Flow structures and heat transfer over a single dimple using hybrid URANS-LES methods

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ABSTRACT

Flow structures and heat transfer of turbulent flow over a single dimple placed at the lower wall in a narrow channel at Reynolds number $\mathrm{Re}_D = 42000$ and $\mathrm{Re}_D = 105000$ for Prandtl number $\mathrm{Pr} = 0.71$ and $\mathrm{Pr} = 3$ have been analyzed using large eddy simulation (LES) and improved delayed detached eddy simulation with improved wall modeling capability (IDDES) of the 2011 Gritskevich version. Based on the simulations of the turbulent channel flow up to $\mathrm{Re}_\tau = 2400$, the LES and IDDES approach are validated and needed mesh resolutions are determined. The heat transfer rates and the skin-friction factors of the IDDES are consistent with empirical correlations whereas LES show large differences for higher Reynolds number using moderate grid sizes. The mean values as well as vortex structures and secondary flow structures using a spherical dimple have been analyzed and compared to experimental data. IDDES reproduces the correct flow physics and mean quantities while reducing the computation time enormously with a satisfying accuracy.

INTRODUCTION

Several methods of heat transfer enhancement like ribs, fins or dimples have been thoroughly investigated in the last decades with the aim to improve the heat transfer at a minimum hydraulic loss. Although the excellent thermo-hydraulic performance of dimples in comparison with ribs or fins is well known, the physics of the flow and the role of heat transfer inside the dimples for higher Reynolds numbers is complicated and still not completely understood. Moreover, it is a big challenge for numerical simulations to predict flow physics and heat transfer.

Afanasyev *et al.* (1993) investigated the pressure drop and heat transfer on a plate with dimples in the turbulent flow regime. The maximum of heat transfer augmentation is found of about 40% with only a low increase of the pressure drop. Most motivating results have been published by Chyu *et al.* (1997), who investigated the pressure drop and heat transfer improvement for the surface roughened by an array of hemisphere and tear-drop shaped cavities in the range of Reynolds number $Re_{Dh} = 10'000 - 50'000$ based on the hydraulic diameter. The heat transfer coefficients on the roughened surface for both configurations were determined to be 2.5 times higher than these on the opposite smooth wall whereas the induced flow resistance was twice as small as that of rib turbulators. The effect of the channel height and dimple depth on heat transfer in the turbulent flow regimes has been studied by Moon *et al.* (2003) and Burgess & Ligrani (2004). It was reported that the heat transfer increases by enlargement of the dimple depth, whereas the flow resistance is also increased. For dimples with a depth to diameter ratio of $H/D \geq 0.22$ a significant augmentation of heat transfer is documented due to dynamic vortex structures with transversal oscillations around the dimple area. Ligrani *et al.* (2003), conducted many studies to establish a data basis including the pressure drop and heat transfer values according to various dimple and channel geometries. Numerical studies of dimpled heat exchangers have been performed by a number of researchers. Park *et al.* (2004) examined turbulent flow over dimples with a ratio $H/D = 0.3$ using RANS method for Reynolds numbers $\mathrm{Re}_H = 2700$ up to $\mathrm{Re}_H = 41000$. This study provides an insight into the development of flow structures produced by the dimples and the increase of heat transfer rates, whereas the integral values were not in good agreement to experimental data. The results showed the existence of a rotating vortex pair generated inside and upstream each dimple and a second pair at the spanwise edges of the dimple. Wang *et al.* (2003) carried out research of low Reynolds number flow in dimpled channels of $\mathrm{Re}_\tau = 180$ based on the friction velocity. The existence of a symmetric horseshoe vortex was documented inside a single dimple. The most detailed numerical investigations have been performed by Isaev *et al.* (2010). Usage of overlapping grids, written in a conservative form, established the possibility to perform calculations with high resolved zones inside and outside the dimple. This method in combination with the $k-\omega$-SST model showed best results in comparison to experiments. On the basis of URANS a series of vortex formations inside have been performed which change quantitative as well as qualitative depending on the ratio dimple depth to dimple print diameter. This result is in accordance with measurements performed by Terekhov *et al.* (1997) for a dimple with a ratio $H/D = 0.26$. Numerical results showed that asymmetric vortex structures provide a higher efficiency than symmetric ones. For the turbulent flow regimes it was suggested to define the dimple geometry for each purpose to generate mono core vortex structures using an asymmetric dimple shape Isaev *et al.* (2003). Previous numerical investigations for dimpled surfaces are performed using steady RANS approach, since LES computations require high grid resolutions and much computational resources. At time the detached eddy simulation (DES) is mostly applied for aerodynamics problems in order to use the advantages of RANS near the wall and to use LES far away from the wall when the turbulent length scales exceed the grid dimensions. Classical DES method has been originally published by Spalart *et al.* (1997) by introducing a characteristic length scale determined by the RANS model to account for switching between RANS and LES region. In addition, several variants of the DES model, have been developed with rather different characteristics to address several kind of flow physics like flow separation or evolving near wall vortex structures. The connection of wall modeled large eddy simulation (WMLES) and delayed detached eddy simulation (DDES) is named IDDES which has been extended to the $k-\omega$-SST by Gritskevich *et al.* (2011). The model extension is used within this study to take the advantages of the near wall modeling regarding vortex structures and heat transfer into account. In addition SST-based formulation has a considerable advantage that the transition from RANS to LES regions does not only depend on the distance to the wall and the mesh resolution, but also on the solution. Previous work from Viswanathan & Tafti (2006) using classical DES formulation proposed by Strelets (2001) of the fully developed flow and heat transfer in a channel with normal ribs showed very good agreement in comparison to the experiments and LES results. It was reported, that the computation time were an order of magnitude less expensive than pure LES approach.

The objective of this study is to clarify the role of the vortex formations with respect to the heat transfer and friction loss of a single symmetric dimple using LES and IDDES method. The hybrid model is validated using experimental data for the single dimple and turbulent channel flows where it is afterwards applied to address higher Reynolds numbers up to $\mathrm{Re}_D = 105000$ and Prandtl numbers up to $\mathrm{Pr} = 3$. In our previous work (see Turnow *et al.* (2010)) vortex structures have already been investigated for a single spherical dimple. LES simulations revealed the formation of asymmetric structures with an orientation switching between two extreme positions. The study is conducted to prove the capabilities of the hybrid method IDDES using $k-\omega$-SST for the RANS regions to capture the asymmetric flow structures and to identify vortex structures for higher Reynolds numbers including heat transfer.

NUMERICAL METHOD

Computations have been performed based on a 3-D finite volume method. The filtered transport equations are solved on a non-staggered Cartesian grid using implicit filtering. The discretization in space and time of the quantities at the cell faces is of second order using central differencing scheme.

The LES equations are obtained by filtering the continuity equation, the Navier-Stokes equations and the transport equation for the non-dimensional temperature θ at the filter width $\widetilde{\Delta}$:

$$\begin{aligned}\frac{\partial \widetilde{u}_i}{\partial t} + \frac{\partial \widetilde{u}_j \widetilde{u}_i}{\partial x_j} &= \\ &-\frac{1}{\rho}\frac{\partial \widetilde{P}}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\nu\left(\frac{\partial \widetilde{u}_i}{\partial x_j} + \frac{\partial \widetilde{u}_j}{\partial x_i}\right) - \tau_{ij}\right]\end{aligned} \tag{1}$$

$$\frac{\partial \widetilde{\theta}}{\partial t} + \frac{\partial \widetilde{u}_j \widetilde{\theta}}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\frac{\nu}{\mathrm{Pr}}\frac{\partial \widetilde{\theta}}{\partial x_j} - J_j^{SGS}\right]. \tag{2}$$

The unclosed subgrid stress tensor $\tau_{ij} = \widetilde{u_i u_j} - \widetilde{u}_i\widetilde{u}_j$ and the subgrid contribution to the scalar dynamics $J_j^{SGS} = \widetilde{\theta u}_j - \widetilde{\theta}\widetilde{u}_j$ are modeled in terms of the filtered quantities $\widetilde{u}_i$ and $\widetilde{\theta}$, using a dynamic one equation eddy-viscosity model proposed by Kim & Menon (1995). The turbulent scalar fluxes are modeled using gradient diffusion approach. In Eq. 2 the non-dimensional temperature θ is defined as $\theta = T - T_{lowerwall}/(T_{lowerwall} - T_{upperwall})$ and further treated as a passive scalar without buoyancy effects. The molecular Prandtl number Pr was set to $\mathrm{Pr} = 0.71$ and to $\mathrm{Pr} = 3.0$ respectively, whereas the turbulent Prandtl number is set to constant $\mathrm{Pr}_t = 0.9$. Investigations revealed that the chosen SGS model shows fairly good agreement with measurements in respect to heat transfer and recirculating flows in contrast to other SGS models.

The IDDES with $k-\omega$-SST approach uses a more extended expression for the turbulent length scales as rather than $\Delta = \Delta_{max}$ as in DES and DDES. This yields to a linear variation of the subgrid length-scale in the vicinity of the wall and becomes Δ_{max} off the wall. It was shown, that this choice is superior to either of the previously used subgrid length-scale definitions for LES (Δ_{max}, $\Delta = V^{1/3}$), with regard to wall-bounded flows. Gritskevich *et al.* (2011) proposed a modified version of the IDDES based on the $k-\omega$-SST model. The recalibration of the shielding function f_d show an improvement for several testcases compared to the previous version without damping of resolved turbulence. The hybrid length scale is given by the following expression

$$l_{IDDES} = \widetilde{f}_d(1+f_e)l_{RANS} + (1-\widetilde{f}_d)l_{LES}, \text{ with} \tag{3}$$

$$l_{LES} = C_{DES}\Delta \tag{4}$$

$$l_{RANS} = \frac{k^{1/2}}{\beta\omega} \tag{5}$$

$$C_{DES} = C_{DES1}F_1 + C_{DES2}(1-F_2) \tag{6}$$

The subgrid length-scale is calculated as:

$$\Delta_{IDDES} = \min\{C_w \max[y, \Delta_{max}], \Delta_{max}\} \tag{7}$$

where $C_w = 0.15$, y_w – wall-Distance and Δ_{wn}-grid spacing in wall normal direction. The full description of the IDDES model can be found in Gritskevich *et al.* (2011).

COMPUTATIONAL SETUP

In a first step the LES and IDDES model have been evaluated using turbulent channel flow simulations up to $\mathrm{Re}_\tau = 2400$ where at the second step both approaches are applied to determine vortex structures, friction loss and heat transfer augmentation for a single spherical dimple placed at the lower wall. In contrast to classical turbulent channel flow simulations, where the dimensions are set in reference to the non-dimensional channel height to $H = 2$, $L_x = 2\pi$ and $L_z = \pi$, the computational domain is down scaled and expanded since in the next step a dimple has to be placed at the lower wall. The channel height H is equal to $H = 0.015m$. The length in axial direction is set to $L_x = 12H$ and in spanwise direction to $L_z = 5.33H$. Thus, the ratio of channel length to channel height is fairly larger than $L_x/H = \pi$ as in the DNS computations. The expansion of the channel dimensions will require much more grid cells using LES method but ensures definitely the decrease of the autocorrelation functions to almost zero in streamwise and spanwise direction. A schematic sketch of the computational domain is presented in Figure 1. The dimple diameter is kept constant with an diameter of $D = 0.046m$ and a ratio of dimple depth-to-diameter

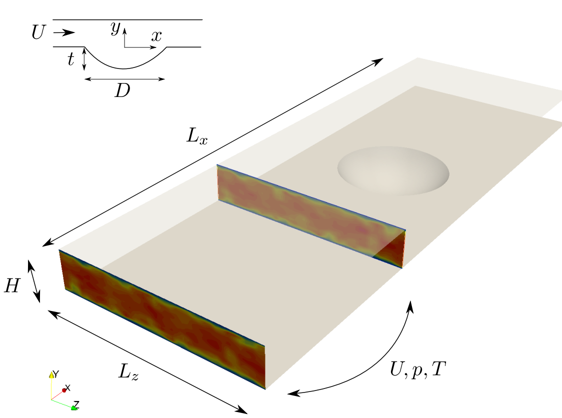

Figure 1. Sketch of the computational domain of the turbulent channel flow including the mapping plane for the recycling method.

$t/D = 0.26$. The recycling method is applied to ensure correlated inflow conditions which copies the values from a given slice located downstream the inlet plane each timestep back to the inlet. Periodic boundary conditions are applied in spanwise direction. To drive the flow with a constant massflow rate, the overall losses within the inlet section are calculated and simply added as driving force to the momentum equation at every time instant. No slip wall conditions were enforced on the lower and the upper solid walls for the velocity whereas the temperature was fixed at the lower (hot surface, $\theta = 1$) and the upper (cold surface, $\theta = 0$) channel wall. Two blockstructured grids are used for the turbulent channel flow (see Table 1).

Table 1. Different grid resolutions for LES of turbulent flow in a plane channel.

Mesh	$N_x \times N_y \times N_z$
C1	$174 \times 48 \times 132$
C2	$272 \times 72 \times 208$

A grid stretching normal to the wall is used where the first grid point is located at a distance of $y^+ = 0.8$ (case C1) and $y^+ = 0.5$ (case C2). A small stretching factor of 4.5 (from the largest to the smallest edge length) normal to the wall was chosen for both cases to place several grid points into the viscous sublayer. The mesh is homogeneous in spanwise and streamwise direction for the plane channel flow.

A block structured curvilinear grid consisting of 1'591'296 cells was chosen on the basis of the turbulent channel flow calculations for further investigations of the dimpled channel shown in Figure 1. Several studies have been carried out (not presented here) to establish the needed grid size for an accurate representation of the flow physics and heat transfer. Special attention is paid to the resolution of the dimple edge to capture the flow separation and shear layer gradients with a proper accuracy. The condition $y^+ \leq 1$ for the first grid point was satisfied even at the highest Reynolds number to ensure a correct estimation of the local heat flux and friction factor.

RESULTS

Turbulent Channel Flow

For validation, reference and for establishment of the grid independency a series of turbulent channel flow simulations within a Reynolds number range from $\mathrm{Re}_\tau = 180$ up to $\mathrm{Re}_\tau = 2400$ have been carried out.
Results for the skin-friction coefficient

$$C_f = -\frac{(\Delta p/L_x)D_h}{0.5\rho U_b^2} \tag{8}$$

have been compared to empiric correlation from Dean

$$C_f = 0.06138\ \mathrm{Re}^{-1/4}, \tag{9}$$

whereas the heat transfer rates in terms of Nusselt number have been compared to empiric correlation of Petukhov-Gnielinski (see Bejan & Kraus (2003))

$$\xi = (0.79\ln(\mathrm{Re}_H) - 1.64)^{-2} \tag{10}$$

$$\mathrm{Nu} = \frac{(\xi/8)(\mathrm{Re}_H - 1000)\mathrm{Pr}}{1 + 12.7\sqrt{\xi/8}\,(\mathrm{Pr}^{2/3} - 1)} \tag{11}$$

for the smooth channel and are further used to determine heat transfer augmentation for the turbulent flow over the single dimple. The results for the friction factor and heat transfer rates obtained for both meshes are presented in Table 2 and Table 3 respectively.

The results for the skin-friction coefficient show, that the pure LES simulations underpredict the empiric values for both grid sizes. The difference increases for higher Reynolds numbers. An underprediction can also be observed using the IDDES as hybrid method, whereas the differences to empiric data are smaller than compared to pure LES results. Up to a Reynolds number of $\mathrm{Re}_\tau = 1024$ IDDES underpredicts the friction coefficient about $C_f/C_{f,IDDES} = 1.06$ compared to $C_f/C_{f,LES} = 1.51$ which show that LES in unemployable even using the fine mesh C2. It has to be noticed, that for the Reynolds number $\mathrm{Re}_\tau = 2400$ both approaches show a big deviation for the chosen mesh sizes. In case of heat transfer it can be seen, that the Nusselt numbers are overpredicted in case of the IDDES and LES method compared to empiric correlations. Especially for higher Prandtl numbers, the deviations are getting higher as for the lower Prandtl number. A clear tendency could not be determined whether to choose LES or IDDES method in case of heat transfer. However, both meshes, C1 and C2, do not fulfill the needed requirements for a pure LES simulations which leads to the IDDES method as the preferable choice.

Turbulent flow and heat transfer of a single dimple

Numerical simulations for turbulent flow over a single dimple at Reynolds number $\mathrm{Re}_D = 42000$ and $\mathrm{Re}_D = 105000$ have been carried out, which corresponds to the Reynolds number based on the friction velocity of $\mathrm{Re}_\tau = 395$ and $\mathrm{Re}_\tau = 1024$ for the smooth channel flow respectively.

DES regions Figure 2 shows the LES (value 1.0) and RANS (value 0.0) regions inside the single dimple. The near wall regions is always resolved using RANS whereas the flow inside the channel and dimple is captured by LES. Since the choice of the region is determined using turbulent length scales, a smooth transition between LES and RANS takes place in dependence of the actual flow features. It can be seen, that the recirculation zone including the shear layer inside the dimple is captured by LES ensuring most accurate calculation. Hence, the IDDES model acts like a wall function in order to capture the high velocity and temperature gradients near the wall which is especially important for higher Prandtl numbers.

Validation Mean values of the velocity and its rms values from the bottom center of the dimple to the upper channel wall in comparison to measurements of Turnow *et al.* (2010) and Terekhov *et al.* (1997) are presented in Figure 3.

A comparison between the mean streamwise velocity profiles and its normalized rms values show small differences between pure LES and IDDES and further a good agreement to experimental LDV measurements for $\mathrm{Re}_D = 42000$. Differences can be observed for $\mathrm{Re}_D = 105000$ in LES and IDDES profiles within the shear layer region around $y/H = 0$. However, no experimental data could be found to evaluate the differences, but since the grid size

Table 2. Friction factor C_f for turbulent channel flow up $\mathrm{Re}_\tau = 2400$ using LES and IDDES method in comparison with Dean correlation.

		$\mathrm{Re}_\tau = 180$	$\mathrm{Re}_\tau = 395$	$\mathrm{Re}_\tau = 1024$	$\mathrm{Re}_\tau = 2400$
Dean		$8.46 \cdot 10^{-3}$	$6.76 \cdot 10^{-3}$	$5.16 \cdot 10^{-3}$	$4.04 \cdot 10^{-3}$
LES	(C1)	$6.87 \cdot 10^{-3}$	$4.93 \cdot 10^{-3}$	$3.33 \cdot 10^{-3}$	$1.51 \cdot 10^{-3}$
	(C2)	$7.37 \cdot 10^{-3}$	$5.79 \cdot 10^{-3}$	$3.40 \cdot 10^{-3}$	$2.12 \cdot 10^{-3}$
IDDES	(C1)	$7.39 \cdot 10^{-3}$	$5.70 \cdot 10^{-3}$	$3.36 \cdot 10^{-3}$	$2.10 \cdot 10^{-3}$
	(C2)	$6.39 \cdot 10^{-3}$	$5.58 \cdot 10^{-3}$	$4.87 \cdot 10^{-3}$	$2.58 \cdot 10^{-3}$

Table 3. Nusselt number for turbulent channel flow up $\mathrm{Re}_\tau = 2400$ for $\mathrm{Pr} = 0.71$ and $\mathrm{Pr} = 3.0$ using LES and IDDES method in comparison with correlation of Petukhov-Gnielinksi.

		$\mathrm{Re}_\tau = 180$		$\mathrm{Re}_\tau = 395$		$\mathrm{Re}_\tau = 1024$		$\mathrm{Re}_\tau = 2400$	
		$\mathrm{Pr} = 0.71$	$\mathrm{Pr} = 3$	$\mathrm{Pr} = 0.71$	$\mathrm{Pr} = 3$	$\mathrm{Pr} = 0.71$	$\mathrm{Pr} = 3$	$\mathrm{Pr} = 0.71$	$\mathrm{Pr} = 3$
Petukhov-Gnielinski		14.66	28.15	31.55	65.15	74.74	165.67	163.31	381.70
LES	(C1)	15.80	37.04	33.18	86.84	64.68	159.74	111.36	253.24
	(C2)	20.80	49.28	38.66	106.72	79.24	216.34	140.34	342.52
IDDES	(C1)	15.94	34.28	37.80	102.18	93.10	146.64	155.64	232.04
	(C2)	20.50	41.98	38.16	101.24	98.30	226.94	160.86	363.92

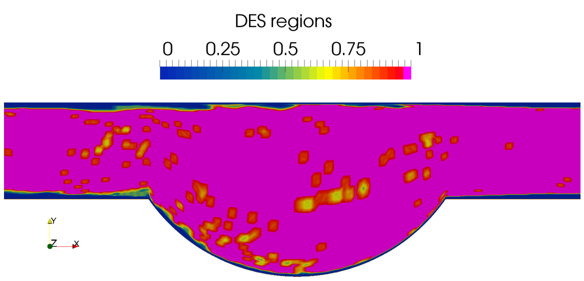

Figure 2. IDDES regions in the transversal center plane of the dimple. The value 0.0 represents RANS and the value 1.0 LES regions at $\mathrm{Re}_D = 42000$.

are too small for a pure LES, the IDDES results are more trustful. The streamwise velocity fluctuations within the shear layer are represented by all simulations and are enhanced by an increasing Reynolds number, whereas the LES reproduces the highest magnitude compared to IDDES. The magnitude of fluctuations is in good agreement to the experimental data. It can be concluded, that the presented profiles as predicted by IDDES are consistent with the profiles predicted by LES.

Heat transfer / friction loss The mean values of the friction losses and heat transfer augmentation for the turbulent flow over a single dimple have been normalized by its values obtained for the smooth channel flow. The mean values for heat transfer are integrated around the dimple within a rectangular box of the dimensions $\pm x/D = 2.5 \times \pm z/D = 1.5$. The results are summarized in Table 4.

Table 4. Skin-friction coefficient and Nusselt number for turbulent flow over a single dimple using LES and IDDES method in comparison to smooth channel flow simulations (denoted with 0) at up $\mathrm{Re}_D = 42000$ and $\mathrm{Re}_D = 105000$ for $\mathrm{Pr} = 0.71$ and $\mathrm{Pr} = 3.0$.

		$\mathrm{Re}_D = 42000$		$\mathrm{Re}_D = 105000$	
		$\mathrm{Pr} = 0.71$	$\mathrm{Pr} = 3$	$\mathrm{Pr} = 0.71$	$\mathrm{Pr} = 3$
LES	C_f/C_{f0}	1.057		1.32	
	$\mathrm{Nu}/\mathrm{Nu}_0$	1.32	1.36	1.11	1.40
IDDES	C_f/C_{f0}	1.144		1.36	
	$\mathrm{Nu}/\mathrm{Nu}_0$	1.35	1.42	1.18	1.45

Results of the skin-friction and heat transfer coefficient for the moderate Reynolds number $\mathrm{Re}_D = 42000$ are in the same range for LES and IDDES method which are in good agreement of results published in literature. The values for heat transfer of both methods are consistent for the investigated Reynolds numbers $\mathrm{Re}_D = 42000$ and $\mathrm{Re}_D = 105000$ whereas a difference for skin-friction coefficient could be observed for the lower Reynolds number. The reason for

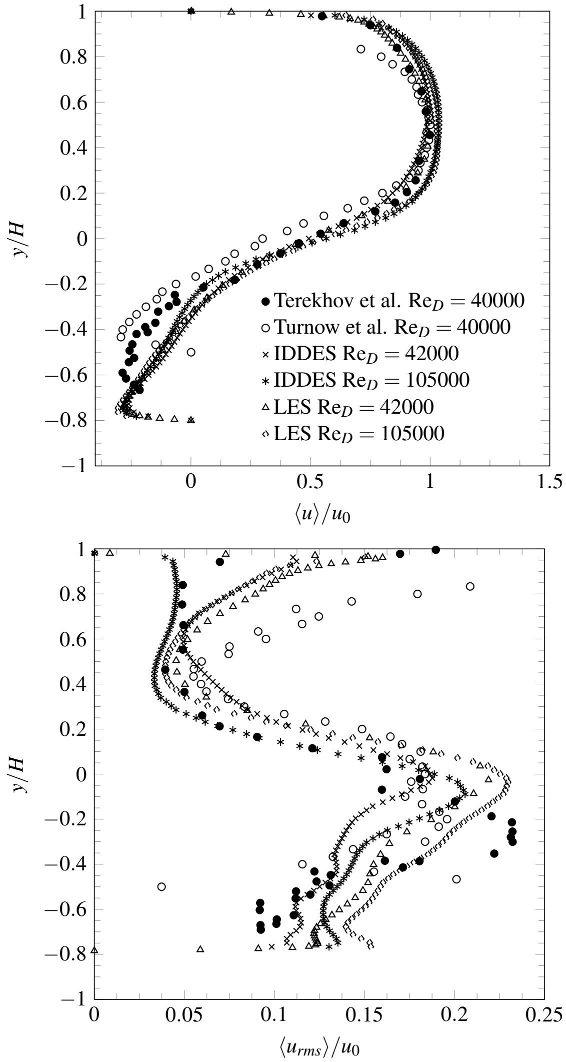

Figure 3. Streamwise velocity profiles for turbulent flow over a single dimple from the center of dimple to the upper channel wall in comparison to experimental LDV measurements for $\mathrm{Re}_D = 42000$ and $\mathrm{Re}_D = 105000$.

the agreement of the friction in case of the higher Reynolds number is found since the dominant losses occur while the flow attaches the dimple trailing side which is well represented by both methods. In case of higher Reynolds numbers, the heat transfer augmentation of 11% in case of LES and 18% in case of IDDES for the lower Prandtl number $\mathrm{Pr} = 0.71$ seem to be questionable and needs further studies. Furthermore, a dramatically increase of heat transfer rates compared to a smooth channel while increasing the Reynolds number could not be observed for Prandtl number $\mathrm{Pr} = 3$. In contrast the ratio of skin-friction coefficients increases for higher Reynolds numbers. Thus, the optimal thermo-hydraulic performance of the dimple determined for moderate Reynolds numbers and is lowered for higher flow velocities.

Flow structures The mean velocity profiles presented in Figure 3 show clearly the creation of a large recirculation zone where the time averaged zone occupies about 80% of the cavity. However, concerning the flow structures within the dimple there is a clear discrepancy between the time averaged and time resolved streamline patterns. Whereas for turbulent flows ($\mathrm{Re}_D > 20000$ for dimples with a small geometric ratio $h/d < 0.2$ the flow structure is found to be fully symmetric, at a ratio of $h/d = 0.2$ the flow experiences a topological change and becomes asymmetric with the creation of a vortex mono structure inclined to the mean flow direction with an angle of approximately $\alpha = \pm 45°$.

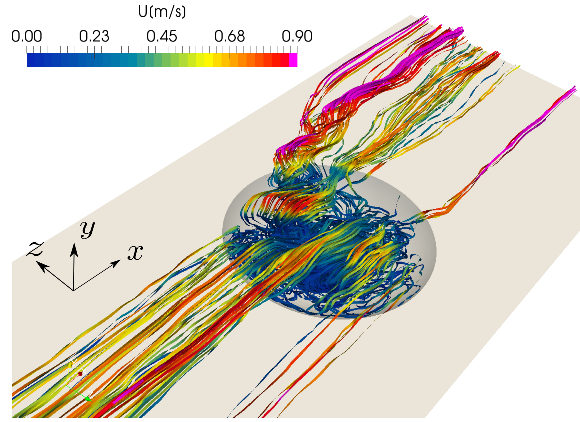

Figure 4. Streamlines for a turbulent flow over single dimple at Reynolds number $\mathrm{Re}_D = 105000$ using IDDES approach.

Streamlines indicate the asymmetrical vortex structures switching its position from $\alpha = +45°$ to $\alpha = -45°$ periodically. As seen from instantaneous streamlines patterns in Figure 4, the fluid enters directly from the channel into the dimple and rotates within the recirculation zone and finally leaves the dimple at one side. Hence, for a dimple with a depth to diameter ratio of $h/d = 0.26$, numerical results from LES and IDDES confirm the generation of unsteady asymmetric monocore vortex structures with a predominant transversal direction. It has to be noticed, that LES and IDDES show the same asymmetric flow behavior for Reynolds number $\mathrm{Re}_D = 42000$ and $\mathrm{Re}_D = 105000$.

However, the time averaged flow field over a sufficiently long time period is nearly symmetric which is in agreement with performed measurements. In this context the results obtained from LES, IDDES and measurement observations contradict to URANS/RANS ones which predict the asymmetry even in the time averaged flow pictures. Thus, LES and IDDES reveals the generation of unsteady asymmetric monocore vortex structures with a predominant transversal direction which are responsible for the appearance of organized self-sustained oscillations. Analysis of time the time signals show a switching of the asymmetric vortex structures with a frequency of $f = 0.2Hz$. The switching of the vortex structures have been confirmed in experiments by Voskoboinick *et al.* (2013).

Another interesting flow feature is the appearance coherent vortex structures inside the shear layer between the main channel and recirculating flow inside the dimple. Vortex rollups are generated within the shear layer enhancing fluctuations in the cavity which are typical for flows with mutual velocities. Isosurfaces of the instantaneous λ_2 structures as predicted by IDDES are presented in Figure 5.

It is observed that LES resolves scales much smaller than the DES computation. However, the vortex structures as predicted by IDDES are consistent with the structures predicted by LES. Typical rollup structures within the shear layer represent a lot of turbulence as the flow progresses downstream which enhance mixing processes within the dimple. The induced turbulence propagates towards the trailing edge enhancing wall normal fluctuations. It was

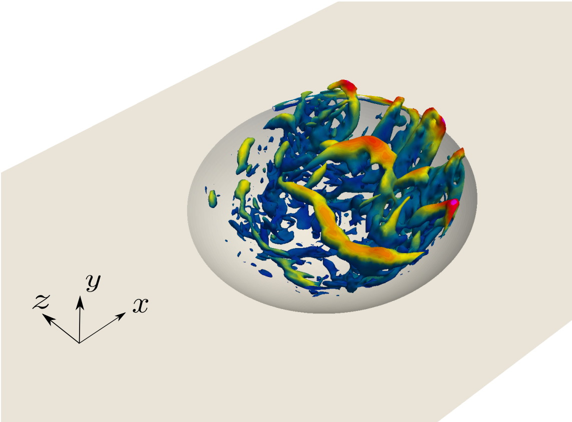

Figure 5. Isosurfaces of λ_2-structures for turbulent flow over single dimple at Reynolds number $\mathrm{Re}_D = 105000$ using IDDES approach colored by the velocity.

found from time resolved heat flux analysis, that the heat transfer is mainly dominated by the direct attachment of the flow at the heated surface and its vortex shedding at the downstream side of the dimple. However, the shifting of the asymmetric structure towards the main flow direction could not be detected in the time history of the heat transfer rates. Thus, a significant enhancement due to the shifting of the asymmetric vortex structure could not be observed. Nevertheless, the vortex shifting permanently drives the hot fluid out of the dimple which is found to be preferable in terms of heat transfer enhancement and also to prevent fouling effects.

CONCLUSION

Comparison of numerical results obtained from LES and IDDES for turbulent channel flow up to $\mathrm{Re}_\tau = 2400$ and the turbulent flow over a single dimple including a variation of the Reynolds number ($\mathrm{Re}_D = 42000$, $\mathrm{Re}_D = 105000$) and Prandtl number ($\mathrm{Pr} = 0.71$, $\mathrm{Pr} = 3$) with the experimental data and empiric correlations confirmed, that the established hybrid method IDDES are capable of predicting flow physics and heat transfer rates with a satisfactory accuracy. Numerical simulations reveal the presence of self-sustained oscillations inside a single dimple. Analysis of flow structures using instantaneous streamline patterns and λ_2-structures show the presence of an asymmetric vortex structure which is inclined in respect to the incoming flow of $\alpha = \pm 45°$. With respect to heat transfer enhancement, the asymmetric vortex structures are preferable in terms of steadily driving the hot fluid out of the dimple. A significant increase of the heat transfer rates due to the long period oscillations could not be found since the time scale of the heat transfer mechanism is much smaller than shifting time of the asymmetric vortex structure.

A drastic augmentation of heat transfer could not be observed for an increasing Reynolds number. The heat transfer augmentation is relatively constant of about $\mathrm{Nu}/\mathrm{Nu}_0 = 1.42$ and $\mathrm{Nu}/\mathrm{Nu}_0 = 1.45$ for Reynolds number $\mathrm{Re}_D = 42000$ and $\mathrm{Re}_D = 105000$ respectively. In contrast, the ratio of skin-friction coefficients increases for higher Reynolds numbers. Thus, the thermo-hydraulic performance of the dimple is optimal for moderate Reynolds numbers and is lowered for higher flow velocities.

In summary it can be concluded, that the IDDES accurately predicts the physics turbulent flow, as p.ex. shear layer induced secondary motions, inside the dimple. The main contribution of this paper is in establishing that IDDES, like LES, can produce good quality results at a much lower computational costs than an equivalent LES simulations.

REFERENCES

Afanasyev, V.N., Chudnovsky, Y.P., Leontiev, A.I. & Roganov, P.S. 1993 Turbulent flow friction and heat transfer characteristics of spherical cavities on a flat plate. *Experimental Thermal and Fluid Science* (7), 1–8.

Bejan, A. & Kraus, A.D. 2003 *Heat Transfer Handbook*.

Burgess, N.K. & Ligrani, P.M. 2004 Effects of dimple depth and on nusselt numbers and friction factors for internal cooling channel. *Proceedings of ASME Turbo Expo* **No. GT2004-54232**.

Chyu, M.K., Yu, Y., Ding, H., Downs, J.P. & Soechting, F.O. 1997 Concavity enhanced heat transfer in an internal cooling passage. *ASME* **No. 97-GT-437**.

Gritskevich, M. S., Garbaruk, A. V., Schuetze, J. M. & Florian, R. 2011 Development of ddes and iddes formulations for the k-omega shear stress transport model. *Flow Turbulence Combustion* pp. 1–19.

Isaev, S.A., Kornev, N., Leontiev, A.I. & Hassel, E. 2010 Influence of the reynolds number and the spherical dimple depth on turbulent heat transfer and hydraulic loss in a narrow channel. *Int J. of Heat and Mass Transfer* **53** (1-3), 178–197.

Isaev, S.A., Leontiev, A.I., Mitjakov, A.V. & Pyshnyi, I.A. 2003 Intensification of tornado-like heat transfer in asymmetric dimples. *J. of Engineering Physics and Thermophysics* **76**, 31–34.

Kim, W & Menon, S. 1995 A new dynamic one-equation subgrid-scale model for large eddy simulation. In *In 33rd Aerospace Sciences Meeting and Exhibit*. Reno, NV.

Ligrani, P.M., Oliveira, M.M. & Blaskovich, T. 2003 Comparison of heat transfer augmentation techniques. *AIAA J.* **41** (3), 337–362.

Moon, H.K., O'Connel, T. & Sharma, R. 2003 Heat transfer enhancement using a convex-patterned surface. *J. of Turbomachinery* **125**, 274–280.

Park, J., Desam, P.R. & Ligrani, P.M. 2004 Numerical predicitions of flow structures above a dimpled surface in a channel. *Numerical Heat Transfer* **45**, 1–20.

Spalart, P.R., Jou, W.-H., Strelets, M. & Allmaras, S.R. 1997 Comments on the feasibility of les for wings, and on a hybrid rans/les approach. In *First AFOSR International Conference on DNS/LES* (ed. C. et al. Liu), , vol. 2. Columbus, OH.: Greyden Press.

Strelets, M. 2001 Detached eddy simulation of massively separated flows. *AIAA 2001-0879* .

Terekhov, V.I., Kalinina, S.V. & Mshvidobadse, Y.M. 1997 Heat transfer coefficient and aerodynamic resistance on a surface with a single dimple. *J. of Enhanced Heat Transfer* **4**, 131–145.

Turnow, J., Kornev, N., Isaev, S. & Hassel, E. 2010 Vortex mechanism of heat transfer enhancement in a channel with spherical and oval dimples. *Heat and Mass Transfer* **47** (3), 301–313.

Viswanathan, A.K. & Tafti, D.K. 2006 Detached eddy simulation of flow and heat transfer in fully developed rotating internal cooling channel with normal ribs. *Int J. of Heat and Mass Transfer* **27**, 351–370.

Voskoboinick, V., Kornev, N. & Turnow, J. 2013 Study of near wall coherent flow structures on dimpled surfaces using unsteady pressure measurements. *Flow, Turbulence and Combustion* **90**, 709–722.

Wang, Z., Yeo, K. & Khoo, B.C. 2003 Numerical simulation of laminar channel flow over dimpled surface. *16th AIAA Computational Fluid Dynamics Conference 23-26* **AIAA 2003-3964**.