refrigerating liquid upwards through orifices and/or nozzles into a refrigerating vessel, thereby creating agitating jets and increasing heat transfer to foods during freezing (Verboven et al., 2003). If the process is adequately controlled, it represents a potential application with remarkable advantages: to use equipments of small size and to improve the freezing of individual pieces of food, besides the advantages related to the ICF process.

Literature related to the design of HF systems is scarce due to it may be considered as an emerging technology. For this reason, few studies have been published: studies using a discontinuous system of lab scale (Scheerlinck et al., 2002; Verboven et al., 2003), and studies using a continuous system of pilot scale (Fikiin, 1985, 1992, 2003; Fikiin and Fikiin, 1998). In this case, knowing how the fluid flow on the food affects the heat and mass transfer is important; therefore knowing how the operation variables affect the surface heat transfer coefficient is necessary. Fikiin (1992) studied a hydrofluidization system called 'fluidized water' and proposed a correlation for the average Nusselt number using fish samples, $Nu = f(Re, Pr, \Phi)$, where Φ is the shape coefficient of the

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Nomenclature

a	coefficients in Eq. (8)
Bi	Biot number ($h\,R\,k_s^{-1}$)
C_p	heat capacity, J kg^{-1} K^{-1}
D	sphere/cylinder diameter, m
d	orifice/nozzle diameter, m
Fo	Fourier number ($\alpha\,t\,R^{-2}$)
h	surface heat transfer coefficient, W m^{-2} K^{-1}
H	distance between orifice/nozzle exit and stagnation point, m
k	thermal conductivity, W m^{-1} K^{-1}
L	jet width at nozzle exit, m
Nu	Nusselt number ($h\,D\,k_b^{-1}$)
Pr	Prandtl number ($C_p\,\mu\,k_b^{-1}$)
r	radial position, m
R	sphere radius, m
Re	Reynolds number ($\rho\,D\,v_o\,\mu^{-1}$)
S	orifice/nozzle spacing, m
T	temperature, K
t	time, s
v	average velocity, m s^{-1}
x	coordinate from stagnation point lateral to jet axis, m

Greek letters

α	thermal diffusivity, m^2 s^{-1}
β	root in Eq. (6)
θ	angle measured from stagnation point, °
μ	viscosity, kg m^{-1} s^{-1}
ρ	density, kg m^{-3}

Subscripts

0	initial
b	bulk
c	central
o	orifice
s	solid

fish. On the other hand, Verboven et al. (2003) showed a correlation for the average Nusselt number for the hydrofluidization system, using stationary aluminium spheres, $Nu = f(Re, Pr, k)$, where k is the turbulent kinetic energy. Although these correlations had good regression parameters, it is necessary to thoroughly study how the mass and heat transfer is affected by the operation variables of HF systems to design, build, model, and efficiently optimize the process.

Taking into account that it is difficult to carry out the experimental measurements in a fluidized system (Scheerlinck et al., 2002), it is necessary to study how the operation conditions affect stationary food samples (Verboven et al., 2003) before to study the fluidized system. The objectives of this work were to design and build a HF system of pilot scale and to study the heat transfer on a stationary copper sphere impinged by a liquid jet generated by the HF system.

2. Materials and methods

2.1. Design considerations for the HF system construction

The HF system should be versatile, allowing easily modifying the sample container, the orifice distribution, the orifice number, etc. The HF system should be easily transportable, small and modular. A batch HF system was chosen because its operation is simpler. Due to the food tank is open, it was necessary to consider that the refrigerating liquid returns by gravity to its reservoir. All the materials used for construction should be chemically compatible with the liquid refrigerant.

Design parameters for HF systems are scarce. Some of the parameters taking into account for the design are listed in Table 1.

Table 1
Design parameters considered for the construction of the hydrofluidization system

Parameter	Value	Reference
Bottom chamber volume/experimental tank volume	1.7	Verboven et al. (2003)
Experimental tank height	100 mm	Verboven et al. (2003)
Temperature	−20 to 20 °C	Fikiin (1992), Verboven et al. (2003)
Velocity at the jet exit	⩾0.5 m s^{-1}	Fikiin (1992), Verboven et al. (2003)
Refrigerant volume/food volume	>70	Lucas et al. (1995, 1998, 1999), Lucas and Raoult-Wack (1996)

2.2. Heat transfer studies using the HF system

2.2.1. Determination of the average surface heat transfer coefficient

Due to the expected h values resulted in $Bi > 0.1$, the lumped-heat capacity method cannot be used (Incropera and DeWitt, 1990) to determine the values of the surface heat transfer coefficient on the sphere.

The one-dimension heat conduction equation in transient state for a sphere immersed in a fluid (Carslaw and Jaeger, 1959) is

$$\frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2\frac{\partial T}{\partial r}\right) = \frac{1}{\alpha}\frac{\partial T}{\partial t} \qquad 0 < r < R, \quad t > 0 \tag{1}$$

with

$$\frac{\partial T}{\partial r} = 0 \qquad r = 0, \quad t > 0 \tag{2}$$

$$\frac{\partial T}{\partial r} + \frac{h}{k_s}(T - T_b) = 0 \qquad r = R, \quad t > 0 \tag{3}$$

$$T = T_0 \qquad 0 \leqslant r \leqslant R, \quad t = 0 \tag{4}$$

The analytical solution of Eqs. (1)–(4) is (Carslaw and Jaeger, 1959; Erdoğdu, 2005)

$$\frac{T - T_b}{T_0 - T_b} = \sum_{n=1}^{\infty} 2\frac{(\sin\beta_n - \beta_n\cos\beta_n)}{(\beta_n - \sin\beta_n\cos\beta_n)}\frac{\sin(\beta_n r/R)}{(\beta_n r/R)}\exp(-\beta_n^2 Fo) \qquad 0 \leqslant r \leqslant R, \quad t \geqslant 0 \tag{5}$$

where

$$\tan\beta_n = -\frac{1}{Bi - 1}\beta_n \quad n = 1, 2, 3, \ldots, \infty \tag{6}$$

Taking into account that $Fo > 0.2$ for all the experimental values, the first term of Eq. (5) was used (Incropera and DeWitt, 1990) and then evaluating the temperature at the geometric center (T_c), Eq. (5) becomes

$$\ln\left(\frac{T_c - T_b}{T_0 - T_b}\right) = f(\beta_1) - \beta_1^2\frac{\alpha}{R^2}t \tag{7}$$

Using the experimental values of T_c and t, and Eq. (7), values of β_1 can be obtained. Finally, Bi and average h values can be obtained from Eq. (6).

2.2.2. Nusselt number correlation

Average Nu dependence with dimensionless numbers such as Re, Pr, and numbers related to the geometry and configuration of systems of liquid or air jets impinging on a surface, are proposed in the literature (Martin, 1977; Jambunathan et al., 1992; Lee et al., 1997, 1999, 2007; Pekdemir and Davies, 1998; Pekdemir et al., 1998; Li and Garimella, 2001; Chan et al., 2002; Sarkar and Singh, 2003; Olsson et al., 2004; Zahran et al., 2006; Robinson and Schnitzler, 2007). Table 2 shows some of the correlations used for jets impinging on surfaces.

Due to a stationary sphere was used in this study, the correlation used for the average Nusselt number was (Lee et al., 1997; Robinson and Schnitzler, 2007)

$$Nu = a_1 Re^{a_2} Pr^{a_3} \left(\frac{H}{d}\right)^{a_4} \left(\frac{d}{D}\right)^{a_5} \tag{8}$$

Eq. (8) takes into account the functionality of Nu with Re, H/d and d/D as proposed by Lee et al. (1997), while the functionality with Pr as proposed by Robinson and Schnitzler (2007) was considered due to the liquid nature of the refrigerant. The thermophysical properties used in Eq. (8) were evaluated at the bulk temperature (T_b). Table 3 summarized the ranges of the dimensionless numbers used.

2.2.3. Temperature measurement

Copper spheres (k_s = 386 W m^{-1} K^{-1}, C_{ps} = 384 J kg^{-1} K^{-1}, ρ_s = 8660 kg m^{-3}) with a central hole that allowed locating the thermocouple (type T, 24 AWG, OMEGA Engineering Inc., Stamford, USA) at the geometrical center, were used for measuring temperature. The thermocouple was sealed with high thermally conductive epoxy Omegabond 101 (OMEGA Engineering Inc., Stamford, USA).

Table 2
Nu correlations found in the literature for jets impinging on surfaces

Source	Correlation	Comments
Jambunathan et al. (1992)	$Nu = Re^{a_1}\dfrac{a_2}{1-(x/d)^{a_3}}$ $a_1 = f(x/d, H/d)$	One round jet Fluid: air Plane surface
Martin (1977)	$Nu = Pr^{a_1}\dfrac{d}{x}\left[\dfrac{1-a_2(d/x)}{1+a_3\left(\frac{H}{d}-a_4\right)\frac{d}{x}}\right]f(Re)$ $f(Re) = a_5 Re^{a_6}\left(1+\dfrac{Re^{a_7}}{a_8}\right)^{a_9}$	One round jet Fluid: air Plane surface
Sarkar and Singh (2003)	$Nu = a_1^{\left[1+a_2\left(\frac{x}{d}\right)^{a_3}\right]} Re^{a_4}$	One round jet Fluid: air Plane surface
Lee et al. (1997)	$Nu = a_1 Re^{a_2}\left(\frac{H}{d}\right)^{a_3}\left(\frac{d}{D}\right)^{a_4}$	One round jet Fluid: air Spherical surface
Robinson and Schnitzler (2007)	$Nu = a_1 Re^{a_2} Pr^{a_3}\left(\frac{S}{d}\right)^{a_4}\left(\frac{H}{d}\right)^{a_5}$	Multiple round jets Fluid: water Plane surface

a_i, Coefficients; D, sphere/cylinder diameter; d, orifice/nozzle diameter; S, orifice/nozzle spacing; x, coordinate from stagnation point lateral to jet axis, and H, distance between orifice/nozzle exit and stagnation point.

Table 3
Ranges of the dimensionless numbers studied

Variable	Range
Re	15,000–115,000
Pr	9.2–11.5
H/d	1.66–16.66
d/D	0.15–0.30

Table 4
Factors and levels considered in the experiments

Factor	Levels
Flow rate (Q) (L s^{-1})	1, 2 and 3[a]
Sphere diameter (D) (mm)	10 and 20
Orifice-sphere distance (H) (mm)	5, 10, 30 and 50
Refrigerant temperature (T) (K)	263.15 (−10 °C) and 258.15 (−15 °C)

[a] It corresponds to v_o equal to 2.36, 4.72, 7.07 m s^{-1}, respectively.

Table 5
Thermophysical properties of the refrigerant (Lugo et al., 2002; Peralta et al., 2007)

Property	Temperature	
	263.15 K (−10 °C)	258.15 K (−15 °C)
Heat capacity (C_{p_b}) (J kg^{-1} K^{-1})	3330.4	3325.9
Density (ρ_b) (kg m^{-3})	1186.4	1188.7
Viscosity (μ_b) (kg m^{-1} s^{-1})	0.001463	0.001796
Thermal conductivity (k_b) (W kg^{-1} K^{-1})	0.528	0.520

The spheres were supported by hollow AISI 304 stainless steel tubes that lead the thermocouple. In no case did the ratio of the diameters of the support and the sphere exceed 20%. In practice, the support is unlikely to have caused a substantial change on the flow pattern around the sphere, since it was located behind the sphere in the direction of the flow and its diameter was probably much less than that of the wake (Isataev and Zhanabaev, 1968; Neve et al., 1981). Data of temperature were obtained through a data acquisition system Data Shuttle DS-16-8-TC (OMEGA Engineering Inc., Stamford, USA) at 4 Hz. Measurements were carried out in quadruplicate.

2.2.4. Experimental design

The factors and levels considered are shown in Table 4.

An aqueous solution of 23.1% w/w NaCl was used as liquid refrigerant. The thermophysical properties of the refrigerant are summarized in Table 5. The copper spheres were immersed in a water bath at 22 ± 1 °C to homogenize their initial temperature. A single central orifice of 3-mm diameter was used for the generation of a single liquid round jet. A total of 193 experiments were carried out to determine the h values.

2.2.5. Statistical analysis

Data were analyzed by ANOVA using Minitab 13.20 (Minitab Inc., State College, PA, USA). When differences between factors were significant ($p < 0.05$), a multiple comparison of means was performed using the Tukey test. Minitab 13.20 was also used to estimate the coefficients in Nu correlation by non-linear regression.

3. Results and discussion

3.1. Hydrofluidization system

A flow chart of the system is shown in Fig. 1.

The HF system consists of two modules (Fig. 2a), a module with the refrigeration unit and the tank of the secondary refrigerant; and a second module with the pump, the valves, the flowmeters, and the hydrofluidization unit.

The main characteristics of the HF system are described below.

3.1.1. Refrigeration unit

The refrigeration unit is a DANFOSS BLUESTAR model HCM 036 (Danfoss do Brasil Ind. e Com. Ltda, São Paulo, Brazil), with a temperature controller DANFOSS model EKC 101 (Danfoss do Brasil

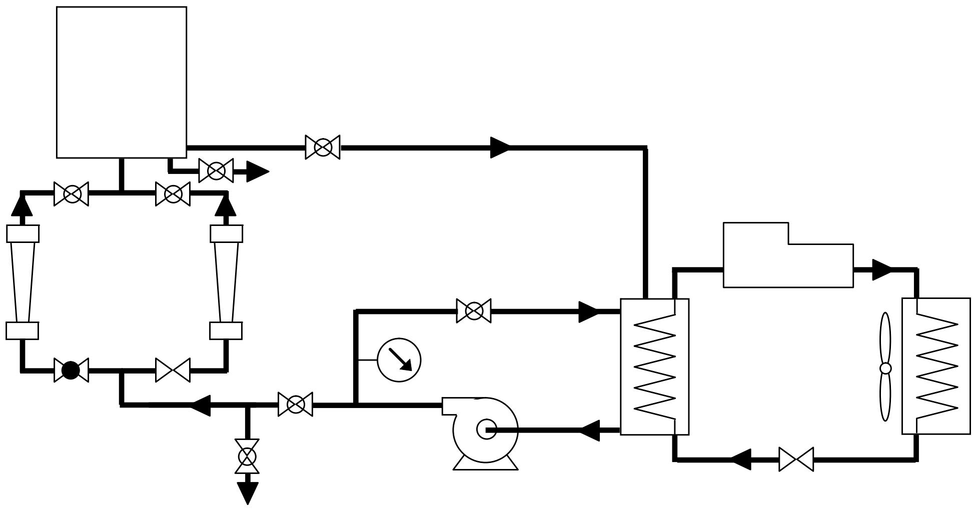

Fig. 1. Flow chart of the hydrofluidization system.

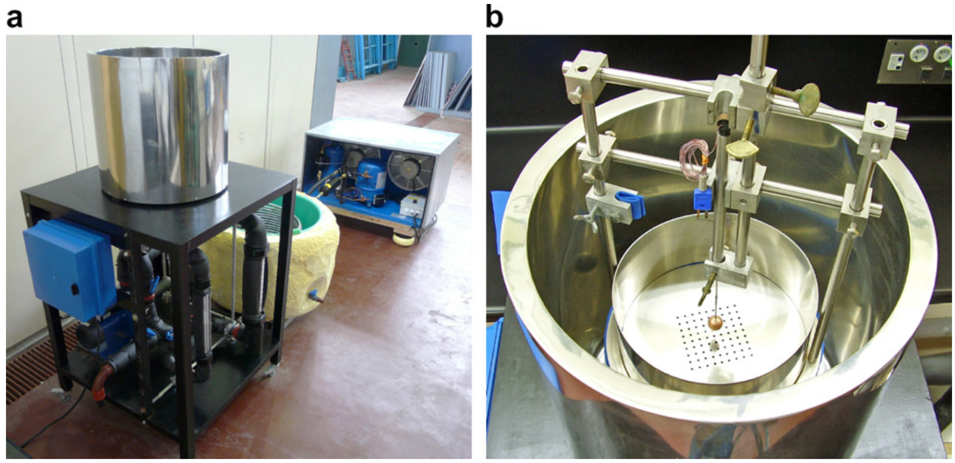

Fig. 2. Hydrofluidization system: (a) view of the complete system and (b) experimentation tank view.

Ind. e Com. Ltda, São Paulo, Brazil) of ±1 °C. This unit has an evaporator coil (bare pipe) made of stainless steel AISI 304 with a length of 25 m. The pipe is used to cool the secondary refrigerant. The refrigeration unit allows reaching −15 °C by removing approximately 2000 W at −15 °C. The secondary refrigerant is cooled in an isolated and stirred tank of 72 L.

3.1.2. Circulating pump unit

The secondary refrigerant is circulated through pipes using a LOWARA pump model CEA(M) 70/3 (Lowara S.R.L., Vincenza, Italy) of 0.5 HP with the casting, impeller, and shaft made of stainless steel AISI 316 L. The pump is controlled by an adjustable speed AC driver controller for asynchronous motors TELEMECANIQUE model ALTIVAR 16 ATV-16 U09M2 (Schneider-Electric, Florida, Buenos Aires, Argentina). The pressure at the pump discharge is measured by a manometer with a gauge reading from 0 to 4 kg cm^{-2}.

3.1.3. Flow rate measurement and control

Flow rate is measured by two rotameters in parallel made of acrylic ODIN (ODIN S.A., Ensenada, Buenos Aires, Argentina) model CMEL-1L and CMEL-5L, with graduations for water at 20 °C and 1 ATA (±2% of full scale) of 0.3–3.5 L min^{-1} and 10–100 L min^{-1}, respectively. The flow rate is controlled by globe and lift valves made of stainless steel AISI 316. Pipes are made of polypropylene of 1/2 and 1 in. nominal diameter.

3.1.4. Hydrofluidization unit

The hydrofluidization unit was made of stainless steel AISI 304. The unit consists of three parts: (a) a bottom chamber of 8.3 L to homogenize the flow before generating the liquid jets; (b) an experimentation tank of 4.9 L above the plate with orifices where the food samples are placed; and (c) an annular tank of 20 L that receives the liquid recirculated to the refrigerant tank. The bottom chamber, the experimentation tank and the plate with orifices are easily removable due to they are attached using flanged joints. A schematic view is shown in Fig. 3.

3.1.5. Sample set-up

A rack set-up allows placing the sample in the experimentation tank. It was made of stainless steel AISI 304 (Fig. 2b).

3.2. Experimental values of Nu

In Fig. 4, values of *Nu* vs. H/d for the different values of *Re* and d/D studied are shown. It can be observed that *Nu* values were in the range of 70–400. Moreover, *Nu* increased when *Re* and *Pr*

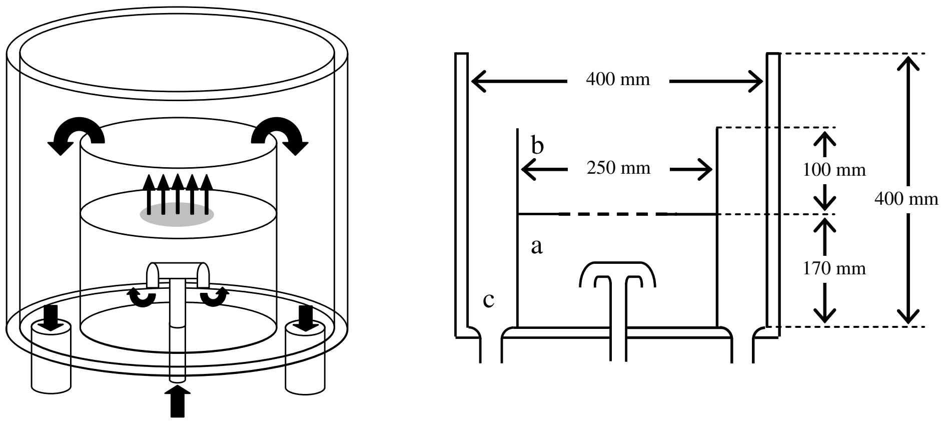

Fig. 3. Scheme of the hydrofluidization unit: (a) bottom chamber, (b) experimentation tank, and (c) annular tank.

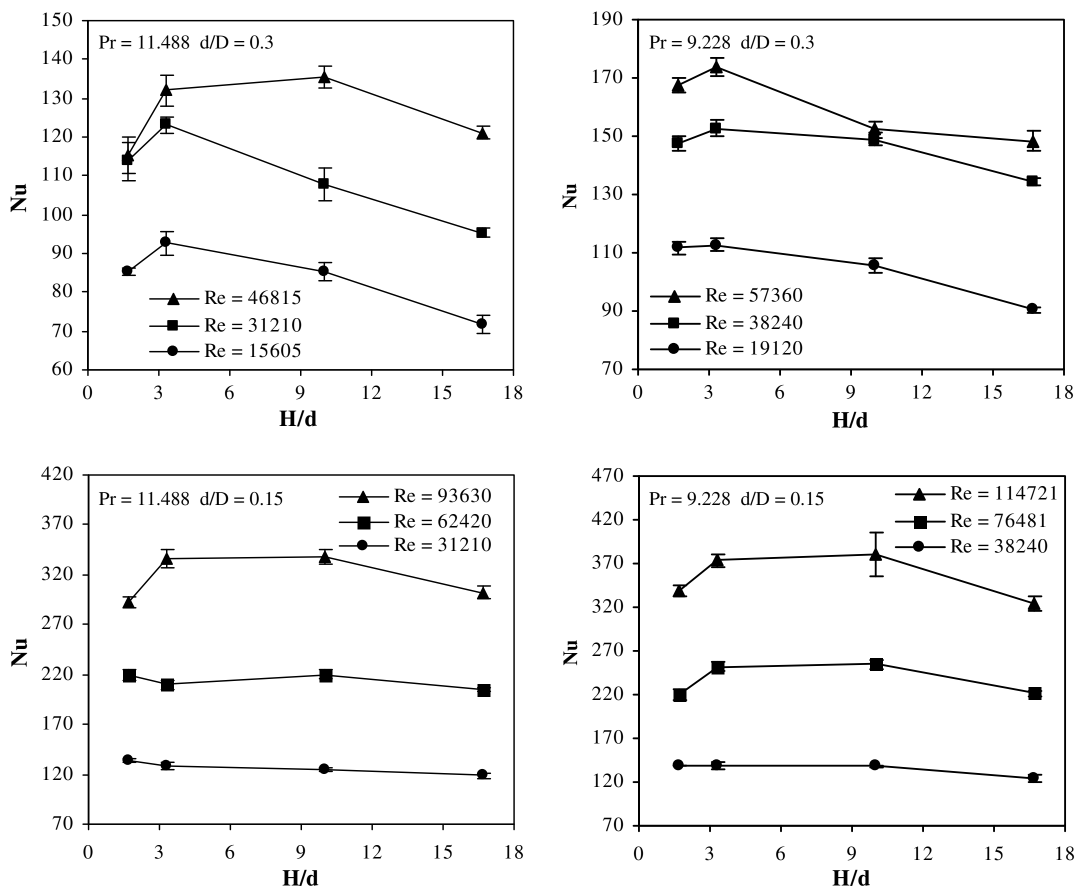

Fig. 4. Effect of H/d distance on the Nu values for different Re, Pr and d/D. Bars indicate standard deviations.

increased. On the other hand, Nu showed a maximum for $3 < H/d < 10$ and decreased when d/D increased.

The average surface heat transfer coefficients were in the range of 3000–10,500 W m^{-2} K^{-1}. These values are greater than the values found for similar systems (Fikiin, 1992; Scheerlinck et al., 2002; Verboven et al., 2003). However, it should be taken into account that the experimental conditions used in each case were different. The h values obtained allow verifying the method proposed in Section 2.2.1 due to the Bi values resulted in the range of 0.03–0.272.

In the literature, the average Nusselt number usually shows the relationships $Nu \propto Re^m$, where $m > 0$ (Martin, 1977; Jambunathan et al., 1992; Lee et al., 1997, 1999, 2007; Pekdemir and Davies, 1998; Pekdemir et al., 1998; Li and Garimella, 2001; Chan et al., 2002; Sarkar and Singh, 2003; Olsson et al., 2004; Zahran et al., 2006; Robinson and Schnitzler, 2007), and $Nu \propto Pr^n$, where $n > 0$ (Martin, 1977; Li and Garimella, 2001; Robinson and Schnitzler, 2007); which agrees with our results.

The maximum for Nu in the range of $3 < H/d < 10$ can be found in the literature for similar systems (Giralt et al., 1977; Lee et al., 1999; Robinson and Schnitzler, 2007). Some authors explained this phenomenon as follows: for $H/d < 3$, the sphere is within the jet potential core region, that is, the centerline velocity is 95–98% of the velocity at the orifice exit. Due to the turbulent intensity in-

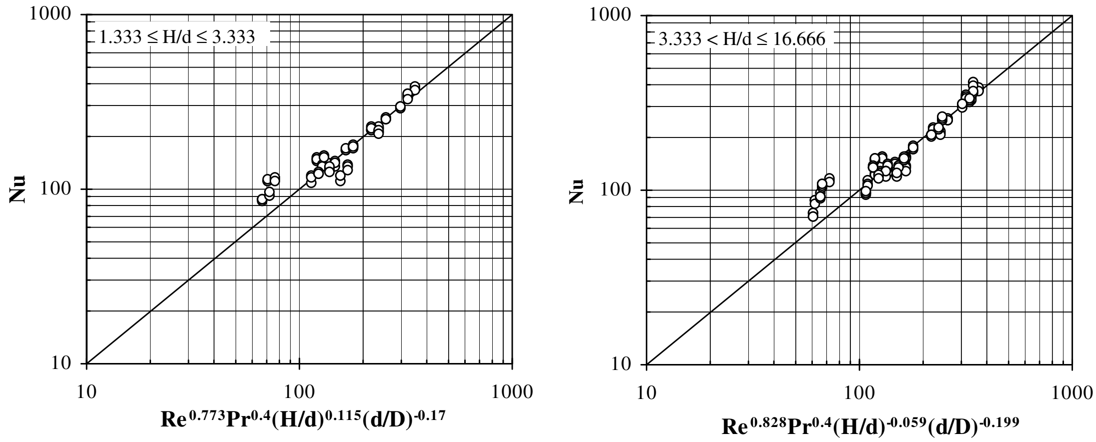

Fig. 5. Theoretical vs. experimental *Nu* values for the ranges of H/d studied.

creases as the fluid moves far from the jet exit, *Nu* increases. For $3 < H/d < 10$, the sphere is still positioned within the jet potential core but the turbulence is fully developed resulting in a maximum for *Nu*. For $H/d > 10$, although the turbulent intensity is maximal, the fluid velocity is degraded because the sphere is positioned in the turbulent shear layer, resulting in a decrease of the *Nu* (Schuh and Persson, 1964; Giralt et al., 1977; Ashforth-Frost and Jambunathan, 1996; Lee et al., 1997; McDaniel and Webb, 2000).

In the literature, *Nu* increases when surface curvature (d/D) increases (Lee et al., 1997) or shows a maximum (Schuh and Persson, 1964). Schuh and Persson (1964) reported a maximum for *Nu* at $L/d = 0.13$ (L: jet width at nozzle exit) for $H/d < 20$ and $35{,}000 < Re < 50{,}000$, while *Nu* decreased when L/d increased for $L/d > 0.13$. Pekdemir et al. (1998) suggested that these two different behaviors may be explained by the L/d range used in each case.

3.3. Nusselt number correlation

The correlations obtained using Eq. (8) and considering two ranges for H/d were

$$Nu = 0.0111 Re^{0.773} Pr^{0.4} \left(\frac{H}{d}\right)^{0.115} \left(\frac{d}{D}\right)^{-0.17} \quad 1.666 \leqslant H/d \leqslant 3.333 \quad (R^2 = 0.936) \tag{9}$$

$$Nu = 0.0072 Re^{0.828} Pr^{0.4} \left(\frac{H}{d}\right)^{-0.059} \left(\frac{d}{D}\right)^{-0.199} \quad 3.333 < H/d \leqslant 16.666 \quad (R^2 = 0.948) \tag{10}$$

Two different ranges for H/d were considered because *Nu* showed a maximum for the conditions studied (Fig. 4). Fig. 5 shows the theoretical vs. experimental *Nu* values, which were very well correlated.

Due to in the literature $0.4 \leqslant a_3 \leqslant 0.44$ (Martin, 1977; Li and Garimella, 2001; Robinson and Schnitzler, 2007), a_3 was assumed to be equal to 0.4 as suggested by Robinson and Schnitzler (2007).

Isataev and Zhanabaev (1968) reported $a_2 = 0.75$ for $d/D \leqslant 0.4$. The authors also suggested that $a_2 = 0.5$ is a characteristic value for $\theta < 90°$ (θ: angle measured from stagnation point), while $0.7 \leqslant a_2 \leqslant 0.8$ are characteristic values for the rear half of the sphere ($\theta > 90°$). This result might indicate that the heat transfer would be mainly affected by the heat transfer in the backside of the sphere. On the other hand, Pekdemir et al. (1998), studying the mass transfer in similar systems, found that for $17{,}000 \leqslant Re \leqslant 400{,}000$, a_1 and a_2 were in the range of 0.033–0.05 and 0.77–0.82, respectively.

Lee et al. (1997) found values of a_4 equal to -0.2 and -0.12 for $2 \leqslant H/d < 6$ and $6 \leqslant H/d \leqslant 10$, respectively, when $11{,}000 \leqslant Re \leqslant 50{,}000$ and $0.034 \leqslant d/D \leqslant 0.089$. Li and Garimella (2001) reported $a_4 = -0.07$ for $1 < H/d < 5$ and $8500 < Re < 23{,}000$.

In summary, the coefficients in the *Nu* correlation proposed in this work are in agreement with the literature findings for similar systems.

4. Conclusions

A hydrofluidization system for chilling and/or freezing of foods in pilot scale was designed and constructed. The system works in a batch mode, is modular and versatile, and allows operating in wide ranges of flow and temperature.

A single round jet impinging on a stationary sphere was studied. An experimental correlation between the average Nusselt number and the dimensionless numbers Re, Pr, H/d and d/D was obtained for the hydrofluidization system. Further studies considering different arrangements of jets and different ranges for the operating conditions are a subject of current study in our research group.

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References

Ashforth-Frost, S., Jambunathan, K., 1996. Effect of nozzle geometry and semi-confinement on the potential core of a turbulent axisymmetric free jet. International Communications in Heat and Mass Transfer 23 (2), 155–162.
Carslaw, H.S., Jaeger, J.C., 1959. Conduction of Heat in Solids. Oxford at the Clarendon Press, London, UK.
Chan, T.L., Leung, C.W., Jambunathan, K., Ashforth-Frost, S., Zhou, Y., Liu, M.H., 2002. Heat transfer characteristics of a slot jet impinging on a semi-circular convex surface. International Journal of Heat and Mass Transfer 45, 993–1006.
Chourot, J.M., Macchi, H., Fournaison, L., Guilpart, J., 2003. Technical and economical model for the freezing cost comparison of immersion, cryomechanical and air blast freezing processes. Energy Conversion and Management 44, 559–571.
Erdoğdu, F., 2005. Mathematical approaches for use of analytical solutions in experimental determination of heat and mass transfer parameters. Journal of Food Engineering 68, 233–238.
Fikiin, A.G., 1985. Method and System for Water Cooling of Fish. Bulgarian Patent No. 40164. INRA, Sofia, Bulgaria.
Fikiin, A.G., 1992. New method and fluidized water system for intensive chilling and freezing of fish. Food Control 3 (3), 153–160.
Fikiin, K., 2003. Nolvelties of food freezing research in Europe and beyond. Institut National de la Recherche Agronomique. Project No. QLK1-CT-2000-00040, 56p.

Fikiin, K.A., Fikiin, A.G., 1998. Individual quick freezing of foods by hydrofluidisation and pumpable ice slurries. In: IIR Proceedings Series "Refrigeration Science and Technology", vol. 6, pp. 319–326.

Giralt, F., Chia, C.J., Trass, O., 1977. Characterization of the impingement region in an axisymmetric turbulent jet. Industrial and Engineering Chemistry Fundamentals 16 (1), 21–28.

Guilpart, J., 2007. An overview on the cold applications throughout the world. In: 5th ALCUEFOOD Seminar. Refrigeration, Energy and Environment: Challenges and Perspectives, São Paulo, Brazil.

Incropera, F.P., DeWitt, D.P., 1990. Fundamentals of Heat and Mass Transfer. John Wiley & Sons, New York.

Isataev, S.I., Zhanabaev, Z.Zh., 1968. Heat transfer of a sphere in a jet flow. Inzhenerno-Fizicheskii Zhurnal 14 (4), 586–592.

Jambunathan, K., Lai, E., Moss, M.A., Button, B.L., 1992. A review of heat transfer data for single circular jet impingement. International Journal of Heat and Fluid Flow 13 (2), 106–115.

Lee, C.H., Lim, K.B., Lee, S.H., Yoon, Y.J., Sung, N.W., 2007. A study of the heat transfer characteristics of turbulent round jet impinging on an inclined concave surface using liquid crystal transient method. Experimental Thermal and Fluid Science 31, 559–565.

Lee, D.H., Chung, Y.S., Kim, D.S., 1997. Turbulent flow and heat transfer measurements on a curved surface with a fully developed round impinging jet. International Journal of Heat and Fluid Flow 18, 160–169.

Lee, D.H., Chung, Y.S., Kim, M.G., 1999. Turbulent heat transfer from a convex hemispherical surface to a round impinging jet. International Journal of Heat and Mass Transfer 42, 1147–1156.

Li, C.Y., Garimella, S.V., 2001. Prandtl-number effects and generalized correlations for confined and submerged jet impingement. International Journal of Heat and Mass Transfer 44, 3471–3480.

Lucas, T., Francois, J., Bohuon, P., Raoult-Wack, A.L., 1999. Factors influencing mass transfer during immersion cold storage of apples in NaCl/sucrose solutions. Lebensmittel-Wissenschoft und Technologie 32, 327–332.

Lucas, T., François, J., Raoult-Wack, A.L., 1998. Transport phenomena in immersion-cooled apples. International Journal of Food Science and Technology 33, 489–499.

Lucas, T., Raoult-Wack, A.L., 1996. Immersion chilling and freezing: phase change and mass transfer in model food. Journal of Food Science 61 (1), 127–132.

Lucas, T., Raoult-Wack, A.L., 1998. Immersion chilling and freezing in aqueous refrigerating media: review and future trends. International Journal of Refrigeration 21 (6), 419–429.

Lucas, T., Raoult-Wack, A.L., Sereno, A., 1995. Control of impregnation phenomena during quick immersion chilling and freezing. In: 19th International Congress of Refrigeration, vol. 11, Hague, The Netherlands, pp. 242–248.

Lugo, R., Fournaison, L., Chourot, J.-M., Guilpart, J., 2002. An excess function method to model the thermophysical properties of one-phase secondary refrigerants. International Journal of Refrigeration 25, 916–923.

McDaniel, C.S., Webb, B.W., 2000. Slot jet impingement heat transfer from circular cylinders. International Journal of Heat and Mass Transfer 43, 1975–1985.

Martin, H., 1977. Heat and mass transfer between impinging gas jets and solid surfaces. In: Hartnett, J.P., Irvine, T.F. (Eds.), Advances in Heat Transfer, vol. 13. Academic Press, New York.

Neve, R.S., Nelson, R., Kotsiopoulos, P., 1981. The drag force on spheres in thin jets. Journal of Fluid Mechanics 107, 521–531.

Olsson, E.E.M., Ahrné, L.M., Trägårdh, A.C., 2004. Heat transfer from a slot air jet impinging on a circular cylinder. Journal of Food Engineering 63, 393–401.

Pekdemir, T., Davies, T.W., 1998. Mass transfer from rotating circular cylinders in a submerged slot jet of air. International Journal of Heat and Mass Transfer 41, 3441–3450.

Pekdemir, T., Davies, T.W., Sara, O.N., 1998. Convective mass transfer from cylinders in a jet flow. Industrial and Engineering Chemical Research 37, 1560–1566.

Peralta, J.M., Rubiolo, A.C., Zorrilla, S.E., 2007. Prediction of heat capacity, density and freezing point of liquid refrigerant solutions using an excess Gibbs energy model. Journal of Food Engineering 82, 548–558.

Robinson, A.J., Schnitzler, E., 2007. An experimental investigation of free and submerged miniature liquid jet array impingement heat transfer. Experimental Thermal and Fluid Science 32 (1), 1–13.

Sarkar, A., Singh, R.P., 2003. Spatial variation of convective heat transfer coefficient in air impingement applications. Journal of Food Science 68 (3), 910–916.

Scheerlinck, N., Jancsók, P., Verboven, P., Nicolaï, B.M., 2002. Influence of shape on the fast freezing of small fruits by means of hydrofluidisation. In: AgEng, No. 02-PH-034, Budapest, Hungary.

Schuh, H., Persson, B., 1964. Heat transfer on circular cylinders exposed to free-jet flow. International Journal of Heat and Mass Transfer 7, 1257–1271.

Verboven, P., Scheerlinck, N., Nicolaï, B.M., 2003. Surface heat transfer coefficients to stationary spherical particles in an experimental unit for hydrofluidisation freezing of individual foods. International Journal of Refrigeration 26, 328–336.

Zahran, R.R., Sedahmed, G.H., Abdelwahab, O.E., El-Sarraf, W.M., 2006. Mass-transfer-controlled impingement corrosion at the jet inlet zone of an annulus under turbulent flow. Industrial and Engineering Chemical Research 45, 1160–1166.