



Numerical Investigation of Thermo-Hydraulic Performance of Staggered Cam-Shaped Tube Bundles

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Article Info

Research article
Received: 14/09/2026
Revision: 19/09/2026
Accepted: 20/09/2026

Keywords

Cam-shaped tube bundles
Thermo-hydraulic
performance
Performance evaluation
criterion (PEC)
Numerical simulation

Makale Bilgisi

Araştırma makalesi
Başvuru: 14/09/2026
Düzeltilme: 19/09/2026
Kabul: 20/09/2026

Anahtar Kelimeler

Kam şekilli boru demetleri
Termo-hidrolik performans
Performans değerlendirme
kriteri (PEC)
Sayısal simülasyon

Grafik Özet (Graphical/Tabular Abstract)

Thirteen staggered tube-bundle configurations were numerically investigated over $Re = 4000-32000$. All cam-shaped models achieved $PEC > 1$. Model M13 ($AR = 2.63$) showed the best thermo-hydraulic performance, with approximately 28% PEC improvement over the circular baseline. / On üç şaşırtmalı boru demeti konfigürasyonu, $Re = 4000-32000$ aralığında sayısal olarak incelenmiştir. Tüm kam şekilli modeller $PEC > 1$ sağlamıştır. Model M13 ($AR = 2.63$), dairesel referansa göre yaklaşık %28 PEC artışıyla en iyi termo-hidrolik performansı göstermiştir.

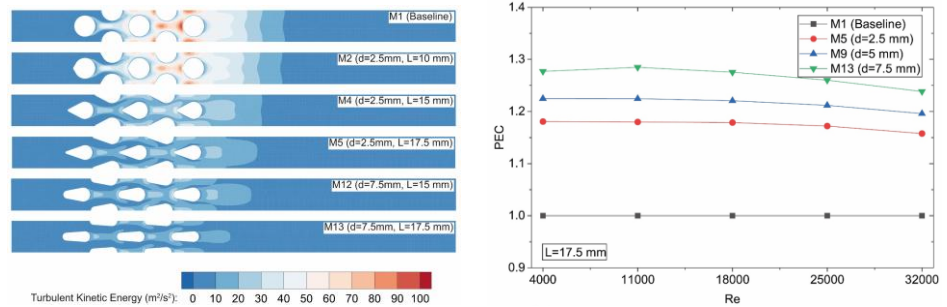


Figure A: Turbulent kinetic energy distributions and PEC variation of the investigated cam-shaped tube bundles. / **Şekil A:** İncelenen kam şekilli boru demetlerinin türbülans kinetik enerjisi dağılımları ve PEC değişimi.

Highlights (Önemli noktalar)

- Thirteen staggered tube-bundle configurations were investigated over $Re = 4000-32000$. / On üç şaşırtmalı boru demeti konfigürasyonu $Re = 4000-32000$ aralığında incelenmiştir.
- All cam-shaped configurations showed lower friction factors and $PEC > 1$ compared with the circular baseline. / Tüm kam şekilli konfigürasyonlar dairesel referansa göre daha düşük sürtünme katsayısı ve $PEC > 1$ göstermiştir.
- Model M13 achieved the highest thermo-hydraulic performance with approximately 28% PEC improvement. / Model M13, yaklaşık %28 PEC artışı ile en yüksek termo-hidrolik performansı sağlamıştır.

Aim (Amaç): This study numerically investigates the effects of leading-edge diameter (d) and center distance (L) on the thermo-hydraulic performance of staggered cam-shaped tube bundles over $Re = 4000-32000$. / Bu çalışma, ön kenar çapı (d) ve merkezler arası mesafenin (L), $Re = 4000-32000$ aralığında şaşırtmalı kam şekilli boru demetlerinin termo-hidrolik performansına etkilerini sayısal olarak incelemektedir.

Originality (Özgünlük): The originality of this study is the systematic evaluation of the individual and combined effects of d and L on the thermo-hydraulic performance of thirteen staggered tube-bundle configurations with identical perimeter and pitch ratios. / Bu çalışmanın özgünlüğü, aynı çevre uzunluğu ve hatve oranlarına sahip on üç şaşırtmalı boru demeti konfigürasyonunda d ve L parametrelerinin ayrı ve birleşik etkilerinin termo-hidrolik performans açısından sistematik olarak değerlendirilmesidir.

Results (Bulgular): Nu increased with Re but generally decreased with increasing d and L , while the friction factor decreased. All cam-shaped configurations showed $PEC > 1$, and M13 achieved the highest PEC improvement of approximately 28%. / Nu , Re ile artarken genel olarak d ve L 'nin artmasıyla azalmış, sürtünme katsayısı ise düşmüştür. Tüm kam şekilli konfigürasyonlar $PEC > 1$ sağlamış, M13 yaklaşık %28 ile en yüksek PEC artışı göstermiştir.

Conclusion (Sonuç): Cam-shaped tubes reduced flow resistance while maintaining favorable heat-transfer performance. M13 achieved the highest PEC and best overall thermo-hydraulic performance. / Kam şekilli borular, uygun ısı transferi performansını korurken akış direncini azaltmıştır. M13 en yüksek PEC ve en iyi genel termo-hidrolik performansı sağlamıştır.



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Abstract

In this study, the thermo-hydraulic performance of staggered cam-shaped tube bundles under crossflow conditions was numerically investigated. The steady-state Reynolds-averaged Navier–Stokes equations were solved using the RNG $k-\epsilon$ turbulence model with enhanced wall treatment. Thirteen configurations (Models M1–M13) were analyzed over a Reynolds number range of 4000–32000. The geometric parameters included the leading-edge diameter ($d = 2.5, 5, \text{ and } 7.5 \text{ mm}$) and the distance between the centers of the two circles ($L = 10, 12.5, 15, \text{ and } 17.5 \text{ mm}$), corresponding to aspect ratios (AR) of 1.00–2.63. The velocity, pressure, and turbulent kinetic energy distributions showed that the cam-shaped geometries modified the downstream flow field and reduced flow resistance compared with the baseline circular tube bundle. The average Nusselt number increased with Reynolds number but generally decreased with increasing d and L , whereas the friction factor decreased as these geometric parameters increased. All cam-shaped configurations exhibited lower friction factors than the baseline circular tube bundle (M1) and performance evaluation criterion (PEC) values greater than unity throughout the investigated Reynolds-number range. Among the investigated configurations, Model M13 ($AR = 2.63, d = 7.5 \text{ mm}, L = 17.5 \text{ mm}$) provided the highest thermo-hydraulic performance, achieving an approximately 28% improvement in PEC compared with the baseline model.

Şaşırtmalı Dizilimli Kam Şekilli Boru Demetlerinin Termo-Hidrolik Performansının Sayısal İncelenmesi

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Öz

Bu çalışmada, çapraz akış koşulları altındaki şaşırtmalı dizilimli kam şekilli boru demetlerinin termo-hidrolik performansı sayısal olarak incelenmiştir. Geliştirilmiş duvar işleme yöntemi ile RNG $k-\epsilon$ türbülans modeli kullanılarak, kararlı Reynolds ortalama Navier-Stokes (RANS) denklemleri çözülmüştür. 4000-32000 Reynolds sayısı aralığında on üç farklı konfigürasyon (Model M1–M13) analiz edilmiştir. Geometrik parametreler, ön kenar çapı ($d = 2,5; 5 \text{ ve } 7,5 \text{ mm}$) ve iki dairenin merkezleri arasındaki mesafe ($L = 10; 12,5; 15 \text{ ve } 17,5 \text{ mm}$) olup, bunlar 1,00-2,63 arasındaki biçim oranlarına (AR) karşılık gelmektedir. Hız, basınç ve türbülans kinetik enerjisi dağılımları, kam şekilli geometrilerin referans dairesel boru demetine kıyasla akış alanını değiştirdiğini ve akış direncini azalttığını göstermiştir. Ortalama Nusselt sayısı Reynolds sayısının artmasıyla artarken, genel olarak d ve L değerlerinin artmasıyla azalmıştır. Buna karşılık, sürtünme katsayısı bu geometrik parametrelerin artmasıyla azalmıştır. Tüm kam şekilli konfigürasyonlar, referans dairesel boru demetine (M1) kıyasla daha düşük sürtünme katsayıları ve incelenen Reynolds sayısı aralığı boyunca birden büyük performans değerlendirme kriteri (PEC) değerleri göstermiştir. İncelenen konfigürasyonlar arasında Model M13 ($AR = 2,63, d = 7,5 \text{ mm}, L = 17,5 \text{ mm}$) en yüksek termo-hidrolik performansı sağlamış ve referans modele kıyasla PEC değerinde yaklaşık %28 iyileşme sağlamıştır.

1. INTRODUCTION (GİRİŞ)

Increasing energy efficiency requirements have positioned the development of compact and high-performance heat exchanger systems as a key focus area in engineering research. In this context, tube bundle heat exchangers have been subjected to

intensive investigation for many years owing to their broad application areas and their thermo-hydraulic significance regarding flow and heat transfer mechanisms [1,2]. These systems play a significant role in engineering, with a wide range of applications, including waste heat recovery

systems, HVAC systems, cooling towers, condensers, and electronic cooling applications [3–5]. However, early boundary-layer separation and extensive wake regions around circular tubes increase drag and pressure drop, limiting thermo-hydraulic performance [6–8]. Extensive research has demonstrated that staggered tube arrangements enhance heat transfer owing to increased turbulence levels; however, this improvement also leads to substantial increases in the drag force and pressure drop [4,9,10]. Flow and heat transfer characteristics are also strongly affected by pitch ratios and tube arrangement [11,12]. Grooved cylinders have been reported to suppress wake vortices, shorten the wake region, and reduce the drag coefficient through modifications of the near-wake flow structure [13]. Furthermore, Söyler [14] demonstrated that fin geometry is a decisive factor in the thermo-hydraulic performance of triangular finned tube banks. Alternative tube cross-sectional geometries provide another approach for reducing flow resistance while maintaining heat-transfer performance. Elliptical, oval, wing-, drop-, and cam-shaped tubes generally produce narrower wake regions than circular tubes and reduce drag [15–17]. Elliptical tubes aligned with the flow direction can reduce pressure drop while preserving thermal performance [18–20]. Park et al. [21] reported an approximately 60% reduction in friction factor for elliptical cylinders with only a 9.3% decrease in average Nusselt number. Deeb et al. [22] showed that a drop-shaped tube delayed boundary-layer separation, reduced wake vortices, and decreased drag and friction factor compared with a circular tube. These studies demonstrate the strong influence of tube cross-sectional geometry on wake development, flow resistance, and thermo-hydraulic performance. Cam-shaped tubes have attracted particular attention because their streamlined geometry can delay flow separation and reduce wake development. Experimental and numerical studies have shown that cam-shaped tubes provide lower drag than circular tubes while maintaining favorable heat-transfer characteristics [23,24]. In particular, the angle of attack plays a critical role in the performance of single cam-shaped cylinders, with both high heat transfer and low drag achieved at optimum angles [25,26]. Akbari et al. [27] reported that cam-shaped tubes provided the highest heat transfer-friction ratio among circular, square, elliptical, and cam-shaped tubes. Lavasani et al. [28] found that in-line cam-shaped tube bundles achieved substantially higher thermo-hydraulic

performance than equivalent circular tubes. Mobedi et al. [29] showed that increasing the aspect ratio reduces the drag coefficient, whereas thermal performance reaches a maximum at specific ratios. Recent studies have further shown that pitch ratios, tube-row number, Reynolds number, and geometric parameters significantly influence the performance of cam-shaped tube bundles [30,31].

Although previous studies have demonstrated the potential of cam-shaped tubes for reducing flow resistance while maintaining heat-transfer performance, the effects of geometric parameters on the thermo-hydraulic performance of staggered cam-shaped tube bundles over a broad Reynolds-number range remain insufficiently characterized. Therefore, the present study numerically investigates staggered cam-shaped tube bundles over a Reynolds number range of 4000–32000 by varying the leading-edge diameter ($d = 2.5, 5, \text{ and } 7.5 \text{ mm}$) and the distance between the centers of the two circles ($L = 10, 12.5, 15, \text{ and } 17.5 \text{ mm}$), corresponding to aspect ratios (AR) of 1.00–2.63. The effects of these geometric parameters are evaluated using the average Nusselt number (Nu), friction factor (f) and performance evaluation criterion (PEC), while the velocity, pressure, and turbulent kinetic energy distributions are used to support the interpretation of the thermo-hydraulic behavior. The original contribution of this study is the systematic evaluation of the separate and combined effects of d and L for thirteen staggered configurations with identical perimeter and pitch ratios. The combined analysis of the flow-field characteristics, Nu , f , and PEC quantifies the thermo-hydraulic trade-off and identifies the most favorable geometry within the investigated design space.

2. MATERIALS AND METHODS (MATERYAL VE METOD)

2.1. Physical Model and Geometric

Configuration (Fiziksel Model ve Geometrik Konfigurasyon)

The thermo-hydraulic performance of cam-shaped tube bundles was numerically investigated. The cam-shaped cross-section was generated by joining two circular sections of different diameters with tangential arcs. All geometries were designed with the same equivalent diameter. The equivalent diameter was defined based on the profile perimeter as follows:

$$D_{eq} = \frac{P_{cam}}{\pi} \quad (1)$$

Accordingly, all cam-shaped geometries had the same perimeter as a circular tube with a diameter of 20 mm. Thirteen configurations (Models M1-M13) were considered. Model M1 represented the baseline circular tube, whereas Models M2-M13 were defined by the leading-edge diameter (d) and the distance between the centers of the two circles

(L). The trailing-edge diameter (D), total profile length (H), and aspect ratio ($AR = H/D$) were determined accordingly. Figure 1 presents (a) the geometric definitions and (b) the corresponding dimensional parameters of all investigated configurations.

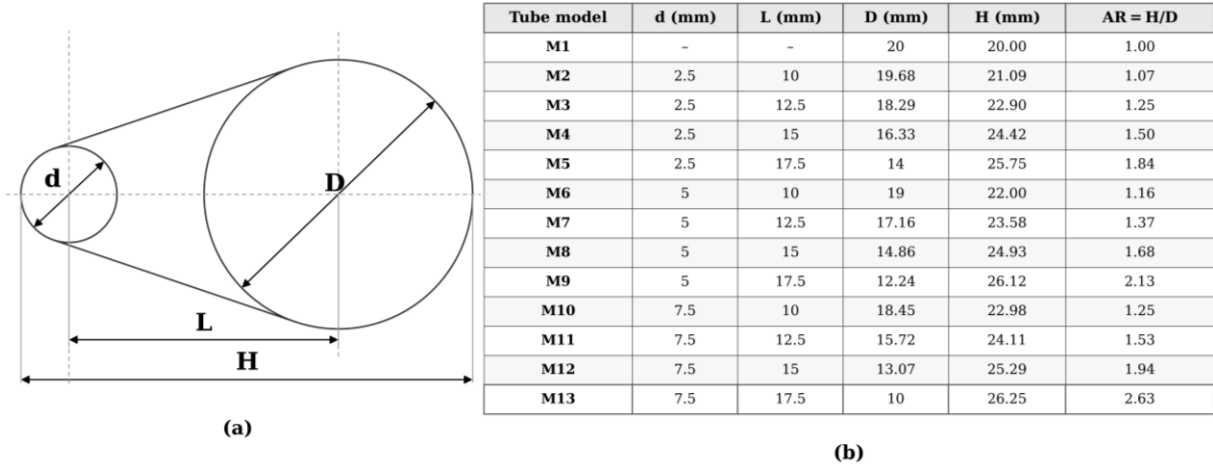


Figure 1. Geometric characterization of the investigated tube configurations: (a) dimensional definitions and (b) parameters of Models M1–M13 (İncelenen boru konfigürasyonlarının geometrik karakterizasyonu: (a) boyutsal tanımlar ve (b) M1–M13 modellerinin parametreleri)

The tube bundles were arranged in a staggered configuration, with transverse and longitudinal pitch ratios of $S_T/D_{eq} = 1.5$ and $S_L/D_{eq} = 1.3$, respectively. The streamwise, transverse, and spanwise directions are denoted by x , y , and z , respectively. Each configuration comprised five tube rows along the streamwise (x) direction. The computational domain extended $3D_{eq}$ upstream of the first tube row and $12D_{eq}$ downstream of the last tube row, as illustrated in Figure 2(a). A three-dimensional view of the computational domain and its boundary surfaces is presented in Figure 2(b).

2.2. Boundary Conditions (Sınır Koşulları)

Air was used as the working fluid in this study. Accordingly, the air properties were considered constant: density $\rho = 1.136 \text{ kg/m}^3$, dynamic viscosity $\mu = 1.884 \times 10^{-5} \text{ kg/ms}$, thermal conductivity $\lambda = 0.027 \text{ W/mK}$, and specific heat $c_p = 1007 \text{ J/kgK}$. A uniform velocity inlet boundary condition was applied, with the inlet temperature set to $T_{in} = 300 \text{ K}$. The inlet velocity was adjusted for each case to achieve the targeted Reynolds number. This approach, combined with the extended outlet length, prevented potential backflow, thereby ensuring numerical stability and

reliable simulation results. A pressure outlet boundary condition with a gauge pressure of 0 Pa was specified at the domain exit. All tube surfaces were treated with a no-slip boundary condition and maintained at a constant wall temperature of $T_s = 353 \text{ K}$. The resulting maximum temperature difference was 53 K. Since the study compared geometries under identical conditions at matched Reynolds numbers, constant reference air properties were used to isolate geometric effects; their possible influence on absolute heat-transfer values is acknowledged as a limitation. A representative segment with a spanwise thickness of 1 mm was modeled. As illustrated in Figure 2(b), the upper and lower surfaces, separated by the transverse pitch S_T , were coupled as a translational periodic pair in the transverse (y) direction. The front and rear surfaces, separated by the 1 mm domain thickness, were coupled as a second translational periodic pair in the spanwise (z) direction. The velocity-inlet and pressure-outlet boundaries were non-periodic and were located along the streamwise (x) direction. These periodic surface pairs represent the repetition of the computational domain in the transverse and spanwise directions. Similar periodic modeling approaches have been used in previous tube-bank studies [32].

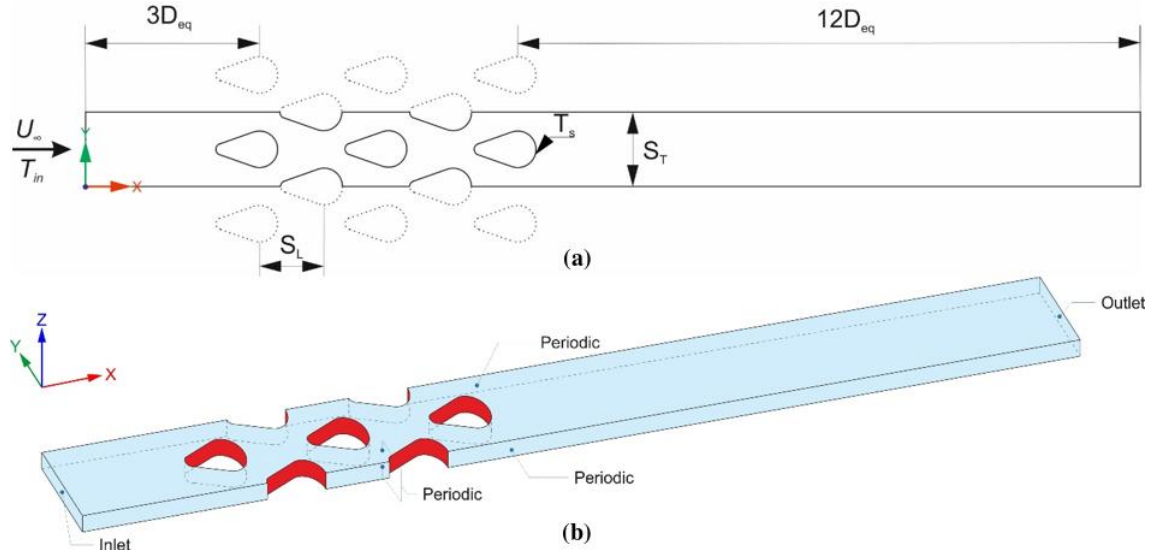


Figure 2. Computational domain of the staggered cam-shaped tube bundle: (a) two-dimensional and (b) three-dimensional views (Şaşırtmalı diziliimli kam şekilli boru demetinin hesaplama alanı: (a) iki boyutlu ve (b) üç boyutlu görünüm)

2.3. Governing Equations and Turbulence Model Selection (Yönetici Denklemler ve Türbülans Modeli Seçimi)

The flow and heat transfer around cam-shaped and circular tube bundles were modeled by solving three-dimensional, steady-state, and incompressible Reynolds-averaged Navier–Stokes (RANS) equations. The fluid was assumed to be Newtonian, and the effects of gravity and thermal radiation were neglected in the analysis. The governing equations are the continuity, momentum, and energy equations. The continuity equation is given as:

$$\frac{\partial}{\partial x_i}(u_i) = 0 \quad (2)$$

where u_i represents the velocity component in the Cartesian coordinate system. The momentum equations in tensor form are written as:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu_{eff} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \frac{\partial}{\partial x_j} (-\rho \overline{u'_i u'_j}) \quad (3)$$

The energy equation is expressed as follows:

$$\frac{\partial}{\partial x_i}(\rho c_p u_i T) = \frac{\partial}{\partial x_i} \left(k_{eff} \frac{\partial T}{\partial x_i} \right) \quad (4)$$

where T denotes the temperature, c_p is the specific heat, and k_{eff} represents the effective thermal conductivity, including both molecular and

turbulent contributions. The Renormalization Group (RNG) k - ϵ turbulence model was employed to account for flow separation, curvature, and turbulent mixing around the tube bundles. Hu and Sun [33] showed that this model accurately predicts the local surface heat transfer coefficients in airflow. Furthermore, Yakhot et al. [34] reported that the RNG k - ϵ model showed good agreement between the numerical and experimental data for backward-facing step flows.

2.4. Mesh Generation and Numerical Method (Ağ Oluşturma ve Sayısal Yöntem)

A structured-dominant computational mesh composed of prismatic and hexahedral elements was generated to resolve the velocity and temperature gradients near the tube surfaces and in the wake regions. An 18-layer inflation mesh was applied to all tube surfaces with a first cell height of 1×10^{-5} m and a growth rate of 1.2, satisfying the specified near-wall resolution criterion. Local mesh refinement was employed within the tube-bundle region, while the mesh was gradually coarsened toward the outer flow domain to maintain a balance between numerical accuracy and computational cost. Only the external flow domain was considered, and the internal tube volumes were excluded. The resulting mesh structure and refinement regions are shown in Figure 3. The governing equations were solved in ANSYS Fluent based on the finite-volume framework. The SIMPLE procedure was adopted for pressure–velocity coupling. Second-order upwind discretization was employed for the

momentum and energy equations, while pressure was evaluated using a second-order scheme. Convergence was achieved when the residuals decreased below 10^{-6} for continuity and momentum

and below 10^{-8} for energy. In addition, the mean Nusselt number and friction factor were monitored during the iterations to ensure that the solution had reached a stable state.

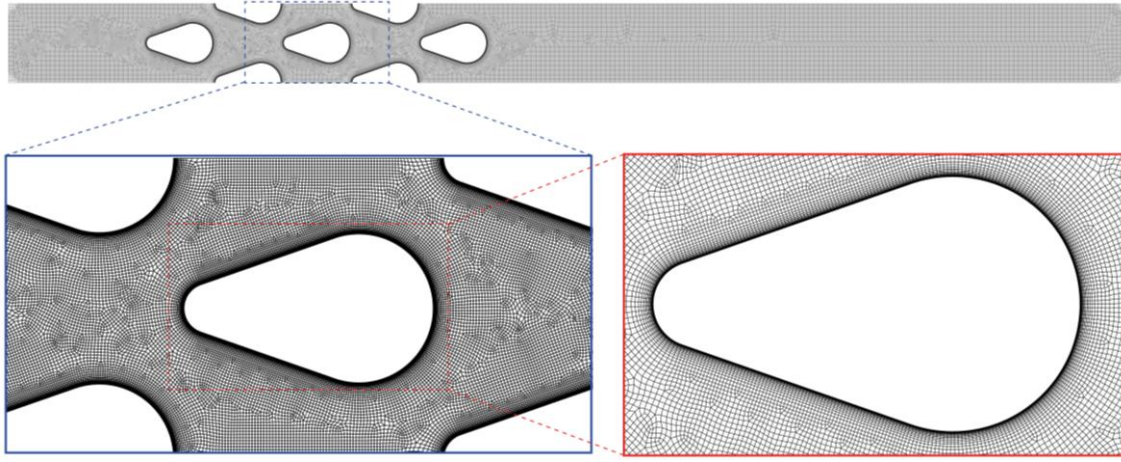


Figure 3. Computational mesh details of the numerical model (Sayısal Modelin Hesaplama Ağı Detayları)

2.5. Parameter Definitions

The thermal and hydraulic characteristics of the tube-bundle configurations were assessed through a set of dimensional and dimensionless quantities widely used in heat-transfer studies. The equivalent diameter (D_{eq}) was adopted as the characteristic length scale for all the configurations. For the circular tube, the equivalent diameter is equal to the tube diameter, whereas for cam-shaped tubes, it is determined according to Eq. (1). The Reynolds number was calculated based on the maximum velocity within the minimum flow area of the tube bundle as follows:

$$Re = \frac{\rho V_m D_{eq}}{\mu} \quad (5)$$

where V_m denotes the maximum velocity through the minimum flow area and is calculated as:

$$V_m = \frac{S_T}{S_T - D} U_\infty \quad (6)$$

where U_∞ is the free-stream velocity at the inlet. The average Nusselt number was calculated based on the equivalent diameter:

$$Nu = \frac{h D_{eq}}{\lambda} \quad (7)$$

The friction factor was calculated from the total pressure drop across the tube bundle according to:

$$f = \frac{\Delta P}{0.5 \rho V_m^2 N_L} \quad (8)$$

where ΔP is the total pressure drop across the tube bundle, ρ is the fluid density, and N_L is the number of tube rows in the flow direction. The performance evaluation criterion (PEC) is defined as follows:

$$PEC = \frac{Nu/Nu_0}{(f/f_0)^{1/3}} \quad (9)$$

where Nu_0 and f_0 denote the average Nusselt number and friction factor of the baseline circular tube model, respectively.

2.6. Grid Independence and Validation (Ağ

Bağımsızlığı ve Doğrulama)

Four different mesh resolutions were examined to determine the sensitivity of the numerical results to grid size, as listed in Table 1. The average Nusselt number (Nu) and friction factor (f) were compared over the Reynolds number range of 4000–32000. Increasing the mesh size from 146,998 to 294,070 cells resulted in changes of less than 1% in both Nu and f . Accordingly, the 146,998-cell grid was adopted for the remaining simulations. The influence of the spanwise domain thickness was also assessed, and increasing this dimension produced variations below 1% for both parameters. Therefore, a thickness of 1 mm was retained. The numerical model was validated against the Nusselt number and friction factor data reported by Zukauskas [35] for staggered circular tube bundles over $Re = 4000$ –32000.

Table 1. Grid independence study based on the average Nusselt number (Nu) and friction factor (f) at selected Reynolds numbers (Seçilen Reynolds sayılarında ortalama Nusselt sayısı (Nu) ve sürtünme katsayısına (f) dayalı ağ bağımsızlığı çalışması)

No. of cells	$Re=4000$		$Re=11000$		$Re=18000$		$Re=25000$		$Re=32000$	
	Nu	f	Nu	f	Nu	f	Nu	f	Nu	f
39900	41.538	0.485	72.818	0.367	99.147	0.325	124.110	0.301	148.986	0.286
84213	41.439	0.485	72.344	0.360	98.244	0.318	123.031	0.296	147.889	0.282
146998	41.385	0.485	72.108	0.359	97.886	0.317	122.550	0.296	147.325	0.282
294070	41.385	0.485	72.090	0.358	97.880	0.317	122.540	0.296	147.310	0.282

As shown in Figure 4, the RNG $k-\varepsilon$ predictions agreed well with the reference data. The maximum deviation in Nu was approximately 8%, while the friction-factor predictions remained within 15% of the experimental values. These results confirm the reliability of the numerical methodology for the analysis of the proposed cam-shaped tube bundles [9,14].

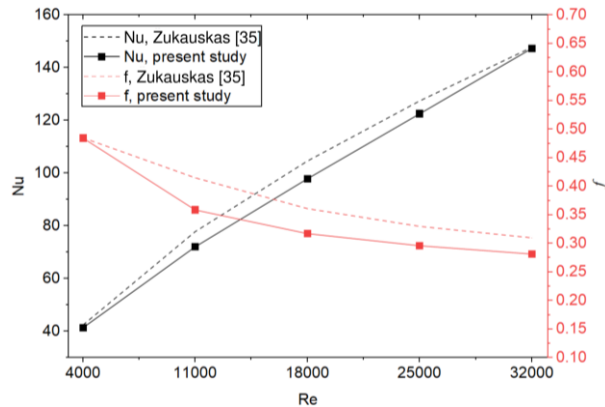


Figure 4. Comparison of the average Nusselt number (Nu) and friction factor (f) obtained in the present study with the reference data by Zukauskas [35] (Bu çalışmada elde edilen ortalama Nusselt sayısı (Nu) ve sürtünme katsayısının (f), Zukauskas [35] referans verileri ile karşılaştırılması)

3. RESULTS (BULGULAR)

In this study, the effects of the leading-edge diameter (d) and the distance between the centers of the two circles (L) on the thermo-hydraulic performance of cam-shaped tube bundles were investigated over the Reynolds number range of 4000–32000. The discussion first examines the velocity, pressure, and turbulent kinetic energy distributions to explain the influence of tube geometry on the flow field. The corresponding effects on heat transfer and flow resistance are then evaluated using the average Nusselt number, friction factor, and performance evaluation criterion (PEC) for the cam-shaped tube bundles (Models M2-M13) relative to the baseline circular

configuration (Model M1, $AR = 1.00$). Figure 5 presents the velocity magnitude contours at $Re = 32000$ for the baseline circular tube bundle (M1) and selected cam-shaped configurations (M2, M4, M5, M12 and M13). Across all configurations, the incoming flow decelerates near the leading-edge stagnation region and accelerates through the passages between adjacent tubes, forming high-velocity regions. Downstream of the tubes, low-velocity wake regions develop, with their size and distribution strongly dependent on the tube geometry. For $d = 2.5$ mm, increasing L from 10 mm (M2) to 17.5 mm (M5) reduces the extent of the low-velocity region downstream. Increasing d from 2.5 mm (M4) to 7.5 mm (M12) produces a smoother velocity distribution around the tube and further reduces the extent of the low-velocity wake. Among the investigated configurations, M13 ($d = 7.5$ mm, $L = 17.5$ mm, $AR = 2.63$) exhibits the smallest low-velocity region and the most uniform downstream velocity field. Figure 6 presents the static pressure contours at $Re = 32000$ for the baseline circular tube bundle (M1) and selected cam-shaped configurations (M2, M4, M5, and M13). The circular baseline exhibits a high-pressure stagnation region at the leading edge and an extended low-pressure region downstream. For M2, the pressure distribution remains similar to that of M1, whereas increasing L to 17.5 mm (M5) produces a more gradual pressure distribution and a shorter downstream low-pressure region. This reduced pressure variation is consistent with the lower friction factors of the cam-shaped configurations. Figure 7 presents the turbulent kinetic energy (TKE) contours at $Re = 32000$ for the baseline circular tube bundle and selected cam-shaped tube bundles (M2, M4, M5, M12, and M13). In M1, the highest TKE levels are concentrated within the shear layers and downstream wake, where strong velocity gradients promote turbulence production. For $d = 2.5$ mm, increasing L from 10 mm (M2) to 17.5 mm (M5) reduces both the extent and intensity of the high-TKE region. Compared with M4, increasing the leading-edge diameter to 7.5 mm (M12) reduces the TKE levels within the wake region. Among all

investigated configurations, M13 ($d = 7.5$ mm, $L = 17.5$ mm) exhibits the smallest high-TKE region and the most compact TKE distribution downstream of the tube bundle. Figure 8 presents the variation of the average Nusselt number with Reynolds number for different leading-edge diameters at fixed values of L . For all investigated configurations, the average Nusselt number increases with Reynolds number.

The highest Nusselt numbers are consistently obtained at $L = 10$ mm (Figure 8(a), M2, M6, and M10). As L increases from 10 to 17.5 mm, the average Nusselt number gradually decreases for all leading-edge diameters. At a fixed L , increasing the leading-edge diameter also gradually reduces the average Nusselt number.

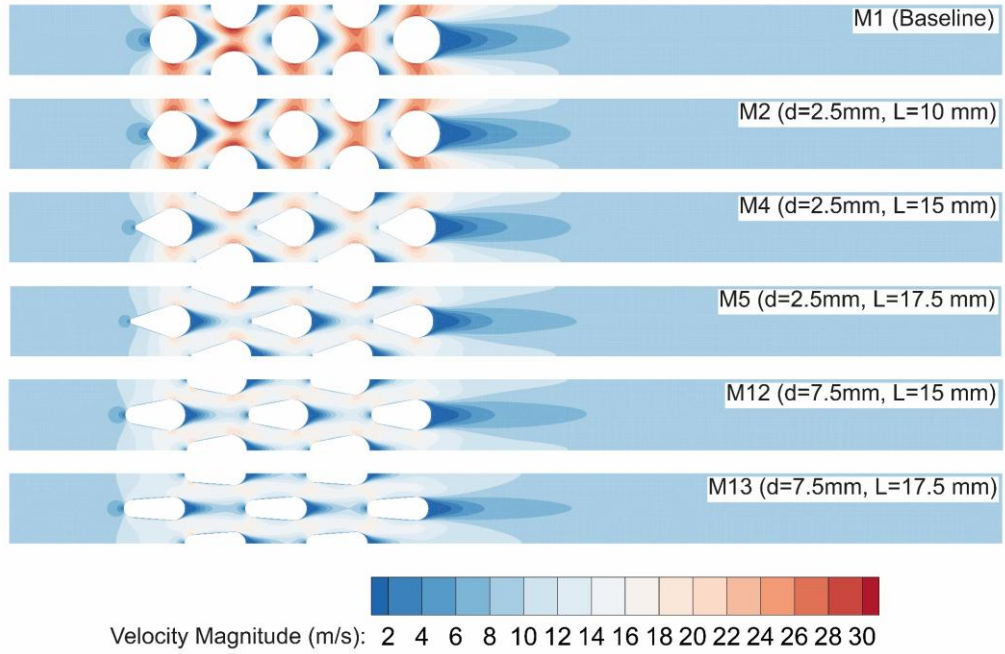


Figure 5. Velocity magnitude contours at $Re = 32000$ for the baseline circular tube bundle (Model M1) and selected cam-shaped tube bundles (Models M2, M4, M5, M12, and M13) ($Re = 32000$ için referans dairesel boru demeti (Model M1) ve seçilen kam şekilli boru demetlerinin (Modeller M2, M4, M5, M12 ve M13) hız büyüklüğü konturları)

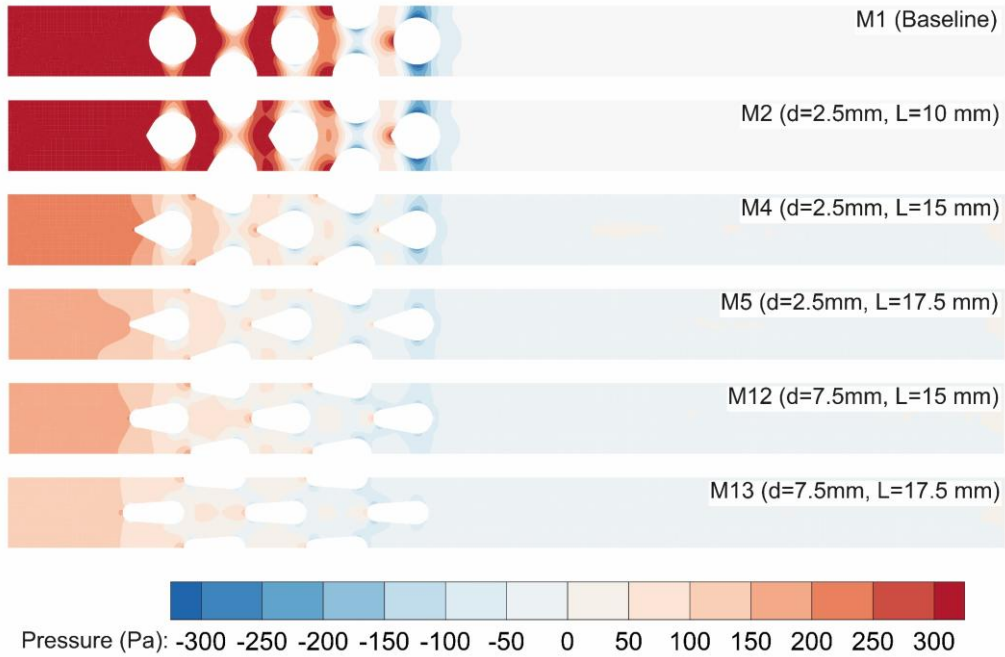


Figure 6. Static pressure contours at $Re = 32000$ for the baseline circular tube bundle (Model M1) and selected cam-shaped tube bundles (Models M2, M4, M5, and M13) ($Re = 32000$ için referans dairesel boru demeti (Model M1) ve seçilen kam şekilli boru demetlerinin (Modeller M2, M4, M5 ve M13) statik basınç konturları)

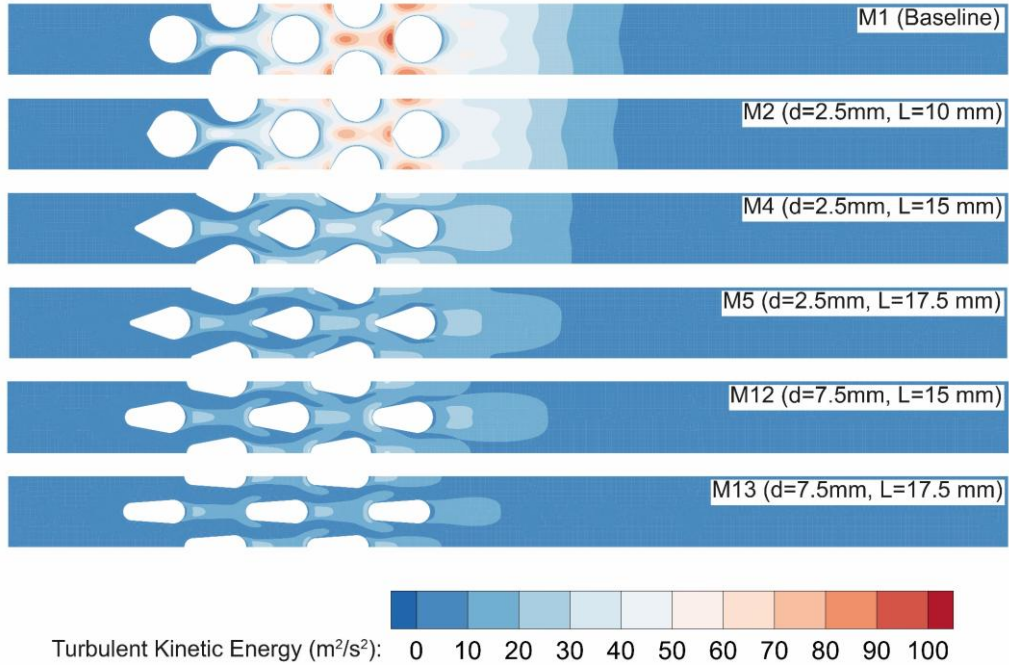


Figure 7. Turbulent kinetic energy (TKE) contours at $Re = 32000$ for the baseline circular tube bundle (Model M1) and selected cam-shaped tube bundles (Models M2, M4, M5, M12, and M13) ($Re = 32000$ için referans dairesel boru demeti (Model M1) ve seçilen kam şekilli boru demetlerinin (Modeller M2, M4, M5, M12 ve M13) türbülans kinetik enerjisi (TKE) konturları)

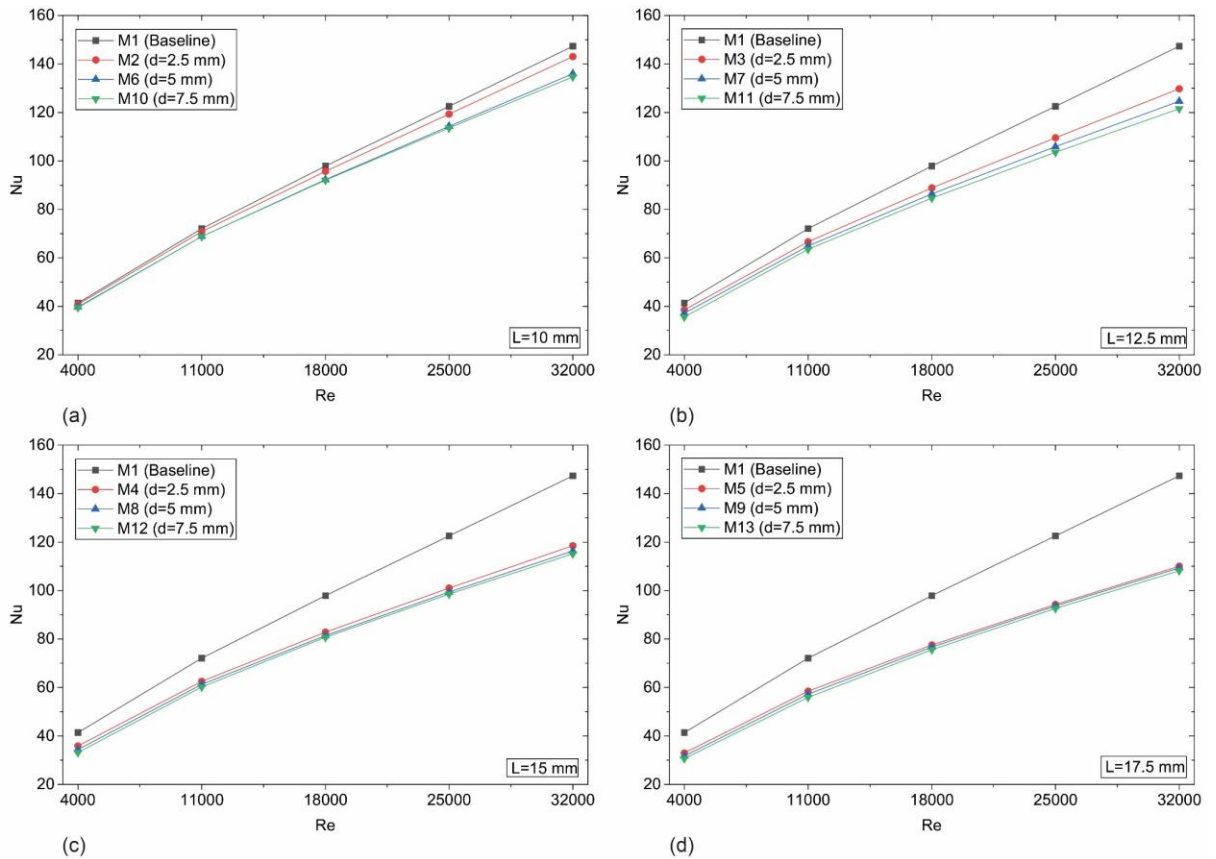


Figure 8. Variation of the average Nusselt number (Nu) with Reynolds number (Re) for different leading-edge diameters (d) at fixed distances between the centers of the two circles (L): (a) $L = 10$ mm, (b) $L = 12.5$ mm, (c) $L = 15$ mm, and (d) $L = 17.5$ mm (iki dairenin merkezleri arasındaki sabit mesafelerde (L), farklı ön kenar çapları (d) için ortalama Nusselt sayısının (Nu) Reynolds sayısı (Re) ile değişimi: (a) $L = 10$ mm, (b) $L = 12.5$ mm, (c) $L = 15$ mm ve (d) $L = 17.5$ mm)

Figure 9 presents the variation of the friction factor with Reynolds number for different leading-edge diameters at fixed distances between the centers of the two circles (L). For all investigated configurations, the friction factor decreases with increasing Re as inertial effects become progressively more dominant than viscous forces. The friction factor decreases systematically as L increases. The highest friction factors are obtained at $L = 10$ mm (Figure 9(a), M2, M6, and M10). As L increases to 17.5 mm (Figure 9(d), M5, M9, and M13), the more elongated geometry results in lower flow resistance and friction factor. At each fixed L , increasing the leading-edge diameter further reduces the friction factor. Consequently, all cam-shaped configurations exhibit lower friction factors than the baseline circular tube bundle (M1). Among the investigated configurations, M13 achieves the lowest friction factor, indicating the most favorable hydraulic performance.

Figure 10 illustrates the variation of the performance evaluation criterion (PEC) as a function of Reynolds number for the investigated cam-shaped and baseline circular tube bundles. The PEC values greater than unity indicate improvement compared with the reference circular tube geometry. For all cam-shaped configurations, the PEC remained above unity throughout the investigated Reynolds number range, confirming that the reduction in flow resistance achieved by the modified tube geometry effectively compensated for the accompanying decrease in heat transfer capability. For $L = 10$ mm (Figure 10(a), M2, M6, and M10), PEC values ranged from 1.02 to 1.08, with Model M10 ($d = 7.5$ mm) providing the highest value in this group. Increasing L further improved the thermo-hydraulic performance. Among the investigated configurations, the maximum PEC was obtained for Model M13 ($d = 7.5$ mm, $L = 17.5$ mm, $AR = 2.63$), providing an improvement of approximately 28% compared with the circular baseline (M1).

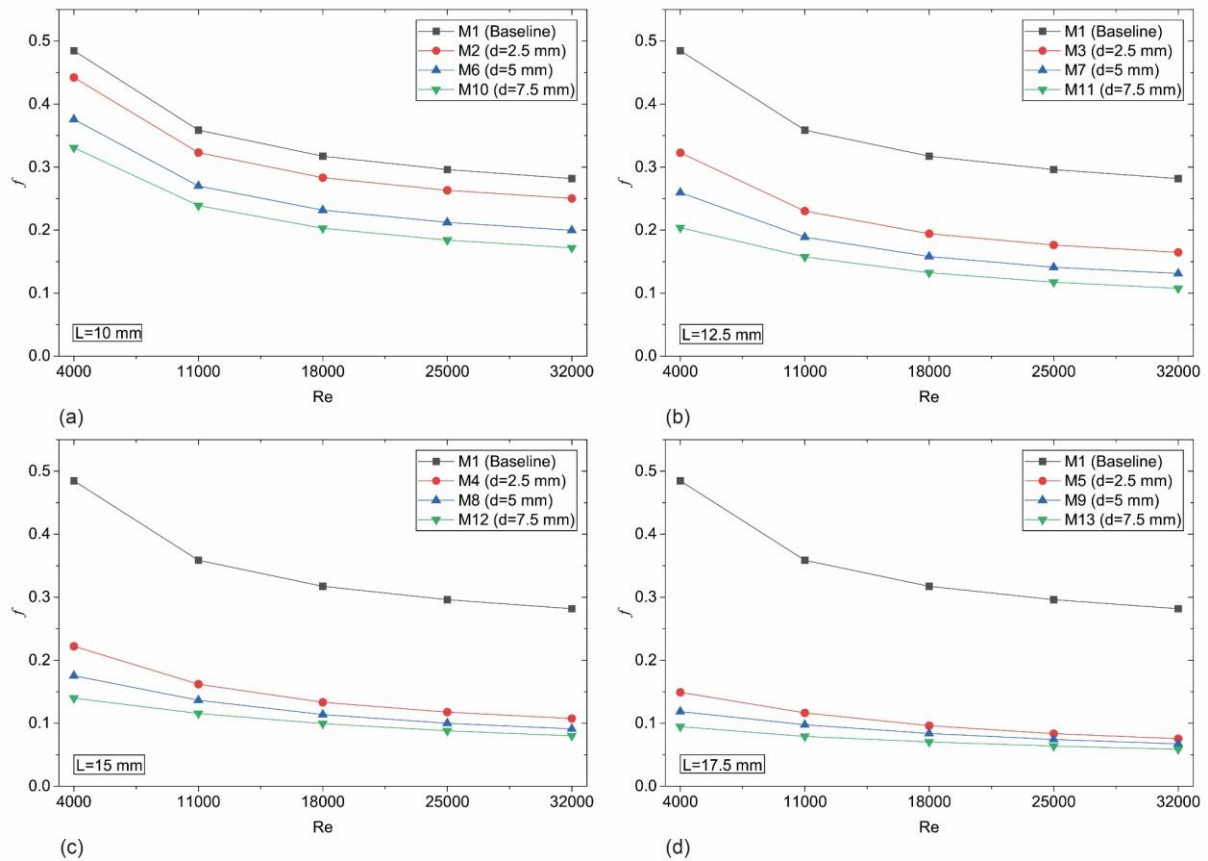


Figure 9. Variation of the friction factor (f) with Reynolds number (Re) for different leading-edge diameters (d) at fixed distances between the centers of the two circles (L): (a) $L = 10$ mm, (b) $L = 12.5$ mm, (c) $L = 15$ mm, and (d) $L = 17.5$ mm (İki dairenin merkezleri arasındaki sabit mesafelerde (L), farklı ön kenar çapları (d) için sürtünme katsayısının (f) Reynolds sayısı (Re) ile değişimi: (a) $L = 10$ mm, (b) $L = 12,5$ mm, (c) $L = 15$ mm ve (d) $L = 17,5$ mm)

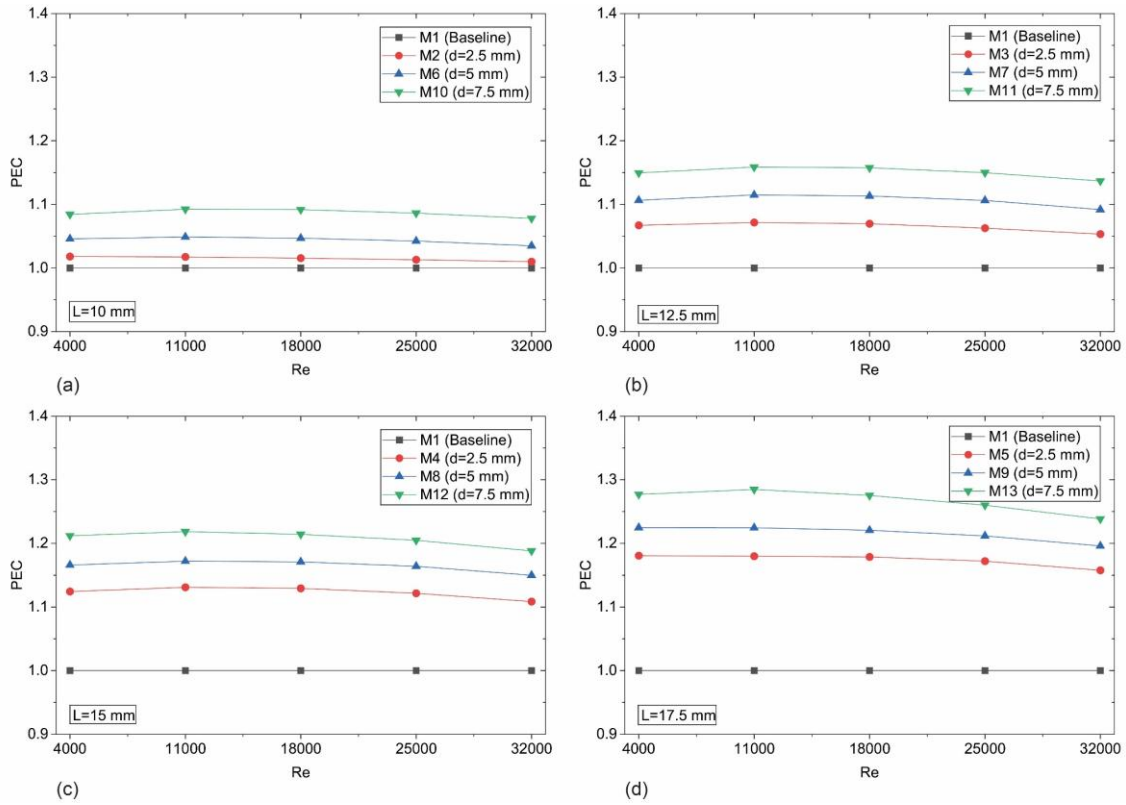


Figure 10. Variation of the performance evaluation criterion (PEC) with Re for the investigated cam-shaped tube bundles and baseline circular tube bundle (İncelenen kam şekilli boru demetleri ve referans dairesel boru demeti için performans değerlendirme kriterinin (PEC) Reynolds sayısı (Re) ile değişimi)

4. CONCLUSIONS (SONUÇLAR)

Within the adopted three-dimensional steady RANS framework, this study investigated the effects of the leading-edge diameter (d) and the distance between the centers of the two circles (L) on the thermo-hydraulic performance of thirteen staggered tube-bundle configurations over $Re = 4000$ -32000.

The velocity, pressure, and turbulent kinetic energy distributions showed that the cam-shaped geometry modified the downstream flow field and reduced the extent of the low-velocity and high-TKE regions compared with the circular tube bundle. The average Nusselt number increased with Reynolds number for all configurations but generally decreased with increasing d and L . In contrast, the friction factor decreased as d and L increased, and all cam-shaped configurations exhibited lower friction factors than the baseline circular tube bundle. Although the cam-shaped tubes generally produced lower Nu values, the reduction in f was dominant, resulting in PEC values greater than unity for all investigated configurations. Among the investigated configurations, Model M13 ($d = 7.5$ mm, $L = 17.5$ mm, $AR = 2.63$) combined the most uniform downstream flow field and the smallest

high-TKE region with the highest thermo-hydraulic performance, providing an approximately 28% PEC improvement relative to the circular baseline.

Overall, the results demonstrate that modifying the cam-shaped tube geometry can significantly reduce flow resistance while maintaining favorable heat-transfer performance. These conclusions are limited to the investigated geometries, pitch ratios, Reynolds-number range, constant-property assumption, and the adopted RNG k - ε turbulence model; broader generalization requires further numerical and experimental validation.

DECLARATION OF ETHICAL STANDARDS (ETİK STANDARTLARIN BEYANI)

The author declares that this study did not require ethical committee approval or any specific legal permission.

Yazar, bu çalışmanın etik kurul onayı veya herhangi bir özel yasal izin gerektirmediğini beyan etmektedir.

AUTHORS' CONTRIBUTIONS (YAZARLARIN KATKILARI)

Mustafa SÖYLER: He designed the study, developed the numerical model, performed the CFD

simulations, analyzed and interpreted the results, and prepared the manuscript.

Çalışmayı tasarlamış, sayısal modeli oluşturmuş, CFD simülasyonlarını gerçekleştirmiş, sonuçları analiz edip yorumlamış ve makaleyi hazırlamıştır.

CONFLICT OF INTEREST (ÇIKAR ÇATIŞMASI)

The author declares that he has no conflict of interest.

Yazar, herhangi bir çıkar çatışması olmadığını beyan etmektedir.

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