INFLUENCE OF FLOW REGIMES ON TEMPERATURE HETEROGENEITIES WITHIN A SCRAPED SURFACE HEAT EXCHANGER

ERIC DUMONT, DOMINIQUE DELLA VALLE and FRANCINE FAYOLLE[1]

Ecole Nationale Supérieure d'Ingénieurs des Techniques Agricoles et Alimentaires
Département Génie des Procédés Alimentaires
Rue de la Géraudière - BP 82225
44322 NANTES Cedex 3 France

AND

JACK LEGRAND

Laboratoire de Génie des Procédés — UPRES EA 1152
Université de Nantes
CRTT — BP 406
44602 SAINT NAZAIRE Cedex France

Accepted for Publication March 13, 2000

ABSTRACT

In industrial applications, fluids processed in scraped surface heat exchangers often show large temperature heterogeneities at the exchanger outlet. Our study deals with the thermal evolution of model fluids, Newtonian and non-Newtonian in heating or cooling conditions and allows us to link the phenomena of appearance and disappearance of temperature heterogeneities with the changes in the flow pattern within the exchanger. Based on literature data dedicated to scraped surface heat exchangers as well as to annular spaces without blades, we have shown that thermally homogeneous products can be obtained when Taylor vortices appear in the exchanger. Studies done on the exchanger with and without blades show that the thermal behavior is basically the same for both geometries but with a difference in critical Taylor numbers value for change in heat transfer regime. The presence of blades promotes the appearance of instabilities at lower values of generalized Taylor number (Ta_g = 10 with blades; Ta_g = 39 without blades). It shows as well, that the value of critical Taylor number in scraped surface heat exchanger closely depends upon the flow-rate even for very low values for Re_{axg} ($Re_{axg} << 1$).

[1] Correspondence should be sent to: Francine Fayolle, ENITIAA, Département GPA, Rue de la Géraudière - BP 82225, 44322 NANTES Cedex 3 France

INTRODUCTION

Scraped surface heat exchangers are used to process highly viscous fluids. They are mostly used in the food industry, to process cream cheese, fruit concentrate and ice cream among others. In this cylinder type heat exchanger, blades fixed on a rotating shaft are scraping the exchange surface and preventing crust formation on that same surface, which leads to an improvement of heat transfer. However, the mechanical fragility of treated products induces working constraints, which do not always guaranty a sufficient rotating speed to allow correct mixing. Industrially, some of the products treated in these exchangers show high temperature heterogeneities. These heterogeneities can reach up to 50C, and have been shown out by Härröd (1990d). For this author, the weak heat transformation of the product is due both to flow regime and shape of the shaft and blades.

In this type of exchanger, the flow is the result of the superposition a Poiseuille flow in an annular space and a Couette flow, on which perturbations created by the blades are added. This flow pattern is particularly complex, and has only been weakly studied directly. Most of the works on the subject tend to model the geometry of the scraped surface heat exchanger as an annular space without blades, where hydrodynamics is well known. The most important question then, is the influence of the introduction of blades. Two schemes are then possible: the first one, most commonly found in the literature (Härröd 1986; Abichandani *et al.* 1987; Naimi 1989), assumes that flow patterns in scraped surface heat exchanger present some similarities with the one obtained in annular spaces without blades. Depending on the rotating speed of the rotor, two different flow regimes can be shown: laminar or vortex flow. Without blades, the change between these two regimes is due to the apparition of vortices at a critical value of Taylor number (Ta=40) which is increasing with Re_{axg}. Vortices appear first at the exchanger outlet and gradually fill the annular space up to the inlet when the rotating velocity increases (Nouar 1986).

The second scheme supposes that introducing blades in the annular space does not allow formation of vortex cells. It has been shown that there can be two different flow regimes, but it has never been clearly shown that the transition was due to Taylor vortices appearance. Therefore, Maingonnat *et al.* (1987) have studied heating capacities of both annular space with an inner rotating cylinder and scraped surface heat exchanger and have shown very different thermal and hydrodynamic behaviors for Taylor numbers less than 100.

The lack of knowledge on the structure of the different flow patterns existing in a scraped surface heat exchanger and their influence on temperature heterogeneities at the exchanger outlet justify our study of the influence of rotating speed and of the blades on the thermal characteristics of products treated in this type of heat exchanger. We have looked for a method that allows following the evolution of the flow as well as to show the appearance of temperature heterogeneities within an industrial type exchanger. We will first explain in what extent literature data on thermal working of scraped surface heat exchangers allowed us to set a satisfying experimental method, the method itself will be detailed in the next paragraph.

Scraped surface heat exchangers follow, with respect to rotating speed of the rotor, two different characteristic thermal regimes. The first regime is characterized A heat transfer limited by poor radial mixing (Benezech 1988; Härröd 1990b). The flow structure is not stable (Härröd 1990d) and the fluid is stagnant in some areas in the exchanger, probably in the inlet and outlet bowls, (Benezech 1988), whereas some changelings appear (Maingonnat 1982). In some cases, the product can go through the SSHE without been neither heated nor cooled (Härröd 1990d). Axial diffusion characterized by back-mixing effects in the inlet bowl is negligible (Benezech 1988; Maingonnat 1982; Härröd 1990c). Finally, heat transfer at the exchanger outlet disappears (Maingonnat 1982) which leads to unstable temperature at the outlet.

The second thermal regime appears when the rotating speed is increased. The heat transfer is then promoted by a more efficient radial mixing (Härröd 1990c), and stable temperature in the exchanger can be obtained. The effect of back-mixing in the inlet bowl is then of great importance (Härröd 1990c): it is characterized by a sharp temperature increase (if heating) or a sharp temperature decrease (if cooling) of the product when it enters the exchanger (in the inlet bowl) and before it flows along the exchange surface itself.

Temperature measurements in both the inlet and outlet bowls seems then the best way to characterize radial and axial mixing in the exchanger. We will then use these mixing properties within the exchanger to emphasize the transition between the different flow patterns in a scraped surface heat exchanger in real thermal working.

MATERIALS AND METHODS

Description of the Pilot-plant

This work is realized on an industrial exchanger Duprat TR 13×60 with two blades diametrically fixed. All dimensions are given in Table 1. Inlet and outlet bowls are equipped in order to measure the temperature in 5 different points on a same flat section at 2 cm of the bowl bottom (Fig. 1). The goal of the SSHE is to evenly heat the product. Then, the place and the choice of an important number of thermocouples allow us to obtain information about the radial evolution of the bowl temperature.

TABLE 1.
GEOMETRICAL CHARACTERISTICS OF SSHE

stator diameter (d_s)	0.13 m
stator length (L)	0.60 m
ratio (L/d_s)	4.61
rotor diameter (d_r)	0.080 m
ratio (d_r/d_s)	0.615
exchange surface	0.24 m^2

Thermocouples are of chromel/alumel type (accuracy: ± 0.1C), and are linked to an AOIP SAM20 data acquisition system. The pilot plant includes: an inlet tank; a volumetric pump; the test exchanger working either in cooling conditions (Glycol water) or in heating conditions (vapor); an electromagnetic flow-meter PICOMAG (DM 16733 type from Tegmag Flowtec). A platinum probe placed before the exchanger measures the entrance product temperature (Tp). The accuracy of this probe is ± 0.1C.

Working Fluids

Two types of fluids were used: a non-Newtonian solution of guar gum (2.3% weight) and two different Newtonian solutions of polyalkylene glycol (Emkarox from I.C.I.). Rheological properties of the different solutions were determined with a rheometer Rheomat 500 (Contraves) with coaxial cylinders (SVDIN type). The rheological behavior of the guar gum can be correctly predicted by a power law $\tau = k\dot{\gamma}^n$ within the range of shear rate in the apparatus ($50<\dot{\gamma}<600 s^{-1}$). The temperature dependence of the rheological properties of guar gum and Emkarox solutions is shown in Table 2.

Operating Conditions

Each experiment is done in the following way: flow rate, rotating speed of the rotor and product temperature Tp before the inlet, are adjusted to the desired values. Five minutes after final adjustment, a first measurement is realized, then 3 new runs are done every 5 min. The average of these 4 runs is then taken. For cooling experiments, guar gum solution was used. The inlet product temperature is Tp = 45C. Three different flow-rates were studied: 55, 90, and 120 $l.h^{-1}$. For each flow-rate, the rotating speed varies from 60 to 1100 $tr.min^{-1}$. Preliminary heating experiments were performed with Emkarox solutions. The inlet product is Tp = 5C. Taking into account the higher viscosity of Emkarox,

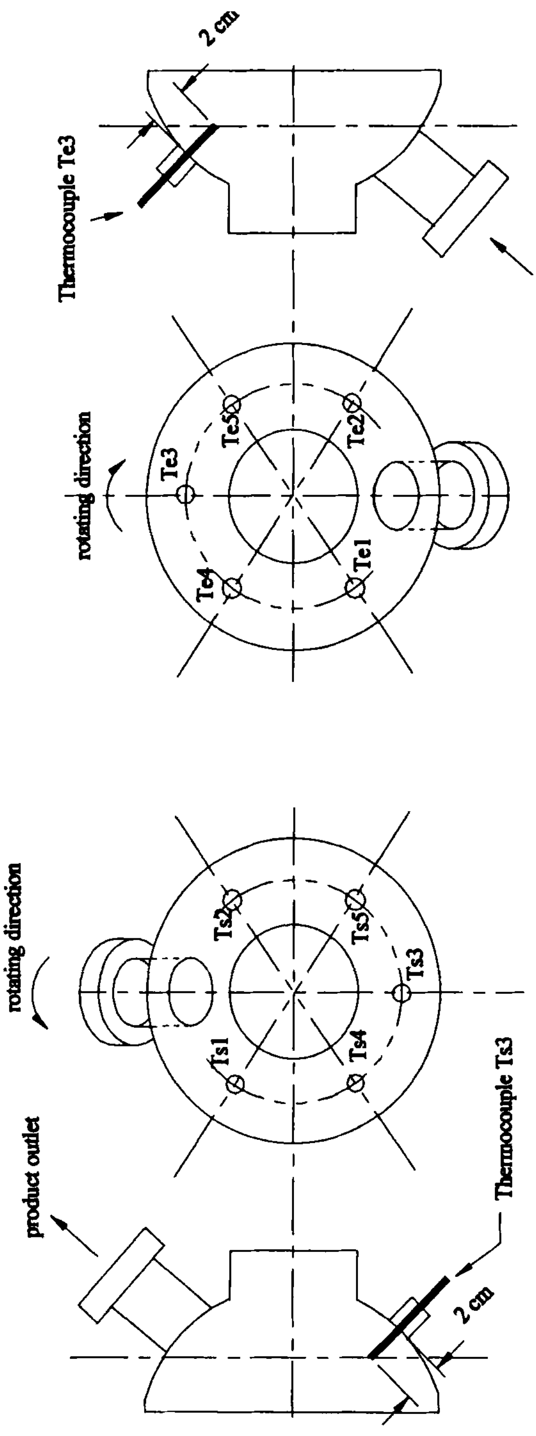

FIG. 1. INSTRUMENTATION OF THE INLET AND OUTLET BOWLS

TABLE 2.
EVOLUTION OF RHEOLOGICAL PROPERTIES OF THE PRODUCTS USED WITH RESPECT TO TEMPERATURE

Guar gum	$n = 0.198 + 0.00115\ T$ $k = 71.7 \exp(-0.013\ T)$	$10 < T < 50C$
Emkarox 40%	$\eta = 5.20 - 1.15 \ln T$ (Pa.s)	$5 < T < 60C$
Emkarox 45%	$\eta = 8.06 - 1.72 \ln T$ (Pa.s)	$5 < T < 60C$

rotating speed of the blades varies from 170 to 1100 $tr.min^{-1}$ for a flow-rate value of 160 $l.h^{-1}$.

Generalized axial Reynolds number (Re_{axg}) and generalized Taylor number (Ta_g) are used to characterize the flow pattern. Physical properties of the three solutions are determined at a mean product temperature in the exchanger: $n = 0.25$, $k = 50\ Pa.s^n$ and $\rho = 1000\ kg.m^{-3}$ for guar gum; $\eta = 1.8$ Pa.s and $\rho = 1057\ kg.m^{-3}$ for Emkarox 40%; $\eta = 3.5$ Pa.s and $\rho = 1065\ kg.m^{-3}$ for Emkarox 45%.

Our study is based on the evolution of temperature in the inlet and outlet bowls with respect to Ta_g. We determine the maximum temperature difference existing within each bowl, ΔT_{max}, which is defined as the difference between the highest temperature and the lowest temperature in the bowl. Of course, the thermocouples placed inside the bowl are going to influence the flow and the temperature gradient by the same manner. However, this influence can only have improving effects on the mixing in the bowl, and a better mixing will lead to a reduction of the temperature differences. The temperature differences measured are then minored values of the real ones. We also use the temperature fall (or raise depending on the case studied) undergone by the fluid when it enters the inlet bowl. This temperature jump is the difference between the mean of the 5 temperatures measured in the bowl and the inlet temperature (Tp):

$$T_{jump} = \left[\left(1/5\sum_{i=1}^{5} Te_i\right) - Tp\right]$$

RESULTS AND DISCUSSION

With Blades

Figure 2 shows the thermal behavior of the SSHE with respect to Ta_g in the case of cooling of guar gum, for a flow-rate of 120 $l.h^{-1}$ (the same remarks could be drawn for flow-rates of 55 $l.h^{-1}$ and 90 $l.h^{-1}$). For $Ta_g < 6.4$, there is a large temperature heterogeneity of the product in the outlet bowl ($\Delta Tmax$ outlet

$\approx$ 10C) and a small heterogeneity in the inlet bowl (around 2C) resulting from the thermocouple in position Te_3, opposite from the inlet area of the product. The four other thermocouples (Te_1, Te_2, Te_4 and Te_5) indicate temperatures nearly equal to Tp. The flow pattern in the SSHE is characterized by a large increase of temperature heterogeneity between the outlet and the inlet, which is due to a poor mixing of the blades. The flow pattern is then pure laminar flow. Naimi *et al.* (1993) point out that in this case, the rotation of the inlet cylinder does not improve heat transfer.

For Ta_g between 6.4 and 13.3, there is an important modification of the flow within the SSHE. Temperature heterogeneities in the inlet bowl are going down to disappear (ΔTmax outlet $\rightarrow$ 0C) and temperature heterogeneities in the inlet bowl increase up to 10C, whereas a significant temperature jump appears at the inlet. This drastic change in the flow pattern might be due to the appearance of vortices just before the outlet bowl. The presence of Taylor vortices, which considerably increase the effects of mixing (Härröd 1990a), would explain the thermal homogeneity of the product at the outlet. Baccar and Abid (1997) show numerically that the pair of rotating vortices appears at the edges of the scraper. Meanwhile, heat axial dispersion would increase in the exchanger with respect to Ta_g (Härröd 1986) and leads to the effect of back-mixing in the inlet bowl. This result is well correlated to the experimental results obtained by Maingonnat *et al.* (1985) and with numerical simulations obtained by Baccar and Abid (1997). Back-mixing effects are confirmed by the evolution of the temperature obtained with the 5 thermocouples placed in the inlet bowl. Temperatures Te_1, Te_3, Te_4 and Te_5 follow an important decrease, dependent on to the thermocouple position; only temperature Te_2 remains equal to the temperature of the product before it enters the exchanger. It can be assumed that the flow structure, dependent on the bowl geometry as well as the rotation of the blades and the flow-rate, is not yet perturbed by the effects of back-mixing in this part of the bowl.

When $Ta_g > 13.3$, Taylor cells are developing throughout the exchanger up to the inlet. These counter-rotated cells contribute to the disturbance of the Te_2 area (from $Ta_g = 23$) then to homogenize (Fig. 2) the temperature in the inlet bowl ($Ta_g = 37$).

For value greater than $Ta_g = 37$, Taylor vortices fill the whole annular space of the SSHE. The temperature jump followed by the product in the inlet bowl becomes very high: 35 to 40% of the thermal treatment are done in this part of the exchanger (Maingonnat 1982). Stable and homogeneous temperatures at the inlet as well as the outlet then characterize the vortex flow throughout the exchanger. The heat transfer performances of the SSHE, which were increased with respect to the rotating speed, may not be able to grow any more at that point (Balasubramaniam and Sastry 1996). Figure 3 shows the results obtained in the case of heating with Newtonian solutions of Emkarox.

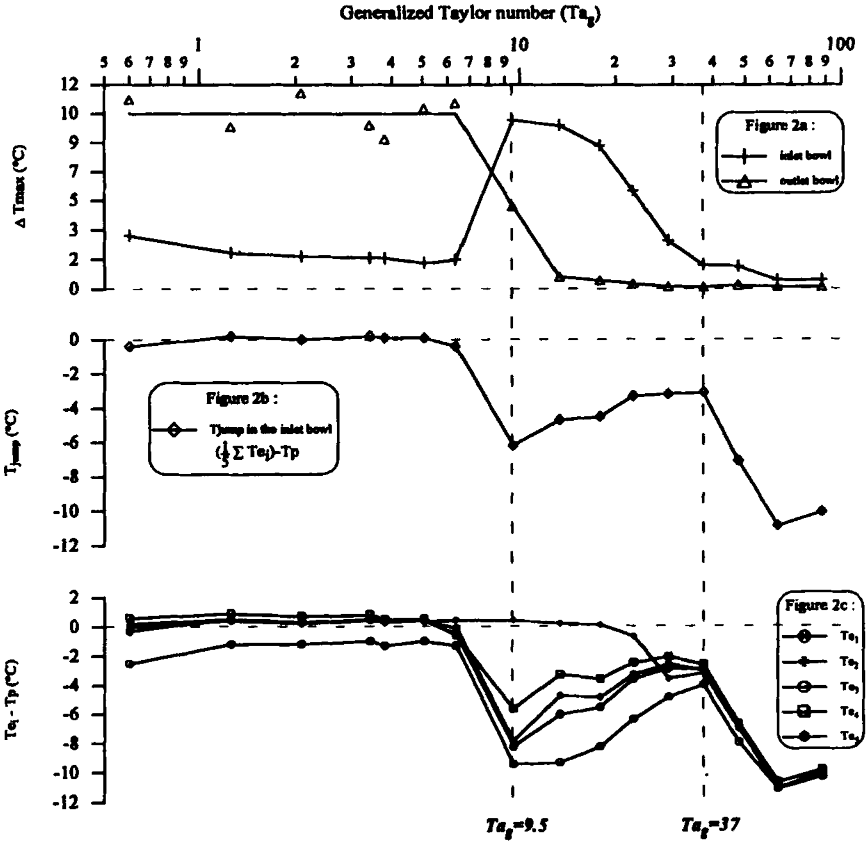

FIG. 2. EVOLUTION OF THE THERMAL BEHAVIOR OF THE SSHE WITH RESPECT TO GENERALIZED TAYLOR NUMBER IN THE CASE OF COOLING OF A GUAR GUM SOLUTION (2.3% WEIGHT); FLOWRATE: 120 $l.h^{-1}$ (Re_{axg}=0.0004)

a) Evolution of the temperature gradient inside the inlet and outlet bowls.

b) Evolution of the temperature jump inside the inlet bowl.

c) Evolution of the temperature gradient measured inside the inlet bowl.

For the cooling of guar gum, it is shown that the type of evolution of temperature jump with respect to rotating speed of the rotor is similar for all values of Re_{axg}. However, vortices occur for Taylor number increasing with respect to Re_{axg} (Fig. 4). It is interesting to note that this well-known result in the literature (Nouar 1986; Benezech 1988) remains true for very small values of the axial Reynolds number and for non-Newtonian fluids.

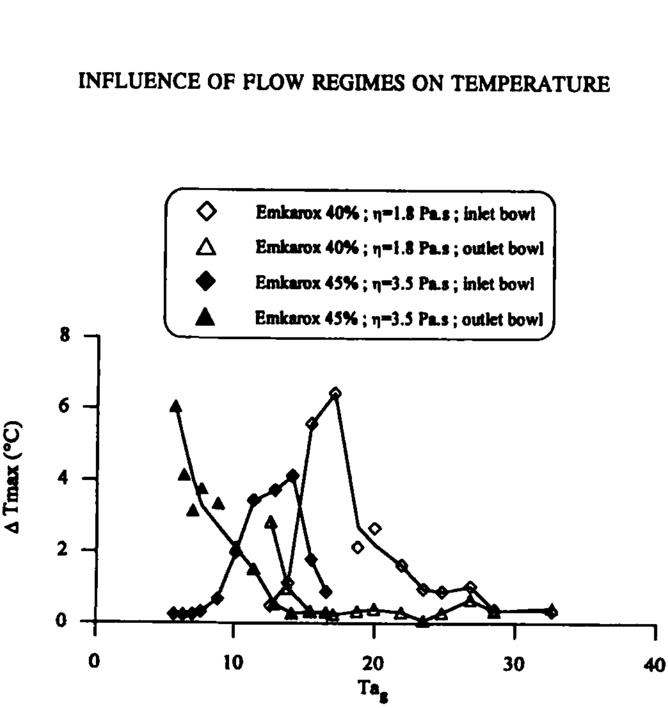

FIG. 3. THERMAL BEHAVIOR OF THE SSHE WITH RESPECT TO GENERALIZED TAYLOR NUMBER IN THE CASE OF HEATING OF EMKAROX SOLUTIONS; FLOWRATE: 160 $l.h^{-1}$ (EMKAROX 40%: $Re_{axg}=0.159$; EMKAROX 45%: $Re_{axg}=0.082$)

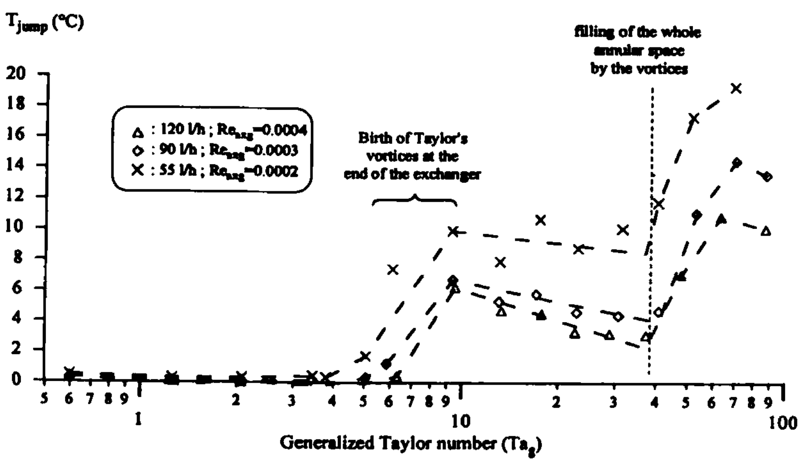

FIG. 4. EVOLUTION OF THE TEMPERATURE JUMP INSIDE THE INLET BOWL WITH RESPECT TO GENERALIZED TAYLOR NUMBER IN THE CASE OF COOLING OF THREE FLOWRATES OF GUAR GUM

The temperature evolution for heating as well as cooling experiments is shown in Fig. 5. A similar evolution according to Ta_g can be noted. The critical value of generalized Taylor number, Ta_{gc}, corresponds to the appearance of vortices at the end of the exchanger (Nouar 1986) and is given for each experiment in Fig. 5. For this critical value, Ta_{gc}, there is a significant temperature gradient in the inlet bowl.

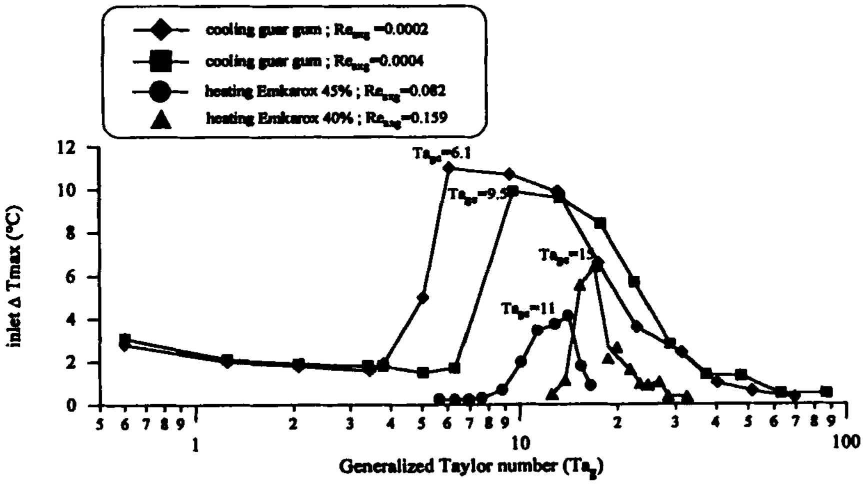

FIG. 5. EVOLUTION OF GENERALIZED TAYLOR NUMBER WITH RESPECT TO Re_{aug}

As a conclusion of this part of the work, we can say that our experimental results dealing with the description of different thermal regimes found in the literature allow us to associate temperature heterogeneity disappearance with Taylor vortices appearance. However, operating conditions needed to obtain Taylor vortices are seldom used in the industry. The high shear stress imposed by the scraping of the blades does not allow working with such high rotation.

Influence of the Presence of Blades on Taylor Vortices Appearance

To obtain a more coherent explanation of the influence of the blades in the flow pattern, we realized cooling experiments in the SSHE without blades and compared the results with the one obtained previously. Measurements obtained in the inlet bowl are shown on Fig. 6.

Evolution of the temperature differences in this bowl with respect to increasing rotating speed seems to have the same profile with or without blades.

However, these profiles are not adjusted to the same Taylor number value. In laminar flow, the temperature differences are more important without blades, which can be explained by the increase of the boundary layer thickness. Values of Ta_{gc} obtained in the case on an annular space without blades are respectively equal to 49 and 28.1 for Re_{axg} = 0.0004 and 0.0002. Undoubtedly, the presence of blades promotes the change in the flow pattern at lower rotating speed. Moreover, the numerical analysis done by Baccar and Abid (1997) confirms that the blade geometry (perpendicular to the wall scrapers or curved scrapers) modifies the thermal performances of the apparatus. The influence of the blades would then create centrifugal instabilities. Visual measurements are now undertaken in order to confirm these results and to characterize the hydrodynamic instabilities due to the blades.

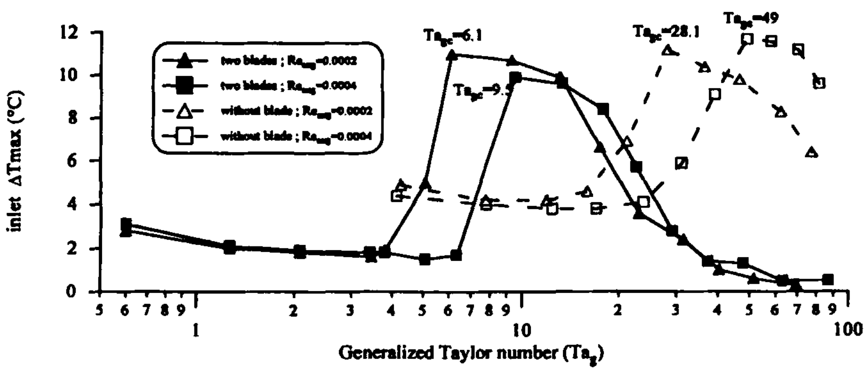

FIG. 6. EVOLUTION OF TEMPERATURE GRADIENT INSIDE THE INLET BOWL WITH RESPECT TO Ta_g AND NUMBER OF BLADES

CONCLUSION

Experimental results obtained show that, either in cooling or heating conditions, under specific operating conditions, food products are not correctly transformed at the SSHE outlet. In laminar flow, large temperature heterogeneity is developing forward from the inlet to the exchanger outlet, which is due to a poor mixing effect of the blades. Our results, associated with interpretation given in the literature let us think that the appearance, and then the extension of Taylor vortices-type instabilities throughout the whole exchanger leads to a progressive homogenization of the temperature from the inlet to the outlet. For the industrial point of view, concerned with the thermal and structural quality of the product, it is necessary to adjust operating conditions (rotating speed in particular) in order to adjust adequate Taylor number (Ta_{gc}) to obtain Taylor vortices at the end of the exchanger.

Our investigation shows that the bowls, because of their geometry, create temperature heterogeneities at low rotating speed and that the blades promote the transition of flow pattern for values of Taylor number less than 40.

Visual analysis as well as studies on the determination of the nature of the flow close to the blades, local as well as global, is actually handled, using electrochemical probes inside the scraped surface heat exchanger. They will help, first to confirm the presence of vortices or of other types of instabilities when using blades, and second to determine the wall shear stress created by the rotation of the blades and deduce the shear rates applied to the treated products.

The originality of the work is to study both cooling and heating conditions in a commercial SSHE and to analyze the relationship between temperature heterogeneity and heat transfer regime. The influence of inlet conditions is also emphasized. To obtain homogeneous temperature at the SSHE outlet, it is necessary to either increase rotational speed or to improve the inlet conditions by changing the bowl shape.

ACKNOWLEDGMENTS

Special thanks are going to F. Tate and C. Veillet who contributed to the study on Newtonian fluids. Acknowledgements to S.A. Duprat for the design of the probed bowls and the lending of one of the SSHE, and especially to B. Touron.

NOMENCLATURE

$$Re_{axg} = \frac{\rho U_d^{2-n} d_h^n}{k}$$

$$Ta_g = \sqrt{\frac{R_s - R_r}{R_r} \frac{\rho d_h^n}{2^n} \frac{(\Omega R_r)^{2-n}}{k}}$$

$\dot{\gamma}$	Shear rates (s^{-1})
η	Dynamic viscosity for Newtonian fluid (*Pa.s*)
ρ	Density (*kg.m^{-3}*)
τ	Shear stress (Pa)
Ω	Rotating speed of the rotor (*rad.s^{-1}*) ; $\Omega = 2 \pi N$
ΔTmax inlet	Maximal temperature gap inside the inlet bowl (°C)
ΔTmax outlet	Maximal temperature gap inside the outlet bowl (°C)
d_h	Hydraulic diameter (*m*); $d_h = (d_s - d_r)$

d_r	Rotor diameter (*m*)
d_s	Stator diameter (*m*)
k	Consistency coefficient of the product (Ostwald's law (*Pa.s^n^*))
L	Stator length (*m*)
n	Flow behavior index of the product (Ostwald's law (-))
R_r	Rotor radius (*m*)
R_s	Stator radius (*m*)
T	Temperature (°C)
Te_i (i from 1 à 5)	Temperatures product inside the inlet bowl (°*C*)
Tp	Temperature of the product before the inlet bowl (°*C*)
Ts_i (i from 1 à 5)	Temperatures of the product inside the outlet bowl (°*C*)
U_d	Mean axial velocity in the annular space ($m.s^{-1}$)

REFERENCES

ABICHANDANI, H., SARMA, S.C. and HELDMAN, D.R. 1987. Hydrodynamics and heat transfer in liquid full scraped surface heat exchangers. A review. J. Food Process Engineering *9*, 121–141.

BACCAR, M. and ABID, M.S. 1997. Numerical analysis on three-dimensional flow and thermal behavior in a scraped surface heat exchanger. Revue Générale de Thermique, *36* (10), 781–790.

BALASUBRAMANIAM, V.M. and SASTRY, S.K. 1996. Fluid to particle convection heat transfer coefficient in a horizontal scraped surface heat exchanger determined from relative velocity measurement. J. Food Process Engineering *19*, 79-95.

BENEZECH, T. 1988. Etude de la distribution des temps de séjour, des performances thermiques et de la consommation de puissance d'échangeurs de chaleur à surface raclée traitant des produits non-newtoniens. Ph.D. Thesis INP Lorraine.

HÄRRÖD, M. 1986. A literature survey of flow patterns, mixing, residence time distribution, heat transfer and power requirement. J. Food Process Engineering *13*, 1-21.

HÄRRÖD, M. 1990a. Methods to distinguish between laminar and vortical flow in scraped surface heat exchanger. J. Food Process Engineering *13*, 39–57.

HÄRRÖD, M. 1990b. Modelling of laminar heat transfer in scraped surface heat exchangers. J. Food Process Engineering *13*, 59–78.

HÄRRÖD, M. 1990c. Modelling of vortical heat transfer in scraped surface heat exchangers. J. Food Process Engineering *13*, 79–111.

HÄRRÖD, M. 1990d. Temperature variations in the outlet from scraped surface heat exchanger. J. Food Process Engineering *13*, 23–28.

MAINGONNAT, J.F. 1982. Contribution à l'étude du comportement thermique d'un échangeur de chaleur à surface raclée. Ph.D. Thesis, ENSIA Massy.

MAINGONNAT, J.F., BENEZECH T. and CORRIEU, G. 1985. Performances thermiques d'un échangeur de chaleur à surface raclée traitant des produits alimentaires newtoniens et non-Newtoniens. Revue Générale de Thermique *279*, 299-304.

MAINGONNAT, J.F., LEULIET, J.C. and BENEZECH, T. 1987. Modélisation de la vitesse de cisaillement apparente dans un échangeur de chaleur à surface raclée. Application aux performances thermiques avec des produits non-Newtoniens. Revue Générale de Thermique, 306–607; 381–385.

NAIMI, M. 1989. Etude des lois d'écoulement et de transfert de chaleur pour des fluides non newtoniens en espace annulaire tournant. Approche réaliste de l'échangeur de chaleur à surface raclée. Ph.D. Thesis ENSEM - LEMTA. INP Lorraine.

NAIMI, M., DEVIENNE, R. and LEBOUCHE, M. 1993. Field flow incidence on the dynamical and thermal performances of scraped surface heat exchangers. Intl. J. Heat Mass Transfer *36*(5), 1413–1421.

NOUAR, C. 1986. Convection thermique pour un fluide rhéofluidifiant. Cas de l'écoulement de Couette-Poiseuille. Thesis ENSEM - LEMTA. INP Lorraine.