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Vortex Dynamics within a Bladed Structure in Mixed Convection

J. F. Torres[1], **F. Ghanadi**[2], **M. Arjomandi**[2] **and J. Pye**[1]

[1]Research School of Engineering, Australian National University,
Canberra, ACT 2601, Australia

[2]School of Mechanical Engineering, Centre for Energy Technology,
University of Adelaide, Adelaide, South Australia 5005, Australia

Abstract

Mixed convection is generally the dominant form of convective heat transfer in large heated structures at mid- to low temperatures or in small heated structures at high-temperatures, as in solar thermal receivers for concentrating solar power, which is the motivation for the present study. Novel bladed receiver designs have been proposed for reducing thermal emission and improving light trapping. However, convective heat losses from these bladed structures (extended surfaces) may increase compared to non-bladed geometries, which is not desirable as thermal efficiencies may drop. In this study, experiment-validated simulations were used to quantify heat transfer coefficients for a varying blade length and number of blades. The back wall pitch angle θ was fixed to $30°$ and $60°$ with headwind of 6 m/s and wall temperature of $300°$C; the turbulence intensity at the inlet was 1%. A numerical simulation based on the three-dimensional SST k–ω turbulence model was performed in OpenFOAM to determine the heat transfer rates and flow behaviour. For a fixed back wall dimension (300×300 mm), simulations were conducted for varying the blade number (N_b) while fixing the blade length to spacing ratio to $R_{BS} = 1$, and varying R_{BS} for $N_b = 5$. Some interesting vortex dynamics were observed in each case. First, for variable aspect ratio, a transition from a single roll to a two-roll flow between the blade spacing was confirmed at $R_{BS} \approx 1.5$ for both pitch angles of $\theta = 30°$ and $60°$. Second, for variable blade number, a transition to a lid-driven-like convection was observed beyond the blade number of $N_b = 7$, which decreased the heat transfer rate. A larger number of vortices between the blades was responsible for the drop in convective heat transfer coefficient, reaching values even less than the flat case for $N_b > 23$.

Introduction

Mixed convection is one of the two dominant forms of heat losses from large heated structures at mid- to low temperatures, the other form being thermal radiation. Due to its highly nonliner behaviour, mixed convection is generally more difficult to predict than thermal radiation in large structures. For example, in solar receivers for concentrating solar power, the receiver surface heats up to temperatures of $\sim 750°$C and can be exposed to a broad range of natural convection (buoyancy) and forced convection (wind) regimes during its required daylight hours of operation. Conventional external receivers have a rather smooth macro-scale structure, either flat or cylindrical, which are exposed to the open air and subject to relatively high thermal losses. To solve this problem, new bladed receiver designs have been proposed for improving light trapping and reducing thermal emission [8, 4].

Predicting mixed convection from bladed solar thermal receivers is challenging due to their complex geometry, large scales involved, and high temperature gradients where the Boussinesq approximation may not be valid. Due to extended surface effects, convective heat losses from these bladed structures are expected to increase compared to the smooth macro-scale case. Nonetheless, convective heat losses for a bladed geometry may also decrease due to a cavity effect in bladed receivers. Torres et al. [5] showed that, for characteristic pitch and yaw angles, flow separation creates stagnation zones or hot pockets of air between the blades that result in a reduction of convective heat losses [5]. The effects of wind direction (yaw angle) and pitch angle on flat isothermal flat [6] and bladed structures [5], as well as non-isothermal bladed case [3], were reported. However, the effects of a variable bladed geometry on convective heat losses are still not well understood.

The aim of this study is to determine convective heat transfer from an isothermal bladed structure under mixed convection, focusing on a variable geometry, which consists of a solid cuboid with an isothermal wall from which horizontal isothermal blades protrude (at the same wall temperature). The parameters of interest are the blade length to blade spacing ratio (R_{BS}) and number of blades (N_b). Mixed convection effects are investigated under a constant freestream velocity U and wall temperature T_w.

CFD Modelling

A Shear-Stress Transport (SST) $k-\omega$ model [2] for turbulent flow was employed in OpenFOAM to investigate the dependence of the convective heat transfer with the variation of R_{BS} and N_b. The governing equations, treatment of thermophysical properties, and discretisation methods are the same as in [6]. An SST $k-\omega$ model was chosen after testing the discrepancy of three different Reynolds averaged Navier–Stokes (RANS) equations with experimental results in [1] for mixed and forced convection. It was found that SST $k-\omega$ model performs reasonably well in both high and low Reynolds mesh conditions. This was also corroborated in [6].

The computational domain and mesh are shown in figure 1 for a rig orientation of $(\theta, \phi) = (35°, 0°)$. Details of this calculation domain, boundary conditions and mesh refinement study (or grid independence) can be found in [5]. These simulations employed a rather coarse mesh to determine the trend in heat transfer as a function of various parameters (L = 0 in [5]). Moreover, only the headwind configuration (yaw angle $\phi = 0°$) was considered. The blade spacing and blade length for a given structure were constant, as shown in figure 1(c). An automatic mesh generation procedure was devised with OpenFOAMs *blockMesh* and *snappyHexMesh* applications and Gmsh, which is a three-dimensional finite mesh generator. The rig has dimensions of $400 \times 300 \times 80$ mm, and a large side of 400×300 mm has a square section or back wall of 300×300 mm from which the blades protrude. The rig dimensions were chosen to compare with previously published results [6] and ongoing wind tunnel experiments. The aspect ratio R_{BS}, defined as the blade length B to spacing S, was increased by $\Delta R_{BS} = 0.25$ from 0 (flat configuration) to 3, keeping the number of blades constant at $N_b = 5$. N_b was then increased from 3 to 37 for a constant $R_{BS} = 1$. The

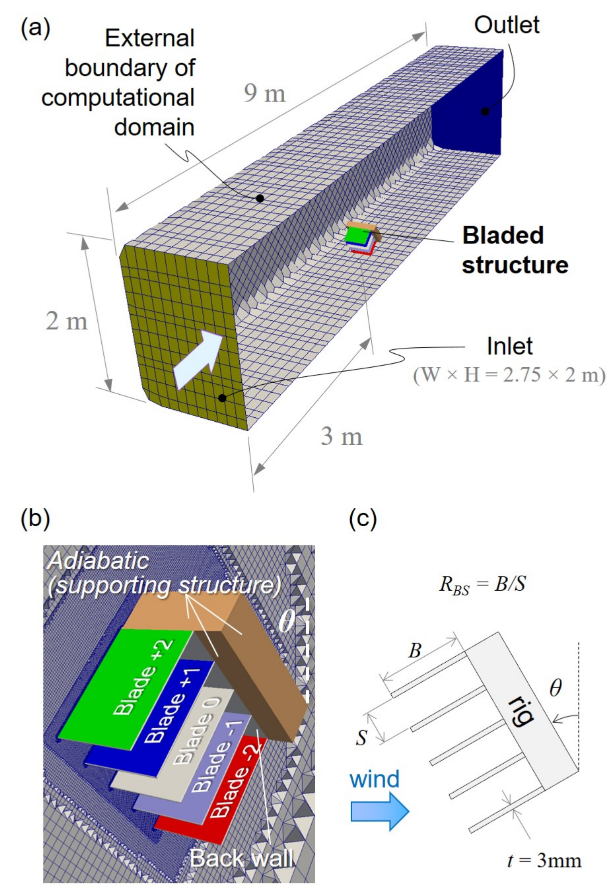

Figure 1: Computational domain for a bladed structure with orientation of $(\theta, \phi) = (35^\circ, 0^\circ)$, $R_{BS} = 3$ and $N_b = 5$. (a) Halved calculation domain indicating the location of the bladed structure; (b) mesh around the bladed structure; (c) geometry.

blade thickness was fixed to $t = 3$ mm in all simulations.

The external boundaries of the calculation domain are indicated in figure 1(a), which have the dimensions of the wind tunnel, large enough to not influence the results. The wind speed at the inlet was set to 6 m/s. The effect of the wind speed is discussed in [5, 6]. The outlet pressure was fixed to the atmosphere pressure and at other boundaries set with a fixed flux pressure. The inlet temperature was an ambient temperature of 25°C. The isothermal sides of the bladed structure, i.e. the blades and back walls, were set to $T_w = 300$°C; the remaining wall surfaces were set adiabatic (supporting structure indicated in figure 1(b)). A wall function with Neumann boundary conditions for the turbulence kinetic k, and a wall constraint on the turbulence specific rate of dissipation ω were used. k at the inlet was determined by the turbulence intensity TI, which was fixed to 1% in this study. The effects of varying the turbulence intensity in the ranges of $1 \leq TI \leq 5\%$ was discussed in [6].

The results are presented in terms of the convection heat transfer rate per unit area per unit temperature, i.e. the heat transfer coefficient h defined as $h = \dot{Q}_{conv} / (T_w - T_\infty) / A$; where $\dot{Q}_{conv}$ is the total heat transfer rate from the bladed structure determined by the CFD simulations, T_∞ is the ambient temperature, and A is the isothermal surface area.

Results

The CFD results are presented in terms of h and corresponding flow patters on the mid- vertical plane, which is the plane of symmetry because the yaw angle is $\phi = 0^\circ$ (headwind), as the parameters of interest are varied, i.e. R_{BS} and N_b.

Variable aspect ratio

The heat transfer coefficient h of the bladed structure is shown in figure 2 as a function of R_{BS} for a constant value of $N_b = 5$ at $\theta = 30^\circ$ and 60° (other parameters are indicated in the caption). The isothermal surface area, also plotted, is linearly proportional to the aspect ratio because N_b was constant.

The results at $\theta = 30^\circ$ show a weak dependence of h on R_{BS}, varying between ~ 30 and 34 W/m^2 K. In contrast at $\theta = 60^\circ$, h is more strongly dependent on R_{BS}, varying between 40 to 28 W/m^2 K (at $R_{BS} \approx 0.75$ and 3, respectively). Note, however, that the mesh used in this study was rather coarse so an error of $\sim 4\%$ can be expected [5]. Interestingly, for both pitch angles, two characteristic aspect ratios seem to exist at which the trend of h changes, i.e. at $R_{BS} \approx 0.75$ and 1.5. These variations in h can be associated with the flow behaviour, which is depicted in figure 3. Here, the flow pattern on the mid-vertical plane of symmetry is plotted as R_{BS} is increased at $\theta = 30^\circ$ and 60°. The number of blades, wind speed, wind direction, wall and ambient temperatures are fixed parameters listed in the caption.

For $R_{BS} \leq 0.5$, there was only one vortex within each blade spacing, as indicated in figure 3. The vortices seem to follow an elliptical pattern that adjusts itself to the rectangular blade spacing. However, there was an exception for $\theta = 30^\circ$ at which the first compartment next to the leading blade has a very weak flow, almost stagnant; in contrast, at $\theta = 60^\circ$, a vortex was observed. This explains why there is a significantly larger h at $\theta = 60^\circ$ than $\theta = 30^\circ$ for small R_{BS} (see figure 2).

Concerning the flow in the first blade spacing, the dynamics of the vortices for both pitch angles significantly shifts for $R_{BS} \geq 0.75$: at $\theta = 30^\circ$ (shown in figure 3(a)) the flow gradually appears, whereas at $\theta = 60^\circ$ (shown in 3(b)) the flow disappears resulting in a stagnant flow, as indicated in figures 3(b) and (c). This behaviour is associated with the flow separation

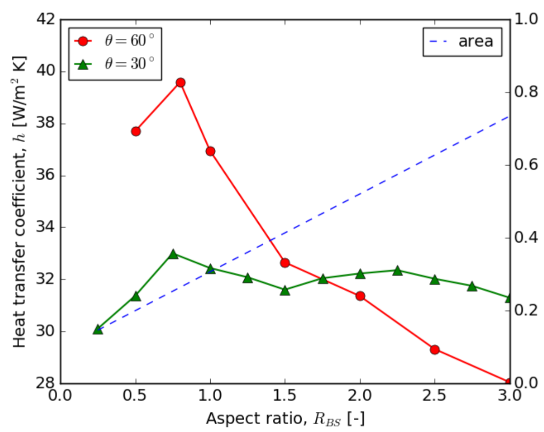

Figure 2: Heat transfer coefficient (left ordinate) of the bladed structure as a function of the aspect ratio R_{BS} for constant values of $N_b = 5$, $U = 6$ m/s, $T_w = 300$°C (573 K) and $\phi = 0^\circ$ (headwind) at two pitch angles of $\theta = 30^\circ$ and 60°. The surface area of the isothermal walls (right ordinate) is plotted by a dashed line.

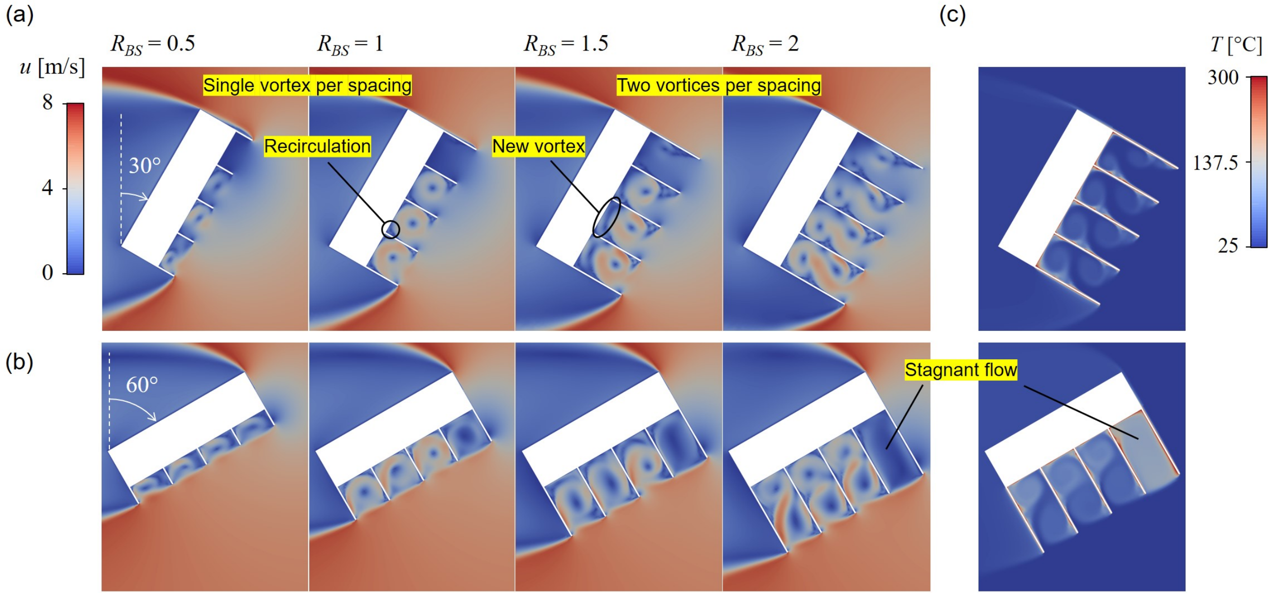

Figure 3: Flow pattern for constant values of $N_b = 5$, $U = 6$ m/s, $T_w = 300°C$ (573 K) and $\phi = 0°$ (headwind). The freestream velocity is from right to left. The velocity contours are plotted on the symmetric plane $y = 0$. (a) Velocity contours at $\theta = 30°$ as R_{BS} is increased; note the transition from one-roll to two-roll flow. (b) For $\theta = 60°$, note the lack of vortices in the front blade spacing. (c) Temperature contours for $R_{BS} = 2$; there is a the stagnation of hot air in the front blade spacing at $\theta = 60°$ due to the lack of airflow.

and reattachment length from the blade front edge. For steep pitch angles, the separation length increases [6], surpassing the blade spacing and hence creating hot pockets of air (discussed in [5] for variable pitch angle). These results suggest that the separation length also depends on R_{BS} because for small values of R_{BS} at $\theta = 60°$ the separation length did not exceed the blade spacing, as suggested by figure 3(b) for $R_{BS} = 0.5$. Therefore, the supporting structure might have influenced the results.

Another interesting observation is a flow bifurcation at $R_{BS} \approx 1.5$, where the flow changes from a single vortex (one-roll) to two vortices (two-rolls) per spacing, as indicated in figure 3. Flow bifurcation for varying aspect ratio has been reported by Torres et al. [7] for natural convection in tilted rectangular enclosures. In this study, flow bifurcation was observed at roughly the same aspect ratio for both pitch angles of 30° and 60°, and was thought to be responsible for the trend change of heat transfer coefficient at $R_{BS} \approx 1.5$ as this parameter was increased (see figure 2).

Variable number of blades

The heat transfer coefficient h and surface area A of the isothermal bladed structure are shown in figure 4 as a function of N_b for $R_{BS} = 1$ at $\theta = 30°$ and 60°. The number of blades was increased in odd numbers to keep a centred blade, i.e. Blade 0 indicated in figure 1(b).

For $N_b < 11$, the heat transfer at $\theta = 60°$ was larger than at 30°, which is consistent with the flow behaviour shown in figure 2 at $R_{BS} = 1$. For $N_b \geq 11$, in contrast, both pitch angles had almost the same value of h. For $N_b \geq 15$, $\theta = 30°$ produced a slightly larger h than 60°. Hence, only the flow behaviour of one pitch angle ($\theta = 30°$) is discussed below. For $N_b > 7$, h decreased almost linearly with N_b. Interestingly, for $N_b \geq 25$, the heat transfer coefficient of the bladed structure became less than the flat non-bladed plate (previously reported in [6] and [5]). It is worth mentioning that the heat transfer rate $\dot{Q}_{conv}$ in watts was still larger than the bladed geometry due to a larger surface area (for the flat case $A = 0.3 \times 0.3 = 0.09$ m^2).

The large decrease of convective heat transfer for $N_b > 7$ is associated with a flow behaviour that exhibits a transition to a flow similar to a lid-driven cavity flow. For a small blade number, e.g. $N_b = 3$ in figure 5(a), a large portion of the airflow exits through the side apertures (normal to the plane plotted in the figure). For a large blade number, the vortices within the blades reduce their velocity and the external flow 'slides' through the front aperture, as indicated in the inset of figure 5(a). This reduction in flow rotation increases the temperature between the blades, which is a phenomenon shown in 5(b). Since the mass transfer through the aperture was reduced, and the thermal conductivity of air is rather small, then the heat exchange between

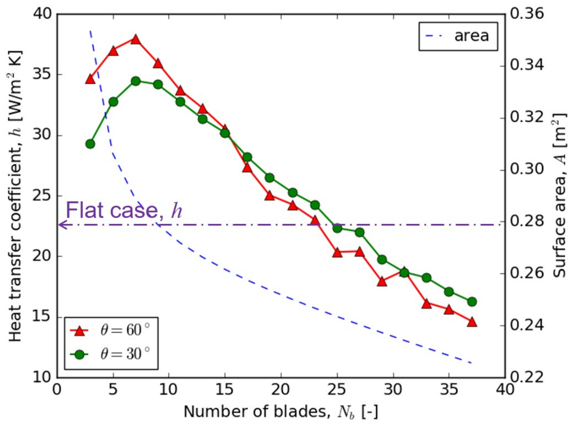

Figure 4: Effect of varying number of blades for constant values of $R_{BS} = 1$, $U = 6$ m/s, $T_w = 300°C$ (573 K) and $\phi = 0°$ (headwind) at two pitch angles of $\theta = 30°$ and 60°. Heat transfer coefficient (left ordinate) and surface area (right ordinate) of the bladed structure are plotted.

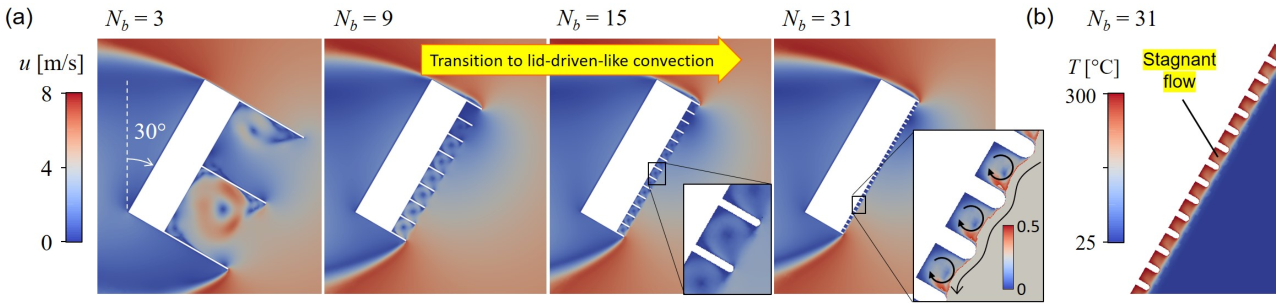

Figure 5: Flow pattern as N_b is increased for constant values of $R_{BS} = 1$, $U = 6$ m/s, $T_w = 300$°C (573 K), $\theta = 30°$ and $\phi = 0°$ (headwind). (a) Velocity contour on the mid-vertical plane of symmetry. The inset of $N_b = 31$ highlights the low values of velocity within the blades and the lid-driven-like behaviour of convection. (b) Close-up of the temperature field between the blades for $N_b = 31$.

the bladed structure and the external air was greatly reduced, resulting in the decrease of convective heat transfer observed in figure 4.

Depending on the application, the design of bladed structures can be tailored to enhance or reduce heat transfer between the bladed structure and its surroundings. If a larger heat exchange per unit area per unit temperature is desired, such as in cooling applications, then few blades (figure 4) and moderate aspect ratio is a suitable design (figure 2). On the other hand, if a lower convective heat transfer coefficient is desired, such as in solar thermal receivers, then a larger number of blades and large aspect ratios would be desired. It is worth mentioning that in most applications, the blades are non-isothermal and the exact heat exchange rate may differ from those reported in this study, although trends should hold for small temperature differences within the structure. Furthermore, at high temperatures, thermal losses usually become significant so a simultaneous consideration of convection and radiation is necessary.

Conclusions

A bladed isothermal structure under mixed convection was investigated motivated by the design of a new solar thermal receivers with blades that improve optical performance. The bladed structure had horizontal blade edges. Focus was given to the effects of varying geometry on the heat transfer coefficient h, i.e. varying the aspect ratio R_{BS} (blade length to blade spacing) while fixing the blade number to $N_b = 5$ and varying the blade number N_b for a constant aspect ratio of $R_{BS} = 1$. The conclusions are as follows.

1. As R_{BS} was increased, h initially increased and then significantly decreased for a large pitch angle of $\theta = 60°$. The large drop ($\sim 30\%$) was associated with a stagnation flow in the front blade spacing. For moderate values of $\theta = 30°$, there was no stagnant air within the blades and hence a mild dependence of h on R_{BS} was obtained.

2. As R_{BS} was increased, a flow bifurcation from a one-roll vortex to two-roll vortices per blade spacing was observed at $R_{BS} \approx 1.5$ for both $\theta = 30°$ and $60°$. This bifurcation slightly modified the dependence of h on R_{BS}.

3. As N_b was increased, h initially increased reaching a maximum value at $N_b = 7$ and then monotonically decreased dropping by $\sim 40\%$ from its maximum value. The drop in h occurred even though the surface area decreased as N_b increased, and h reached values even lower than the flat plate when $N_b \leq 25$ for both pitch angles of 30° and 60°.

4. As N_b was increased, a transition in flow behaviour was observed from a configuration where airflow mainly exited through the side apertures to a configuration where the airflow 'slid' along the front aperture. For a large N_b, low rotation vortices appeared between the blades, which acted as a 'thermal barrier' that decreased convective heat transfer.

5. Depending on the application, the bladed geometry can be tuned to enhance or decrease convective heat transfer. The values of h reported here aid the design of bladed structures where mixed convection is important.

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References

[1] Harris, J.R., Lance, B.W. and Smith, B.L., *Journal of Fluids Engineering*, **138**, 2016, 011401.

[2] Menter, F.R., Kuntz, M. and Langtry, R., *Turbulence, Heat and Mass Transfer*, **4**, 2003, 625–632.

[3] Nock, I., Logie, W., Coventry, J. and Pye, J., *Proceedings of the 2016 Asia-Pacific Solar Research Conference*, December 2016, Canberra, Australia.

[4] Pye, J., Coventry, J., Ho, C., Yellowhair, J., Nock, I., Wang, Y., Abbasi, E., Christian, J., Ortega, J. and Hughes, G., *AIP Conference Proceedings*, **1850**, 2017, 030040.

[5] Torres, J.F., Ghanadi, F., Arjomandi, M. and Pye, J., *Proceedings of the 11th Australasian Heat and Mass Transfer Conference*, July 2018, Melbourne, Australia.

[6] Torres, J.F., Ghanadi, F., Nock, I., Arjomandi, M. and Pye, J., *International Journal of Heat and Mass Transfer*, **119**, 2018, 418–432.

[7] Torres, J.F., Henry, D., Komiya, A., Maruyama, S. and Ben Hadid, H., *Physical Review E*, **88**, 2013, 043015.

[8] Yellowhair, J., Ho, C. K., Ortega, J. D., Christian, J. M. and Andraka, C. E., *High and Low Concentrator Systems for Solar Energy Applications X*, **9559**, 2015, 95590A.