Numerical Investigation on the Performance of Fin and Tube Heat Exchangers using Rectangular Vortex Generators

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Abstract. In the present work, a 3-D numerical investigation has been performed to explore the effect of attack angles on the thermal-hydraulic performance of fin and tube heat exchanger (FTHE) using rectangular winglet pairs (RWPs). RWPs are placed adjacent to the tubes and three attack angels are considered for the study i.e. 5°, 15° and 25°. The effect of attack angles are examined on the heat transfer characteristics as well as in pressure drop penalty with airside Reynolds number Re_a ranges from 500 to 900. Two performance evaluation criteria namely PEC1 i.e. area goodness factor (j/f) and PEC2 i.e. heat transfer rate per unit fan power consumption (Q/P_f) are considered for the performance evaluation. Furthermore, MOORA method is applied to obtain the performance order of FTHE configurations by taking PEC1 and PEC2 as beneficial attributes and fan power P_f as a non-beneficial attribute, keeping equal importance to each attribute. The results show that 5° attack angle provides the better performance in terms of PEC1 as heat transfer coefficient is increased by 27.70% at Re_a=500 and 32.73% at Re_a=900 respectively with 13.01% increased pressure drop penalty at Re_a=500 and 14.26% at Re_a=900 respectively. In terms of PEC2, though the 5° attack angle provides the high values of Q/P_f factor among the 15° and 25° attack angles, but it is found insignificant to replace the baseline configuration i.e. plain fin and tube heat exchanger configuration without vortex generators. Moreover, in MOORA optimization analysis also, it is found that 5° attack angle provides the better thermal-hydraulic performance.

Keywords: Fin and tube heat exchanger; Vortex generators; Heat transfer enhancement; MOORA method

INTRODUCTION

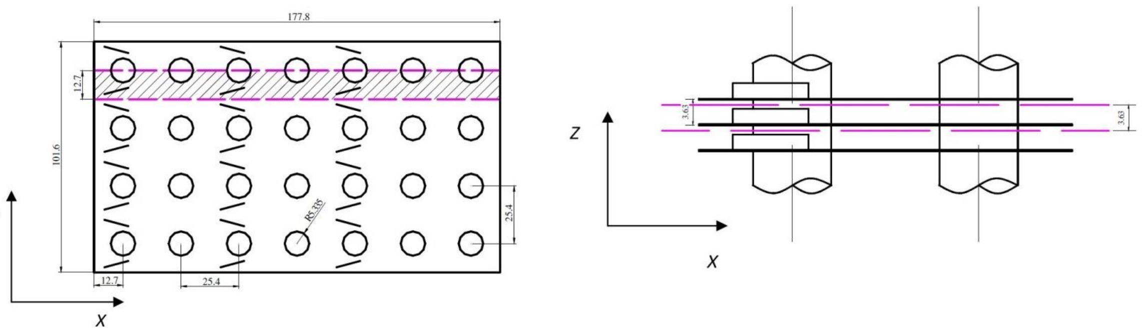
Fin-and-tube heat exchangers have widespread use; either individually or as components of a large thermal system, in a wide variety of commercial, industrial and household applications, e.g. power generation, refrigeration, ventilating and air-conditioning systems as well as in car radiators. As the air side thermal resistance is the most dominant part representing 90% of the total thermal resistance because of the low heat transfer coefficient and thermo-physical properties of air. Therefore, efforts to improve the performance of these heat exchangers mainly focus on the air-side surfaces. Placing strategy of vortex generator, attack angle, fin pitch, and vortices direction affect the performance of heat exchanger and thus become the interesting field of research and optimization.

A number of experimental as well as numerical research works has been done to explore the effect of vortex generators on the performance of fin and tube heat exchangers. Jacobi and Shah [1] presented a review on heat transfer enhancement by surface modification for deriving the well zone for further research in the field of using longitudinal vortex generators. Flat tube heat exchanger with vortex generators has much higher heat transfer value than conventional fin-circular tube heat exchanger [2]. Placing strategies of vortex generators is also explored and compared with fin and circular tube heat exchanger to obtain the better performance [3]. It is found in the past studies that using vortex generators with oval tubes augmented the heat transfer rate with moderate pressure drop [4]. It is also pointed out that winglets in staggered arrangement bring larger heat transfer enhancement than in in-line arrangement. Several studies are also available which demonstrate the clear promise of using vortex generators with augmented surfaces i.e. wavy, louvered and slit fins [5]. Furthermore, in implementation of vortex generators, placing strategy and angle of attack also affects the performance of heat exchanger, He *et al.* [6] and Sinha *et al.* [7, 10] performed numerical analysis to obtain the position and angle of attack in implementation of vortex generators. Some non-conventional type vortex generators i.e. full dimples, semi dimples, wavy up and wavy down type rectangular winglet vortex generators also studied and compared with conventional vortex generators to obtain high heat transfer coefficient [8-9]. Oneissi *et al.* [11] performed numerical analysis with a novel vortex generator named inclined projected winglet pair (IPWP) which shows superior performance relative to the conventional delta winglet pair (DWP).

It is found from the literature review that performance evaluation of heat exchangers has been done either in terms of Colburn factor j and Friction factor f or in terms of area goodness factor j/f in most of the work. As area goodness factor criterion is only based on providing minimal frontal area of heat exchanger, some other performance criterion should also be considered in evaluating the performance which can give the heat transfer rate per unit fan power consumption explicitly. In the present study RWPs are placed adjacent to the tubes and three attack angels are considered i.e. 5°, 15° and 25°. Two performance evaluation criteria namely PEC1 i.e. area goodness factor (j/f factor) and PEC2 i.e. heat transfer rate per unit fan power consumption (Q/P_f factor) are considered in the present study. Furthermore, MOORA method [12] is applied to obtain the performance order of FTHE configurations by taking PEC1 and PEC2 as beneficial attributes and fan power P_f as non-beneficial attribute, keeping equal importance to each attribute.

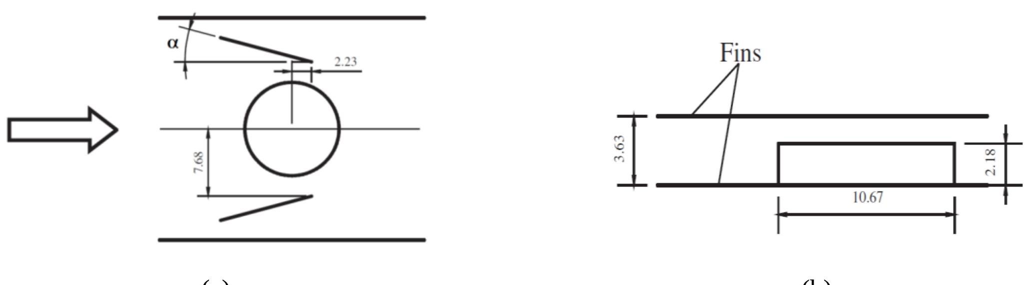
MODEL DESCRIPTION

In this study, fin and tube heat exchanger with RWPs mounted symmetrically in common flow up orientation on the fin surface adjacent to the tubes is considered. Figure 1(a) and 1(b) shows the top view and side view of the of fin-and-tube heat exchanger with RWPs respectively. Because of the geometry character of symmetry of the fin-and-tube heat exchanger, the region (in X-Y plane) outlined by the dashed lines in Fig. 1 (a) is selected as the computational domain. Two adjacent fins form a channel of height H = 3.63 mm, width B = 12.7 mm, and length L = 177.8 mm. Diameter of first tube is D = 10.67 mm and is located at X = 12.7 mm from the channel. Both the transverse tube pitches P_s and the longitudinal tube pitch P_l are 25.4 mm. To maintain the uniformity of the inlet velocity, the actual computation domain is extended by 5H at the inlet and to ensure a recirculation-free flow and the domain is extended by 30H at the outlet. Height of the winglet is equal to 60% of the channel height (H). Figure 2 shows the dimensions of rectangular winglets and their placement with respect to the tube.



(a) (b)

FIGURE 1. Computational domain: (a) top view of the computational domain; (b) side view of the computational domain (All dimensions are in mm)[6]



(a) (b)

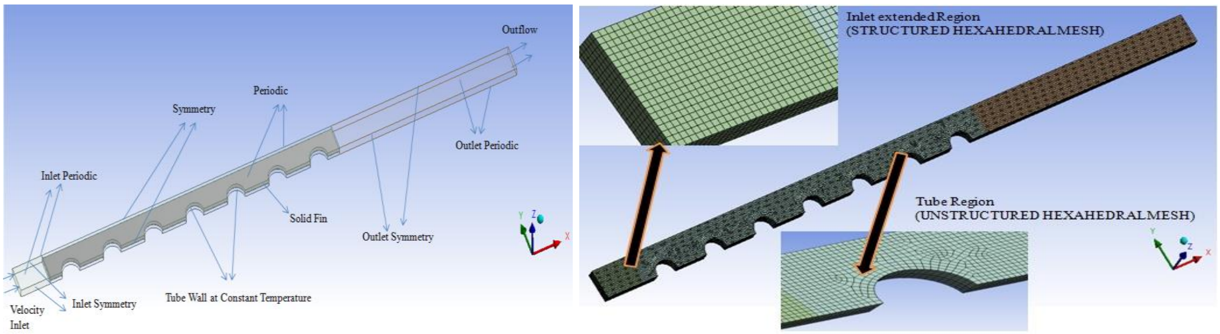
FIGURE 2. Dimensions of vortex generators and their placement with respect to the tube (in mm): (a) top view of RWPs placed at adjacent to the tube; (b) side view of the RWPs [6]

GOVERNING EQUATIONS AND BOUNDARY CONDITIONS

In the present study, the fluid is considered as incompressible with constant properties. Generation of longitudinal vortices is a quasi-steady phenomenon. As a result, due to the low inlet velocity and the small fin pitch, the flow in the channel of the compact heat exchanger is assumed to be laminar and steady. Fin thickness and heat conduction in the fins and vortex generators are taken into consideration. The distribution of the temperature over the fins can be determined by solving the conjugate heat transfer problem in the computational domain. Figure 3 shows the 3-D model of interest and a detailed description of governing equations and boundary condition can be seen in He *et al.* [6].

NUMERICAL METHODS

Due to symmetry and periodicity of geometry only one eighth part of fin plate with tube is modeled in Design Modeler ANSYS 14.0. At inlet and outlet extended region a structured hexahedral mesh is generated and near the tube wall an unstructured hexahedral mesh is generated as shown in Figure 3(b). Grid has been generated by workbench ANSYS 14.0 (ICEM CFD). The governing equations along with the boundary condition equations are solved in FLUENT ANSYS 14.0. The convective terms in governing equations for momentum and energy are discretized with the second order upwind scheme. The coupling between velocity and pressure is performed with SIMPLE algorithm.



(a) (b)

FIGURE 4. computational domain (a) 3D model of with boundary conditions (b) Mesh generation for circular inline arrangement (Baseline case)

VALIDATION OF NUMERICAL RESULTS

In order to have a solution independent from grid system effect, three grid systems having 850950, 1179341 and 1681899 elements have been investigated for the circular tube with inline arrangement. The predicted averaged Nusselt number for three grid systems are calculated as shown in Figure 5. For reduction in computational efforts grid system having 11799341 elements is selected finally and similar validations are also carried out for other cases.

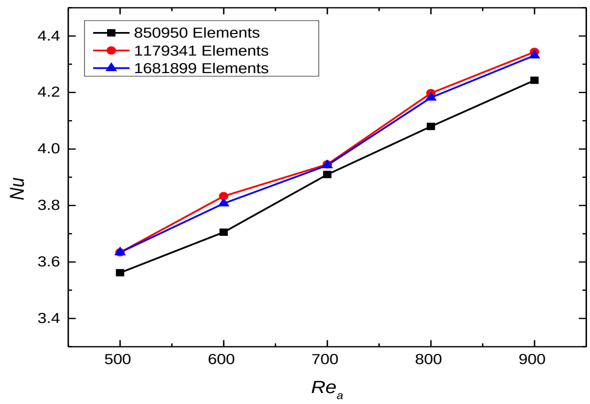


FIGURE 3. Variation of the predicted Nusselt number with different grid number

In order to validate the reliability of numerical method, the results are compared with the available experimental results. The numerical simulation is conducted for a fin-and-tube heat exchanger with the same geometrical configurations as presented in Joardar and Jacobi [3]. The predicted air side heat transfer coefficient and pressure drop are compared with experimental results and represented in Figure 7. The inlet air velocity ranges from 1.04 m/s to 1.88 m/s and the corresponding Re_a number ranges from 500 to 900. Predicted results slightly differ from experimental results as the numerical solution is close to ideal condition. Moreover, in experimental study, the leakages of test specimen and contact resistance are also inevitable. A good agreement is found in the results which show the reliability of numerical model for further predictions.

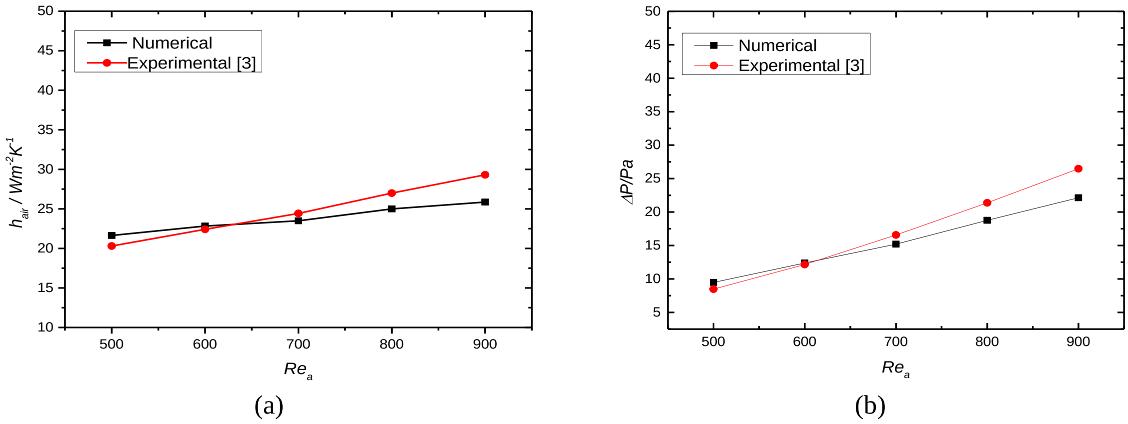


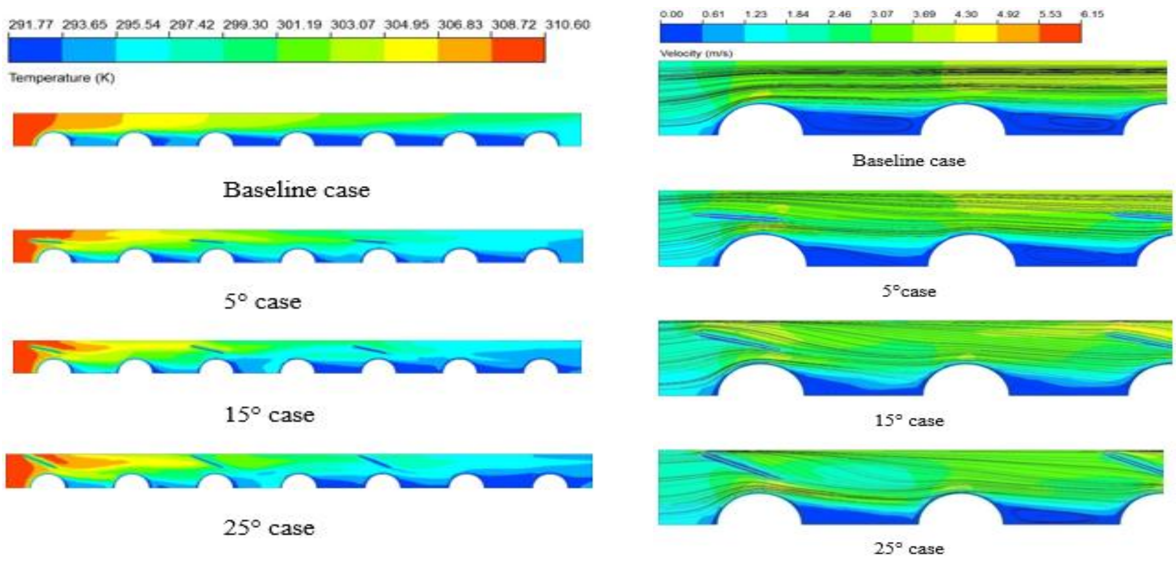
FIGURE 7. Comparison of present numerical result with existing experimental result [3], (a) effect of Re_a on h_{air} (b) effect of Re_a on ΔP

RESULTS AND DISCUSSION

The parameters of simulation are summarized as follows:
Inlet temperature: 310.6 K; Re_a ranges from 400 to 900; tube wall temperature: 291.77 K.

As the incoming flow move toward the RWPs in the flow course, pressure difference between the upstream side and the downstream side of the RWPs facilitate the generation of longitudinal vortices. The axes of the vortices are parallel with the main flow direction, thereby capturing the fluid from the near wall and swirl it to the core of the vortices. This strong swirling flow interrupts the developing of thermal boundary layer on the fin surface and thus augments heat transfer rate. Furthermore, nozzle like passage adjacent to the tube because of RWPs, accelerates the fluid flow towards the wake zone to increase the heat transfer rate because of separation delay. As the attack angle increases, wake zone also reduces and enhance the heat transfer characteristics that can be easily seen in Figure 8 (a) and (b). The improved mixing between tube wake and main flow by the swirling longitudinal vortices, the temperature gradient increases with increase in attack angle and thus enhances heat transfer. Longitudinal vortices induced by the RWPs are the main cause of increased pressure drop, thus at 25° attack angle pressure drop is increased more significantly. Figure 9 shows the variations in heat transfer coefficient and pressure drop at different attack angles. It can be seen that 25° attack angle provides high heat transfer rate with high penalty of pressure drop, whereas in 5° attack angle, pressure drop penalty is less with comparatively lower heat transfer rate. Therefore to obtain the best configuration, performance is evaluated by PEC1. Variation of PEC1 is depicted in Figure 9(b) that shows high thermal hydraulic performance by 5° attack angle case as heat transfer coefficient is increased by 27.70% at Re_a=500 and 32.73% at Re_a =900 with 13.01% increased pressure drop penalty at Re_a =500 and 14.26% at Re_a =900. Moreover, the performance of 25° attack angle case is lower than the baseline case as pressure drop penalty is increased by 120% at Re_a = 900.

PEC1 is well accepted performance evaluation criterion to obtain the minimal frontal area of heat exchanger but PEC2 (Q/P_f) provides the heat transfer rate per unit fan power consumption P_f. Vortex generators augment heat transfer characteristics but with some pressure drop penalty and thus increased fan power, which increase the cost of heat exchanger. If the pressure drop is too high using vortex generator then heat transfer rate per unit fan power is lowered significantly and thus use of vortex generators becomes uneconomical. Figure 9 (c) represents the variation in PEC2 at different Re_a. It can be concluded from the variations that at higher attack angles pressure drop increment rate higher so it is insignificant to use vortex generators if the criterion is to increase heat transfer rate per unit fan power consumption.



(a) (b)

FIGURE 8. Distribution of (a) temperature and (b) velocity, on the middle cross-section at $Re_a = 900$

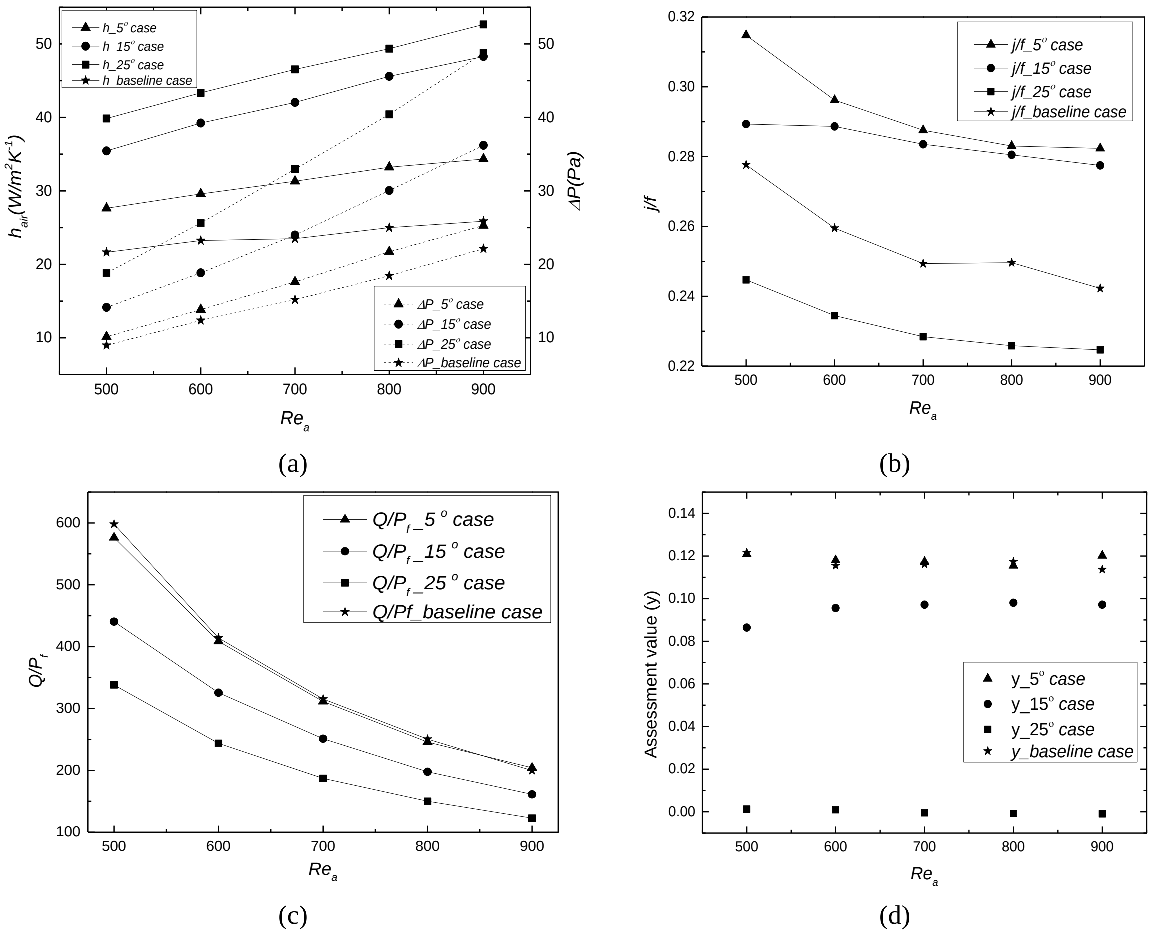


FIGURE 9. Variation of different performance parameters versus Rea (a) hair & ΔP, (b) PEC1 (*j/f*), (c) PEC2 (*Q/Pf*) and (d) Assessment values (y) for different attack angles and baseline case

Furthermore, the results from MOORA analysis [12] as depicted in Figure 9(d), it can be concluded that 5° attack angle configuration provides the high *j/f* and Q/P_f factors at moderate fan power consumption and can be considered as an optimized configuration.

CONCLUSIONS

The major conclusions for the present study are drawn as follows:

- Thermal mixing of the fluid is enhanced by RWPs generated vortices. It also delays the boundary layer separation and reduces wake zone, which enhance the heat transfer. The common-flow-up orientation of the RWPs provides a constricted nozzle-like passage to accelerate the fluid between the RWPs and the aft region of the tube. The accelerated fluid flow delays the boundary layer separation and reduces the tube wake. It also impinges fluid directly on the downstream tube, which results significant augmentation of local heat transfer.
- In comparison to baseline case, heat transfer coefficient is augmented by 27.70-32.73%, 63.76-86.71% and 84.17-103.59% for 5°, 15° and 25° attack angle respectively but with significant pressure drop penalty. However, 5° attack angle provides the increased heat transfer coefficient at moderate pressure drop penalty and thus exhibits better thermal-hydraulic performance.
- Area goodness factor PEC1 (j/f factor) is higher for 5° whereas 25° shows the lowest values of PEC1. Hence, it can be conclude that 5° attack angle provides better thermal hydraulic performance and requires less frontal area. Furthermore, using PEC2 (Q/P_f factor), it is found that though 5° attack angle provides highest heat transfer coefficient and higher j/f factor values but fan power consumption is also increased which shows the limitation of vortex generator implementation.
- MOORA method is applied to obtain the performance order of FTHE configurations by taking PEC1 and PEC2 as beneficial attributes (which are to be maximized) and fan power P_f as non-beneficial attribute (which is to be minimized), keeping equal importance to each attribute. It is found that 5° attack angle provides the better thermal-hydraulic performance in comparison to other considered cases.

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