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### ► **To cite this version:**

L. Leal, Marc Miscevic, Pascal Lavieille, M. Amokrane, François Pigache, et al.. An overview of heat transfer enhancement methods and new perspectives: Focus on active methods using electroactive materials. *International Journal of Heat and Mass Transfer*, 2013, 61, pp.505-524. 10.1016/j.ijheatmasstransfer.2013.01.083 . hal-01459453

**HAL Id: hal-01459453**

**<https://hal.science/hal-01459453v1>**

Submitted on 20 Feb 2023

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# An overview of heat transfer enhancement methods and new perspectives: focus on active methods using electroactive materials

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## Abstract

The main techniques for the enhancement of heat transfer between a solid wall and a fluid are reviewed for both single phase (liquid and gas) and two-phase (boiling and condensation) systems. First, a brief description of the commonly used passive techniques is given. For each of them, we report the values of the enhancement factor given in the literature. The principal active methods, i.e. methods involving the supply of external energy, are then detailed. The physical mechanisms leading to the heat transfer enhancement are identified from the analyses published to date. The paper then focuses on the techniques that use periodic deformation of a wall over time. Such a wall deformation enhances heat transfer by disrupting the boundary layer and simultaneously setting the fluid in motion. The piezoelectric materials that can be implemented to generate the channel wall dynamic deformation are reviewed. As deformation of a wall is generally of low amplitude, the technique is well suited to micro channel systems: i) in single-phase configuration, imposing a deformation traveling wave to a micro channel wall is found to simultaneously enhance the heat transfer and to set in motion the fluid; ii) boiling in a narrow space is found to involve both boiling and cavitation phenomena in the nucleation process.

*Key words:* heat transfer enhancement, passive techniques, active techniques, nanofluid, jet, spray, electrohydrodynamic, acoustic, ultrasound, vibration, piezoelectric, phase change, boiling, condensation, single phase

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## Nomenclature

$A$	amplitude ( $m$ )
$d_{33}$	longitudinal PZT coupling factor ( $m.V^{-1}$ )
$D$	diameter ( $m$ )
$E$	confinement distance ( $m$ )
$e$	electric field ( $V.m^{-1}$ )
$\vec{E}$	electric field strength ( $V.m^{-1}$ )
$f$	frequency ( $Hz$ )
$F$	fan width ( $m$ )
$\vec{F}_e$	electrohydrodynamic force ( $N.m^{-3}$ )
$G$	mass velocity ( $kg.m^{-2}.s^{-1}$ )
$h$	heat transfer coefficient ( $W.m^{-2}.K^{-1}$ )
$H$	axial distance nozzle to plate ( $m$ )
$L$	exposed blade length ( $m$ )
$L_0$	overall blade length ( $m$ )
$Nu$	Nusselt number
$P$	remanent polarization ( $C.m^{-2}$ )
$Pe$	Peclet number
$q''$	heat flux density ( $W.m^{-2}$ )
$q_c$	free electric charge density ( $C.m^{-3}$ )
$t$	time ( $s$ )
$T$	temperature ( $K$ )
$x$	radial distance from stagnation point ( $m$ )
$y$	position along the length of the plate wall deformation ( $m$ )
$z$	axial distance nozzle to plate ( $m$ )

### Greek symbols

$\Delta P$	pressure difference ( $Pa$ )
$\epsilon$	fluid permittivity ( $F.m^{-1}$ )
$\lambda$	wavelength ( $m$ )
$\rho$	fluid density ( $kg.m^{-3}$ )
$\sigma_e$	electrical conductivity ( $S.m^{-1}$ )
$\phi$	heat flux density ( $W.m^{-2}$ )

### Subscripts

$EHD$	electrohydrodynamic
$ref$	reference
$sat$	saturation

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### 1. The challenges and the different techniques of intensification

The intensification of heat transfer is an important societal challenge in terms of energy saving and materials, sustainable development, thermal control, compactness, etc. The fields of application are numerous. Examples include micro and power electronics, nuclear energy, air conditioning, habitat, transportation, space and aeronautics industries, renewable energy, chemical engineering and industrial processes, etc.

Heat and mass transfers between fluid and wall are influenced by the thermal, mass and hydrodynamics boundary layers. During the last century, numerous works have been conducted to better

control and/or to increase parietal heat transfer. Therefore, an abundant literature has accumulated.

To reduce the boundary layers, intensification techniques have been used but often in a very empirical manner. Thus, the number of studies (publications and patents) undertaken on the subject has been constantly growing since the early 1960s. Intensification techniques are numerous (Webb [? ], for example, has identified 13 of them) and can be classified into two categories: passive and active.

Passive techniques consist, in most cases, of increasing the exchange surface area. Initially they were mainly developed to intensify the transfer between a wall and a gas. Indeed, the low conductivity of the fluid in the gaseous state implies a low heat transfer coefficient. This leads to a requirement for complex surface geometry (surface treatment, fins, etc.) to increase the exchange surface area, and also to disrupt the boundary layers and thus increase the convective heat transfer coefficient. Nevertheless, compactness is another challenge which is not consistent with the increase of the heat exchange area.

Another way to enhance heat transfer, which is also the subject of lot of works, is to use the liquid-vapor phase change. In this case, the complexity of the geometry used to reduce the boundary layer thicknesses is not only connected to the wall but also to the distribution of the phases within the fluid. Although it is now generally accepted that the use of phase change is one of the most effective way to intensify the heat transfers, the understanding of the phenomena involved remains incomplete and the approaches are mainly empirical. For this reason, the control of the heat transfers remains rough. Nevertheless, when a fluid changes phase on a wall, the heat transfers are generally found to be increased significantly. The reason leading to such an increase in the intensity of heat transfer can be analyzed regarding the boundary layer thickness. Indeed in the case of boiling, the liquid is overheated near to the wall. This overheated layer is systematically disrupted or destroyed by nucleation at the wall and the detachment of bubbles. Intensification of the heat transfer is then observed when the number of nucleation sites and the frequency of detachment of the bubbles increase due to the inception of thin liquid film. In the case of condensation, similar mechanisms occur near the wall with high intensification of heat transfer in the presence of thin liquid films.

The need for intensification is becoming increasingly important both in high-tech and conventional industries. This requires more efficient methods to be addressed. The active techniques are seen as a solution to respond to the increased need for intensification and need for flow and heat transfer control with or without phase change. Compared to studies involving passive techniques, very few works have been carried out using active techniques, i.e. requiring the input of external power.

Whatever the technique used to achieve phase change, works on heat transfer intensification have often been conducted in an empirical manner. In most of cases, reliable theories allowing predictive results are difficult to obtain owing to the numerous interfering mechanisms and scales. Even through an improvement of heat transfer is often observed, it remains unpredictable when the experiment is carried out in another configuration. Indeed, results obtained in similar configurations are strongly scattered, highlighting the dependence on experimental device characteristics.

The aim of this paper is to review heat transfer methods in particular those using electroactive materials. First, outlines of passive techniques are briefly recalled. The active techniques are then classified in two categories: active methods with static walls and active methods using dynamic deformation of a wall. The latter configuration involves mainly microfluidic thermal systems. A part of this section is so devoted to the techniques most suited to the types of wall deformation considered in the context of microfluidics. Paths for future works in single-phase configuration

are proposed in the final section of the paper.

## 2. Passive techniques

The heat exchange surface area, the physical properties of the surface and/or the fluid can be modified in order to enhance the heat transfer coefficient. These two effects (increase of exchange area and enhancement of heat transfer coefficient) can be simultaneous and are briefly discussed in the following paragraphs. Hydrodynamic cavitation phenomena can also appear in certain situations (e.g. dramatic reduction of flow cross section), and strongly impact heat transfer as discussed in the last part of this section.

### 2.1. Surface coating at the micron scale

Surface roughness at micron scale, which can either be obtained by coating or machining, increases convective heat transfer because it promotes turbulence, which improves mixing of the fluid, and reduces the boundary layer thickness. This increase of the heat transfer coefficient is not significant in laminar natural convection because the roughness does not cause major disruptions. This technique is widely used for both gases and liquids with slightly different technologies. Finned plates, tubes with internally fins, externally finned tubes [? ], porous [? ] or fibrous [? ] inserts are commonly used to enhance heat transfer with or without phase-change. Dalkilic and Wongwises [? ] reviewed condensation heat transfer intensification using extended surfaces. For instance, Cavallini et al. [? ] achieved an increase of the heat transfer coefficient of up to 140% with micro-fin tubes while Miyara et al. [? ] achieved an increase of 350% with herringbone tubes. Khan et al. [? ] reported that surface roughness is the most influential factor for boiling heat transfer, as nucleation sites are closely related to surface topology. Both heat transfer coefficient enhancement and critical heat flux increase may be obtained, as for example in the work of El Genk and Ali [? ]. These authors used micro-porous surface layers on a copper substrate. The critical heat flux is found to be enhanced up to 70% while the heat transfer coefficient is increased by a factor greater than 17 compared to the one obtained on a plain surface. Another effect of the micro-porous layer is the consequent decrease of the wall temperature at the onset of boiling. More details on the effect of micro-structuration of the surface are available in Poniewski and Thome [? ].

### 2.2. Nanoscale coating

Nanoscale structuration (produced using e.g. coating, etching, directional growth.... ) change chemical and physical properties of surfaces. Here, the small dimensions (nanometric) of roughness have little effect on the heat transfer surface area. An effect on the heat transfer coefficient can be obtained, in particular in phase change conditions: the augmentation of the nucleation site number and/or a change in the wettability of the liquid on the wall may significantly improve heat transfer. Khan et al. [? ] reviewed pool boiling heat transfer enhancement using surface coating. Numerous ways to make such coatings exist: chemical etching, electroplating, painting, sintering, spraying, making three-dimensional cavities. Coating a surface with porous metal is one of the best ways to enhance the heat transfer. The nucleation sites are numerous and easily activated due to the presence of gas trapped in the coating pores. Phan [? ] reviewed the effects of surface wettability on boiling heat transfer. Two main ways can be used to change the wettability of a liquid on a substrate: coating the surface to change the molecular interactions between the fluid and the solid, or changing the nanoscale surface topography to generate

a Cassie and Baxter state [?] that can be considered as similar to a bed of nails (a Fakir's carpet [?]). Increasing the contact angle between liquid and solid reduces the energy needed to form a nucleus. However, for highly hydrophilic surfaces, the radius of departing bubbles is high which favors coalescence. This phenomenon should deteriorate heat transfer. For very wettable surfaces, the liquid microlayer underneath the bubble is larger and thinner which enhances heat transfer. According to Phan et al. [?], the competition between these two opposite effects leads to the highest heat transfer coefficient for a contact angle close to both  $0^\circ$  and  $90^\circ$ , in the case of water. For a contact angle close to  $90^\circ$ , the enhancement is due to higher frequency of bubble detachment.

There are many reviews dealing with heat transfer enhancement using nanofluids. Nanoparticles in suspension in a fluid improve the apparent thermal conductivity, and also change the viscosity and the density of the fluids [?]. A large proportion of the papers deals with the thermal conductivity enhancement (see for instance the review of Özeriç et al [?]). Kakaç and Pramuanjaroenki [?] made a review about convective heat transfer with nanofluids. They cited Heris et al. [?] who carried out experiments with laminar flow and constant wall temperature. Intensification results (compared to the case without nanoparticles) in fully developed conditions with various nanoparticle materials and concentrations were reported versus the Peclet number. Intensification of about 40% is achieved for a Peclet number of 7000.

Few papers concern convective boiling heat transfer enhancement using nanofluids. An update on the state of the art has been made by Taylor and Phelan [?]. Some results demonstrate the effectiveness of this technique in boiling situation: compared to pure water, the wall superheat at the onset of boiling is 2-3°C lower and the heat transfer coefficient is increased by about 25-40%, while in subcooled boiling condition the heat transfer coefficient is found to decrease. Other authors obtained higher enhancements as reported in table x of the review of Taylor and Phelan [?].

### 2.3. Hydrodynamic cavitation

Hydrodynamic cavitation is the vapor (bubble growth) or air (gaseous cavitation) nucleus development due to local static pressure reduction below a critical value. This situation can appear, for example, in a dramatically reduced flow section. Peles and coworkers published several papers dealing with this phenomenon in microchannels [? ? ? ? ? ?]. Figure 1 reports

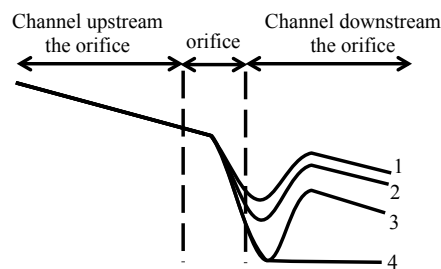


Figure 1: Qualitative pressure profiles in the flow through a micro-orifice for a fixed inlet pressure and several outlet pressures according to Schneider et al. [?]

the liquid flow pressure change in a microchannel, upstream, inside, and downstream a micro-orifice. Curve 1 corresponds to a low pressure difference between the inlet and outlet of the

microchannel. The outlet pressure is then progressively decreased ( $2 \rightarrow 4$ ) while the inlet pressure is maintained constant. High static pressure reduction occurs in the zone where the channel section decreases. When static pressure falls below a threshold value, cavitation is initiated. This threshold value depends on the quantity of dissolved gas within the liquid. For static pressure above saturation pressure, the main reason for the growth of nuclei is the presence of dissolved gas in the liquid which diffuses into these nuclei (gaseous cavitation). Increasing the pressure difference leads to greater static pressure reduction in the micro-orifice to reach the saturation pressure considering curve 3. **Below this pressure difference value between the ends of the microchannel, it is not possible to further reduce the static pressure in the micro-orifice (choked flow). This phenomenon is called supercavitation.**

Schneider et al. [?] studied the effect of hydrodynamic cavitation downstream a micro-orifice on heat transfer with R-123 as the working fluid. Non-cavitating flow and cavitating flow were both studied. Non-cavitating flow was obtained by adjusting both inlet and outlet pressures while cavitating flow was obtained by changing only the outlet pressure. These authors reported some results (heat flux versus wall temperature) highlighting significant enhancement of the heat transfer for mass velocities  $G = 622 \text{ kg.m}^{-2}.\text{s}^{-1}$  and  $G = 1368 \text{ kg.m}^{-2}.\text{s}^{-1}$ . The maximum enhancement in the heat transfer coefficient was 84 %, obtained at the highest mass velocity ( $G = 1368 \text{ kg.m}^{-2}.\text{s}^{-1}$ ). This enhancement was attributed to the presence of bubbles which mix the fluid and reduce hydraulic diameter and thus the thickness of the boundary layer. When supercavitation was reached, decrease of the outlet pressure no longer increased heat transfer any more.

Thus, the cavitation process appears as an interesting way to enhance heat transfer. It is also noticeable that the onset of boiling in the presence of cavitation occurred without temperature hysteresis. This will be discussed in more detail in section 4.3 considering cavitation induced by a dynamic morphing of the wall in confined pool boiling configuration.

#### 2.4. Turbulence and mixing promoters

**IL MANQUE LES CITATIONS POUR LES REFS DE CETTE PARTIE** Enhanced surfaces can create one or more combinations of the following conditions that produce a desirable heat transfer rate increase and unfortunately also increase friction factor : (1) boundary layers development interruption as well as increase of turbulence intensity , (2) extension of heat transfer surface area , and (3) swirling and/or secondary flows generation. Reverse/swirl flow devices (rib, groove, wire coil, conical ring, snail entry, twisted tape, wingleet, etc.) form an important group of passive transfer enhancement technique. The major function of such devices is producing recirculation in an existing axial flow, leading to an improvement of fluid mixing and thus an efficient reduction of the thickness of a boundary layer. Twisted-tape inserts have been used since 1896, according to Manglik and Bergles(1993) There are many shapes available and currently marketed (Thakur et al, 2003) and, impact of such devices -as well as their optimization- is still under investigation. For decades, lots of the wire coil and twisted tape devices employed for augmentation of laminar and turbulent flow heat transfers have been reported and an abundance of data is available see e.g. Bergles, A.E. 1969, Kalinin and Dreitser (1998), Dewan et al. (2004), Promvongse, 2008, Manglik and Bergles (2002), Uttarwar and Raja Rao (1985), Ravigururajan and Bergles (1996)[? ] .



#### 2.4.1. *Single phase flow*

**IL MANQUE LES REFERNCES AUSSI** The swirl induced heat transfer enhancement, the shear stress and pressure drag in a tube with the coiled wire or twisted tape insert are accordingly increased. For this type of enhanced tube, the Reynolds number, the coil pitch ratio and the twist ratio of twisted tape are the governing parameters to signify the pressure drop and heat transfer characteristics. However, few comparisons between the different geometries are available as pointed out in Garcia et al (2007), or shoji et al. (2003) and studied by Wang and Sunden.

As reported in several research papers, twisted tapes were combined with several enhancement devices for further improving heat transfer rate, for example: twisted with spirally grooved tube, converging-diverging tube, corrugated tube, dimpled tube, conical-ring, wire coil turbulators and twisted tape with wire-coil. The use of combined wire coil and twisted tape inserts may be fruitful since the continuous swirling flow generated by the twisted tape can help to wash up the trapped flow in the coil groove leading to higher heat transfer rate. Promvong and Eiamsa-ard investigated the effects of compound turbulators (conical-nozzle, conical-ring or V-nozzle together with a swirl generator) and found an increase in heat transfer rate of about 20/50% when using a single enhancement device but also substantial rise in pressure losses. They assessed the results of the heat transfer and pressure drop in a heat exchanger tube fitted with regularly spaced helical tapes or helical screw-tape with/without core-rod inserts. It was found that at similar operating conditions, the heat transfer rates provided by the helical screw-tape with core-rod were higher than those given by the tape without core-rod. In addition, heat transfer rate increased with decreasing spacing.

Mostly, heat transfer rates associated by the combined techniques or the modified twisted tapes are higher than those given by typical twisted tapes. However, an improvement in heat transfer is unavoidably accompanied by an increase of pressure drop. It is still a challenge to design optimized geometries in order to achieve an excellent heat transfer with reasonable pressure drop penalty. Studies of modified twisted tape geometries to induce more perturbation of longitudinal fluid flow are nowadays numerous, e.g. broken insert (Chang et al. 2007), peripherally cut (Eiamsa-ard et al. 2010), serrated (Eiamsa-ard and Promvong 2010), with oblique teeth (Saha 2010), perforated (Thianpong et al. 2012) and many other twisted tape shapes (delta-winglet, trapezoidal-cut, square-cut), with wire-nail with alternate-axes and variously shaped wings (triangle, rectangle, trapeze).

#### 2.4.2. *Boiling and condensation*

The usage of helical horizontal micro-fin tubes is the most common passive enhancement device for condensers in use nowadays due to their high heat transfer performance and moderate increase in pressure drop. Manglik and Bergles (1993), Shatto and Peterson (1996), Dalkilic and Wongwises in 2009 or Kanizawa et al. (2011) presented comprehensive reviews concerning single and two-phase flows inside tubes containing twisted-tape inserts or various shaped fins. Determinations of the heat transfer and pressure drop have major significance in design practice. Twisted-tape presents the main advantages of low manufacture and installation costs, easy maintenance, and according to the possibility of being installed in heat exchangers already in use (Thome and Ribatski 2005). According to Agrawal et al. (1982), for the same pumping power and an optimum heat exchanger design, heat transfer capacity enhancements up to 20% can be achieved by using twisted-tape inserts. Shatto and Peterson (1996) highlighted the fact that the negative result of increasing the pressure drop gradient by the use of inserts can be overcome by a reduction in the heat exchanger size due to the heat transfer enhancement.

During the last 20 century decades experiments showed that use of twisted tapes as swirl flow promoters in water-subcooled flow boiling is very effective in flow boiling critical heat flux (CHF) enhancement, allowing increases in the CHF up to a factor of 2.0 (Nariai et al 1991, 1992, Cardella,1992). CHF correlations developed for simple tubes were adapted to such situations (e.g. Celata 1995). Although initially formulated for operating conditions typical of the thermal hydraulic design of fusion reactor high heat flux components, the model is proved to give a satisfactory answer for the prediction of the CHF under more general conditions, provided local thermodynamic conditions of the bulk flow at the CHF are sufficiently far from the saturated state.

Bergles et al. (1971) suggested the use of twisted-tape insert also for wispy flows as the effects of liquid droplets impingement are enhanced by the centripetal acceleration and, consequently, higher heat transfer coefficients are expected. Based on this phenomenon, Jensen (1985) suggested the use of twisted-tape inserts only in the dryout region, corresponding to high vapor qualities. In the case of liquid-vapor flows, Shatto and Peterson (1996) suggested that the heaviest phase is driven to the tube periphery by the centripetal acceleration; during convective boiling, increasing the uniformity of the liquid film thickness along the tube perimeter may promote an increase in the heat transfer coefficient. According to Thome and Ribatski (2005), the centripetal acceleration also favors the bubble detachment, increasing the heat transfer performance. For in-tube condensation, the centripetal acceleration acts to segregate the colder parcel of the fluid near the wall and the hotter in the tube core, increasing the thermal resistance. The twisted-tape inserts promote a decrease in the convective effects and are not recommended for such applications, according to Kanizawa et al. (2011).

## 2.5. Summary

### 2.5.1. Single-phase free and forced convection

With the exception of the familiar technique of providing extended surfaces, the passive techniques have little to offer in the way of enhanced heat transfer for free convection. This is because the velocities are usually too low to cause flow separation or secondary flow. The restarting of thermal boundary layers in interrupted extended surfaces increases heat transfer so as to more than compensate for the lost area.

One of the motivations for studying such situations is to assess the effect of an inherent condition on heat transfer, e.g. roughness produced by standard manufacturing, degassing of liquids with high gas content, surface vibration resulting from rotating machinery or flow oscillations, fluid vibration resulting from pumping pulsation, and electrical fields present in electrical equipment.

The promoters produce a sizable elevation in the Nusselt number, or heat transfer coefficient, at constant Reynolds number, or velocity. However, there is an accompanying large increase in the friction factor. Surface roughness has been used extensively to enhance forced convection heat transfer. In view of the infinite number of possible geometric variations, it is not surprising that, even after several hundred studies, no satisfactory unified treatment is available. In general, the maximum enhancement of laminar flow with many of the techniques is the same order of magnitude, and seems to be independent of the wall boundary condition. The enhancement with some rough tubes, corrugated tubes, inner-fin tubes, various static mixers, and twisted-type inserts is about 200%. The improvements in heat transfer coefficient with turbulent flow in rough tubes (based on nominal surface area) are as much as 250%.

Extended surfaces can be considered "old technolog" as far as most applications are concerned.

The real interest now is in increasing heat transfer coefficients on the extended surface. Compact heat exchangers of the plate-fin or tube-and-center variety use several enhancement techniques: offset strip fins, louvered fins, perforated fins, or corrugated fins. Coefficients are several hundred percent above the smooth-tube values; however, the pressure drop is also substantially increased, and there may be vibration and noise problems.

### 2.5.2. Convective Boiling/Evaporation

The structured surfaces are generally not used for in-tube vaporization, because of the difficulty of manufacture. One notable exception is the high-flux surface in a vertical thermosiphon reboiler. Helical repeated ribs and helically coiled wire inserts have been used to increase vaporization coefficients and dryout heat flux. Numerous tubes with internal fins, either integral or attached, are available for refrigerant evaporators. Average heat transfer coefficients (based on surface area of smooth tube of the same diameter) for typical evaporator conditions are increased by as much as 200% and the average evaporation boiling coefficient is increased from 30 to 80% usually with a lower percentage increases in pressure drop. Twisted-tape inserts are also used to increase the burnout heat flux for subcooled boiling at high imposed heat fluxes (up to  $10^7 - 10^8 \text{ W/m}^2$ ). Increases in burnout heat flux of up to 200% were obtained at near atmospheric pressure.

### 2.5.3. Condensation

Surface extensions are widely employed for enhancement of condensation. The integral low fin, used for kettle boilers, is also used for horizontal tube condensers. With proper spacing of the fins to provide adequate condensate drainage, the average coefficients can be several times those of a plain tube with the same base diameter. These fins are normally used with refrigerants and other organic fluids that have low condensing coefficients, but which drain effectively, because of low surface tension. The fin profile can be optimized to take advantage of the Gregorig effect, whereby condensation occurs mainly at the tops of convex ridges. Surface tension forces then pull the condensate into concave grooves, where it runs off. The average heat transfer coefficient is greater than that for an axially uniform film thickness.

The typical applications of in-tube condensation are horizontal kettle-type reboilers, moisture separator reheaters and air-conditioner condensers. Internally grooved or knurled tubes, deep spirally fluted tubes, random roughness, conventional inner-fin tubes have been shown to be effective for condensation. As in the case of evaporation, the substantial heat transfer improvement is achieved at the cost of an increase in pressure drop. Still, general correlations are not yet available but usually, the best performance tubes for evaporation are found to perform also well in condensation. Twisted-tape inserts and coiled tubular condensers give only modest increases in heat transfer coefficient for complete condensation.

Passive techniques enhance heat transfer by modifying surface exchange or fluid properties, by changing surface geometry, by disrupting boundary layers, by promoting phase change. Nowadays, most heat exchange systems use passive techniques to enhance heat transfer. Nevertheless, owing to the constant growth of demand for heat transfer, passive technique performances are insufficient to reach the required performance so the use of active techniques becomes necessary to meet the required transfer characteristics.

Active techniques consist in increasing the heat transfer coefficient using the input of external

power which can improve heat exchange compared to passive methods. Active methods are becoming more frequently considered and numerous techniques have been proposed which could be classified into two categories: active methods where the channel walls are immobile and active methods where at least one solid wall moves periodically.

### 3. Active methods: motionless channel wall

#### 3.1. Electrohydrodynamics

Electrohydrodynamic (EHD) heat transfer enhancement refers to the coupling of an electric field with the flow field in a dielectric fluid medium. In this technique, an electric field characterized by a high voltage and low current is applied to the fluid. Several reviews giving the state of the art have reported the main results obtained with this technique such as those of Laohalertdecha et al. [? ]. One of the major aims of the EHD technique is to convert electric energy into kinetic energy: EHD forces set the dielectric fluid in motion. Fluid displacement is due to volumetric electric force which can be expressed as follows [? ] :

$$\vec{F}_e = q_c \vec{E} - \frac{1}{2} E^2 \vec{\nabla} \epsilon + \frac{1}{2} \vec{\nabla} (\rho E^2 \left[ \frac{\partial \epsilon}{\partial \rho} \right]_T) \quad (1)$$

with  $\vec{E}$  the electric field strength ( $V/m$ ),  $\vec{F}_e$  the electrohydrodynamic force ( $N/m^3$ ),  $q_c$  the free electric charge density ( $C/m^3$ ),  $\epsilon$  the fluid permittivity ( $F/m$ ),  $\rho$  the fluid density ( $kg/m^3$ ), and  $T$  the temperature ( $K$ ).

The first term is the Coulomb force which is an electrophoretic force. It represents an electrostatic force density induced by the presence of net charge in the dielectric fluid. The Coulomb force is the main reason underlying the ionic winds (ionic flow of charged particles). Heat transfer enhancement in single phase is mainly due to Coulomb force. Indeed, for single phase heat transfer, thermal gradients in the fluid lead to non-uniformity in electrical conductivity  $\sigma_e$  and in dielectric permittivity  $\epsilon$ . The permittivity characterizes the polarisability of the dielectric fluid. Assuming that dielectric permittivity and electrical conductivity are only functions of the temperature, electric charge density can be expressed as [? ] :

$$q_c = \vec{E} \cdot \vec{\nabla} T \frac{d\epsilon}{dT} - \frac{\epsilon}{\sigma_e} \vec{E} \cdot \vec{\nabla} T \frac{d\sigma_e}{dT} \quad (2)$$

Equation 2 shows that gradients of permittivity and electrical conductivity induce a change of free charge density which can imply an increase in the electrophoretic force.

The second term of equation 1 is the dielectrophoretic force. It results from non-homogeneity or spatial change in the permittivity of the fluid. It can be present in the single phase if there is a temperature gradient which brings about a permittivity gradient. This second term could be expanded as follows showing that the force appears in the presence of permittivity change with fluid density and temperature [? ] :

$$-\frac{1}{2} E^2 \vec{\nabla} \epsilon = -\frac{1}{2} E^2 \left[ \left( \frac{\partial \epsilon}{\partial T} \right)_\rho \vec{\nabla} T + \left( \frac{\partial \epsilon}{\partial \rho} \right)_T \vec{\nabla} \rho \right] \quad (3)$$

The third term of equation 1 is the electrostrictive force which involves a phenomenon of electrostriction based on the presence of different electric fields within the dielectric fluid. These domains attract each other in the presence of an electric field because they are charged differently. This force can affect the flow if there is a liquid/vapor interface.

High values of enhancement can be reached thanks to these forces. For example, in single phase oil flow, Fernandez and Poulter [?] obtained a heat transfer coefficient up to 20 times the fully developed laminar flow heat transfer coefficient. This enhancement results from a strong increase in the radial motion of the fluid which disrupts the thermal boundary layer. The same order of magnitude of heat transfer enhancement may be reached in a two-phase heat transfer system. The main mechanisms involved in EHD boiling and condensation heat transfer improvement are briefly discussed below.

Wang et al [?] reported the differences between nucleate boiling mechanisms with and without EHD. In nucleate boiling conditions without EHD, heat transfer is ruled by three mechanisms :

- Conduction in the boundary layer of the interface;
- Evaporation of the microlayer located between the heating wall and the bubble bottom;
- Convection around the bubble and the heating surface.

During nucleate boiling with EHD, heat transfer is governed by different mechanisms. The electric field is not uniform in the liquid near the interface because density and temperature are not uniform. This generates electrostrictive forces. The electric force of the liquid far from the heating wall is greater than the electric force of liquid near the heating wall which generates electroconvective movements. These serve to increase heat transfer. Moreover, when boiling occurs, as liquid permittivity and vapor permittivity are different, a dielectrophoretic force is induced. This force acts on the vapor bubbles and consequently agitating the boundary layer and causing heat transfer enhancement [?]. Furthermore, the shape of the bubbles can be changed by electrohydrodynamic forces. For instance, Chen et al. [?] injected a single nitrogen bubble in a uniform electric field. The bubble became elongated along the direction parallel to the applied electric field. Electrostriction stress compresses a bubble while dielectrophoretic stress expands a bubble. The EHD stress exerted on the bubble makes it more slender in the presence of an electric field. The number of bubbles thus increased both by breaking up large bubbles and by lowering the bubble detachment diameter [? ? ?].

Another effect to be noted is the influence of the electric field on contact angle and surface tension. As the wall superheat needed for the onset of boiling depends on these two parameters, the presence of an electric field may lead to a decrease of wall superheat at boiling incipience down to the disappearance of boiling hysteresis [? ? ? ? ? ? ?].

Furthermore, some papers dealt with the effect of electrohydrodynamic forces on pool film boiling. For example, the work of Carrica et al. [?] and the one of Cipriani et al. [?] showed experimentally the effect of electric fields on pool film boiling regimes and on heat transfer coefficient. These two papers reported that two different pool boiling film regimes exist and they are separated by an additional boiling transition. The heat transfer coefficient obtained during the first pool film boiling regime strongly depends on the electric field effects : high heat transfer coefficient could be obtained during this regime. Concerning, the second pool film boiling

Source	Working fluid	$\frac{h_{EHD}}{h_{ref}}$ (max)
Bochirol et al. [?] ]	Tircholethylene and ethyl ether	2.7
	Methyl alcohol	6
Watson [?] ]	n-hexane	2.6
Schnurmann and Lardge[?] ]	n-heptane	7
	20% isopropyl alcohol in n-heptane	7
	Perfluoromethylcyclohexane	2
Blachowicz et al. [?] ]	Benzène	2
Allen and Cooper [?] ]	R-114	> 60
Kawahira et al. [?] ]	R-11	4
Karayiannis et al. [?] ]	R-114	4
Cooper [?] ]	R-114 + huile	13
Damianidis et al. [?] ]	R-114 (tube lisse)	1
	R-114 (low-fin tubes)	2.5
Ohadi et al. [?] ]	R-123 / R-11	5.5 / 1.7
Ogata et al. [?] ]	R-123	7
Yabe et al. [?] ]	R-123	6
Paper et al. [?] ]	R-123	6
Neve and Yan [?] ]	R-114	3
Cheung et al. [?] ]	R-134a	5.1
Salehi et al. [?] ]	R-134a	10
Yamashita and Yabe [?] ]	Acetone, benzene	1.82
Karayiannis [?] ]	R-123 and R-11	9.3
Feng and Seyed-Yagoobi [?] ]	R-134a	4.5
Liu et al. [?] ]	R-123	2.1
Posew and Yabe [?] ]	R-134a	1.8
Wang et al. [?] ]	$LN_2$	0.5
Kamkari et al. [?] ]	$H_2O$	7.6

Table 1: Boiling heat transfer enhancement using EHD (derived from the Laohalertdecha et al's review [?] ] and completed by other references).

regime, the heat transfer coefficient weakly depends on the electric field.

Gravity has an important effect on pool boiling in reason of the buoyancy forces. In microgravity, buoyancy forces can no longer drive the vapor away of the heated wall. Large bubbles are formed near the heated wall by coalescence. This effect could deteriorate heat transfer coefficient. Di Marco and Grassi [?] ] realized an experimental study both at terrestrial gravity conditions and at reduced gravity conditions. They reported that in micro gravity situation, the heat transfer coefficient is degraded. To compensate the lack of buoyancy force, electric field is imposed. The heat transfer degradation is strongly reduced by the effect of the applied electric field because the latter improve the vapor bubble removal. Nevertheless, for high heat flux, the electric field can not improve heat transfer in micro gravity situation even for high value of voltage applied (during the experiments).

Table 1 reports some of the boiling EHD heat transfer enhancement values available in the literature.

In condensation, applying an electric field may also have a strong effect, as indicated for instance in the reviews of Laohalertdecha et al. [?] ] and Omidvarborna et al. [?] ]. The physical mechanisms involved in the intensification of the heat transfer during EHD condensation are numerous. Effective removal of liquid condensate from the heat transfer surface may be obtained utilizing EHD-induced forces, thus producing a decrease in the condensate film thickness [?] ? ? ]. The dispersion of the condensate film from the condensing surface by electrostatic atomization induces a change in the flow regime from film condensation to pseudo-dropwise

Source	Working fluid	$\frac{h_{EHD}}{h_{ref}}$ (max)
Choi and Reynolds [? ]	R-113	2
Velkoff and Miller [? ]	R-113	1.5
Choi [? ]	Freon-113	2
Holmes and Chapman [? ]	R-114	> 10
Seth and Lee [? ]	R-113	1.6
Smirnov and Lunev [? ]	R-113, dyethyether	3.6
Didkovsky and Bologa [? ]	polar and non-polar dielectric fluids	20
Yabe et al. [? ]	eau, R-113	2.24
Cooper and Allen[? ]	R-12 et R-114	2.9
Trommelmans and Berghmans [? ]	R-11, R-13, R-114	1.1
Bologa et al. [? ]	Hexane-CO2	5
	R-113-helium / R-113-air / Hexane-air	1.8 / 2.6 / 5
Damianidis et al. [? ]	R-114	1.08
Sunada et al. [? ]	R-123	6
Yamashita et al. [? ]	perfluorohexane, R-114	6
Bologa et al. [? ]	R-113	20
Jia-Xiang et al. [? ]	R-11	1.85
Wawzyniak and Seyed-Yagoobi [? ]	R-113	6.1
Bologa et al. [? ]	Dietyl ether / R 113, hexane	20 / 10
Singh et al. [? ]	R-134a	6.5
Cheung et al. [? ]	R-134a	7.2
Da Silva et al. [? ]	R-134a	3
Butrymowicz et al. [? ]	R-123	2
Gidwani et al. [? ]	R404a	18.8
Yabe et al. [? ]	Silicone, oil, R-113 n-perfluorohexane	4.5 4
Laohalertdecha and Wongwises [? ]/[? ]/[? ]	R-134a	1.15 / 0.3 / 1.1
Cotton [? ]	R-134a	2.5

Table 2: Condensation heat transfer enhancement using EHD (from the review of Laohalertdecha et al's [? ] plus other references).

condensation [? ? ? ].

Beyond these main effects on heat transfer enhancement when applying an electric field during condensation, several other effects can be noted: the EHD force disturbs the accumulation of non-condensable gases at the liquid-vapor interface, induces perturbations and waviness into the condensate film [? ? ], and increases convection due to the electrophoretic force on the individual phases and electrostrictive force on the vapor phase [? ].

Table 2 reports some of the condensation EHD heat transfer enhancement values available in the literature.

Electrohydrodynamic forces significantly enhance heat transfer. The maximum values of intensification obtained in boiling and condensation heat transfer are respectively > 60 [? ] and 20 [? ? ]. Another way to enhance heat transfer is the use of jet as described in the following section.

### 3.2. Jets

The projection of a fluid at high speed toward a surface increases the convective heat transfer coefficient because the jet breaks the boundary layer and mixes the fluid near the surface. Jets are used in many industrial applications such as cooling of internal combustion engines or thermal control of high-flux devices (electronics, X-rays, optics, gas turbines, etc...). Jets are well known to give a high heat transfer coefficient near the stagnation region where the jet impinges on the target plate. In general, this technique uses multiple impingement jets to heat or cool the surface

uniformly and widely [? ]. In the following, the main mechanisms of flow and heat transfer enhancement are described for both single and multiple jets.

### 3.2.1. Single steady impingement jet

The single steady impingement jet i. e. the simplest configuration has been widely studied because it constitutes the academic case of the multiple-jet cooling process.

The flow and heat transfer characteristics of a jet impinging on surface orthogonal to the main flow axis is usually described as in figure 2 [? ? ? ] where:

1. Zone 1 is called the flow development region. It is made up of a free jet region and a core region. The core region is where the velocity is nearly equal to the nozzle exit bulk velocity. The end of the core region is commonly defined as the point where the velocity of the center of the jet is equal to 95 percent of the nozzle exit velocity. Gautner et al. [? ] suggested that the core length is six nozzle diameters. The free jet region widens as the distance from the nozzle increases due to shear interaction with the surrounding fluid. The surrounding fluid is entrained into the jet inducing a decrease of the fluid velocity in the free jet zone.
2. The fully developed region (zone 2) appears at the end of the core. Axial jet velocity decreases as the distance from the nozzle increases.
3. The region near the wall (zone 3) is characterized by an increase in the static pressure and a sharp decrease in the axial velocity. This region is called the deflection zone. The stagnation point is at the center of the deflection zone. Jambunathan et al. [? ] used the data from Gardon et al. [? ] to plot the Nusselt number versus the distance from the stagnation point. For very short distances between the jet nozzle and the impingement plate ( $z/D < 6$ ), the flow is laminar near the stagnation point inducing a local minimum in heat transfer at the stagnation point. Nevertheless, heat transfer is highest in this area due to tangential acceleration (in the x direction). When the impingement plate is placed at the end of the core region ( $z/D = 6$ ), maximum heat transfer intensity is reached at the stagnation point owing to the intrusion of Kelvin-Helmholtz vortices. For higher separation distances ( $z/D > 6$ ), an increase of the separation distance induces a decrease of the jet velocity due to the shear stress between the jet and the surrounding fluid that progressively reduces heat transfer at the stagnation point. Heat transfer then decreases as the distance from the center increases.
4. The wall jet region (zone 4) is characterized by an increase in the transverse velocity up to a maximum value, followed by a continuous decrease of this transverse velocity. This deceleration brings a transition from laminar to turbulent flow because the stabilizing effect of the acceleration decreases. For low values of the separation distance, this laminar-to-turbulent transition in the flow regime increases the heat transfer and a secondary peak can be observed on the Nusselt number variation.

So, the heat transfer coefficient depends on multiple parameters. The main ones are nozzle geometry, nozzle-to-plate distance, jet Reynolds number, heat flux, velocity profiles, turbulence intensity.

A major conclusion of single jet studies is that the heat transfer is very high close to the impact zone, but decreases rapidly away from it. One solution to overcome this problem is to use multiple jets as described below.



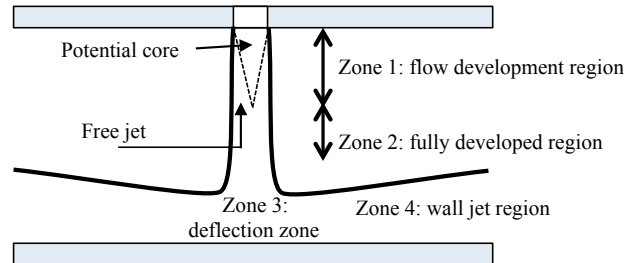


Figure 2: Schema of the various flow zones in an impinging jet: 1- initial mixing region; 2 - established jet; 3 - deflection zone; 4 - wall jet

### 3.2.2. Multiple steady impingement jet heat transfer

The multiple steady jet regions are the same as for single steady jets regions. If there is no interaction between the jets, heat transfer for multiple jets is similar that of a single jet. In the other case, heat transfer is changed by two principal jet interactions: jet interaction prior to impingement and head-on collision of surface flow after impingement.

Jet interaction prior to impingement is due to the entrainment of the fluid present between two adjacent jets by shear stress. It occurs for small inter-jet spacing and high separation distances. Such an interaction can weaken the jet strength and shorten the jet core so that the overall heat transfer on the target plate may be degraded compared to the case with no interaction between two adjacent jets prior to their impingement on the substrate.

Head-on collision of surface flow after impingement can cause fountain flow depending on the strength of the interaction. Fountain flow is particularly strong for small inter-jet spacing, small separation distances, and large jet velocities. Recirculating vortices may then exist between two adjacent jets. This recirculating flow re-enters the cores of the neighboring jets, degrading heat transfer performance. However, turbulence is generated by wall jets collision (secondary stagnation zones) which locally increases heat transfer. Furthermore, turbulence may diffuse towards the centre of each jet which also enhances heat transfer [? ].

To summarize, multiple jets enhance heat transfer in several manners. The heat transfer coefficient is greatly enhanced near the impingement zone. Thus, increasing the number of impingement zones improves heat transfer and temperature homogeneity on the target plate. For small jet-to-jet spacing, a second type of intensification is induced by the appearance of secondary stagnation zones due to interactions between jets which generate turbulence. Instabilities can also be obtained by imposing flow rate modulations and heat transfer improvement can be expected. This is briefly addressed in the next part.

### 3.2.3. Unsteady jet heat transfer

Pulsating flow could enhance heat transfer by disrupting hydrodynamic and thermal boundary layers through flow instability and higher turbulence. Thus, the pulsing impingement jet was studied by several authors in order to evaluate their heat transfer performance. For instance, in their numerical study, Xu et al. [? ] reported a significant heat transfer enhancement due

to the intermittent pulsation in a turbulent impinging jet. They also noticed that the on-to-off jet time ratio and jet-to-plate distance have a significant effect on heat transfer enhancement. They explained the heat transfer enhancement by several mechanisms: higher turbulence, larger vortices, higher flow entrainment and mixing promoted by flow instabilities and reduction of the instantaneous boundary layers.

Zhou et al. [?] indicated in their study the following main parameters influencing the heat transfer characteristics :

- Nozzle geometry
- Nozzle-to-plate distance
- Jet Reynolds
- Heat flux
- Velocity profile
- Turbulence intensity
- Shape of unsteadiness (e. g. frequency, amplitude)

Furthermore, the target plate geometry influences on the heat transfer. Zhou et al. [?] studied heat transfer on smooth and non-smooth surfaces. For the range of parameters tested, roughness had a weak influence on heat transfer for steady jets whereas it had a significant influence on heat transfer for unsteady jets. The authors pointed out that this result cannot be generalized without a systematic study of the influence of the roughness parameters on heat transfer on a non-smooth plate.

Another type of unsteady jet was recently developed in which the net mass flux is nil. Such jets are also called "synthetic jets". As the pulsating flow of a synthetic jet is generated by a moving a membrane, this technique will be detailed in section 4 in accordance with the classification chosen in this paper.

### 3.3. *Spray*

A spray consists of liquid droplets generated by air or by a pressure assisted atomizer. Impinging on the heated surface with droplets enhances the heat transfer [?] because the drops spread over the surface and evaporate or form a thin film of liquid. By evaporation, convection and secondary nucleation (explained below) heat transfer is high even at low temperatures [?]. The heat transfer characteristics (heat flux as a function of the wall temperature) using spray cooling present the following behavior :

- For low wall temperature, the heating plate is cooled by a single-phase regime. In this temperature range, the liquid does not have time to warm up enough to reach the boiling onset condition at the heating wall because the flow is rapidly expelled from the heating surface. Convection heat transfer is enhanced because droplets impinge on the liquid film thus mixing the fluid better.
- Below the saturation temperature, the curve slope may increase due to the evaporation of a thin liquid film formed on the heating surface [?].

- When liquid near the heating wall becomes overheated, heat transfer is increased by secondary nucleation and/or boiling which enhance the heat transfer coefficient, the evaporation rate and the turbulence [? ]. Secondary nucleation occurs within the liquid contrary to boiling which occurs at the heating wall. The first of these two phenomena can be explained as follows : when the droplets enter the liquid film, the layers of adsorbed gas molecules surrounding the droplets are dissolved and new nucleation sites are formed. The number of nucleation sites is proportional to the number of liquid droplets forming the spray [? ]. Yang et al. [? ] indicated that the higher the number of nucleation sites, the higher the heat transfer coefficient is. Nevertheless, the augmentation of the droplet flow-rate also increases the liquid film thickness having a negative effect on heat transfer. So, heat transfer is not proportional to the mass flow-rate.  
In this range of temperatures (superheated liquid), boiling can also begin at the wall. Generally, no temperature hysteresis (no temperature excursion prior to the boiling incipience) is observed [? ]. It can be noted that boiling and secondary nucleation can be observed simultaneously leading to a higher heat transfer coefficient. Indeed, secondary nuclei are added to boiling bubbles. The bubble density and the turbulent mixing within the film are thus increased as is convection and evaporation [? ]. Furthermore, the spray also has an impact on bubble puncturing: the bubble life time and the puncturing diameter significantly decrease [? ]. This mechanism has to be considered in particular in studies focusing on dry out.
- When the flow increases further, the surface begins to dry. The critical heat flux is higher for spray cooling than for pool boiling because the vapor generated by phase change is quickly removed.

The spray cooling is very complex because it depends on many parameters such as, for example, nozzle to heating surface distance, incident angle, droplet size, droplet velocity, droplet density, surface roughness and orientation, non-condensable gas, air pressure, etc. [? ]. The effects on cooling performance of some of these parameters are described below.

### 3.3.1. *Effect of nozzle to heating surface distance*

Cheng et al. [? ] reported that heat transfer is very sensitive to spray height. Different zones can be defined within the spray as illustrated in figure 3. At low heights, the mainstream region (i.e. the region where the droplet density is high) forms a ring between the entrainment region and the exterior spray region (i.e. regions where the droplet density is very low). The spray impinges on a small part of heating surface. Nevertheless, the fluid forms a liquid film moving outwards from the heating surface. The liquid film covers nearly the whole heating surface, consequently the heating surface is cooled effectively and uniformly. A small amount of liquid is trapped inside the ring of the high droplet density impact zone forming a stagnation region. With the increase of the spray height, this zone expands from the center of the heating surface, causing a temperature rise at the center of the surface. The surface temperature is not uniform and heat transfer is reduced. If the spray height continues to increase, the volumetric flow-rate decreases while the coverage area increases. Some bubbles appear and stick to the surface center within the stagnation region, increasing the local heat transfer resistance. Heat transfer and temperature non-uniformity get worse. For the greatest distances, the spray area oversteps the heating surface. Liquid film and dry wall coexist on the heating surface: heat transfer deteriorates sharply.

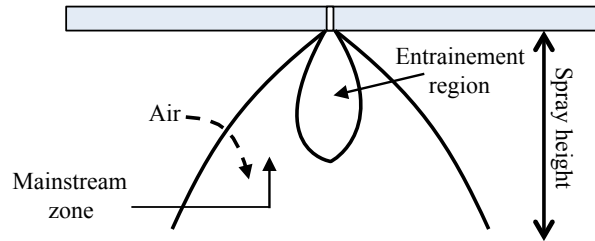


Figure 3: Spray flow field and definition of the various zones

Mudawar and Estes [?] proposed a critical heat flux (CHF) model developed around the concept of spray volumetric flux, which is defined as the flow rate of coolant divided by the spray impact area. They also developed experiments where the CHF measurements were made maintaining the flow rate constant and varying the spray height. Spray impact area decreases as nozzle to heating surface distance decreases yielding small CHF values. When the spray impact area is bigger than the heating surface (high nozzle to heating surface distance), CHF decreases because part of the liquid is not used to cool the surface. CHF maximum value is reached when the spray impact area just inscribes the heating surface.

### 3.3.2. Effect of incident angle on heat transfer

Li et al. [?] reported the effect of the spray incident angle on heat transfer. The model of Mudawar and Estes [?] was also modified to take into account the spray incident angle effect. Heat transfer was found to increase slightly from  $0^\circ$  to a critical angle value from which the heat flux decreases rapidly. For incident angles greater than this critical value, only part of the droplets impinges on the heat surface and single droplet momentum decreases, explaining the heat transfer decrease.

Silk et al. [?] reported an enhancement of the heat transfer as the spray incident angle increases. The spray inclination promotes liquid drainage in the stagnation zone that can appear under the spray. This conclusion is consistent with Wang et al's study [?] in which heat transfer is found to be enhanced when the incident angle is increased due to the reduction of the stagnation area and to the strengthening of the washing effect.

### 3.3.3. Effects of droplet diameter, flux, velocity and density

According to the model of Tilton and Pais [?], droplet diameter must be as small as possible and the impact surface area must be as large as possible to form a thin liquid film promoting evaporation and thus enhancing heat transfer. In addition, the droplet velocity must be chosen to favor the spreading of the droplets and to avoid rebounds. The same trend is observed as regards the critical heat flux. CHF increases as droplet diameter decreases because smaller droplets have a greater surface-to-volume ratio than larger droplets [?]. Thus, the smaller is the diameter, the greater is the cooling efficiency [?].

Similar conclusions were obtained by Chen et al. [?] concerning the droplet velocity effect: both heat transfer coefficient and critical heat flux are enhanced by an increase of the droplet

velocity because secondary nucleation is closer to the heated surface and mixing of the fluid is increased. Increasing droplet velocity also helps droplets to overcome the rising vapor flow to reach and wet the surface, thereby increasing CHF. Droplet velocity is the dominant parameter influencing the critical heat flux, changes in the diameter of the droplets having less effect.

As droplet density increases, more droplets break up the bubbles before the microlayer dries up. The action of these droplets helps replenish the heating surface with liquid and brings secondary nuclei into the liquid enhancing boiling heat transfer as well as convective heat transfer. However, it should be noted that excessively dense sprays do not help enhancing cooling efficiency because liquid is expelled over the heater without being vaporized or significantly heated [? ]. To reach high critical heat flux at a fixed flow rate, it is better to have a light spray with a high velocity than a dense spray with a low velocity. A small volumetric flux reduces liquid buildup: droplets can directly impinge directly on the heated surface. Nucleation is easier when the frequency of droplet impingement upon the surface is low that is to say for light spray. A large part of the spray evaporates which brings a large heat transfer enhancement upon transition from single phase liquid cooling to nucleate boiling [? ].

### 3.4. Summary

In this section, active techniques where the channel walls are immobile have been detailed. Electrohydrodynamics heat transfer process consists in coupling electric field with a flow field medium which generates several forces in both single-phase and two-phase flows. These forces lead to physical mechanisms (e.g. motion of the fluid, modification of contact angle and surface tension) that enhance heat transfer. Another active method consists in impinging a fluid at high speed (jet) or droplets (spray) on a heated surface. Jets break boundary layers and mix the fluid near the heated surface leading to the heat transfer enhancement. In the case of sprays, the heat transfer coefficient is increased by convection, evaporation and secondary nucleation.

These three techniques consist in directly acting on the fluid. **another** type of active methods consists in acting on the fluid through the deformation of a solid wall. They are described in the following section.

## 4. Active methods: case where a solid wall is periodically moving

Three main heat transfer enhancement techniques involving periodic movement of a solid wall are considered here :

- acoustic waves generated by high frequency oscillations of a membrane;
- synthetic jet where the flow is imposed by the motion of a diaphragm bounding a cavity;
- dynamic deformation of a solid at high amplitude.

In each case, the main expected effects are fluid mixing and boundary layer disruption created by surface displacement. In the presence of a single phase, these techniques favor convection intensity in particular in regions where there would otherwise be natural convection. Thereby, a forced flow generated by a vibrating beam (acoustic streaming) [? ? ] proves to be particularly interesting. The solid wall movement can also produce local cavitation due to pressure decrease. The use of acoustic vibrations to increase heat transfer has been a topic for research since the sixties whether in natural convection [? ? ], in pool boiling [? ], or in forced convection with

and without boiling [? ?]. Synthetic jet and dynamic deformation of the channel wall have been developed more recently as heat transfer enhancement techniques in order to respond mainly to the challenge for compactness in micro electronic applications.

The physical mechanisms involved and the enhancement potential of the three techniques are detailed below. A review on the ways used to generate the periodic movement of a solid wall considered in each of these techniques is then presented.

#### 4.1. *Ultrasound waves*

This section focuses on the effect of ultrasound waves on heat transfer when nucleation processes are involved. According to Legay et al. [?] ultrasound is defined as acoustic waves at frequencies higher than the upper limit of human hearing that is to say 16 or 20 kHz. Nucleation can be activated by cavitation and/or by boiling. One way to distinguish cavitation from boiling is to define cavitation as nucleation in a liquid at a pressure below the saturation pressure and to define boiling as nucleation in a liquid in which the temperature increases until the saturation temperature is reached [?].

The great majority of works done on convective and boiling heat transfer enhancement using ultrasound have been reviewed by Legay et al. [?]. In it, the reader interested in these techniques can find further details and data.

##### 4.1.1. *Non-boiling configurations*

The effect of acoustic vibration depends mainly on the heat transfer rate and on the quantity of gas dissolved in the liquid [?]. In natural convection, the bubble created by gaseous cavitation plays an important role in heat transfer. The best intensification of heat transfer is achieved where the acoustic vibrations cause a violent movement of the bubbles generated by cavitation. Uhlenwinkel et al. [?] investigated heat transfer from a circular cylinder and reported a heat transfer coefficient intensification of up to 25 times. Li and Parker [?] studied the effect of sound waves on natural convection heat transfer in a horizontal heating wire. They noticed that the enhancement was greatest when the fluid was subcooled. Wong and Chon [?] found that this intensification can reach up to 800% in the most favorable cases. This huge enhancement is due to the motion of cavitation bubbles in the vicinity of the heating wire, which creates disturbances in the stagnant liquid film near the wire. The experiments conducted by Iiad and Tsutsui [?] in natural convection indicate similar trends. The maximum intensification they found was of the order of 200%. Nomura et al [?] examined the influence of flow induced by ultrasonic cavitation on a flat surface. In tap water, ultrasonic waves induce a jet that can increase heat transfer at the hot plate. When non-condensable gases are present, ultrasonic waves generate cavitation and heat transfer is enhanced up to ten times due to small scaleless perturbations.

Cai et al. [?] performed a numerical study of the natural convection heat transfer of a liquid in a square enclosure with or without acoustic cavitation. The cavitation model takes into account the effects of phase-change, of the dynamics of the bubbles and of the non-condensable gases. To model sound waves, periodic pressure fields are imposed in the area where the wave transmitter is. The results show that cavitation intensifies heat transfer and improves the homogeneity of the temperature of the liquid. These results can be explained by two main mechanisms:

- the impact of the jets formed by the collapse of cavitation bubbles on the heating wall which reduces the boundary layer;
- the increase of liquid mixing.

In forced convection, the gain obtained with acoustic vibrations on the heat transfer is more moderate. Lemlich and Hwu [?] observed different levels of maximum heat transfer intensification in laminar and turbulent regimes. They obtained maximum enhancements of 51% in laminar flow and of 27% in turbulent flow.

#### 4.1.2. Boiling configurations

Regarding experiments with boiling, the impact of the acoustic wave on heat transfer has been the subject of numerous publications. Bergles [?] became interested in this technique in the early sixties. He showed that for wall temperatures of 30 K below the saturation temperature, the heat transfer coefficient is invariant. When the wall temperature is increased close to the saturation temperature, the heat transfer coefficient gradually increases to reach an enhancement value of up to 100%. Intensification is then reduced on fully developed boiling. Several years later, Wong and Chong [?] confirmed that acoustic waves do not have much influence in a fully developed boiling regime. The increase of heat transfer found for low wall superheating decreases with the increase of wall superheat until it becomes negligible in fully developed boiling. More recently, Liad and Tsutsui [?] conducted experiments to determine the influence of ultrasonic waves on heat transfer in nucleate boiling and film boiling using an acoustic transducer. They noticed an increase in heat transfer during film boiling, but not during nucleate boiling. The maximum intensification they found was of the order of 150% in the case of film boiling. The studies of Kim et al. [?] and Sun et al. [?] consolidate this behavior. Kim et al. [?] used a vessel filled with liquid in which a heating wire is immersed. The liquid was degassed beforehand or not. When the fluid was not degassed prior to the experiments, the cavitation bubbles were present throughout the volume of liquid. When the liquid was degassed before, there were no nuclei in the liquid to onset the nucleation. For liquid temperatures close to saturation temperature, the solubility of non-condensable gases is low, and there is nearly no difference in the results obtained with degassed liquid and non-degassed liquid. So, the heat transfer enhancement due to the ultrasound is low in the developed boiling regime, the maximum achieved is less than 10%. Nevertheless, for higher heat flux, film boiling is delayed increasing the critical heat flux [?]. For instance, Jeong and Kwon [?] found an increase of critical heat flux of up to 126% in the case of pool boiling.

The intensity of the sound wave as well as the distance from the sound source to the surface to cool also have an effect on boiling heat transfer. Zhou et al. [?] found that the boiling heat transfer coefficient can be increased or decreased according to the values of these parameters. Cavitation bubbles provide embryos within the cavities of the heating surface. In addition, the cavitation bubbles can be projected as a jet on the heating surface. But if the sound source is too far from the heating surface, heat transfer decreases as the jet phenomenon disappears and the cavitation bubbles have the effect of compressing the liquid which restricts heterogeneous nucleation.

Another peculiarity that must be noted concerning the effect of acoustic waves on boiling heat transfer characteristic is drastic reduction of the wall temperature excursion at the onset of boiling. Park and Bergles [?] studied the effects of ultrasound on heat transfer in the case of boiling in a dielectric liquid. They noted that the hysteresis in the boiling characteristic curve was negligible and explained this behavior by the formation of cavitation bubbles near the heating surface which allows the activation of the boiling process at lower wall superheat.

To conclude, boiling heat transfer using ultrasonics waves has many advantages such as:

- High increase of heat transfer
- High increase of critical heat flux
- Disappearance of boiling incipience temperature hysteresis

#### 4.2. Synthetic jet heat transfer

A synthetic jet, also named zero-net mass flux jet, is formed when the fluid is alternately sucked into and ejected from a small cavity. The orifice flow is forced by periodic pressure varied by the motion of a diaphragm bounding the cavity. This technology is very attractive because no external fluid source is required. It is well suited for microelectronic cooling applications as it is coherent with the compactness challenge [? ].

Relatively few papers are available on synthetic jet impingement cooling. Chaudhari et al. [?] studied the heat transfer characteristics of the process. The behavior of the average Nusselt number is found to be similar to that obtained for a continuous jet. The maximum heat transfer coefficient with the synthetic jet is found to be up to 11 times the natural convection heat transfer coefficient. Chandratilleke et al. [?] performed a numerical study of flow and heat transfer in a configuration where the synthetic jet is perpendicular to a flow of air inside a microchannel. They showed that the pulsating jet interacts with the microchannel flow and they obtained a 4.3 fold increase in the heat transfer coefficient.

The heat transfer characteristics depend on numerous parameters such as orifice diameter, thickness of the orifice plate, cavity depth, orifice shape, excitation frequency and amplitude, flow parameters, distance between heated plate and jet. Chaudhari et al. [?] studied the effect of orifice shape on heat transfer enhancement using synthetic jet. In reference [? ], Chaudhari et al. identified the governing non-dimensional parameters on the Nusselt number. They found that the Nusselt number is a function of Reynolds number, the Prandtl number, the normalized axial distance (ratio between the distance of the heated plate to the jet orifice and the orifice diameter), the normalized length of the orifice (ratio between the length of the orifice and the orifice diameter) and the normalized heater size (ratio between the half-length of the heated plate and the orifice diameter). Chaudhari et al. [?] also proposed to use a multiple synthetic jet formed by a single cavity connected to 5 orifices (1 center orifice and 4 satellite orifices) and a single actuator. They showed that, for the same input power, this configuration enhances heat transfer and reduces the optimal jets-to-plate distance compared to the single synthetic jet, which is an advantage in terms of compactness.

#### 4.3. Heat transfer enhancement using high amplitude vibratory motion

Among the solutions using the deformation of a wall to increase transfers, one of which is particularly studied. It consists in evaluating the impact of an oscillation of a flexible blade constituting a "piezoelectric fan". Such a piezoelectric fan could be miniature which is consistent with the objective of compactness. The oscillation of the blade causes motion of the surrounding fluid. Generally, piezoelectric fans are used to impose motion of a fluid locally in a globally stagnant fluid area, increasing heat transfer in hot spot zone, and so reducing local temperature. A review of studies about heat transfer enhancement using piezoelectric fans available in the literature is reported in table 3.

Other studies about heat transfer enhancement using wall morphing have been conducted. For



example, using a plate oscillating in a channel enhances natural and forced convection. In the same manner, a vibration can be imposed directly on the wall of a channel. Table 4 reports examples of studies investigating about the effect of the oscillation of a solid (plate inside the channel or wall channel itself) on the heat transfer.

For the boiling condition, studies dealing with the use of high amplitude vibratory oscillation to enhance heat transfer are less numerous. However, the use of phase change processes in a narrow space is a promising solution to enhance heat transfer. One major problem in thermal management systems involving boiling is the high temperatures that must be reached to obtain the onset of boiling. To avoid this problem, Léal et al. [?] proposed to increase the liquid metastability level by temporarily decreasing the pressure of the liquid. In a confined space, dynamic deformation of the surface with low absolute amplitude (of the order of hundreds of micrometers) but with high relative amplitude may lead to significant changes of the liquid pressure in the channel. The idea is thus to combine cavitation and boiling in order to obtain boiling incipience at a lower wall temperature and/or to control the onset of nucleation.

Their experimental device was mainly composed of a heating plate having a  $0.1 \mu m$  mean roughness and a confinement plate deformed by a piezoelectric actuator. The confinement wall was placed above and parallel to the heating surface. It was dynamically deformed at its center whereas its periphery was maintained fixed. Figure 4 reports schematically the deformation cycle of the confinement wall. The frequency of the dynamic deformation ranged from 0 (i.e., without actuator) to 100 Hz.

The experiments were carried out using n-pentane at atmospheric pressure. The fluid temperature was maintained slightly below the saturation temperature (at 308.5 K) to reduce the occurrence of boiling phenomena other than at the heated surface. Before and during experiments the fluid was degassed to avoid the presence of non-condensable gas which could have an effect on pool boiling incipience temperature. The temperature was followed over time when a heat flux step was applied with and without powering-on the actuator.

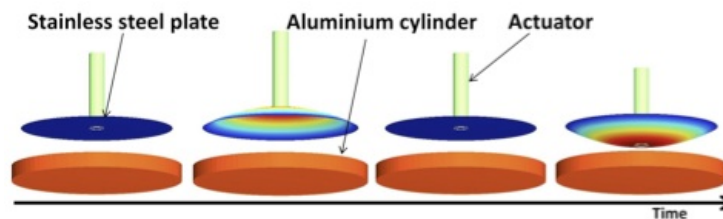


Figure 4: Deformation cycle of the confinement wall used by Léal et al. [?] to combine cavitation and boiling in the nucleation process.

Without dynamic deformation, pool boiling incipience wall superheat is about 21.7 K whereas with dynamic deformation at a frequency of 100 Hz boiling incipience wall superheat is dramatically reduced to about 2.3 K. Thus, the decrease in pool boiling incipience wall superheat is almost 90% for this experiment. The wall superheat at the onset of boiling was found to decrease continuously as a function of the frequency.

Published year	Authors	Brief study description	Results
1979, 1981	Toda [? ? ]	Theory for air flow generation, vibration prediction, heat transfer experiments. Vibrating $PVF_2$ are used in resonant condition	Decrease the temperature ( $-17^\circ C$ ) of the radiation panel of a power transistor in a TV receiver by fans. Prediction of air flow velocity and air volume flow rate
1994	Ihara and Watanabe [? ]	Study of the flows around the ends of oscillating flexible cantilevers by numerical simulation and experimental results. Authors reported the effect of the distance between two plates oscillating in-phase and counter-phase.	Validation of numerical model with flow visualization experiments. Distance has an important effect at counter-phase whereas the distance affected scarcely the flow at in-phase case.
2004	Açikalin et al. [? ]	Feasibility of incorporating piezoelectric fans in actual electronic devices. The aim is to determine the optimal location of the fans.	Supplement cooling in hot spots and stagnant areas. Application for small device where rotary fans are not practical and where heat flux densities are high.
2005	Wu and Ro [? ]	Prototype of a miniature cooling system with vibrating piezoelectric bimorphs : study of flow pattern and cooling effect on microelectronics	Enhancement up to 210%. A temperature drop of $25.9^\circ C$ from $92^\circ C$ was obtained.
2007	Açikalin et al. [? ]	Parametric study about the effect of fan vibrations near a discrete heat source (distance to target, vibration amplitude, resonance or not) by computational model	Enhancement up to 375% of heat transfer relative to natural convection
2007	Kimber et al. [? ]	Experiments about heat transfer due to the flow induced by vibrating cantilevers	Models to evaluate heat transfer coefficient in functions of vibration amplitude and distance from target. The authors reported that this behavior shares a number of characteristics with jet impingement parameters.
2008	Kimber et al. [? ]	Experiments to evaluate the flow and the pressure induced by vibrating cantilevers	Tools predicting the flow and pressure generated
2009	Kimber and Garimella [? ]	Study of the cooling performance of piezoelectric fans array in their first resonant modes. Two piezoelectric fans are set near constant heat flux surface. Some parameters are tested: vibrating amplitude, distance from the heated source, pitch between fans.	They found an optimum spacing between fans for heat transfer (about 1.5 times the vibration amplitude) and proposed a correlation to describe the enhancement of the heat transfer in terms of governing parameters.
2009	Kimber and Garimella [? ]	Quantify the influence of each parameters and its impact on heat transfer performance	Concluded that maximum performance, for a given fan, can be described by amplitude and frequency oscillations; Frequency has a more important influence than amplitude in determining the maximum heat transfer. Proposed correlations for determining heat transfer
2009	Dey et al. [? ]	Computation study on the enhancement of forced convection	The authors found a dependance between Nusselt number on Reynolds number, oscillation frequency, oscillation amplitude. They reported the existence of critical threshold of frequency and amplitude. Under these threshold value, oscillations do not affect Nusselt number.
2009	Abdullah et al. [? ]	Numerical and experimental studies of the cooling capability of piezoelectric fans for electronic devices. Piezoelectric fans are oriented horizontally.	Reduction of the temperature of the heat source by as much as $68.9^\circ C$ . Good agreement between experimental and simulation results (difference of 11%).
2009	Liu et al. [? ]	Experimental work about thermal performance of piezoelectric fans. Six piezoelectric fans with various blade geometries and arrangements are tested.	Two modes of heat transfer enhancement (same order of magnitude): entrained air flow and jet-like air stream. Proposition of a correlation about thermal performance for vertical arrangement
2009	Açikalin and Garimella [? ]	The aim is to develop and validate a numerical model for the effect of piezoelectric fans on flow and heat transfer.	Numerical model agreed to the heat transfer experiments (20%) in most of cases.

Table 3: Summary of studies about heat transfer enhancement using piezoelectric fans

#### 4.4. Summary

Periodically moving a wall leads to high heat transfer enhancement values, either with or without boiling of the working fluid. Acoustic waves improve cavitation phenomena which in

Published year	Authors	Brief study description	Results
2002	Yang [? ]	Numerical studies the use of a vibrating bar upstream of heated components	Heat transfer enhancement by vortex generation. Increase of frequency and amplitude oscillation implies the increase of heat transfer
2007	Florio and Harnoy [? ]	Numerical studies about heat transfer performance for different orientations of vibrating beam placed in the center of a channel. Oscillating plates are set on cross flow in vertical channel	Enhancement of heat transfer relative to natural convection.
2011	Florio and Harnoy [? ]	Numerical investigation of natural convection enhancement by a vibrating plate and a cross flow opening.	Up to 50% enhancement in the local heat transfer coefficient for vibrating plate amplitudes higher than 30% of the mean clearance space and frequency higher than 82 rad/s.
2007	Florio and Harnoy [? ]	Analysis of main parameters is obtained by numerical simulations. The system consists in a vertically oriented flow path with one uniformly heated wall and one transversely oscillating insulated wall.	Supplement natural convection. Enhancement of 340% in local heat transfer coefficient.

Table 4: Some studies about heat transfer enhancement using wall morphing and discrete vibrating plates set on a channel

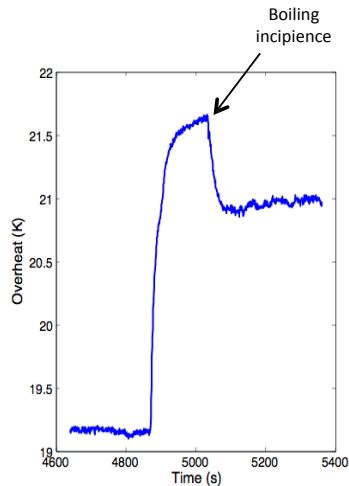


Figure 5: Wall superheat as a function of time when the actuator was powered-off. Onset of boiling of the working fluid (n-pentane) was found to occur when the wall superheat reached 21.7 K.

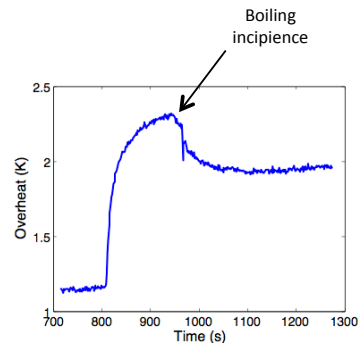


Figure 6: Wall superheat as a function of time when the actuator was powered-on (amplitude and frequency of the deformation of the confinement wall were equal to  $210 \mu\text{m}$  and  $100 \text{ Hz}$ , respectively). Onset of boiling of the working fluid (n-pentane) was found to occur when the wall superheat reached 2.3 K.

turn enhance heat transfer by mixing effect. In boiling configuration, acoustic waves enhance both heat transfer and critical heat flux. Increasing the amplitude (at lower frequencies) sets the fluid in motion and thus enhances the convective effect. Several ways may be considered to generate such dynamic wall deformations. Among them, the use of piezoelectric actuators appear as one of the most suitable solution. Indeed, important developments have been made in this field during the last decades so that amplitude of several hundred micrometers can be easily obtained with actuators developing strong forces. The following section detailed these piezoelectric actuators characteristics and performances to demonstrate they constitute an effective solution and to provide the informations necessary to their implementation in a thermal device.

#### 4.5. Means of actuation for heat transfer active methods - The piezoelectric solution

Whether the generation of acoustic wave in the fluid [? ], synthetic jet generation [? ] or deformation of a channel wall [? ], the actuation mechanism is still the same. It is initiated by setting in motion at least one solid body in contact with the fluid medium. Therefore, the distinction between these active methods simply relies on the geometry of the containment area, the operating frequency and the displacement amplitude. Depending on these constitutive parameters, a macroscopic motion of the fluid may occur or not.

While there are numerous technological solutions for moving a solid body, the most commonly used in applications involving solid/fluid interfaces is the piezoelectric technology. Indeed, the power densities, the operating frequencies and compactness of piezoelectric actuators are particularly suitable for these applications. Besides, piezoelectric actuation devices are countless. That is the reason why the illustrations in the following part only concern studies involving a solid/fluid interface.

Firstly, the piezoelectric principle is briefly introduced. Then, because there are few studies about active methods of heat transfer, the different application fields involving piezoelectric transducers compatible with heat transfer requirements are investigated. Different categories of piezoelectric transducers are finally detailed.

##### 4.5.1. Piezoelectric principle

The piezoelectric property is the capability of some natural or synthetic materials to produce an electric field between its electrodes when a mechanical stress is applied to the volume of the piezoelectric solid. Sensors such as strain gauges, accelerometers and many others are commonly built on this direct piezoelectric effect. Likewise, a mechanical displacement is generated when a sufficiently high electric field is applied to its surface by electrodes. This is in fact called the piezoelectric reverse effect. Figure 7 illustrates the various types of electromechanical coupling for a piezoelectric material with a parallelepipedic shape, depending on the direction of its remanent polarization  $P$  and the direction of the applied electric field  $E$ .

Many materials present a piezoelectric property but the industrial domain mainly relies on the use of LZT ceramics (lead zirconate titanate) or barium titanate ( $BaTiO_3$ ) because of their high electromechanical coupling factor (ex: longitudinal LZT coupling factor  $d_{33} \approx 500.10^{-12}m/V$ ). By applying some tens or hundreds of volts at its surface, the resulting elementary deformation of the material is of the order of a few micrometers or less. Fortunately, these displacements can be strongly increased in different ways: stacking of ceramics or multilayers, by mechanical amplification (like lever arms, deformable elliptical shapes, etc.) or by using the resonant property of the mechanical structure.

##### 4.5.2. Actuation principles

An interesting review about piezoelectric transducers in sonar systems has been published by K. Benjamin in [? ], describing the widespread technical solutions. Many of the architectures presented are transposable to active heat transfer enhancement methods.

short

The piezoelectric actuators have very different performances depending on the architecture chosen, the type of ceramics used or the mode of operation (quasistatic, inchworm, resonant, etc). They cover a wide range of velocities, displacement amplitudes and force densities. Considering

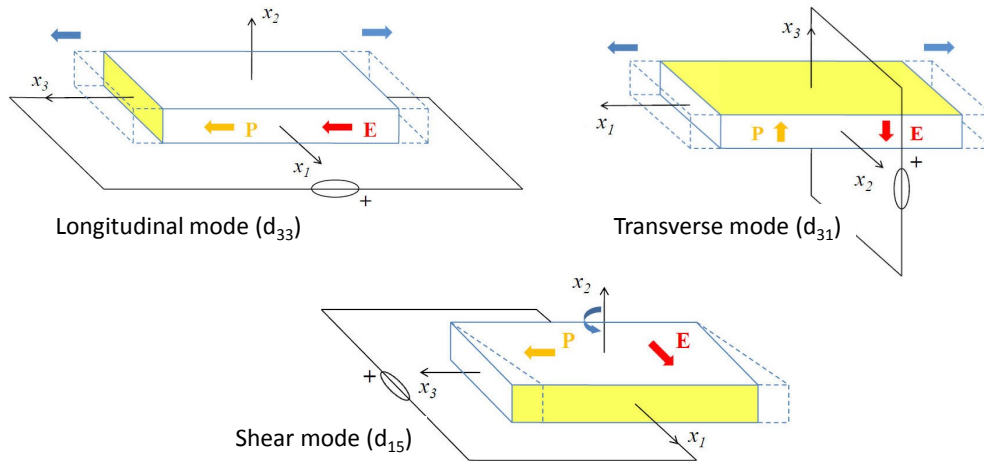


Figure 7: Different electromechanical coupling in piezoelectric ceramic

situations where there is an interaction between solid and fluid, the piezoelectric devices may be grouped into three main categories: mass actuation, the flextensional displacement and acoustic wave in solid.

*Mass actuation.* The principle simply involves setting in motion a solid mass completely or partially submerged in fluid. The simplest structure is called a Langevin transducer as attributed to Paul Langevin [? ]. Its elementary architecture consists of piezoelectric ceramics sandwiched between two passive mechanical parts and assembled by screws or glueing. The displacement can be strongly increased by matching the supply voltage frequency with the resonant mechanical frequency of the assembly. Among this kind of actuators is the Tonpilz actuator which can be designed for use in a wide range of operating frequencies (1-50 kHz) depending on its dimensions and mass. This actuator provides dynamic sub-millimetric displacements in a narrow frequency band [? ? ]. **The head mass of the Tonpilz illustrated on figure 8 is commonly a mushroom shape giving better acoustic impedance matching between solid and fluid for higher acoustic power. These actuators were notably used for the study of ultrasonic cavitation phenomena by Gachagan et al. [? ? ].**

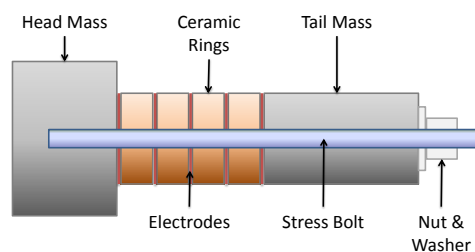


Figure 8: Cross section view of a Tonpilz transducer

Other shapes of transducers are also conceivable, depending on the expected application and the size constraints. As an illustration, an annular ultrasonic projector can be used like in [? ]. The acoustic energy is obtained from the alternative radial stretching and contraction of the ring.  
short

An interesting comparison between different low frequency underwater projectors was done by Decarpigny et al. [? ] transducers with different active technologies such as piezoelectric or magnetostrictive. The transducers described are dedicated to high power ocean sonar applications, but the architecture is easily transposable to active heat transfer enhancement methods.

*Flextensional actuators.* When both compactness and high amplitude displacement are necessary, mass actuation does not meet these requirements well. A simple deformation of one or several confinement walls may be sufficient to affect the heat transfer behavior as illustrated in figure 4 from Léal et al. [? ]. The flexion of an elastic beam or plate (comparable to a membrane) is a convenient solution commonly used in the pump and micropump systems or in the piezoelectric fans (Tab. 3) can also be grouped with flextensional actuators.

They typically operate at low frequency (100 Hz up to 1 kHz) with a displacement amplitude up to several millimeters. They are commonly constituted by piezoelectric ceramics which bend a flexible passive element. The geometry of the passive piece can vary, depending on the expected operating frequency and the desired displacement amplitude. Figure 9 shows two typical geometries. The simplest structure is a bimorph bender constituted by ceramic glued on a resonator. short Different deformation shapes are possible depending on the frequency and the edge conditions (fixed or free to move). Among the existing mechanical structures, the elliptic amplifier is one of the most widespread. It provides high displacement amplitude and while being conveniently compact owing to the deformation relation in the orthogonal direction. In the case of the solid/fluid interface configuration, the confinement wall may be directly composed of the flexible element or attached to it. short

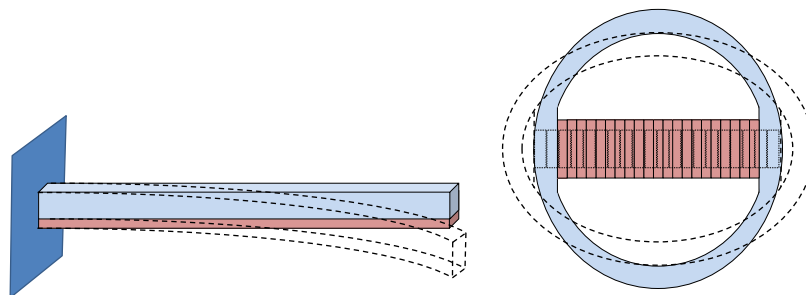


Figure 9: Bender and elliptic actuators

Obviously, the flexural mechanical actuation has to be sufficiently powerful to not be significantly influenced by the fluid pressure. Moreover, the wall must also remains in its elastic mechanical behavior without crossing the plastic transition limit (irreversible deformation). More exhaustive enumerations of flextensional actuator types are available in [? ? ]. For instance, from a sonar application [? ], a flexible membrane configuration has been developed. This architecture is quite similar to that of a loud speaker. The trilaminar membrane disc is composed

of piezoceramics and a resonator glued together, and fixed all along its edge. The bending of the membrane by the piezoeffect can promote cavitation and acoustic wave propagation into the confined fluid. Moreover, in the case of a thin channel configuration, the resulting deflection leads to the control of the thickness dimension in quasistatic or dynamic conditions as required.

*Mechanical acoustic wave in a solid.* Another approach to influence the thermal behavior of a fluid is to disturb the boundary layer by creating waves on the wall in contact with it. They can be standing or progressive waves. Among the available techniques, the typical one is the transducer SAW. The Surface-Acoustic-Wave transducers are called IDT (Inter-Digital Transducers) because of the shape of their interlaced electrodes deposited on a piezoelectric substrate [? ]. JR Klauder was one of the first authors to speak about SAW transducers in the 60's. Such structures were used for the actuating and sensing the position of a droplet by Alzuaga et al. (2003) [? ]. These authors built a structure able to move a droplet of liquid in two directions, demonstrating the ability to promote fluid movement in the vicinity of the wall.

Besides, other works based on LZT self-focusing acoustic transducers (SFATs) studied the ability of mixing and transporting fluids in microfluidic subsystems [? ]. The research describes noninvasive acoustic micromixers and transporters as well as a microfluidic subsystems (on a single silicon chip) based on the two microfluidic actuators. Figure 10 illustrates the interdigital structure in a narrow channel generating acoustic waves about 10-12.5 MHz, and an acoustic streaming effect. This study has proved the ability of driving fluids along the channels (fluid flow speed up to 123 mm/s) and mixing liquids that are contained in a glass or plastic wafer.

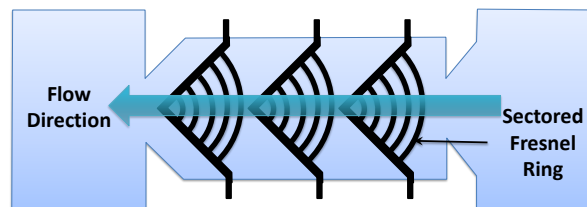


Figure 10: Schematic view of the fluid driver composed of a linear array of acoustic transducers with sectored Fresnel annular rings developed by Yu et al. (2006) [? ].

Another way to promote an acoustic wave is to use massive actuators attached to a deformable body. Figure 11 illustrates the elementary architecture with two actuators [? ? ]. The phase shift between the actuators leads to generate standing or travelling waves. The physical principle of droplet transport has been pointed out by J. Scortesse et al [? ]. This motion relies on the droplets location in the antinode and there moving with the traveling waves.

The piezoelectric actuation of solid/liquid interfaces can be done by different methods that can largely match with the constraints of the heat flux exchange intensification devices.

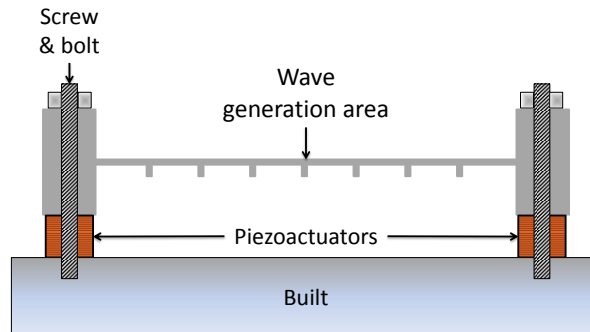


Figure 11: Schematic view of the periodic vibrating device for moving droplet with piezoelectric stacks developed by Alzuaga et al. (2005) [? ].

### 5. Recent works: High amplitude deformation of the wall, compact heat exchanger integrating the pumping function

The increasing demands of new and effective mixing and heat transfer technologies for various industrial fields (e.g. chemical engineering) associated to the increasing need to control highly exothermic or explosive chemical reactions as well extract high heat fluxes density on various objects, have contributed to the development of new and effective technologies involving sub-millimetric channels. They should allow terms to increase the selectivity of process, the purity of products, safety and give access to new products and processes in many fields as they allow extracting high heat flux density and local control of temperature. Several groups of research institutes and many industries are currently active in this field. During the last twenty years, new needs have emerged, linked to the development of mini and micro devices (microfluidic, micro reactors). These systems can exacerbate the surface effects and reduce the volumes involved. The development of miniature heat exchangers is also due to the need to integrate the cooling systems as closest as possible of the dissipaters in electronic systems. If these systems can increase the compactness, one major drawback is the difficulty to disturb the boundary layers, as flow is mostly laminar in such small devices. Consequently, the net gain in terms of mass or heat transfer is mainly due to the exchange surface extension as the mass and heat transfer coefficients are usually reduced compared to classical systems. Moreover such devices lead usually to very high-pressure drops. Indeed, to address the problem of mass and heat transfer coefficients; high velocities of the fluid must be achieved. Many techniques have been developed to enhance heat transfer in classical as discussed in previous sections. Applying these techniques at very low scale remains challenging so that they appear not (yet) suitable for micro thermal devices. Moreover, they involve an additional increase in the pressure drop and so an increase in the mechanical power necessary to generate the flow of the working fluid within the micro channel.

To address the problem of heat transfer coefficient decrease in low diameter channel high velocities of the fluid must be achieved, leading to high pressure-losses and consequently to large mechanical power consumed by the pump. The latter is generally located more or less further upstream the heat exchanger. Because of the externalization of the pump, the flow distribution in the channels of the exchanger is often ill controlled and additional pressure drop are generated



as well as heat transfer reduction. Integrating the pump in the heat exchanger appears thus as a solution to architectural optimization of thermal systems. Besides the performance issue, the operating conditions of heat exchangers are generally constrained, since the heat transfer coefficient is proportional to the fluid velocity: when the flow rate is imposed in a given process, the amount of heat extracted is not easily adjustable. It is therefore not possible to control dynamically and independently the temperature and the flow rate within the process.

Literature, discussing the effects of wall dynamic deformations (channel size varied with both space and time) on flow and heat transfer, is sparse although work dealing with static corrugated channel are numerous. Several authors have considered flow inside squeezed thin films like Languis [? ], however, only few of them have analyzed heat transfer inside squeezed thin films such as Hamza [? ], Bhattacharyya and al. [? ] or Debbaut [? ]. But, in these works, the squeezing was not of oscillatory type.

Nakamura et al. [? ] investigated numerically the influence of the wall oscillation on the heat transfer characteristics of a two dimensional channel. In their study, the moving boundary problem was transformed into a fixed boundary problem using the coordinate transformation method. Khaled and Vafai [? ] considered flow and heat transfer inside incompressible oscillatory squeezed thin films. These studies involved boundaries that move with a prescribed motion (e.g., displacement, pressure, or velocity). They studied the effects of both external squeezing and internal pressure pulsations on flow and heat transfer inside non-isothermal and incompressible thin film that has a small thickness compared to its length. The lower plate of the thin film was fixed while the vertical motion of the upper plate was considered to assume a sinusoidal behavior when the thin film gap is not charged with the working fluid. They report influence of non-dimensional parameters defined specifically on both heat transfer and fluid flow, correlations are obtained for Nusselt numbers and their corresponding amplitudes for two different thermal conditions: constant wall temperature and uniform wall heat flux. They concluded that, fluctuations in the heat transfer and the fluid temperatures can be maximized at relatively lower frequency of internal pressure pulsations.

Kumar and al. [? ] recently studied heat transfer phenomena inside the tubes with static and moving sinusoidal corrugated walls. They focused on intermediate size tubes (mm) as other dimensions have been previously studied for static cases. Numerical analysis were performed to study the effect of spatial wavelengths ( $\lambda = 1/2, 2/3, 1, 2 \text{ mm}$ ), Reynolds number (1-120) and amplitude (1%-20% of tube diameter,  $D_0$ ) on heat transfer and pressure drop in static and moving sinusoidal corrugated wavy walls. They compare a static deformed wall tube to one whose wall is moved by a stationary radial sine wave. For static wall case, they have shown that, heat transfer is enhanced (compared to straight tube) only when amplitude is less than 5%.  $j/f$  factor is better than established straight tube flow but when amplitude increases, heat transfer performance decreases and is always lower than the performance of straight tube of small length ( $L/D \sim 10$ ) but higher than fully developed one ( $L/D < \infty$ ) and, indeed, heat transfer increases with increase in the Reynolds number. Moreover, pressure drop increases due to obstruction produced within fluid flow by increasing number of sine waves ( $\lambda$  decreases) on wavy wall but heat transfer coefficient remains almost unchanged. For the moving wall case, Nusselt number, is higher (for all frequencies) compared to static wall case and heat transfer increases up to 35-70% are observed. At higher amplitudes, there is a sharp decrease in pressure drop by a factor of 1.2 to 5 compared to static case. Moreover, heat transfer and pressure drop values apparently change erratically with wall frequency and no general trend in respect with operating parameters has been found. They classified different structures of tubes according to their use in different engineering applications.

During the last few months, Leal et al [?] numerically studied fluid flow and heat transfer behavior in a dynamically deformed channel in order to gain a better understanding of operating parameters influence on heat transfer and pumping performances. They also realize a laboratory experiment whose design is similar and that use piezoelectric actuators to generate the wall deformation. But they do not yet report experimental results in terms of heat transfer enhancement and pressure losses reduction. The virtual prototype is rectangular (width = 30mm, length = 100mm, variable height  $e$ ). It constituted of two parallel plates and three different average channel heights are studied:  $e = 0.3, 1$  and  $3$  mm. The bottom one is fixed and it is heated at constant heat flux and the top one is adiabatic and dynamically deformed by a progressive sinusoidal wave figure 12). Constant fluid properties were assumed and the flow was considered three-dimensional and laminar. A constant pressure difference was imposed between the ends of the channel to obtain significant fluid flow whatever are the deformation wave characteristics (including the reference case where the two walls of the channel are immobile).

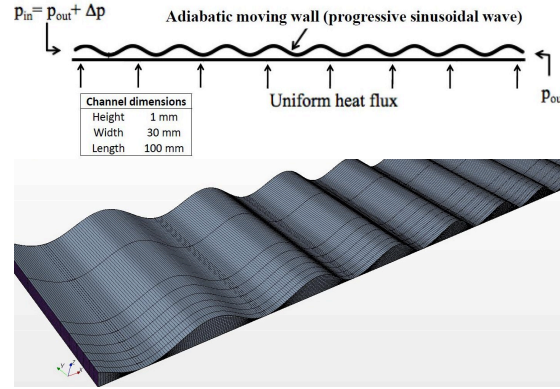


Figure 12: studied geometry, boundary conditions and mesh view. In order to observe the fully developed regime ten wavelengths are imposed along the length of the channel (10 cm), corresponding to  $K = 100m^{-1}$ . The relative amplitude ( $A_0/\delta$ ) varies from 2% to 98.5% while the frequency varies from 1 Hz to 50 Hz. The thickness of the channel (without deformation) is set to  $\delta = 1mm$ . The pressure difference between inlet and outlet is fixed to  $\Delta p = 20Pa$ . At the entrance section, homogeneous temperature is imposed.

The influence of the movement parameters i.e. frequency ( $f$ ), amplitude ( $A_0$ ) and wavelength ( $\lambda$ ) is analyzed. The calculations are performed using the commercial software Starccm+ (CD-Adapco).

Constant fluid properties (water at 20C) are assumed and the flow is considered three-dimensional and laminar. The transient conjugate flow and heat transfer problems are solved simultaneously using a segregated approach with implicit second order temporal discretization. The time step is chosen small enough to allow an accurate description of the wall movement: typically 20 up to 30 time steps per period.

The geometry of the upper adiabatic wall  $z(x, t)$  as a function of time  $t$  and longitudinal abscissa  $x$  is :

$$z(x, t) = \delta - A_0 \sin(2\pi(ft - Kx)) \quad (4)$$

where  $\delta$  is the mean distance between the two plates,  $f$  is frequency (Hz) and  $K$  is the number of wavelengths per unit length ( $m^{-1}$ ).

The first advantage of this device is to integrate the pumping function of the coolant within the heat exchanger, which then becomes a multifunctional system. Indeed, the flow is induced by the wave of deformation that causes the fluid to flow through viscous and pressure forces allowing thus an increase of the compactness of the processes. Moreover, reversing the wave direction allows reversing the flow direction. Thus this device presents a strong potential for chemical applications such as reverse flow reactors.

The prescribed pressure difference across the channel leads to a flow rate of about  $18.2 \text{ kg.m}^{-2}.\text{s}^{-1}$  (superficial velocity  $0.02 \text{ m.s}^{-1}$ ) for the flat channel. Mass flux ranging from 1 to  $510 \text{ kg.m}^{-2}.\text{s}^{-1}$  can be obtained by varying amplitude between 0 and 98.5% and frequency between 1 and 50 H z . The pumping function is thus clearly demonstrated as the flow rate is multiplied by more than 25 compared to the static channel case. The best pumping performances are obtained for the most important relative amplitudes, making the proposed system well suited for micro/mini-channels. The choice of the pertinent parameter used to control the mass flux is governed by the expected heat transfer intensity.

The second advantage of such device is the heat transfer enhancement due to local perturbations of the boundary layers from the changes of topology, velocity and temperature. The local heat transfer coefficient is obtained by the evaluation of the averaged fluid and wall temperatures on one lengthwave and on one temporal period. Dynamic deformation greatly enhances heat transfer and strongly reduces the establishment time.

From a quantitative point of view, considering the conditions shown in figure 12, gains (relative increase of mean heat transfer coefficient compared to the flat channel) of up to 500% on the intensity of heat transfer can be achieved for the highest values of the relative amplitude ( $A_0/\delta$ ) and frequencies of a few dozen of Hertz. The gain sensitivity to amplitude is greater than to the frequency variations.

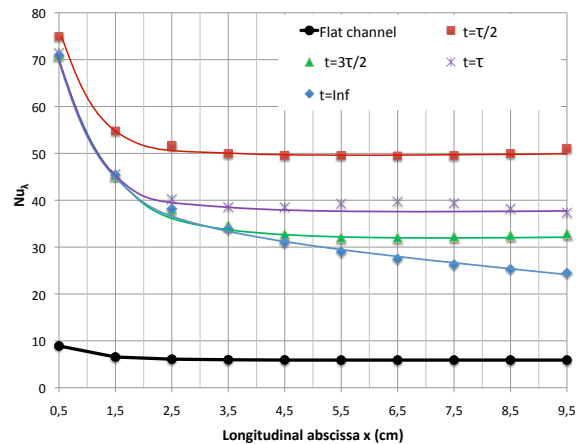


Figure 13: Evolution of  $Nu_x$  versus the longitudinal abscissa at different times until the established regime for static and dynamic cases. Wall dynamic deformation enhances heat transfer throughout the entire heating plate. Near the end of the channel and at steady state, the gain on local heat transfer is about 3. Without deformation, establishment time is about one minute whereas dynamic deformation sharply reduces this duration : there is no more temporal evolution of  $Nu$  for time superior to 2 periods (about 0.01 s).

The third advantage of the dynamic deformation is to uncouple flow and heat transfer rates, allowing to impose independently the fluid flow and the heat transfer coefficient within a single apparatus. This unique properties could be explained by the facts that flow rate is proportional to

frequency whereas heat transfer depend mainly on relative amplitude. This property constitutes a major advance in terms of process control as a simple deformation command according on the process constrains (e.g. wall temperature or outlet fluid temperature) could be easily derived.

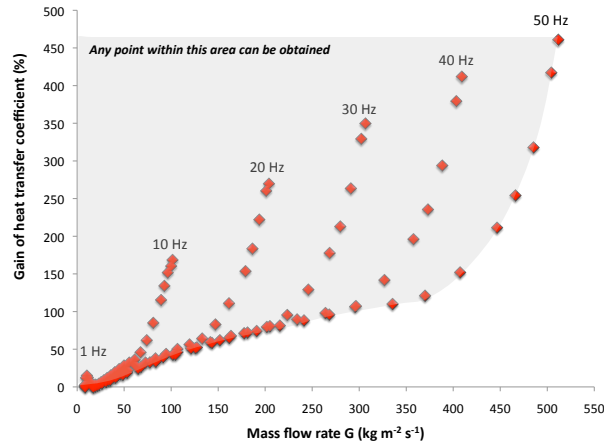


Figure 14: Heat transfer enhancement versus mass flux rate produced by varying the deformation amplitude and frequency for a constant pressure difference. The obtained points are spread in the accessible zone (greyed). For example, an gain of 200% in heat transfer compared to the non-deformed case any mass flux below  $450 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$ . The mass flow rate and the heat transfer coefficient could be imposed independently.

So, according to these various studies, the dynamic deformation of a wall efficiently enhances heat transfer. It also allows integrating the pumping function within reactors and, for some configuration, the mass flow rate and the heat transfer coefficient could be imposed independently. It therefore appears possible to realize a technological breakthrough using such a deformable device.

## 6. Conclusions and potential perspectives

Many ways exist to enhance heat transfer :

- Large gains can be obtained using passive techniques, and developments are still possible, mainly for phase-change heat exchangers.
- Considerable enhancement of heat transfer is reported in the literature for active techniques.
- Among them, the technique using dynamic deformation of a solid wall appears promising. The key features of wall deformation are multiple :
  - First, such wall movement generates local boundary layer perturbations and strongly modifies temperature and velocity fields in the channel. A strong amplification of wall heat transfer is thus obtained.
  - Secondly, the wall deformation produces a pumping effect and the heat exchanger becomes a multifunctional device. This feature allows considerable gain in terms of compactness and direct integration of heat exchange devices especially miniaturized devices. This effect can be used to eliminate the need for an external pump or to

control or enhance the flow rate for a given pressure drop. The flow direction can also be controlled (e.g. reverse flow, alternative discharge of the fluid along different part of the outlet section).

- Third, the wall heat transfer coefficient in such devices is (largely) independent of the flow rate unlike in a classical heat exchanger. Thus either heat flux (respectively fluid or wall temperature) and flow rate can be adjusted individually. This feature allows, for example, the wall temperature and homogeneity to be controlled even with variable temperature or inlet flow rate. Reciprocally, the flow rate could be fixed and the wall temperature (or outlet fluid temperature) could be modulated. This devices could also possibly be used to control residence time (or thermal history) of reactant by varying local velocity or trajectory of the fluid. The same approach could be used to control and enhance mixing.

Another aspect is that heat transfer enhancement studies, in particular those involving phase change process, are often performed empirically. Furthermore, experimental results are often scattered even under similar configurations and physical mechanisms are not always well described. Thus it is not possible to obtain reliable correlations of heat transfer coefficient. More works about basic physical mechanisms involved in these different enhancement techniques are so needed.

### **Acknowledgements**

Financial support from CNRS Energie CITAMPE PR09-3.1.3-2 and FNRAE SYRTIPE are gratefully acknowledged.