Heat transfer from a square tube bank with different hydraulic diameters at unsteady turbulent flow

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Abstract

The purpose of this study is heat transfer from a square tube bank with different hydraulic diameters at unsteady turbulent flow using numerical methods. By placing the smaller cylinders on larger cylinders upstream of a square tube bank, vorticity control of the flow through the cylinders, as well as its effect on drag force, pressure drop and heat transfer have been investigated. By reducing the longitudinal and transverse distance between the cylinders and resizing the smaller cylinders, six different arrangements of tube banks have been created that the comparison has been made with a tube bank of equal tubes. Unsteady turbulent incompressible flow is 40000 Reynolds. The finite volume method and FLUENT software have been used to solve Navier - Stokes and energy equations. The results show that the unequal tubes and reduce of longitudinal distance between them result in significant reduction of drag coefficient and flow Nusselt increase, but little change takes place in pressure drop.

Keywords: Square tube bank, heat transfer, vorticity, unequal tubs, drag

Introduction

The problem of flow through a cylinder with a square cross-section has been considered by many researchers in the field of aerodynamic because in real life, most architectural structures such as buildings, bridges and historical monuments and etc. have the square or rectangle cross-sections. Many theoretical and practical flows, such as shell and tube heat exchangers, aerodynamics and cooling of electronics show the phenomenon of vortex shedding in particular condition. Vortex shedding is the main factor of structure vibration which results in the failure caused by fatigue, the significant increase in the mean drag and lift fluctuating, acoustic noise and resonance. As a result, control and neutralization of this phenomenon have great importance in engineering. In recent decades, many affords have been made to control the flow vortex through the cylinder with circle and square cross-section, which is obtained by placing a small extra element on the master tube upstream. Vortex and wake flow control result in drag force reduction which in turn might have an impact on the rate of heat flow. In 2005, Zhang, Wang, Lu and Mi [1] investigated the aerodynamic characteristics of the turbulent flow around a square cylinder placed at a circular rod downstream. They found that in the inline arrangement of rods and cylinder, drag reduction is higher than staggered arrangement and this reduction is about 10% of an alone square cylinder drag. Zhang, Wang and Huang [2] in 2006 simulated a laminar flow around a circular cylinder located at a circular rod downstream with inline arrangement. In this study it was found that the flow characteristics are strongly depended on the ratio of the distance between two cylinders and the master cylinder diameter. In the same year, Huang, Olsen, Kerekes and Green [3] simulated the laminar flow of water around two columns of circular cylinders. The diameter of the first column cylinders was lower. They examined the impact of the cylinder arrangement on the vortex flow. In 2008, Chuan Ping Shao and Qing Ding Wei [4] using small circular, square, and thin-strip cross-sectional elements controlled the vortex of turbulent flow through a square cylinder. They evaluated the effect of cylinder size, position and the angle of the elements with the square cylinder on vortex and founded that if the element is placed in the effective zone of downstream cylinder, the suppressed flow, mean drag and cylinder lift fluctuating will be reduced. In 2008, the Strouhal number, forces and flow structures around two tandem circular cylinders of different diameters were investigated by Mahbub Alam and Zhou [5] so that small cylinders were at larger cylinder upstream and it was determined that by reducing the ratio of small cylinder diameter to larger one, time-averaged drag on the downstream cylinder increases, whereas fluctuating forces droppes. In 2009, Kuo and Chen [6] tried passive control of the wake behind a circular cylinder by two small circular cylinders placed symmetrically at the master cylinder downstream. This resulted in a significant reduction (70% -80%) of fluctuating lift and reduced drag by 5% in all cylinders. In 2012 Sofia, Mavridou, Demetri and Bouris [7] began the numerical evaluation of heat exchanger with tandem cylinders of different sizes in order to reduce fouling. By changing the size of the odd column cylinders in a circular tube bank and also the distance between cylinders, they evaluated two new arrangements of tube banks and founded that by neutralizing the flow vortex, the particles residue amount is influenced. In 2013 AkshoyRanjan Paul, Shrey Joshi, Aman Jindal, Shivam P Maurya,. Tiwari, Anuj Jain and Prakhar K [8] studied the cross-section shape effect of the rod upstream of a square cylinder on the flow drag. For this purpose square, circular and triangular rods with the same cross section size were investigated. The circular rod reduced about 78% and the square rod about 84% of the flow drag but the triangular rod showed abnormal behavior of the drag. In the same year, YeJunChen and ChuanPingShao [9] attemted to neutralize the vortex of a rectangular cylinder at low Reynolds by control rods with square and rectangle cross sections and it was found that the square rod has the largest effective zone and the triangular rod creates the smallest effective zone. They also concluded that if the rod is in the effective zone, the vortex will be neutralized and drag and lift fluctuating forces will be reduced significantly. In this paper by changing the arrangement of a tube bank in a square tube bank, it is tried to control the flow vorticity and its impact on drag force, pressure drop and heat transfer. In this study, the heat transfer at unsteady turbulent incompressible air flow around a square tube bank with different structures and square layouts has been simulated numerically and in two dimensions. Tube wall temperature is a constant of 70 degrees Celsius. The air is assumed with constant properties and inlet temperature of 15 ° C. In the arrangement A, tubes are the same as width of 2 cm and with 4 cm longitudinal and transverse distances. On the arrangement B, the cylinder cross-section width of the even columns (second, fourth, sixth and eighth columns) is 2 cm and the cylinder cross-section width of the odd columns (first, third, fifth and seventh columns) is smaller. The longitudinal distance between them has been reduced by 3 cm. In the structure of C, in addition to changes formed in the structure B, the cylinder transverse distance has been reduced to 3 cm. In both the B and C structure the ratio of larger to the smaller square width d / D has been considered three different values of 0.5, 0.3 and 0.2 that are called respectively B1, B2, B3, C1, C2 and C3.

The governing equations

Equations of continuity, Navier-Stokes, energy and also equations of the turbulence model are given below.

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0$$

(1)

$$\frac{\partial \bar{u}_i}{\partial t} + \bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\upsilon \frac{\partial \bar{u}_i}{\partial x_j} - \overline{u_i' u_j'}\right)$$

(2)

$$c_p\left(\frac{\partial T}{\partial t}+\bar{u}_j\frac{\partial T}{\partial x_j}\right)=\frac{\partial}{\partial x_j}\left(\Gamma\frac{\partial T}{\partial x_j}-c_p\overline{u_j' T'}\right)$$
(3)

Using turbulence model K-ε (RNG) and its two extra transport equations, one for kinetic energy and the other for turbulent dissipation rate, two Reynolds stress $\overline{u_i' u_j'}$ and scalar flux $\overline{u_j' T'}$ are modeled.

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k\bar{u}_i)=\frac{\partial}{\partial x_j}\left(\alpha_k\mu_{eff}\frac{\partial k}{\partial x_j}\right)+G_k+S_k \quad (4)$$

$$\boldsymbol{\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho\varepsilon\bar{u}_i)}=\frac{\partial}{\partial x_j}\left(\alpha_\varepsilon\mu_{eff}\frac{\partial\varepsilon}{\partial x_j}\right)+C_{1\varepsilon}\frac{\varepsilon}{k}G_k-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon+S_\varepsilon \quad (5)$$

$$R_\varepsilon=\frac{C_\mu\rho\eta^3\left(1-\frac{\eta}{\eta_0}\right)}{1+\beta\eta^3}\frac{\varepsilon^2}{k}$$
(6)

The model constants are obtained by the RNG theory:

$$\boldsymbol{\eta=\frac{Sk}{\varepsilon},\eta_0=4.38,C_{1\varepsilon}=1.42,C_{2\varepsilon}=1.68,\beta=0.012,}$$
$$\boldsymbol{C_\mu=0.0845}$$

More information is available in source [10].
Reynolds and Nusselt numbers are calculated based on the hydraulic diameter:

$$Re_D=\frac{u_\infty D_h}{\upsilon}$$
(7)

$$Nu=\frac{hD_h}{k_f}$$
(8)

Boundary conditions

on inlet boundary, the air enters at a speed of 15 meters per second and about 4% turbulent intensity and temperature of 288 K. On outlet boundary, the relative air pressure has been considered zero Pa. Tube wall temperature is 343 K and a no-slip condition is governed. At symmetrical boundaries, symmetrical options have been considered. The upstream face distance of the cylinder from the inlet has been considered 5 times and the downstream face distance of the cylinder from the outlet 30 times larger than the width of the larger square.

Numerical solution method

Fluent software in the form of two dimensional has been used for double precision for the numerical solution. Geometry and boundary conditions used are shown in Figure 1. To solve the problem, SIMPLEC algorithm has been applied for pressure-velocity coupling scheme. The dimensionless time step is 0.375. To mesh the problem geometry mapped quadrilateral mesh has been used and after examining different grids, approximately 200,000 square meshes have been considered suitable for solving the problem. Turbulent model K-ε (RNG) has been used and to achieve more accurate calculations near the wall, the enhanced wall treatment option has been used. To interpolate the pressure, the standard method has been used and for discretization the momentum equations, turbulent kinetic energy, turbulent dissipation rate, energy and Reynolds stresses equations, the second order Upwind has been used. Convergence criterion has been considered 10^{-5} for continuity equation, for energy equation 10^{-9} and for the rest of the equations 10^{-6}.

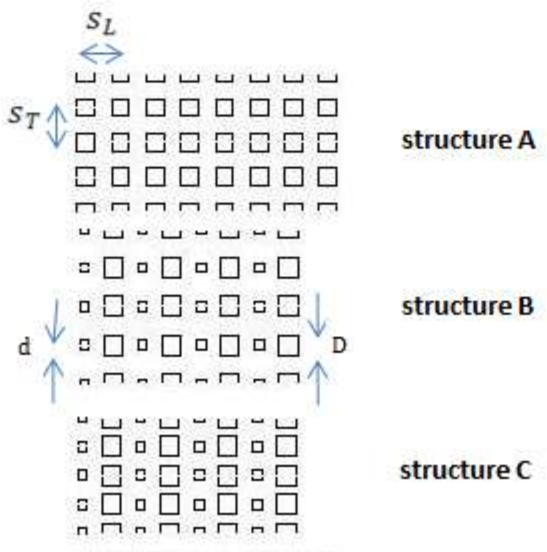


Figure 1: Geometry of the problem and the different structures of a square tube bank

Validation of the numerical solution

Two tandem square cylinders with the width of 60 mm and distance of 3 times the square width placed under the unsteady air cross flow with the Reynolds of 50000 have been simulated for the validation of numerical solution. Figure 2 shows the schematic representation of the two cylinders and measured points of the pressure coefficient. The results of numerical solution for time averaged and fluctuating pressure coefficient at different points on the cross section of two cylinders have been compared with the experimental results [11] in Table 1, which show a relative match.

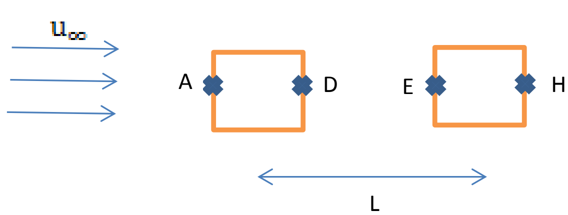


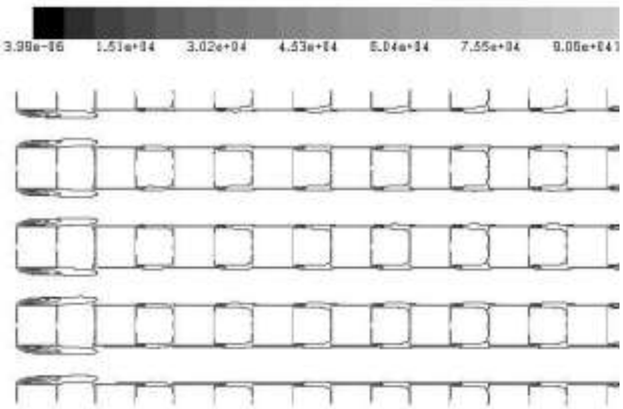
Figure 2: Schematic view of two square cylinders and measured points of the pressure coefficient

	A	D	E	H
$\overline{C_P}$ numerical	1.1	-1.06	-1.04	-0.38
$\overline{C_P}$ experimental	1	-1	-1	-0.4
$\acute{C}_P$ numerical	0	0	0	0.1
$\acute{C}_P$ experimental	0	0	0	0.1

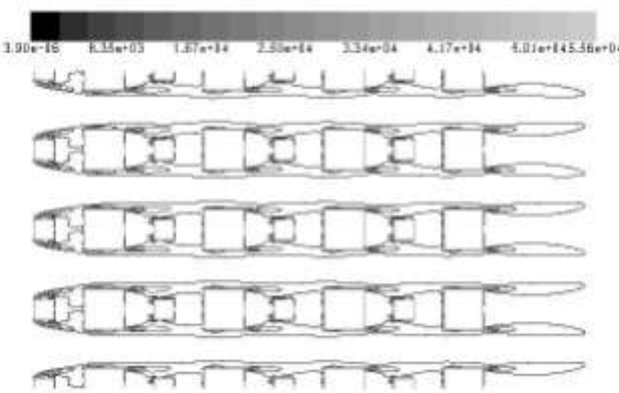
Table 1: Comparison of time averaged and fluctuating pressure coefficients obtained from the experimental resolution [11] and the present study

Results and Discussion

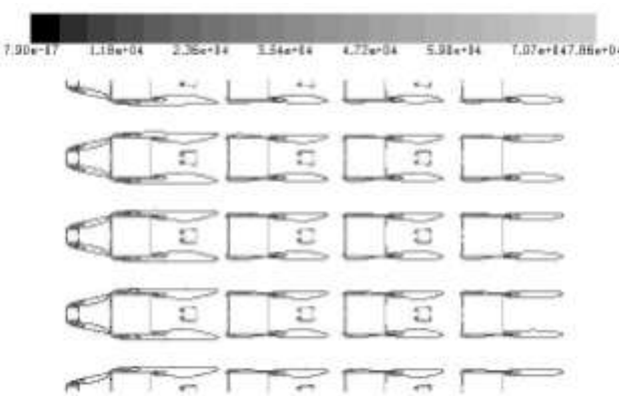
Figure 3 shows the vortex flow distribution around the cylinders for seven different tube bank structures. In the arrangement C, larger vortices are formed behind the cylinders compared to structures of A and B. This could be due to the transverse distance reduction between the cylinders which reduces the cross-section of the flow and more encounters of turbulent flow with cylinders wall and this creates more reverse flow and vorticity. In the arrangement B, smaller vortices are formed behind the cylinders compared to structures of A. This is because of placing the smaller cylinders on larger cylinders upstream of a square tube bank, which help control of vorticity. In structure B with decreasing d / D ratio, vortices reduce as in the model B3 the lowest vorticity exists compared to the model A.



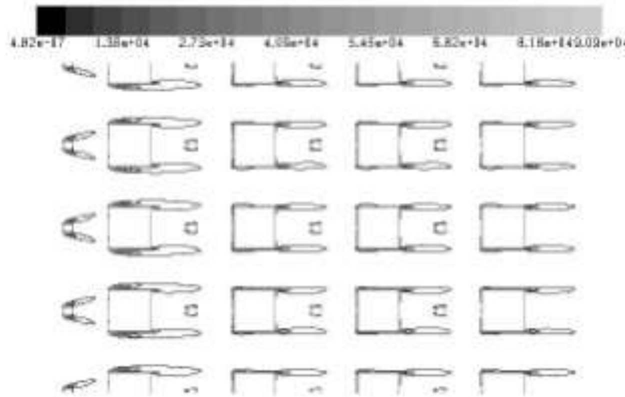
A



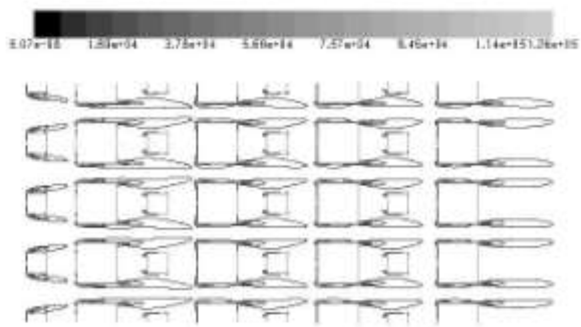
B1



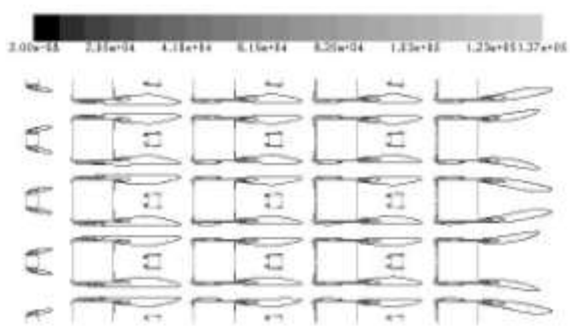
B2



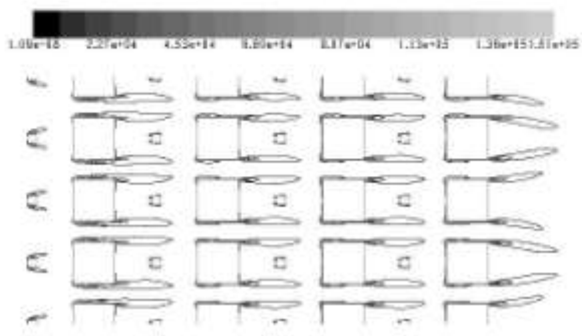
B3



C1



C2



C3

Figure 3: Vortex flow distribution around the cylinders for different tube bank structures for Reynolds 40000

In Figure 4, the flow Nusselt in different structures has been drawn depending on the tube bank columns. Increased flow Nusselt is only in the arrangements of B2 (9%) and B3 (26%) relative to the arrangement A and in the rest of the models, flow Nusselt has dropped in all of the columns. This increase is due to the smaller size of the odd column tubing which results in more flow trapped between the two tubes placed in adjacent even columns that its majority is reverse and this increases the heat transfer between the fluid trapped behind the tube and the tube wall and Nusselt increases too. But in the arrangement C due to the reduction of transverse distance and existence of flow turbulence, less flow is trapped behind the tubes and less heat transfer is occurred.

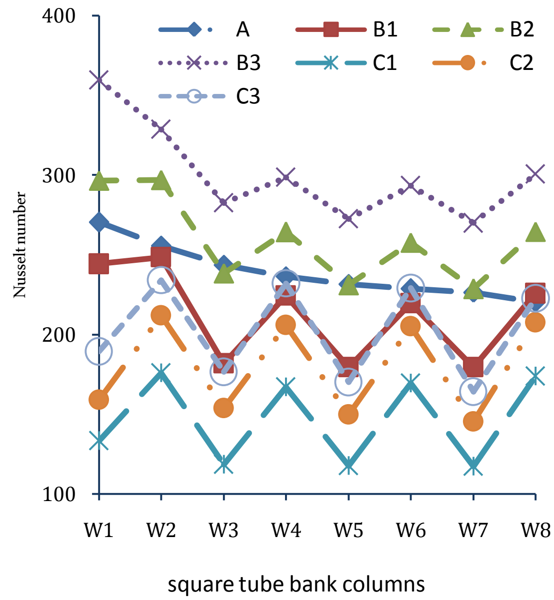


Figure 4: Distribution of the flow Nusselt number according to the square tube bank columns with different arrangements for Reynolds 40000

In Figure 5 the flow drag coefficient is plotted versus the tube bank columns for different arrangement. Due to the reduction of flow vorticity in B arrangements, drag coefficients have dropped about 50% compared to the model A, but in C arrangements due to the vorticity increase, drag coefficients have averagely increased about 30%.

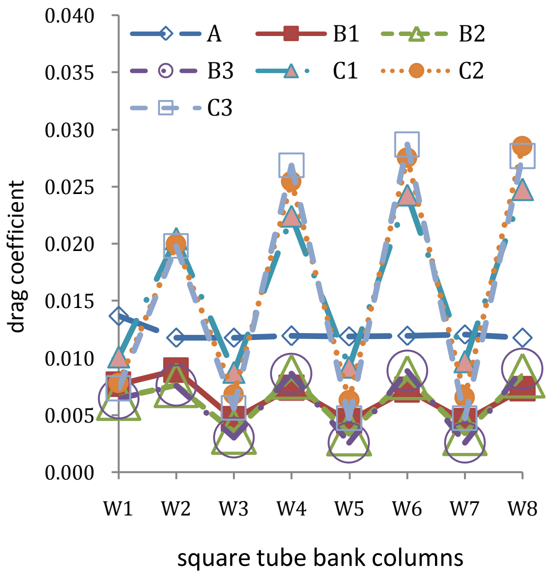


Figure 5: Distribution of the flow drag coefficient according to the square tube bank columns with different arrangements for Reynolds 40000

Figure 6 shows the pressure drop per flow length unit that A and B arrangements have almost the same pressure drop, but in C arrangements due to the reduction of transverse distance between the cylinders, a significant increase has been occurred in flow pressure drop (about 4 times more than arrangement A).

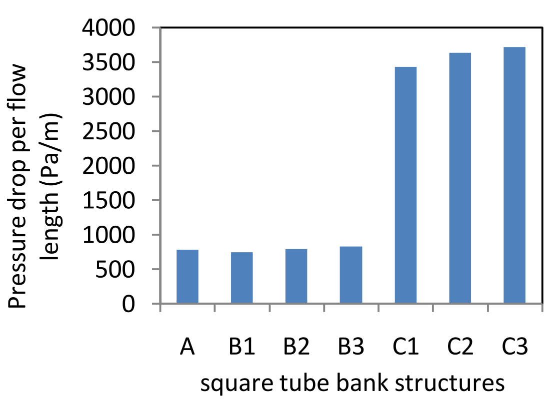


Figure 6: Pressure drop per flow length unit in different arrangements for Reynolds 40000

Discussion and conclusion

in this study by changing the arrangement of a square tube bank which has been done by placing the smaller cylinders on larger cylinders upstream and reducing the longitudinal and transverse distance between the cylinders and resizing of the smaller tubes, the flow vorticity control through the tubes and its impact on the drag force, pressure drop and heat transfer have been investigated. The flow is unsteady, turbulent and incompressible air with Reynolds of 40000. In order to do this study, the finite volume method has been used for solving the equations of Navier - Stokes and energy. The results show that the existence of unequal tubing and reduction of longitudinal distance between them, significantly reduce the drag coefficient (about 50%) and increase the flow Nusselt (an average of 9% in B2 arrangement and 26% in the arrangement B3), while the pressure drop does not change much. B2 and B3 arrangements are preferred over the arrangement A, due to the more heat generation and less vortices and drag coefficient. In addition they have smaller size and physical dimensions that will save material and cost.

Nomenclature

Cp	specific thermal capacity (J/kg K)
$\acute{C}_P$	fluctuating pressure coefficient
$\overline{C_P}$	time averaged pressure coefficient
d	small square width (m)
D	large square width (m)
D_h	hydraulic diameter (m)
h	convection coefficient (W/m^2k)

K	kinetic energy (m^2/s^2)
k_f	thermal conductivity (W/m K)
L	flow length (m)
NU	Nusselt number
$\overline{P}$	time averaged pressure (pa)
Re	Reynolds number
s_T	transverse distance (m)
s_L	longitudinal distance (m)
t	time (s)
T	temperature (K)
$\acute{T}$	fluctuating temperature (K)
$\mathbf{u_i'}$	fluctuating velocity (m/s)
$\overline{\mathbf{u}}_\mathbf{i}$	time averaged velocity (m/s)
$\mathbf{u_\infty}$	inlet velocity (m/s)
$\mathbf{W}$	tube bank columns

Greek symbols

α_ε	inverse effective prandtl number related to ε
α_k	inverse effective prandtl number related to $\mathbf{k}$
$\Delta\mathbf{p}$	pressure drop (Pa)
ε	dissipation rate (m^2/s^3)
μ_{eff}	effective viscosity (Kg/m.s)
$\boldsymbol{\rho}$	density (kg/m^3)
$\boldsymbol{\upsilon}$	kinematic viscosity (m^2/s)
$\boldsymbol{\Gamma}$	thermal diffusion rate

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