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OSCILLATING DISCS FOR TURBULENT DRAG REDUCTION

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The modification to a turbulent channel flow subjected to sinusoidal oscillations of flush-mounted discs is investigated through direct numerical simulations (DNS). The discs are positioned on the channel walls in a square-packing scheme, as shown in figure 1. Neighbouring discs in the streamwise (x) direction have opposing sense of rotation, neighbouring discs in the spanwise (z) direction have the same sense of rotation. The strategy is an extension to that of Ricco & Hahn (2013), wherein the disc motion is steady. The Reynolds number is R_τ=180, based on the friction velocity of the stationary-wall case and the half-channel height. The effect of the disc forcing is analysed by separating the total flow in channel $\mathbf{u}$ as $\mathbf{u}$=$\mathbf{u}_m$+$\mathbf{u}_d$+$\mathbf{u}_t$. Here, $\mathbf{u}_m$ is the mean flow averaged in time and in the homogeneous directions, $\mathbf{u}_d$ is the disc flow, and $\mathbf{u}_t$ is the turbulent component. Particular attention is paid to the effect of the disc parameters on the performance quantities, i.e. the skin-friction drag reduction $\mathcal{R}$, the power spent to actuate the discs $\mathcal{P}_{sp,t}$, and the net power savings $\mathcal{P}_{net}$. The skin-friction drag reduction is given by $\mathcal{R}(\%)$=100$(C_{f,s}$−$C_f)/C_{f,s}$, where C_f is the skin-friction coefficient and the s subscript denotes the stationary-wall reference case. The power spent to actuate the discs is expressed as a percentage of the power spent to drive the fluid in the streamwise direction, and is given by

$$\mathcal{P}_{sp,t}(\%) = -\frac{100R_p}{R_\tau^2 U_b}\left(u_d \widehat{\frac{\partial u_d}{\partial y}}\bigg|_{y=0} + w_d \widehat{\frac{\partial w_d}{\partial y}}\bigg|_{y=0} \right), \quad (1)$$

where U_b is the bulk mean flow, R_p is the Poiseuille Reynolds number, and the $\hat{\cdot}$ notation refers to an average in time and the homogeneous directions. The net power saving $\mathcal{P}_{net}$ is then the difference between the power saved due to the disc actuation and the power spent maintaining their motion, defined as $\mathcal{P}_{net}(\%)$=$\mathcal{R}(\%) - \mathcal{P}_{sp,t}(\%)$. Full details of the technique have been published in Wise & Ricco (2014). The primary effect of the oscillating disc forcing is the sustained reduction in skin-friction drag, which reaches a maximum of $\mathcal{R}$=20%. Notable time modulation of this drag reduction is shown, in particular for cases which achieve high $\mathcal{R}$. The amplitude of these modulations is dependent on the oscillation period. This significant modulation which is not apparent for spanwise oscillating-wall forcing is likely to be due to the discs forcing the wall turbulence in the streamwise direction. The wall-shear stress modulation has a period equal to half of the disc oscillation period, due to symmetry of the unsteady forcing with respect to the streamwise direction.

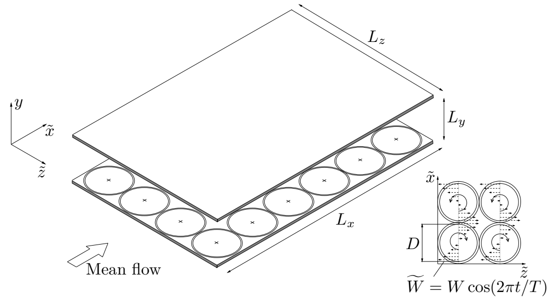
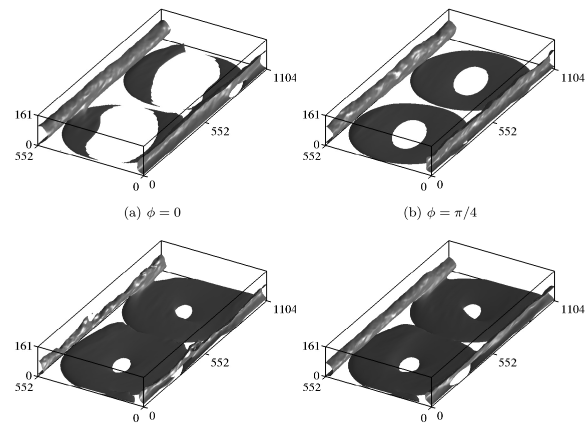


Figure 1: Schematic of the flow domain showing the positioning and sense of rotation of the disc actuators



(a) $\phi = 0$ (b) $\phi = \pi/4$ (c) $\phi = \pi/2$ (d) $\phi = 3\pi/4$

Figure 2: Disc flow visualizations of q=$\sqrt{u_d^2 + w_d^2}$ at different phases ϕ of the disc oscillation. u_d and w_d are the space and time-ensemble averaged disc flow in the stream and spanwise directions.

The Fukagata-Iwamoto-Kasagi (FIK) identity (Fukagata *et al.* , 2002) has been extended to take into account the oscillating disc flow. The modified identity is given by

$$C_f = \frac{6}{U_b Re_p} - \frac{6}{U_b^2}\left[(1-y)\left(\widehat{u_t v_t} + \widehat{u_d v_d}\right)\right]_g, \quad (2)$$

where the $[\cdot]_g$ notation indicates a volume average. Using this modified identity it is possible to quantify the contributions to drag reduction coming either from changes to the turbulent Reynolds stresses relative to the stationary wall case, or from additional time-averaged disc Reynolds stresses. These contributions to drag reduction are denoted by $\mathcal{R}_t$ and $\mathcal{R}_d$ respectively, and are given by

$$\mathcal{R}_t(\%) = 100\frac{R_p\left[(1-y)\left(\widehat{u_t v_t} - \widehat{\langle u_{t,s} v_{t,s}\rangle}\right)\right]_g}{U_b - R_p\left[(1-y)\widehat{\langle u_{t,s} v_{t,s}\rangle}\right]_g}, \quad (3)$$

$$\mathcal{R}_d(\%) = 100\frac{R_p\left[(1-y)\,\widehat{u_d v_d}\right]_g}{U_b - R_p\left[(1-y)\widehat{\langle u_{t,s} v_{t,s}\rangle}\right]_g}. \quad (4)$$

Figure 2 shows visualizations of the disc flow at different phases, ϕ, of the oscillation period. For ϕ=0 the disc-tip velocity is zero, and for $\phi{=}\pi/2$ the disc-tip velocity is at its maximum value. The oscillating-disc forcing, like the steady disc forcing of RH13, creates streamwise-elongated tubular structures between the discs. These structures are largely unaffected by the oscillation of the discs and are only weakly modulated in time. Thin circular patterns are also evident above the surface of the discs. The large time modulation of the circular patterns is expected as they are directly related to the disc motion. Through analysis of the time-averaged disc flow it is shown that the tubular interdisc structures are the sole contributor to the disc-flow Reynolds stresses term occurring in the modified FIK identity, and that they have a beneficial effect on the drag reduction.

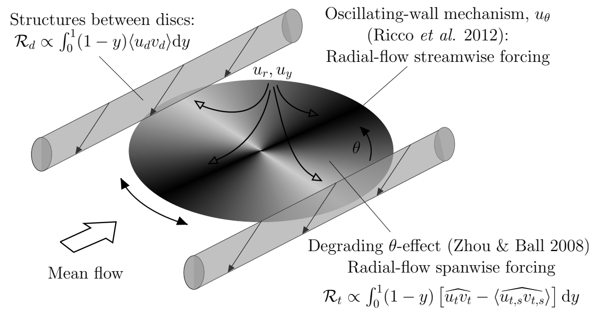


Figure 3: Schematic of the two mechanisms responsible for the drag reduction from the oscillating discs. The mechanism related to the attenuation of the turbulent Reynolds stresses is termed $\mathcal{R}_t$, whilst the second mechanism caused by the structures occuring between discs is termed $\mathcal{R}_d$.

The separated mechanisms contributing to drag reduction are shown schematically in figure 3 and are further analysed with respect to the disc forcing parameters. Through aid from the laminar solution of Rosenblat (1959) it is shown that the contribution due to the attenuation of the turbulent Reynolds stresses is dependent on the penetration thickness of the disc boundary layer δ. $\mathcal{R}_t$ vs. δ is presented in figure 4 (left). This mechanism can be thought of as analogous to the one proposed by Ricco *et al.* (2012) for oscillating-wall forcing. There are however key differences, notably the radial and wallward flows generated by the disc motion. As documented by Zhou & Ball (2008), oscillations of the wall which are not spanwise oriented achieve less than optimal drag reduction. The decay in the performance of this mechanicsm is therefore indicated in figure 3 by the gradiated shading on the disc surface. The drag reduction from the second mechanism comes from the interdisc structures which persist throughout the disc oscillation. The contribution to drag reduction from these structures scales linearly with a function of the disc-tip velocity and oscillation period, shown in figure 4 (right). This is a result expected from the Rosenblat laminar flow solution. The linear fit for the second mechanism is excellent when the disc parameters are scaled using outer units. This therefore means that the interdisc structures are not influenced by the dynamics of the near-wall turbulence.

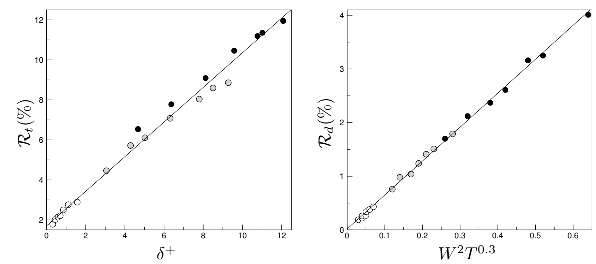


Figure 4: Left: $\mathcal{R}_t$, the contribution to drag reduction due to turbulent Reynolds stress attenuation, vs. δ^+, the penetration depth. Right: $\mathcal{R}_d$, the contribution to drag reduction due to the disc-flow Reynolds stresses, vs. $W^2T^{0.3}$.

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