Asociación Argentina
de Mecánica Computacional

Mecánica Computacional Vol XXXIV, págs. 1891-1905 (artículo completo)
Sebastián Giusti, Martín Pucheta y Mario Storti (Eds.)
Córdoba, 8-11 Noviembre 2016

FLUID FLOW AND HEAT TRANSFER NUMERICAL PREDICTION OF CROSS FLOW HEAT EXCHANGER

Hugo D. Pasinato; Jonathan J. Dorella; Alejandro R. Gorosito and Hernán M. Solier Z.

Dpto. Ing. Electromecánica, FRP-Universidad Tecnológica Nacional, Avda. Almafuerte 1033, 3100 Paraná, Argentina; hpasinato@frp.utn.edu.ar

Keywords: Staggered Tube-Bundle; Turbulent Heat Transfer; RANS, URANS, DES, LES

Abstract.

Heat exchangers design is a major challenge problem in the process industry today. Heat exchanger applies to all equipment used to transfer thermal energy between two streams. Namely equipments in which two process streams exchange heat with each other. One of these equipments -cross flow heat exchanger-, used in numerous industry processes, uses as heat exchanger a normal flow to arrays of circular tubes arranged in staggered or in-line formation. These configurations, for instance, are typical in heat exchanger designs used in fossil-fuel and nuclear power plants and, indeed, over many other sectors of thermal and process engineering. Cross flow heat exchangers are designed based on experimental correlations. Depending on which correlation is used -and on the Reynolds number- , however, the heat exchanger area can vary from 30 to 50 percentage. According to the turbulence modeling technique and other details of the simulation, numerical prediction can be a useful tool for improving designs accuracy, in addition to gives more information about the physics of temperature and velocity field. This study presents the computational modeling of flow and thermal field in a cross flow heat exchanger, with staggered formation, and longitudinal and lateral pitch equal to 2, for Reynolds number about 40,000 and Prandtl number equal to 0,744. The simulation uses a non-periodic computational domain in the longitudinal direction with periodic domains in transverse and along the tube directions. Furthermore, a second computational domain periodic in x,y,z, is used for comparison. Four turbulence modeling techniques are examined: Reynolds-averaged Navier-Stokes (RANS), unsteady Reynolds-averaged Navier-Stokes(URANS), Detached Eddy Simulation(DES), and a very Large Eddy Simulation(vLES) -in addition to simulations without any turbulence model(WTM)-. On the other hand, RANS and URANS techniques are examined in 2D and 3D. Results from this study show that 3D RANS is superior to 2D RANS, that vLES is the best simulation from a physical point of view, and DES presents the best agreement with experimental data.

1 INTRODUCTION

As stated by technical and scientific literature the three major challenges in the process industry today are the heat exchanger design, the heat transport and the heat storage. Heat exchanger applies to all equipment used to transfer thermal energy between two streams. One of these equipment, called heat exchanger of tube bundle or tube banks, uses as heat exchanger a normal flow to arrays of circular tubes arranged in staggered or in-line formation. Tube banks heat exchangers are used in a great variety of industrial, thermal, commercial and household applications. In some of these applications the efficiency of heat exchangers are essential.

For example, in recent years, utilizing low-grade waste heat for energy production has attracted attention (waste heat being released into the environment, such as exhaust gases from turbines and engines and waste heat from industrial plants) for its potential in reducing the fossil fuel consumption. However, although renewable energy sources such as solar thermal and geothermal, and vast amounts of industrial waste heat, are potentially promising energy sources, the moderate temperature from these sources cannot be converted efficiently to electrical power; unless the different power generation processes be improved, especially the transfer of heat from the source.

The design of industrial heat exchangers are based on experience and correlations, which are determined by experiments. These correlations are valuable tools in the design process but they have its limitations. Heat exchanger have multiple design and thus multiple parameters, and correlations with many parameters are often not very accurate. A recent comparative study on these correlations has revealed errors up to 50% in the mean Nusselt number, which means differences in the design area also up to 50% (Pysmennyy et al., 2010).

Numerous experimental study (some purely thermal performance studies of cross-flow tube banks) have been carried out (Zukauskas, 1972; Zukauskas and Ulinskas, 1988; Tsunoda et al., 1996; Simonin and Barcouda, 1986), as also numerical studies (Benhamadouche and Laurence, 2003; West et al., 2015; Johnson, 2008; Afgan, 2007), on heat transfer in cross tube banks. Most experimental studies have examined the dependence of the mean Nusselt number (Nu_{av}) from the geometric arrangement and the Reynolds(Re) and Prandtl(Pr) number, and numerous correlations has been proposed (Pysmennyy et al., 2010).

The following references are related with the present study. Johnson (2008) presented the 2D numerical solution, using RANS and URANS simulations, of the staggered tube bundle measurement of Simonin and Barcouda (1986). His predictions are in reasonable agreement. He claims that a 2D simulation is enough in comparison with 3D URANS and LES simulations.

West et al. (2015) have presented the results of a study on numerical simulation of cross flow with heat transfer on an in-line configuration. They used a small cluster of 2×2 tubes with periodic boundary condition in the three directions. They used RANS, URANS and LES, being the best result from LES, with results from URANS with close agreement with LES. They also investigates the influence of the number of tubes of the cluster in the flow structures.

Afgan (2007) found, in one of his LES of an in-line configuration of tube-bundle with 1.5×1.5, '... a purely asymmetric pattern with the flow as a whole displaying an upward tilt ...'. In other words, one remain question is whether a small cluster of tubes actually generates the same flow structures as in a package of thousands of tubes like those used in practice.

Rollet et al (1999) in other numerical simulations suggested that for LES there is little difference between the results of Smagorinsky and dynamic Smagorinsky models.

Accurate flow and heat transfer modeling involves determining suitable modeling strategies, including the turbulence simulation technique (Reynolds averaged Navier-Stokes equations (RANS), large eddy simulation (LES), detached eddy simulation (DES)), whether the simulation should be unsteady, or steady is enough, whether the simulation should be 3D or 2D is enough, the grid density and the wall treatment, and so on. As regarding steady or unsteadiness, for the Reynolds number simulated in this study, the flow is turbulent -which is 3D and unsteady-, but the mean flow is not steady. Although for engineering porpoise it is necessary to predict the mean values of the transport coefficient at the wall, a steady simulation would not take account of this mean flow unsteady effect.

The main goal of this study is aiming at to test different aspects of numerical prediction of heat transfer of cross-flow heat exchangers. Basically the importance of different aspects of numerical prediction, turbulence techniques, etc. of practical turbulent flow with heat transfer are examined. Much of the previous studies has been oriented to investigate mean values, turbulence and structure of mean flow. Although it is of fundamental importance the prediction of flow field to predict thermal field, this study present a comparison of different simulation techniques, with different grids, and computational domain, but focused on heat transfer.

2 NUMERICAL DETAILS

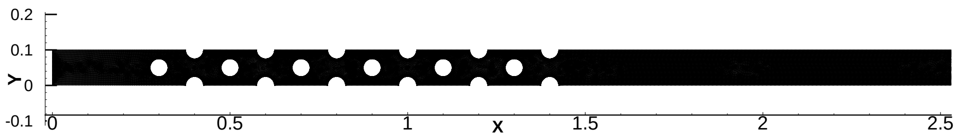
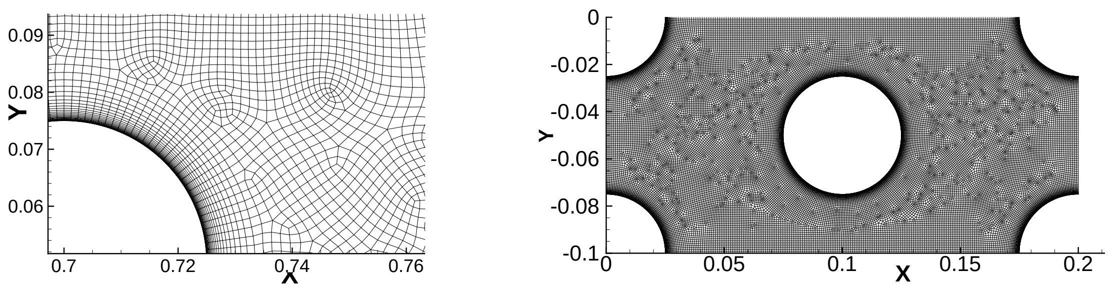


Figure 1: Top view of the full computational domain with $L_x = 51D$, $L_y = 2D$, and $L_z = 2D$, with periodic boundary conditions in y and z.

The tubes in a tube bundle are usually arranged in either in-line or staggered arrangements. In the present study results from a staggered configuration is presented with 12 rows of tubes in the stream-wise direction and two row in the transverse direction. Benhamadouche and Laurence (2003) in a previous numerical study found that a width of two diameters was sufficient in the cross-stream direction to obtain correct vortex shedding characteristics such as fluctuating lift and drag forces and Strouhal number. Also results from a computational domain with 2×2 tubes, with periodic boundary conditions in x,y,z, are presented for comparison (Figure 2(b)).

The definition used by Zukauskas and co-workers is adopted to describes distances between



(a) (b)

Figure 2: (a)Details of the grid distribution near the wall for a very-coarse grid for the 3D full computational domain; (b)Top view of the 3D computational domain used with periodic boundary conditions in x, y, z, with 2×2 tubes.

Table 1: Main features of the numerical grids.

Grid	$NE \times 10^{-3}$	$y_{min} \times 10^{3}$	BL	REF	N_{θ}	N_y	$N_{x,e}$	$N_{x,bt}$	$N_{x,o}$	N_z
CG2D	53	0.050	18	1,24	76	38	120	66	400	–
MG2D	92	0.025	26	1,18	100	50	160	88	500	–
DG2D	130	0.015	28	1,18	132	60	200	116	600	–
CG3D	905	0.050	22	1,24	76	38	120	66	400	24
MG3D	2,650	0.025	26	1,18	100	50	160	88	500	32
DG3D	6,520	0.015	28	1,18	132	60	192	104	600	48
PER3D	3,157	0.015	28	1,15	256	124	—	256	—	80

Table 2: Grids discretization in wall units.

Grid	y_1^+	$(R\Delta\theta)^+$	Δz^+	Δx^+
CG3D	4,00	139	280	100
MG3D	2.00	106	210	75
DG3D	1.00	80	140	64
PER3D	1.00	41	85	25

cylinders, and the domain is based on the experimental set-up of Meyer (1994), with values $a = S_1/D = 2$ and $b = S_2/D = 2$ for the longitudinal and transverse spacings between tubes, respectively. The configuration of the twelfth-row of staggered tubes is shown in Figure 1. And Figure 2(a) presents a near-wall detail of a very-coarse grid with similar parameters of the grids used in the study. The full computational domain with the 12 row of tubes has 51D, 2D, and 2D, in x, y and z-direction, respectively. The entrance region or distance between the inlet and the center of the first tube has 6D, and the distance between the center of the last tube and the output section 23D, where the diameter D of the tubes is 0.05 meters. Also a computational domain with dimension ($2D \times 2D \times 2D$) and periodic boundary condition in x,y,z, respectively, with $a = b = 2$ is used in order to do comparison with the results of the full computational domain, Figure 2(b).

u, v, and w are the instantaneous velocities in the stream-wise(x), transverse(y), and cylinder-direction(z), respectively. The plus symbol is used sometimes to denote dimensionless values using the wall parameters u_τ and ν, where u_τ is evaluated at the maximum skin-friction point around the fourth tube.

The Reynolds number examined, $Re = U_m \rho D/\nu$, is in the $40,000 - 43,000$ range, where U_m is the mean largest velocity between tubes (mean velocity in the cross section between tubes where the mean velocity is largest). The temperature is considered to be a passive scalar; thus, the results are also valid for mass transfer. All results are based on a molecular Prandtl number Pr equal to 0.744, and the physical properties of the fluid ν, ρ, and k are considered constant.

Across the entrance of the computational domain the velocity and temperature profiles are assumed to be uniform with a % of turbulence intensity. To account for the appropriate velocity and temperature profile and turbulence intensity at the entrance would necessitate a separate simulation to generate boundary conditions. Some tests, however, were done showing that solution was not sensitive to profiles and turbulence levels at the entrance. The same has been tested and published by Liang and Papadakis (2007).

Table 3: Turbulence techniques.

Name	Simulation	Dimensions	Computational domain
$\kappa-\epsilon$	steady/unsteady	2D	y periodic, x developing
$\kappa-\epsilon$	steady/unsteady	3D	y,z periodic, x developing
S-A	steady/unsteady	2D	y periodic, x developing
S-A	steady/unsteady	3D	y,z periodic, x developing
WTM	steady/unsteady	2D	y periodic, x developing
WTM	steady/unsteady	3D	y,z periodic, x developing
DES	unsteady	3D	y,z periodic, x developing
vLES	unsteady	3D	y,z periodic, x developing
WTM	unsteady	3D	y,z,x periodic
DES	unsteady	3D	y,z,x periodic
LES	unsteady	3D	y,z,x periodic

In Table 1 the main characteristics of the grids are presented. N_θ are the nods around the tubes, and N_y and N_z are nods in the span-wise and tubes axes direction, respectively, while $N_{x,e}$, $N_{x,bt}$, $N_{x,o}$ are nods at the entrance, between two tubes and at the output region, respectively. BL is the number of elements in the boundary layer around the tube, REF is the radial-expansion factor used from the wall to resolve the near-wall sublayer, and y_{min} is the distance between the center of the first volume and the wall.

Almost all elements of the grids are of hexahedral type. In Table 2 the grid discretization in wall units are presented. The grids are coarse, medium or dense; e.g. CG2D is a coarse grid for a two dimensional simulation. The minimum distance at the wall is in the range 0,000050-0,000015, which means approximately $y^+ \simeq 4$ for coarse, $\simeq 2$ for medium, and $y^+ \simeq 1$ for dense grids (there are differences among the skin friction for different turbulence techniques, therefore also exceptions are for those simulations WTM which presented higher maximum skin friction), evaluated with the friction velocity, evaluated with the maximum shear stress at the wall for the fourth tube row.

In the present study an old version of the commercial code Fluent from Ansys Inc, was used (Ansys, 2010). Table 3 shows the turbulence techniques used, where $\kappa-\epsilon$ refers to the RANS model known in the literature as the standard $\kappa-\epsilon$ with enhanced wall treatment (Launder and Spalding, 1972); S-A refers to the RANS model proposed by Spalart and Allmaras (1992); WTM means a simulation without-turbulence-model; DES refers to the Detached-Eddy Simulation technique, the RANS-LES turbulence technique proposed by Spalart et al. (2006); vLES is a very-Large Eddy Simulation using the Smagorinsky's model proposed by Smagorinsky (1963), with a coarser grid than that required by LES; and LES is a Large Eddy Simulation with a medium grids with the Smagorinsky's model.

WTM deserves some comments. WTM is a simulation which uses the same numerical mesh used by RANS, vLES, LES, etc., but without any modeling of turbulence (a pseudo-DNS). However, since its grid is not enough in order to simulates the energy flux through the different scales of turbulence, it is an imperfect simulation of turbulence. For this reason part of this energy remains trapped in the smaller scales of the grid, hindering the convergence to the stationary solution. For this reason all simulations WTM are unsteady and its results are mean time-averaged values(similar to Reynolds averaged values of a DNS). But, despite of this shortcoming, WTM have been a very valuable tools to judge the quality of those numerical predictions with turbulence models.

Steady and unsteady simulations with the $\kappa - \epsilon$ and S-A models, in 2D and 3D, did not presented differences. Therefore the $\kappa - \epsilon$ and S-A predictions in 2 and 3D presented here belong to steady simulations. The only techniques that has presented a clear convergence with mesh refinement has been vLES(Figure 10). All RANS models, included DES which is a RANS/LES techniques, presented only small improvements with mesh refinement. These improvements were basically in the cylinder rear (at the separation bubble and posterior reattachment behind the tubes). It is thought that such behavior is proper of RANS techniques, but also is the consequence that transport of turbulent flow in tube banks is mainly due to big turbulent scales. Note, however, that near-wall distance in the 3 grids used in the present study are reasonably small. In other words, it seems that for RANS techniques the minimum distance at the wall for the coarse grid is enough for this flow.

3 RESULTS AND DISCUSSION

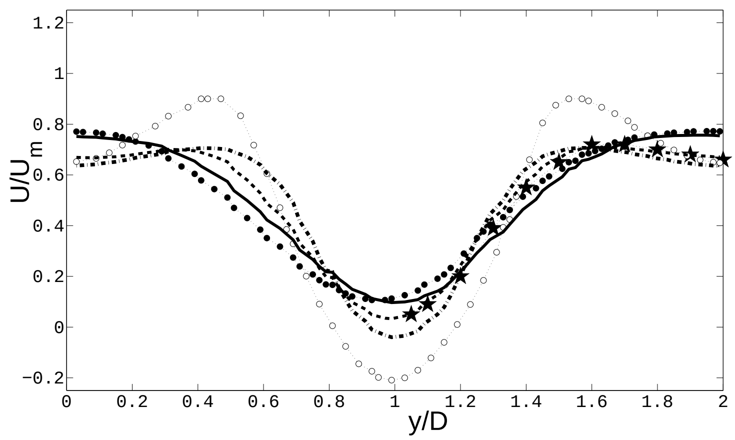


Figure 3: U/U_m at $x/D = 1$ downstream from the center of the 7th row, for different 3D simulations for the full domain, in comparison with experimental data. Solid line, vLES; $----$, DES; $\cdot - \cdot - \cdot$, $\kappa - \epsilon$; $\bullet \cdot \bullet \cdot \bullet$, WTM; $\circ \cdot \circ \cdot \circ$, S-A; $\star \cdot \star \cdot \star$, Meyer (1994).

As it was stated in the abstract, the design of a heat exchanger is a multi-step complex process which can result in area differences of $30 - 50\%$ depending on which correlation is used. It is expected, therefore, that numerical simulations can help in this process. There are, however, two aspects related with numerical prediction of cross flow heat exchanger that deserves be mentioned in advance. First of all, in the last decades many improvement have been achieved in numerical prediction of turbulent fluid flow, but the turbulent heat transfer prediction has not received the same research effort -although turbulence fluid flow modeling is not a solved problem, the modeling of heat transfer is lagging behind-. And second, the turbulent flow around tube bundle is a complex flow which presents many difficulties related with the turbulence modeling, as also many physical phenomena related with longitudinal lateral pitch between tubes, with in-line or staggered formation, and so on. For these reasons the expectation should be reasonable.

In order to develop more knowledge on tube bundles, therefore, the following results show the level of precision and physical sound of 2D in comparison with 3D simulations, of steady with unsteady simulations, and comparisons between different turbulence techniques.

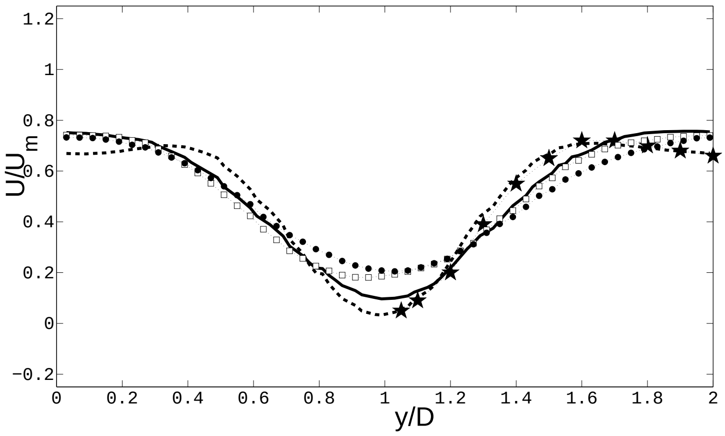


Figure 4: U/U_m comparison of 3D simulations with and without periodic boundary conditions. Solid line, vLES; $- - --$, DES; $\bullet \cdot \bullet \cdot \bullet$, LES(x-periodic); $\square \cdot \square \cdot \square$, DES(x-periodic); $\star \cdot \star \cdot \star$, Meyer (1994).

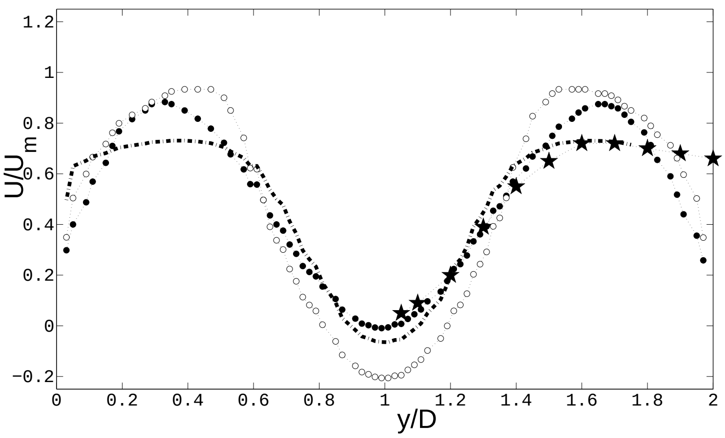


Figure 5: U/U_m comparison with experimental data for 2D simulations. $\cdot - \cdot - \cdot$, $\kappa - \epsilon$; $\bullet \cdot \bullet \cdot \bullet$, WTM; $\circ \cdot \circ \cdot \circ$, S-A; $\star \cdot \star \cdot \star$, Meyer (1994).

In the range from $Re \approx 1000$ and up to the critical $Re \approx 2 \times 10^5$ the flow regime around a cylinder is called sub critical. Staggered tube banks in the sub critical regime is probably the most used flow configuration in heat exchanger. This kind of flow develop a boundary layer in the front of the cylinder which then separates, forms a separation bubble that reattaches to the surface and then separates again.

Since turbulent fluid flow techniques are mature enough, only a few data of the velocity field are presented for comparison. Figures 3, 4, and 5, show the mean longitudinal dimensionless velocity prediction in the middle of 7th and 8th row of tubes for same techniques together with the experimental data of Meyer (1994). Figure 3 shows U/U_m for the 3D full computational domain, for the different turbulence techniques. In this Figure DES presents the best performance, S-A the worst, while WTM prediction has a similar performance to vLES and better than $\kappa - \epsilon$. Also in this Figures is noted that the minimum velocity of WTM is slightly tilted to the left (mean flow is not symmetric). This behavior was noted by West et al. (2015) in their simulations for an in-line tube ban. None of the other techniques showed that asymmetry. Figure 4 presents a comparison with experimental data of vLES and DES using the full domain, with

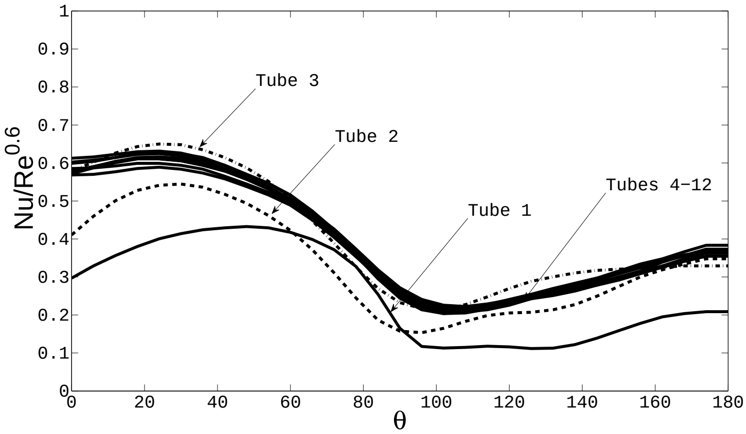


Figure 6: Nu for the 12 tube rows for vLES. Solid line, 1ft-row; $- - --$, 2nd-row; $\cdot - \cdot - \cdot$ 3th-row; solid line; 4-12th rows.

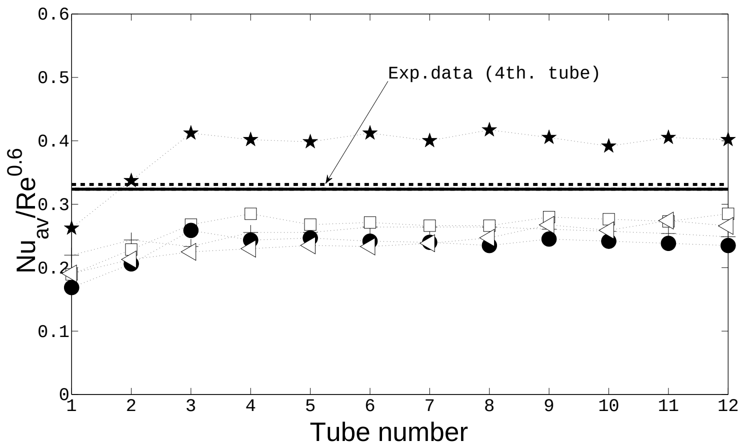


Figure 7: Nu_{av} for the different tubes of the 3D simulations. Solid line, Meyer (1994), Re=34100; $\cdot - \cdot - \cdot$, Meyer (1994), Re=41500; $-.-$, Baughn (1986), Re=34500; $\star \cdot \star \cdot \star$, vLES; $\bullet \cdot \bullet \cdot \bullet$, WTM; $\square \cdot \square \cdot \square$, DES; $+ \cdot + \cdot +$, $\kappa - \epsilon$; $\triangleleft \cdot \triangleleft \cdot \triangleleft$, S-A.

LES and DES for the domain with periodic boundary conditions in x,y,z. Although all these predictions are acceptable, DES using the full computational domain show the best agreement. Since a denser grid is used in the periodic domain, it is thought that the lower performance of DES with this denser grid can be a consequence of the computational domain size or that its LES modeling is not efficient enough (Note that a denser grid with DES means that the RANS region is thinner). If the computational domain size is the problem, probably a larger computational domain in x should be used with periodic boundary condition for tube bundle for $a = b = 2$. This was not tested in the present study.

Figure 5 shows a comparison for 2D prediction of longitudinal mean velocity with experimental data. WTM and S-A show a poor performance, but $\kappa - \epsilon$ prediction is reasonable. As discussed in the literature and verified in this study, tube-banks is a turbulent flow where momentum and heat seem to be transported mainly by big scales in the x-y plane. For example, the numerical refinement of meshes away from the wall does not improve the prediction of the mean velocity or heat transfer for 2D or 3D RANS techniques. This seems to be the consequence that

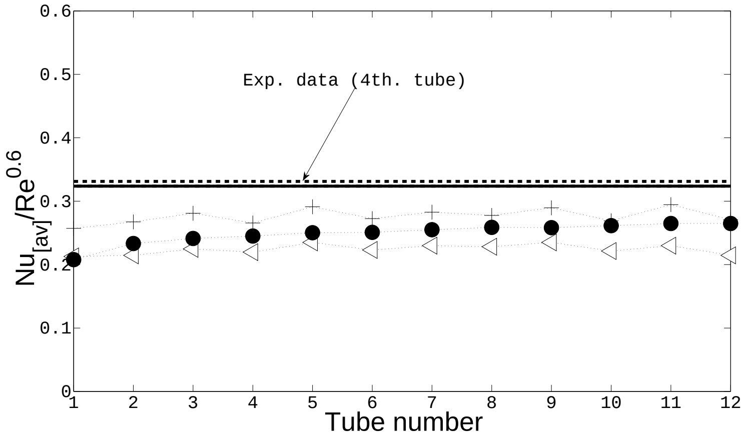


Figure 8: Nu_{av} for the different tubes of the 2D simulations. Solid line, Meyer (1994), Re=34100; $\cdot - \cdot - \cdot$, Meyer (1994), Re=41500; $-.-$, Baughn (1986), Re=34500; $\bullet \cdot \bullet \cdot \bullet$, WTM; $+ \cdot + \cdot +$, $\kappa - \epsilon$; $\triangleleft \cdot \triangleleft \cdot \triangleleft$, S-A.

momentum and heat transport are mainly due to big bi-dimensional scales. Therefore, from this point of view seems that a 2D simulation can be enough for tube-banks. But a close look show that in many ways a 2D is a deficient simulation. Below an explanation is given to this apparent good performance of 2D turbulence techniques for tube-banks.

The local Nusselt number Nu is the dimensionless expression of the local advection heat transfer coefficient (h), $Nu = hD/k$, where k is the thermal conductivity of the fluid. h is evaluated from the wall heat transfer (q_w) using a bulk temperature (T_b) afar from the cylinder surface, in regions where the temperature of the fluid is not perturbed

$$q_w = -k\frac{\partial T}{\partial \eta} = h(T_w - T_b) \tag{1}$$

where η is the normal to the cylinder surface, T_w is the temperature at the cylinder wall and T_b is the bulk temperature.

It is important to comment that the evaluation of T_b in a numerical simulation with many tubes is critical, because small differences in its evaluation can represent important errors in the evaluation of the Nu. In the present study, using many tubes, the mean temperature afar from the tube walls is not constant along the computational domain, thus a T_b for every tube should be evaluated. However a small error in the evaluation of T_b could represent a large error in the evaluation of Nu (e.g. a relative error in the order of 0.2% in T_b can generate a relative error of 6% in Nu). An alternative is to make a separate simulation for every cylinder, where only one cylinder is heated at a time as it is done in experimental work. But using many turbulence techniques these simulations would require an enormous amount of time. For this reason in the present study only one simulation has been done for the 12 tubes, and T_b is evaluated in the region upstream of every tube.

The mean Nusselt number (Nu_{av}) is the mean value around the tube of the local value. The following correlation for the Nu_{av} published by Zukauskas and Ulinskas (1988) is used in this study for comparison

$$Nu_{av} = 0.35\ (\frac{a}{b})\ Re^{0.6}\ Pr_{\infty}^{0.36}\ (\frac{Pr_{\infty}}{Pr_w})^{0.25} \tag{2}$$

Table 4: Mean Nusselt number for tube at the 4th row for air with $Pr = 0.744$ (The Er(%) is evaluated using the mean of the first 5 values of the column ($Nu_{av}/Re^{0.6}$), which belong to experimental data and correlations (3) by Zukauskas and Ulinskas (1988), and correlation (4)) by Incropera and DeWitt (1985).

Re	Technique	Sim.	Grid	$Nu_{av.}$	$Nu_{av}/Re^{0.6}$	Er(%)	Author
41000	eq.(3)			188.2	0.3213		
41000	eq.(4)			188.4	0.3216		
34100	exp.			168.5	0.3213		Meyer (1994)
41500	exp.			190.7	0.3232		Meyer (1994)
34500	exp.			170.9	0.3236		Baughn (1986)
43700	$\kappa - \epsilon$	steady	DG2D	150.3	0.2469	23	present
41109	$\kappa - \epsilon$	steady	DG3D	149.1	0.2542	21	present
43700	S-A	steady	DG2D	128.7	0.2272	29	present
41143	S-A	steady	DG3D	137.8	0.2347	27	present
41212	WTM	unsteady	DG2D	147.7	0.2506	22	present
41212	WTM	unsteady	DG3D	143.1	0.2435	24	present
41212	WTM	unsteady	DG3DP	171.7	0.2923	9	present
41040	DES	unsteady	DG3D	171.7	0.2929	9	present
41040	DES	unsteady	DG3DP	183.0	0.3122	3	present
41171	vLES	unsteady	DG3D	236.0	0.4019	24	present
41075	LES	unsteady	DG3DP	226.0	0.3854	19	present

which for $a = b = 2$, and using $Pr_w = Pr_\infty = 0.744$ for air is

$$Nu_{av} = 0.3213\ Re^{0.6} \tag{3}$$

Also the following correlation published by Incropera and DeWitt (1985) is used for comparison

$$Nu_{av} = 0.229\ \ Re^{0.632} \tag{4}$$

Figure 6 shows the Nu divided by $Re^{0.6}$ for the 12 tubes for vLES, with profile overlap from the fourth tube onwards. With small differences all 3D simulations have shown almost similar behavior to the vLES prediction, which means that thermal development occurs basically through tubes 1-4(DES presents a slight difference). Figures 7 and 8 show the Nu_{av} for the 12 rows of tubes for 3D and 2D, respectively, together with the experimental values of Meyer (1994) and Baughn (1986). Figure 7 confirms for all 3D techniques the results presented by Figure 6 that, with a small difference in the DES prediction, Nu_{av} approaches an almost constant value from the 4th tube onwards. In other words thermal development occurs through tubes 1-4. For DES the thermal development seems to be slightly longer, through tubes 1-5. After the thermal development all 3D simulations underestimate the Nu_{av} with the exception of the vLES, which overestimates it. Figure 8 shows the equivalent results for the 2D simulations. This Figure shows that for 2D simulations is not clear the thermal development as in 3D. It seems that 3D turbulent transport play an important rule at least in the thermal development through the first tubes.

Table 4 presents an overall resume of the comparisons of Nu_{av} of all numerical predictions for the fourth tube, with experimental data from Meyer (1994) and Baughn (1986) and correlations (3) and (4). Since there is small differences in the Re of the numerical predictions and experiments, all Nu_{av}, included the experimental values and those predicted from correlations (using a $Re = 41000$), are divided by $Re^{0.6}$. In order to compute the $Er(\%)$ a mean value of these experimental values, together with the values of the correlations, is used ($Nu_{av}/Re^{0.6} = 0.3222$).

Regarding the values in Table 4 some aspects should be commented. The Nu_{av} is an important parameter used in the design of heat exchangers, therefore the Er is a good index to judge the overall quality of a numerical prediction. However a low Er of the mean value not always necessarily mean a greater physical sound of the flow and heat transfer of a particular turbulence techniques. For example, the $\kappa - \epsilon$ 2D prediction presents almost the same Er than its counterpart in 3D, but it does not mean that for this model the 2D is actually almost the same predictions to 3D. Checking the local Nu and the velocity field of 2D and 3D for the $\kappa - \epsilon$ model one can see that that this Er of Nu_{av} seems to reflects that different errors are canceled, not owing to the fact that the 2D is equivalent to 3D simulation. Having said that, the best prediction from this Table is from DES for the periodic domain, and for the full domain. But one more things should be said about DES; since its thermal development is actually from tube 1 to tube 5, its under prediction in the thermal development region is actually larger. And, it should be remarked also again, as in many part of this study, that WTM has presented results with physical sound and good agreement with experimental data, similar to every one of the best predictive techniques used here.

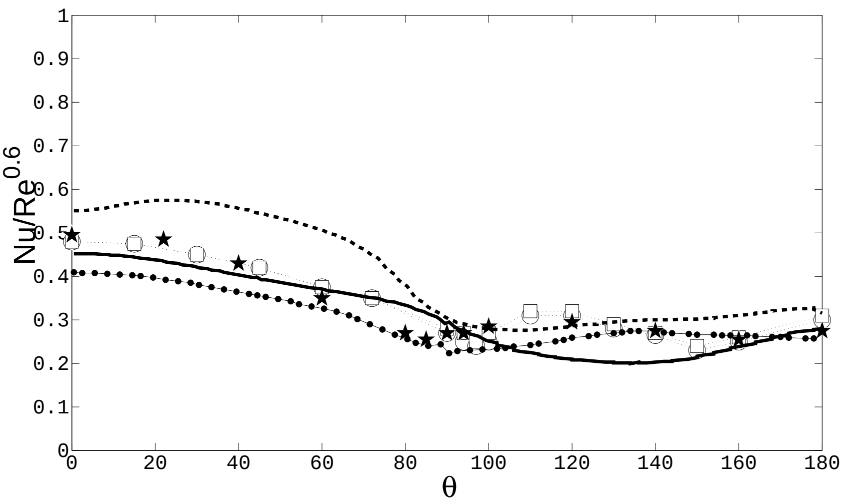


Figure 9: Comparison of Nu for the 4th tube for LES, DES and WTM for the computational domain with periodic boundary conditions in x,y,z, with experimental data. Solid line, DES; $- - - -$, LES; $\bullet \cdot \bullet \cdot \bullet$, WTM; $\circ \cdot \circ \cdot \circ$, Meyer (1994), Re=34100; $\square \cdot \square \cdot \square$, Meyer (1994), Re=41500; $\star \cdot \star \cdot \star$, Baughn (1986), Re=34500.

Figure 9 presents the comparison of the results for LES, DES and WTM for the computational domain with periodic boundary conditions in x, y, z, with experimental data. First of all, experimental data from Meyer (1994) and Baughn (1986) present, between them, a good agreement in most of the surface of the cylinder, showing two local minimum values in the rear of the cylinder and two local maximum at their stagnation points. They have, however, an important difference in the location of the first minimum (more or less at 85^o for Baughn (1986) and more or less at 95^o for Meyer (1994)). The first minimum of Nu is related with the boundary layer

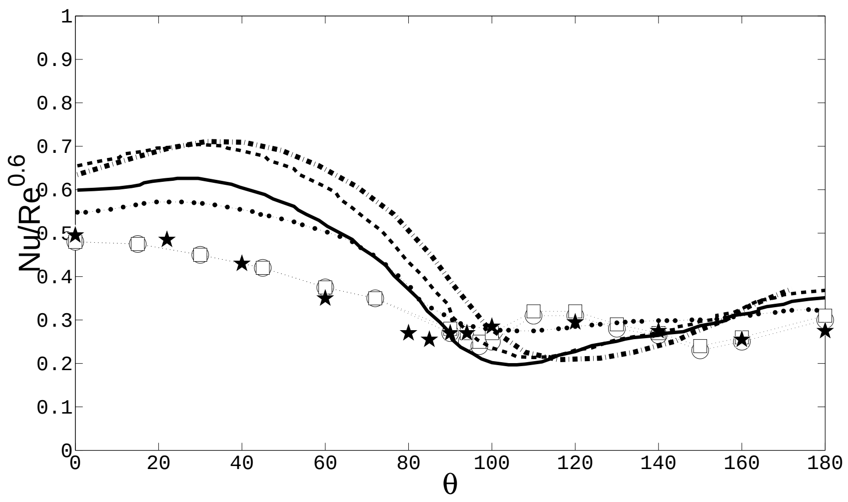


Figure 10: Comparison of Nu for the 4th tube for vLES with different grids and LES, with experimental data. ∘ · ∘ · ∘ , Meyer (1994), Re=34100; □ · □ · □, Meyer (1994), Re=41500; ⋆ · ⋆ · ⋆, Baughn (1986), Re=34500; solid line, vLES(DG3D); — — ——, vLES(MG3D); · — · — · , vLES(CG3D); • · • · •, LES(DG3DP).

separation. Looking now to the numerical prediction, LES and WTM present a first minimum in good agreement with the experimental data of Meyer, but the whole prediction of WTM is remarkably good if it is thought that it does not use any turbulence model. DES, on the other hand, presents only one minimum more or less at 130^o in clear disagreement with experimental data. Therefore, as previously commented, DES presents the best prediction of longitudinal mean velocity (Figure 3) and the minimum Er of Nu_{av}, but the worst local value of Nusselt in the rear of the tube. vLES, LES and WTM, which present greater Er, show a better distribution of the local Nusselt from the boundary layer separation to the second stagnation point.

Figure 10 shows a comparison of results from vLES using different grids and from LES. And Figures 11 and 12 show the distribution of the Nu for 3D and 2D simulations. In Figure 10 is clear how the vLES improves the Nu prediction with denser grids. This is a good news since shows that better predictions can be achieved with a denser mesh, proper of a LES. From Figures 10 and 11 a comparison of all 3D predictions of the Nu around the 4th tube show that DES presents a relative good agreement in the front of the tube, but a poor performance in the rear. In contrast, vLES presents a poor prediction in the boundary layer in the front of the cylinder and a better prediction in the separation region in the back of the cylinder. The results from the $\kappa - \epsilon$ and the S-A models have similar characteristics to the DES result, with lower precision. Note that $\kappa - \epsilon$, S-A, are RANS models, and DES is a URANS model in the near-wall region. And all these models present a poor prediction in the separation region in the back of the tube. This is the region, on the other hand, where vLES (or LES for the periodic domain) shows a better prediction. In other words, it seems that an eddy viscosity model as a function of the grid size, like as the Smagorinsky's model, can better model the turbulence afar from the cylinder wall in the detached region in the back of the cylinder. But, in contrast, it seems that a RANS model can be the best option in the attached near wall region in the front of the tube. This is actually the concept of the DES technique, but, surprisingly, it seems that its performance is not so good in the detached region.

The WTM prediction deserves a separate comment. As it is explained above, WTM does not

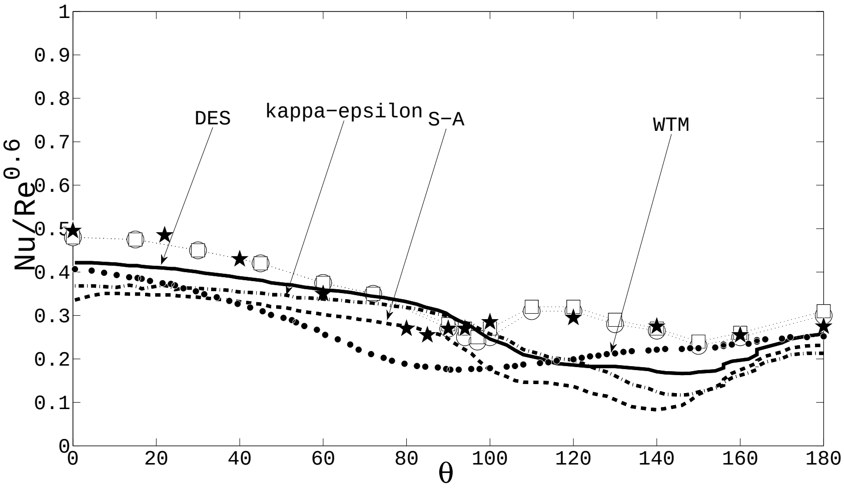


Figure 11: Comparison of Nu for the 4th tube for the 3D simulations in the full computational domain, with experimental data. Solid line, DES; $---\,-$, S-A; $\bullet\cdot\bullet\cdot\bullet$, WTM; $\cdot-\cdot-\cdot$, $\kappa-\epsilon$, Meyer (1994), Re=34100; $\square\cdot\square\cdot\square$, Meyer (1994), Re=41500; $\star\cdot\star\cdot\star$, Baughn (1986), Re=34500.

use a turbulence model, but also it does not use a grid fine enough to simulate the energy flux through the different turbulence scales. But despite of all these shortcoming, WTM have shown a prediction level with the same precision to the other turbulence techniques.

The 2D predictions of heat transfer, sometime more or less good, need also a tentative explanation. There was not a clear better performance of 3D over 2D, although in the thermal development is clear the deficient modeling of the 2D simulations. One reason for the, sometimes, good performance of 2D techniques can be the following. RANS techniques, for some reason, under predicted the heat transfer in the present study (maybe a lower value of Pr_t should be used). On the other hand, as it is commented above, it seems that a tube-banks generate a turbulent flow with big bi-dimensional scales in the longitudinal-transverse plane. Therefore, since it is expected that in a bi-dimensional turbulent flow there is an energy flux from the small to the larger scales, the deficiency of the 2D simulations end up being an improvement of the heat transfer predictions. Note that tube-banks has big vortexes (that for RANS are steady) which are not strictly part of the turbulence.

4 CONCLUSIONS

The main goal of this study was to test different aspects of numerical prediction of heat transfer through cross-flow heat exchangers, in order to know the level of accuracy that can bring to heat exchanger design. Basically the importance of different aspects like as turbulence techniques, computational domains, mesh refinement, etc. were tested. The following are the main conclusions.

The DES prediction presented the lower differences with experimental Nu_{av}, but the distribution of the local Nusselt shows that DES does not correctly model the turbulence, and thus the heat transfer, behind the tube.

vLES presented an improvements of prediction with mesh refinement, showing that better results can be achieved with denser grids; however more was expected from this techniques with the meshes used in the present study.

The experimental data used for comparison with the mean longitudinal velocity was in the

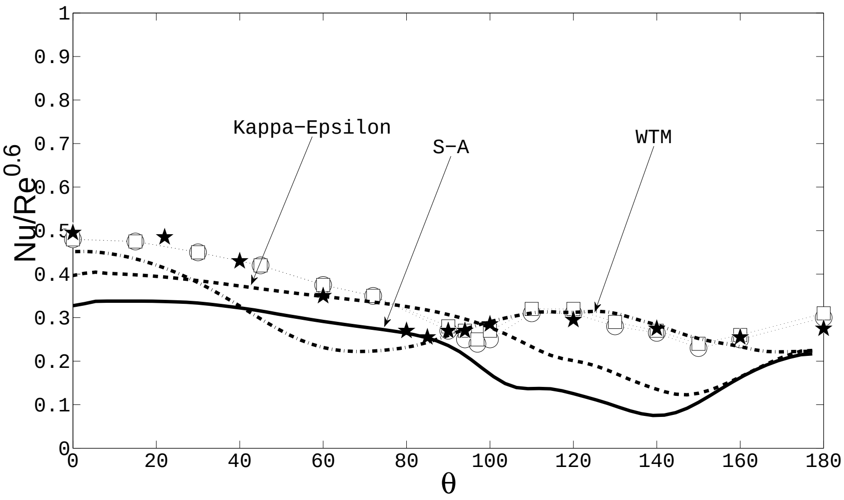


Figure 12: Comparison of Nu for the 4th tube for the 2D simulations in the full computational domain with experimental data. Solid line, S-A; $\cdot - \cdot - \cdot$, WTM; $- - - -$, $\kappa - \epsilon$, Meyer (1994), Re=34100; $\square \cdot \square \cdot \square$, Meyer (1994), Re=41500; $\star \cdot \star \cdot \star$, Baughn (1986), Re=34500.

middle of 7th and 8 th tube rows. However the use of a computational domain for LES with periodic boundary conditions with a denser grid does not presented better results than vLES in the full domain, showing that probably 2 rows of tubes are not enough in x-direction for periodic domain.

2D techniques, mainly $\kappa - \epsilon$, have shown relative good performance. It is thought that this is not actually based on its virtues but rather in its defects (since bi-dimensional flow increases bi-dimensional big scales).

Simulations WTM has presented reasonable values of Nu_{av}, in same cases with differences in the order of 10% with experimental data. But more important, the prediction of the distribution of the local Nu has more physical sound than RANS and DES predictions. It is hard to say whether this is a merit of this numerical technique without turbulence model, or it is a shortcoming of the turbulence techniques used in the present study.

Finally, based on the differences with experimental data this study shows that numerical predictions can bring improvements to heat exchanger design; but this improvement is only reasonable and will depend on the quality of the numerical simulations.

REFERENCES

Afgan, I. 2007 Large-Eddy Simulation of Flow over Cylindrical Bodies Using Unstructured Finite Volume Methods (Ph.D. thesis), The University Manchester, Manchester.

Fluent 2010, Ansys Inc., ESSS, São Paulo, Brazil.

Baughn, J.W., M.J. Elderkin, and A.A. Killop 1986 Heat transfer from a single cylinder, cylinders in tandem, and cylinder in the entrance region ofa tube bank with a uniform heat flux, Journal of Heat Transfer, **108**, pp. 386-391.

Benhamadouche S. and D. Laurence 2003 LES, coarse LES and transient RANS comparisons of the flow across a tube bundle, *Int. J. Heat Fluid Flow*, **24** pp. 470ÂŰ479.

Incropera F.P and D.P. DeWitt 1985 Fundamentals of Heat and Mass Transfer, John Wiley and Sons, 2th. Ed, New York.

Johnson R.W. 2008 Modeling strategies for unsteady turbulent flows in the lower plenum of the

VHTR, *Nucl. Eng. Des.*, **238**, pp. 482ÂŰ491.
Launder, B.E. and D.B Spalding 1972 Lectures in Mathematical Models of Turbulence, Academic Press, London.
Liang, C. and G. Papadakis 2007 Large-eddy simulation of cross-flow through a staggered tube bundle at sub-critical Reynolds number, *J. Fluids Struct.*, **23**, pp. 1215ÂŰ1230.
Meyer, K.E. 2000 A theory for turbulent pipe and channel flows. *J. Fluid Mech.*, **421**, pp. 115-145.
Meyer, K.E. 1994 Experimental and Numerical Investigation of Turbulent Flow and Heat Transfer in Staggered Tube Bunddle, AFM 94-04, Technical University of Denmark.
Pysmennyy Y., G. Polupan, I. Carvajal-Mariscal, F. Sánchez-Silva 2010 Estudio comparativo de los métodos del cálculo de la transferencia de calor en bancos de tubos, Científica, **(14)**1, pp. 17-23.
Rollet-Miet P., D. Laurence and, J. Ferziger 1999 LES and RANS of turbulent flow in tube bundles. *Int. J. of Heat and Fluid Flow*, **20**, pp. 241-254.
Simonin, O., and Barcouda, M. 1986 Measurements of Fully Developed Turbulent Flow across Tube Bundle, *Proceedings of the Third International Symposium on Applications of Laser Anemometry to Fluid Mechanics*, Lisbon, Portugal, pp. 21.5.1-21.5.5.
Smagorinsky, J. 1963 General Circulation Experiment with the Primitive Equations. I the Basic Experiment, Monthly Wea. Review, **91**, pp. 99-164.
Spalart, P. and S. Allmaras 1992 A one-equation turbulence model for aerodynamics flows, Technical Report, AIAA 92-04-39, American Inst. of A. and Astronautics.
Spalart, P., S. Deck, M. Shur, K. Squires, M. Trelets, and A Travins 2006 A new version of detached eddy simulation, resistant to ambiguous grid densities, Theo. and Computational F. Dynamics, **20**, pp. 181-105
Tsunoda, K., S. Okamoto, M. Kijima, S. Higashi, N. Abe, 1996 Measurement of flow around two-dimensional circular cylinder bundles with LDV, in: *Proc. 8th Int. Symp. On Applications of Laser Techniques to Fluid Mechanics*, Lisbon.
West, A., B. E. Launder, and H. Iacovides 2015 On the Computational Modelling of Flow and Heat Transfer in In-Line Tube Banks, *Advances in Heat Transfer*, **46**, pp. 1-46.
Zukauskas A. 1972 Heat Transfer from Tubes in Cross Flow, *Adv. Heat Transfer*, **8** pp. 93-160.
Zukauskas A. and R. Ulinskas 1988 Heat Transfer in Tube Banks in Cross Flow *Hemisphere Pub. Corporation*, Washington DC.