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Experimental Investigation of Forced Convection Enhancement by Acoustic Resonance Excitations in Turbulated Heat Exchangers

The present research deals with enhancing the thermal performance of turbulated heat exchangers through the application of sound pressure waves at acoustic resonance frequencies. Extending the findings of prior wind tunnel studies, where a standing wave greatly improved the forced convection in reattaching flows, this paper exploits such a phenomenon in a practical heat exchanger setting. The current experiments are conducted in representative turbulated plate and double-pipe heat exchanger geometries, mounted in a dedicated facility. After identifying the inherent acoustic resonance frequencies of the passageways, the impact of excitation is studied in various sound pressure levels, blockage ratios, as well as Strouhal and Reynolds numbers. The acoustic resonance excitation resulted in heat transfer enhancement of 20% and 10% in the plate and double-pipe designs, respectively, absence of additional pressure penalties. To the best knowledge of the authors, this is the first demonstration of acoustic forced convection enhancement in turbulated heat exchanger geometries. Such a technology can pave the way toward future designs that require low-pressure losses, minimal form factor, and/or process controllability. [DOI: 10.1115/1.4045659]

Keywords: experimental heat transfer, forced convection, heat exchangers, heat transfer enhancement, aero-thermal flow control, acoustic resonances

Introduction

In the modern technological cycles, many components require heat to be either added or dissipated toward maintaining their operability and enhancing their overall thermodynamic efficiency. This change in gas or liquid temperature is typically achieved in a heat exchanger, which operates by associating two streams of different thermal potential. Due to form factor limitations of many size restrained applications, the state of the art is advancing toward more compact designs. This forms the basis toward higher performance and efficiency heat exchangers, which enable more heat transfer for the same size unit. This can be achieved by means of designs having sets of various perturbators in the heat exchange passageways (i.e., ribs, pins, fins), providing larger heat transfer surface area per unit of volume and promoting turbulence in the flow. Among them, the technologies that provide the same heat transfer at lower pressure penalty are considered to be superior.

In order to further promote heat transfer, active flow control methods can be used to create aero-thermal flow modulations. The majority of prior literature focuses on local excitation via miniaturized mechanical devices, which is not practical for the confined passages of compact heat exchangers. Addressing this issue, acoustic excitations have prior been considered mostly in the scope of the attached flow heat transfer.

The first experimental effort to determine whether acoustic perturbations imposed on a gaseous medium would have an appreciable effect on heat transfer in a smooth pipe has been realized by Jackson et al. [1]. Soon after, Westervelt [2] hypothesized that the effect of sound waves on heat transfer is mostly due to the modification of the inner streaming boundary layer, known to occur when the sound particle displacement amplitude is larger than the acoustic boundary layer thickness. This theory was experimentally validated by Holman [3]. Although the acoustic streaming phenomenon is well documented and significant efforts were invested to understand its underlying physics, there was only a limited number of works that focused on the application-relevant environment. Lemlich and Hwu [4] were the first to conduct a study of the acoustic excitation effect on forced convection in a simple double pipe heat exchanger. In addition to increasing with the acoustic amplitude, the enhancement was most prominent when the flow was excited at the channel acoustic resonance frequencies. More recently, Yao et al. [5] presented a 17% heat transfer enhancement in water-to-water smooth shell-and-tube heat exchanger under ultrasonic excitation. However, the thermal boundary layer streaming is only relevant in smooth wall surfaces, and the potential of aero-acoustic coupling has not been explored in turbulated heat exchangers, where periodic flow reattachment occurs.

Considering more fundamental geometries, there exists a modest amount of scientific effort addressing the aero-thermal impact of acoustic excitations. For example, in backward-facing step geometries, sound waves reduced the reattachment length (by around 20%) and were reported to yield increased heat transfer (by up to 30%) in the step wake region [6]. However, studies on acoustic flow control in ribbed flow geometry are extremely scarce. The only known relevant investigations were purely aerodynamic [7,8] and reflected a reduction of the turbulent reattachment length by up to 50%. Although this geometric configuration received much less attention from the scientific community, it is more representative of the aero-thermal conditions encountered in real heat exchange applications.

More recently, the Technion research group studied the convective heat transfer ramifications of sound excitation over an isolated

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rib obstacle in a wind tunnel [9]. The excitations in-tune with the passage acoustic resonance frequencies resulted in the generation of a standing pressure wave, which in a conducive Strouhal regime (0.17–0.22) promoted reattachment length reduction of up to \sim 35% and local heat transfer enhancement of \sim 25%. In the meanwhile, the integral static pressure drop remained unchanged. Moreover, a minimum source amplitude threshold was found (\sim 121 dB); above this level, the aero-thermal effectiveness of the forcing rises monotonously with increased sound pressure. Conversely, when the flow was excited with acoustic frequencies that do not correspond to resonances (traveling wave forcing), the local heat transfer distributions remained unchanged.

Motivation. Building upon these promising findings in wind tunnel testing of sound excited turbulators, the current effort focuses on the experimental demonstration of acoustic resonance heat transfer enhancement in realistic heat exchangers. The study is conducted in a dedicated facility using exemplary air-to-air heat exchangers in the plate and double-pipe configurations. According to the best knowledge of the authors, this is the first experimental demonstration showcasing the potential of acoustic resonance excitation to increase convection in turbulated heat exchangers, in absence of any additional pressure penalty. The dependency of the phenomena to sound pressure level (SPL), blockage ratio, and Strouhal and Reynolds numbers is demonstrated. The beneficial impact is observed over a broad parameter space. The outcomes of this work are intended to provide a basis for the future development of advanced compact heat exchangers.

Experimental Methodology

Heat Exchanger Test Facility. A dedicated experimental facility is designed to include a modular heat exchanger test bed and two separate air supply lines for the hot and cold sides (Fig. 1). The streams are provided by separate compressors, which are capable to deliver a mass flow rate of up to 110 kg/h each. The supply of both lines is adjusted by Samson electric valve, which is connected to a Micro Motion Coriolis mass flow meter and controlled via TROVIS 6493 compact controller. The hot stream is treated by a 24 kW OSRAM Sylvania inline electric heater, which can bring the air in the test section to temperatures of up to 150 °C. In order to reduce the heat losses to the atmosphere, the test section and the hot airline are covered by a 25 mm thick Armaflex XG

insulation sheet, which has a thermal conductivity (λ) of 0.036 W/m K and is resistant up to 150 °C.

The temperatures of both streams are measured using S + SREGELTECHNIK 4-wire PT1000 resistance temperature detectors (RTDs), mounted in the center of test section supply and exhaust manifolds in a way that ensures measurement of mixed flow quantity. Pressure drops in the cold and hot channels of the tested geometry are measured by Huba Control differential pressure sensors. Separate units with full-scale value of 2.5 kPa and 30 kPa are used for lower and higher mass flow rates, respectively. Honeycombs are placed in the exhaust lines to ensure flow uniformity and better resolve the downstream pressures and temperatures. The acoustic excitation is generated separately in the outlet port of each stream, using two Mackie DLM8 2000 W powered loudspeakers connected to a function generator. The resultant SPLs are recorded using G.R.A.S high-precision condenser microphone and high-temperature pre-amplifier.

The accuracies and measurement ranges of all devices used during facility operation are summarized in Table 1. All experimental values are sampled, stored, and processed using a dedicated National Instruments data acquisition system.

Tested Plate and Double-Pipe Heat Exchangers. To demonstrate the potential acoustic enhancement, a common plate heat exchanger (PHE) and double-pipe heat exchanger (DPHE) geometries are considered in the scope of the current work. Both designs are assembled in counterflow configuration. For low-pressure drop heat exchangers, the rib blockage ratio (BR) is typically below 0.15; similar values are implemented in the scope of this work. The optimal pitch-to-height ratio that maximizes heat transfer is typically around 6, which is also adapted in the current study. It is reasonable to expect that the rib height should be higher than the

Table 1 Range and accuracy of measurement devices

Measurement device	Range	Accuracy
Coriolis mass flow meter 4-wire PT1000 RTD	10–110 kg/h –35–180 °C	$\pm 0.75\%$ of measured value According to 1/10 DIN
Differential pressure sensors	0–2.5 kPa 0–30 kPa	$\pm 0.75\%$ of full scale $\pm 0.5\%$ of full scale
Microphone	4 Hz–100 kHz 30–168 dB	±0.08 dB



Fig. 1 Schematic description of the experimental facility

viscous and the buffer sublayers in the turbulent regime. In order to confirm that the rib height is confidently above the linear sublayer [10], the following constrains can be considered:

$$r_h \ge 150 \,\nu/\nu \tag{1}$$

where ν is the kinematic viscosity, ν is the free-stream velocity, and r_h is the rib height. This dictates a value of more than 0.2 mm in the current configuration. Nevertheless, the available manufacturing tools limited the rib height to a minimum of 1 mm and the plate thickness to a minimum of 3 mm. In order to reduce the weight, and the effect of thermal inertia on measured values, aluminum is selected as the raw material due to its density and machinability. In addition, the selection of conducive acoustic excitation frequency stems from previous wind tunnel findings [9]. Therefore, the Strouhal number is selected to be in the range of 0.1–0.25, which creates constraints on mass flow rate and geometry that prescribes suitable resonance frequencies.

Under these considerations and limited available air supply, PHE parameters are selected, being number of plates, plate width, plate length, plate thickness, plate spacing, rig height, and pitch-to-height ratio. Similarly, the design parameters of DPHE are pipe length, inner pipe radius and thickness, outer pipe radius and thickness, rib height and pitch-to-height ratio. The exact parameter values for both heat exchanger types can be found in Table 2 and Table 3 for PHE and DPHE, respectively. An exploded view of the PHE is presented in Fig. 2. The unit has been designed in a way to avoid any brazing or gasketing by using pipe sections to set the distance between plates and enclosing the volume with sidewalls. Moreover, the design can accommodate different numbers of internal plates in order to control the Reynolds number in the heat transfer passages. Similarly, an exploded view of the DPHE is depicted in Fig. 3. After assembly and sealing, the two designs are pressurized and tested for both external and internal (channel-to-channel) leakages.

Facility Validation. To validate that the facility is properly insulated during its operation, the heat balance between the hot and cold sides is measured at steady-state, in absence of acoustic excitation (Fig. 4). As there is no latent load involved in the tested units, the total heat transfer rates (\dot{Q}) for each side are determined by

$$\dot{Q}_{meas,cf} = \dot{M}_{cf}c_p(T_{ex,cf} - T_{su,cf})$$
(2)

$$\dot{Q}_{meas,hf} = \dot{M}_{hf}c_p(T_{ex,hf} - T_{su,hf})$$
(3)

where c_p is the specific heat capacity at constant pressure, \dot{M} are the mass flow rates, and T are the measured temperatures. Specific heat

Table 2	PHE	characteristics
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Number of plates	5/3
Plate width	130 mm
Plate length	445 mm
Plate thickness	3 mm
Plate spacing	8 mm/19 mm
Rib height	1 mm
Pitch-to-height ratio	6

Table 3 DPHE characteristics

Pipe length	1150 mm
Inner pipe radius	23 mm
Inner pipe thickness	3 mm
Outer pipe radius	34.5 mm
Outer pipe thickness	10 mm
Rib height	1.5 mm
Pitch-to-height ratio	6

capacity is determined based on the average of supply and exhaust temperatures in each stream. Considering that the measured heat transfer values of the hot and cold streams are almost equal, the test facility and the insulation are proven to suit the experimental campaign.







Fig. 4 Heat balance of heat exchangers in steady-state

Data Reduction. In each experiment, an equal mass flow rate is imposed in cold and hot streams. Achieving steady-state, the effectiveness is measured using

$$\varepsilon_{meas,hf} = \frac{\dot{Q}_{meas,hf}}{\dot{C}_{min}(T_{su,hf} - T_{su,cf})} \tag{4}$$

$$\varepsilon_{meas,cf} = \frac{\dot{Q}_{meas,cf}}{\dot{C}_{min}(T_{su,hf} - T_{su,cf})}$$
(5)

The value of \dot{C}_{min} is obtained from

$$\dot{C}_{min} = \min(\dot{C}_{cf}; \dot{C}_{hf}) \tag{6}$$

where \dot{C}_{cf} and \dot{C}_{hf} are the heat capacity rates $(c_p \dot{M})$ for the cold and hot flows, respectively.

In addition, the heat exchanger thermal performance is measured as a function of Reynolds number, calculated based on hydraulic diameter. Nusselt number is estimated from averaged heat exchanger effectiveness using the number of transfer units (NTUs) method. In this approach, the effectiveness is

$$\varepsilon_{avg,HE} = \frac{\varepsilon_{meas,hf} + \varepsilon_{meas,cf}}{2} = \frac{1 - \exp[-NTU(1 - w)]}{1 - w \cdot \exp[-NTU(1 - w)]}$$
(7)

where *w* is the ratio of heat capacity rates:

$$w = \frac{\dot{C}_{min}}{\dot{C}_{max}} \tag{8}$$

$$\dot{C}_{max} = \max(\dot{C}_{cf}; \dot{C}_{hf}) \tag{9}$$

NTU can be calculated from Eq. (7) and assuming that the heat transfer area (A_{HE}) is known, the overall heat transfer coefficient (U) can be found from

$$NTU = \frac{UA_{HE}}{\dot{C}_{min}} \tag{10}$$

The product UA_{HE} is inversely proportional to the sum of the heat transfer resistances (*R*):

$$UA_{HE} = \frac{1}{R_{cf} + R_{hf} + R_w} \tag{11}$$

which are determined based on the heat transfer coefficients (*h*) and the thermal conductivity (λ_w) for the plates or the pipe:

$$R_{cf} = \frac{1}{A_{HE} \cdot h_{cf}} \tag{12}$$

$$R_{hf} = \frac{1}{A_{HE} \cdot h_{hf}} \tag{13}$$

$$R_w = \frac{t}{A_{HE} \cdot \lambda_w} \tag{14}$$

In the case of PHE, a symmetric heat exchanger with balanced mass flow rates is considered. In light of small temperature differences across hot and cold streams bounding Prandtl variations, the heat transfer coefficients for both sides are assumed constant $(h_{cf} = h_{hf})$ [11]. For DPHE, since symmetry can not be assumed, the values of heat transfer coefficient are calculated from the experimentally determined surface temperature at the cold supply side (RTD location defined in Fig. 3) using the thermal-electrical analogy:

$$\frac{(T_w - T_{su,cf})}{R_{cf}} = \frac{(T_{ex,hf} - T_{su,cf})}{R_{cf} + R_{hf} + R_w}$$
(15)

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Now, the Nusselt number can be determined from

$$Nu = \frac{hD_h}{\lambda_a}$$
(16)

where D_h is the hydraulic diameter of the channel. In order to correlate it to Reynolds number, power-law approach is employed [12]:

$$Nu = a \cdot Re^b \cdot Pr^c \tag{17}$$

where Re is the Reynolds number and Pr is the Prandtl number. In this formulation, the coefficient c is set to 1/3, while a and b are identified from experimental data for a range of working conditions. The influence of the temperature-dependent viscosity is neglected.

In order to determine the hydraulic performance, tests are conducted in ambient temperature to reach a faster steady-state. The friction factor (f) is then determined as a function of the measured pressure drop (ΔP), the air density (ρ), the mean channel length (L), the hydraulic diameter (D_h), and the mean velocity (v), using the approach described in Ref. [13]:

$$\Delta P = \frac{f\rho L v^2}{2D_h} \tag{18}$$

The dependency of the friction factor on the Reynolds number can be described using a common correlation [14]:

$$f = a \cdot \mathrm{Re}^{-b} \tag{19}$$

Depending on the geometry, the a and b coefficients are identified from experimental data.

Uncertainty Analysis. The uncertainties of the reported findings are evaluated from the known sensor accuracies using a method described in Ref. [15]. The purpose of this technique is to calculate how the uncertainty in each of the directly measured variables $x_1, x_2, ..., x_n$ propagates into the value of the calculated quantity $y = f(x_1, x_2, ..., x_n)$. According to this method, the uncertainty of the calculated quantity (U_y) can be determined as a function of the measured variables' uncertainties $(U_{x,i})$ via

$$U_{y} = \sqrt{\sum_{i=1}^{n} \left(\frac{\partial y}{\partial x_{i}}\right)^{2} \cdot U_{x,i}^{2}}$$
(20)

The errors in each variable are summarized in Table 4, within a 95% confidence interval. Moreover, every presented chart contains error bars; however, the typical values are too small to be visualized within the data markers.

Experimental Results

After obtaining a proper thermal balance, the unexcited reference thermal and hydraulic performances of each heat exchanger are measured in a series of tests. In the following, a parametric experimental analysis is conducted to investigate the influence of

Table 4 Errors in each variable

Variable	Origin	Error
Mass flow rate (\dot{M})	Measured	±0.75%
Heat transfer rate (\dot{Q})	Calculated	±0.96%
Temperature (T)	Measured	±0.15 °C
Pressure drop (ΔP)	Measured	±18.75 Pa
Friction factor (f)	Calculated	±1.3-11%
Reynolds number (Re)	Calculated	±0.75%
Nusselt number (Nu)	Calculated	$\pm 1.8\%$
Effectiveness (ε)	Calculated	$\pm 0.6\%$
Heat transfer enhancement (Nu/Nu ₀)	Calculated	±2.5%

sound pressure level, blockage ratio, Strouhal number and Reynolds number, indicative of the flow regime.

the same friction factor and heat transfer coefficient. This yields the following friction factor correlation:

Unexcited Performance. The thermal performance of the tested PHE is investigated in a mass flow range of 15–95 kg/h, which corresponds to an open channel velocity range of 1.7–12 m/s with the atmospheric exhaust pressures. This results in Reynolds variation from 1300 to 10,500. Due to the heat balance of the facility, the hot and cold stream effectiveness is very close; averaging the two values, the mean is charted in Fig. 5. A linear dependency on the mass flow rate in each stream can be observed.

The aero-thermal performance is characterized using least-squares fit, as

$$Nu_{PHE} = 0.09 Re^{0.83} Pr^{1/3}$$
(21)

The obtained relationship is compared with Nusselt evolution in smooth rectangular ducts of similar dimensions by implementing the correlations from Ref. [11] (Fig. 6). Expectedly, the addition of ribs promotes turbulence and increases heat transfer by up to 500%.

In the following, the pressure drop for both channels is measured (Fig. 7). The closeness of the values justifies the symmetry assumption of PHE design and validates the notion that both streams have



Fig. 5 PHE hot flow, cold flow, and average effectiveness as a function of mass flow rate, in absence of acoustic excitation



Fig. 6 Comparison of Nusselt number as a function of Reynolds number in PHE and in a smooth rectangular duct, in absence of acoustic excitation

 $f_{PHE} = 20.8 \text{Re}^{-0.24} \tag{22}$

depicted in Fig. 8. Contrasting with a similar size smooth duct [11], the surface roughening resulted in an order of magnitude friction factor increase.

Acoustically Excited Performance. In order to investigate the impact of acoustic resonance on the heat exchanger performance, both cold and hot streams are excited by a sound that creates standing waves within the confines of the passage. The exact acoustic resonances of the heat exchanger (fundamental and harmonics) are identified by noting peak values in microphone measurements for a broad range of excitation frequencies. In order to ensure that the structural integrity of the heat exchanger is maintained in the process, the values are confirmed to not correspond to the structural (mechanical) modes of the setup via accelerometer measurements.

For the 325 Hz resonance, when the mass flow rate and the Reynolds number are kept constant at 11.5 kg/h and 1300, respectively, Strouhal number of 0.2 can be achieved. With the constant supply of cold and hot streams at 30.3 °C and 128.2 °C, respectively, the SPL is incrementally varied from 123 to 139 dB, reaching steady-state at each step. The temperature evolution at the supply and exhaust ports of both streams is depicted in Fig. 9.



Fig. 7 PHE pressure drop as a function of mass flow rate, in absence of acoustic excitation



Fig. 8 Comparison of friction factor as a function of Reynolds number in PHE and in a smooth rectangular duct, in absence of acoustic excitation



Fig. 9 Supply and exhaust temperature evolution in time under increasing SPL at constant St = 0.2



Fig. 10 Comparison of pressure drop with and without acoustic resonance excitation of 325 Hz and 135 dB as a function of mass flow rate

As the exhaust temperatures of both streams change with increasing SPL, the heat exchanger effectiveness is further enhanced.

In addition to measuring the heat transfer enhancement associated with the acoustic excitation, the influence of the forcing on the heat exchanger pressure penalty is also of significant interest. Thus, pressure drop evolution is measured over a range of mass flow rates for both excited and non-excited flows (Fig. 10). For the 135 dB standing wave at a frequency of 325 Hz, no additional pressure drop could be observed within the accuracy of the differential pressure sensor.

Correlating the thermal performance of 325 Hz excitation (St = 0.2) in terms of Nusselt number (Fig. 11), a linear dependency is observed between the SPL and the heat transfer enhancement, reaching the highest measured value of ~1.2 at 139 dB. The unexcited Nusselt number is ~30 (Fig. 6). In the following, by reducing the excitation frequency to a lower acoustic resonance, the Strouhal number is adjusted to 0.07. It can be seen that the enhancement during the reduced Strouhal excitation is smaller across all SPLs. However, the general trends remain the same. This is expected, as the lower Strouhal number is no longer within the conducive range identified in the wind tunnel experiments of isolated turbulators [9].

In order to quantify the effect of blockage ratio on the benchmark St = 0.2, the number of plates is reduced from 5 to 3 for the same mass flow rate. This maintains the same flow velocity within the passage, while decreasing the blockage ratio from 12.5% to 5.4% (Fig. 12). Although the hydraulic diameter-based Reynolds



Fig. 11 Heat transfer enhancement as a function of SPL for St = 0.07 and St = 0.2 at a constant Re = 1300



Fig. 12 Heat transfer enhancement as a function of SPL for BR = 12.5% and BR = 5.4% at St = 0.2

number increases from 1300 to 3000, the inertial effects on the rib (Reynolds number based on rib height) remain the same. Therefore, the dominant mechanism is the drop in blockage ratio, which in turn results in reduced acoustic enhancement of heat transfer across all SPLs. This is mostly associated with the rib contributing less to the aggregate heat exchange, and therefore, the local enhancement around its vicinity has a smaller impact globally.

Double-Pipe Heat Exchanger

Unexcited Performance. Analyzing a second geometry, the DPHE is mounted on the test bench. Its thermal performance is investigated in a mass flow range of 15–95 kg/h, corresponding to internal and external pipe velocity ranges of 2.9–18 m/s and 2.6–15 m/s, respectively. Due to mass flow limitations, the internal and external pipe data spans across Reynolds ranges of 5000–34,000 and 1800–16,000, respectively. The resulting average effectiveness is presented as a function of the mass flow rate in Fig. 13. In the following, the Nusselt–Reynolds correlations are obtained and contrasted to a turbulent flow in a smooth pipe [16] (Fig. 14). Based on the least-squares method, the correlations are found to be

$$Nu_{DPHE,internal} = 0.03 Re^{0.84} Pr^{1/3}$$
(23)

$$Nu_{DPHE,external} = 0.11 Re^{0.81} Pr^{1/3}$$
 (24)



Fig. 13 DPHE hot flow, cold flow, and average effectiveness as a function of mass flow rate, in absence of acoustic excitation



Fig. 14 Comparison of Nusselt number as a function of Reynolds number in internal and external passages of DPHE and in a smooth pipe, in absence of acoustic excitation

As expected, the ribs greatly promoted heat transfer in both heat exchange passages. In the following, the pressure drop in the two pipes is correlated across the mass flow range in terms of frictions factor and contrasted against a smooth pipe [17] (Fig. 15). The f-Re correlations are found to be

 $f_{DPHE,internal} = 3.9 \text{Re}^{-0.16}$

$$f_{DPHE,external} = 13.1 \text{Re}^{-0.35}$$
 (26)

Acoustically Excited Performance. When conducting heat transfer measurements in DPHE, the Reynolds numbers are initially kept constant at 5100 and 2000 for internal and external pipes, respectively. The first resonance frequencies are identified as 258 Hz and 205 Hz in internal and external streams, respectively, both corresponding to a Strouhal number of 0.14. In this configuration, the assumption of equal heat transfer coefficient in both channels is no longer valid. Thus, only one of the heat exchanger sides is acoustically excited at any given time. In this way, the heat transfer coefficient in the unexcited channel can still be estimated from the correlations obtained in Fig. 14, while h in the excited passage is calculated using Eqs. (12)–(14).

First, in order to highlight the effect of acoustic excitation directly on temperature, the temporal evolution of the acoustically excited internal hot flow exhaust is presented in Fig. 16. Changes in the temperature can be observed when the flow is stimulated at various SPLs at the resonance frequency, suggesting that a higher



Fig. 15 Comparison of friction factor as a function of Reynolds number in internal and external passages of DPHE and in a smooth pipe, in absence of acoustic excitation



Fig. 16 Internal hot flow exhaust temperature temporal evolution under various SPLs for St = 0.14 and Re = 5100

amount of heat passes into the cold stream—increasing the heat exchanger thermal performance. However, the enhancement of the cold flow stream can not be directly attributed to the acoustic excitation, but rather is a manifestation of the changing thermal balance. In the following, the pressure penalty is considered for acoustic resonance excitations of internal and external flows at 258 Hz with 139 dB and at 205 Hz with 133 dB, respectively (Fig. 17). Evidently, there is no observable effect of the acoustic excitation on pressure drop.

Summarizing the findings in terms of Nusselt number, Figs. 18 and 19 depict the heat transfer as a function of SPL for the internal and external pipes, respectively. In the maximum attainable SPL of 139 dB and 133 dB in each passage, clear heat transfer enhancement of up to ~10% from the unexcited value of ~60 (Fig. 14) is observed. With a reduction in SPL, the enhancement seems to drop linearly until a minimum threshold is reached. Below this value, there is no noticeable impact on heat transfer. When the Strouhal number is varied by changing the resonance excitation frequency, similar trends are observed. However, the minimum activation threshold decreases if the channel is excited in a more conducive frequency. As indicated in Fig. 19, this value appears to be in the range of $0.14 \le \text{St} \le 0.2$. This is also consistent with the previous wind tunnel experiments over a single rib [9].

Concerning the influence of Reynolds number on the external pipe performance, no acoustic enhancement was observed for values beyond 2400. In addition, due to current test setup limitations, it is impossible to accurately measure the aero-thermal

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Fig. 17 Comparison of pressure drop with and without acoustic resonance excitation of internal and external flows at 258 Hz with 139 dB and at 205 Hz with 133 dB, respectively, as a function of mass flow rate



Fig. 18 Heat transfer enhancement as a function of SPL for different Strouhal numbers at a constant Re = 5100 in the internal pipe



Fig. 19 Heat transfer enhancement as a function of SPL for different Strouhal numbers at a constant Re = 2000 in the external pipe

performance for Reynolds numbers lower than 2000, where heat transfer enhancement is observed. Therefore, although a Reynolds dependency clearly exists, it is not fully characterized. This is corroborated by the measurements in the internal pipe for two different Reynolds numbers at changing resonance excitation frequencies



Fig. 20 Heat transfer enhancement in the internal pipe as a function of SPL for two Strouhal-Reynolds numbers combinations

(Strouhal numbers; Fig. 20). The increase in Reynolds number for the same SPL resulted in a decrease in heat transfer enhancement; this is expected from the increased dominance of the momentum terms.

Summary and Conclusions

Concerning the influence of Reynolds number on the external pipe performance, no acoustic enhancement was observed for values beyond 2400. In addition, due to current test setup limitations, it is impossible to accurately measure the aero-thermal performance for Reynolds numbers lower than 2000, where heat transfer enhancement is observed. Therefore, although a Reynolds dependency clearly exists, it is not fully characterized. This is corroborated by the measurements in the internal pipe for two different Reynolds numbers at changing resonance excitation frequencies (Strouhal numbers; Fig. 20). The increase in Reynolds number for the same SPL resulted in a decrease of heat transfer enhancement; this is expected from the increased dominance of the momentum terms.

The work is conducted in a dedicated heat exchanger research facility over two representative test cases. Common in various applications, plate, and double-pipe heat exchanger units are considered. The heat transfer enhancement due to the acoustic resonance excitation is observed for both geometries, in absence of any additional pressure penalty in the passages. Moreover, the dependencies to sound pressure level, resonance excitation frequency (Strouhal number), blockage ratio, and Reynolds number are demonstrated.

Although an active acoustic excitation source is employed in the experimental campaign, it is possible to create the same effect by tuning inline passive generators (horn, whistle, Helmholtz type, etc.) to the acoustic resonances of the heat exchanger, eliminating the need for any electrical input. Moreover, depending on the size of the heat exchange passage, the resonance excitation is not necessarily in the audible frequency range.

Overall, the findings of this study prove that acoustic resonance enhancement of forced convection in heat exchangers is possible. Thus, the outcomes of this work can be used in the development of next-generation heat exchanger designs that could be particularly optimized to benefit from this phenomenon. This is particularly relevant toward enhancing heat transfer in various common applications that require low-pressure losses, minimal form factor, and/or process controllability.

Information Contained in Figures

- Figure 1—Schematic representation of the experimental setup.
- Figure 2—Exploded view of tested PHE.

- Figure 3-Exploded view of tested DPHE.
- Figure 4—Validation of thermal balance in the setup.
- Figure 5-Effectiveness of PHE in tested mass flow range.
- Figure 6—Unexcited PHE thermal performance.
- Figure 7—Validation of PHE symmetry assumption via pressure drop measurements.
- Figure 8—Unexcited PHE hydraulic performance.
- Figures 9 and 10—Impact of acoustic resonance excitation in PHE.
- Figures 11 and 12—Heat transfer enhancement in PHE as a function of SPL in different regimes.
- Figure 13—Effectiveness of DPHE in tested mass flow range.
- Figure 14—Unexcited DPHE thermal performance.
- Figure 15—Unexcited DPHE hydraulic performance.
- Figures 16 and 17—Impact of acoustic resonance excitation in DPHE.
- Figures 18–20—Heat transfer enhancement in DPHE as a function of SPL in different regimes.

Lessons Learned

- (a) Through acoustic resonance excitations, heat transfer in the plate and double-pipe heat exchangers is enhanced by up to 20% (Fig. 11) and 10% (Fig. 18), respectively, without additional pressure penalties (Figs. 10 and 17).
- (b) Increase in the sound pressure level results in further enhancement of heat transfer with a linear dependency (Figs. 11 and 18).
- (c) The most conducive Strouhal number regime exists in the range of $0.14 \le \text{St} \le 0.2$ (Figs. 11, 18, and 19).
- (d) The minimal sound pressure level necessary for acoustic heat transfer enhancement depends on the Strouhal number (Fig. 19).
- (e) Reduction in blockage ratio diminishes the heat transfer benefit of acoustic resonance excitation (Fig. 12).
- (f) Increasing the Reynolds number decreases the heat transfer enhancement for the same sound pressure level (Fig. 20).

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Nomenclature

Symbols

- f = friction factor
- h = heat transfer coefficient (W/m²K)
- t =plate or pipe thickness
- v = free-stream velocity (m/s)
- w = heat capacity ratio
- y = calculated quantity
- L = channel length (m)
- R = heat transfer resistance (W/K)
- $T = \text{temperature } (^{\circ}\text{C})$
- \dot{C} = heat capacity rate (W/K)
- \dot{M} = mass flow rate (kg/s)
- Q = heat transfer rate (W)

- c_p = specific heat capacity at constant pressure (J/kgK)
- r_h = rib height (m) x_i = measured variable
- heat transfer are-
- A_{HE} = heat transfer area (m²) D_h = hydraulic diameter (m)
- $U_i =$ uncertainty
- $N_u = Nusselt number$
- Pr = Prandtl number
- Re = Reynolds number based on hydraulic diameter
- St = Strouhal number based on rib height
- BR = rib blockage ratio
- ΔP = pressure drop (Pa)
- $\varepsilon = \text{effectiveness}$
- λ = thermal conductivity (W/mK)
- ν = kinematic viscosity (m²/s)
- $\rho = \text{density (kg/m^3)}$

Subscripts

- 0 = unexcited value
- a = value in air
- cf = cold flow
- ex = heat exchanger channel exhaust
- hf = hot flow
- meas = measured value
 - su = heat exchanger channel supply
 - w = value at wall

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