STUDIA UBB CHEMIA, LVIII, 4, 2013 (p. 265-274)
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*Dedicated to Professor Liviu Literat*
*On the occasion of his 85th birthday*

# INTENSIFICATION OF CONVECTIVE HEAT TRANSFER IN STRAIGHT PIPES BY USING SOME TURBULENCE PROMOTERS

**ANDRA TĂMAŞ[a,*], SORINA BORAN[a]**

**ABSTRACT.** In this paper was studied the influence of some turbulence promoters (annular or spiral baffles), placed in the inner tube of a double pipe heat exchanger, on heat transfer intensification. These studies were performed in order to enhance heat transfer by convection of the fluid that moves through these spaces, flowing in transient or turbulent regime. The quantitative determination of heat transfer intensification in the presence of turbulence promoters was performed by calculating, from the energy balance data, the total or partial heat transfer coefficients and comparing these values to the values calculated in the absence of promoters.

***Keywords:*** *baffles, heat exchanger, heat transfer, partial/total heat transfer coefficient, turbulence promoters*

## INTRODUCTION

Due to an insufficient high value of the heat transfer coefficient by convection, the heat transfer represents a limiting factor for the performance of heat exchangers. For this reason, any technical measure is useful, functionally and economically justified, which is able to improve the level of these performances. From among the methods that enhance convection, studied and partially implemented in practice, the following can be mentioned: increasing fluid velocity, vibration/pulsation of the fluid or of the heat exchanger surface, the use of electrostatic or electromagnetic fields, injection and absorption of

[a] *Universitatea "Politehnica" Timisoara, Facultatea de Chimie Industriala şi Ingineria Mediului, Bd. V. Parvan Nr. 6, RO-300223 Timisoara, Romania*
** Corresponding author: andra.tamas@upt.ro*

the boundary layer, the use of additives or turbulence promoters (uniform or discrete artificial roughness, turbulence generators) [1-5].

Turbulence promoters cause an additional local turbulence which leads to the transfer speed increase. They can be surface or moving fluid promoters.

In the first case, heat transfer can be enhanced by modifying the surface state, respectively, by introducing artificial roughness (channel type, rib type or granular). These modify the boundary layer flow, which leads to the increase of superficial heat transfer and of friction coefficient as well.

Moving fluid promoters create an artificial turbulence throughout its mass. The elements are placed inside the pipe and can be flat spirals, turbulence pads, conical surfaces, twisted metalic bands. In practice, the highest efficiency is achieved by using turbulence pads, to Reynolds number values lower than $10^4$. Also, by placing turbulence promoters on the outer surface of the inner tube of an annulus, the heat flow increases up to 60 % compared to the case of their absence [1,2,6].

By using static mixer elements (non-moving elements with a helical form, combining alternating right and left hand) in each heat exchanger tube, film build-up on the inside walls is greatly reduced. Process fluid is continuously pushed from the center of each tube to the wall and back to the center, eliminating thermal gradients and boosting the inside film coefficient. Kenics static mixer elements produce a more uniform, consistent transfer process, with three to seven time greater heat transfer rates than empty tubes alone [7,8].

Addition of milli or micro sized particles to the heat transfer fluid is one of the techniques employed for improving heat transfer rate. This method presents some disadvantages: high pressure loss, clogging, erosion of the material of construction. These problems can be overcome by using nanofluids - dispersion of nanosized (below 50 nm) particles ($Al_2O_3$, CuO, $Cu_2O$, $TiO_2$, SiC, $SiO_2$, copper, gold, silver) in a base fluid. Nanoparticles increase the thermal conductivity of the base fluid which in turn increases the heat transfer rate. The extent of enhancement depends on the nanoparticle material type and volume fraction. Unfortunately, nanoparticles also increase the viscosity of the base fluid, resulting in higher pressure drop for the nanofluid compared to the base fluid [9-13].

In this paper was studied the effect of some turbulence promoters placed on the air flow path of a coaxial tube heat exchanger on the intensification of heat transfer to it. The quantitative evaluation of heat transfer intensification was performed by calculating, from the energy balance data, the total heat transfer coefficient in the absence and presence of turbulence promoters, respectively.

The pressure loss caused by air flow through the central tube was measured with an inclined-tube manometer [14].

## RESULTS AND DISCUSSION

The total heat transfer coefficient $K$ is determined from the energy balance equation:

$$Q_1 = Q_2 + Q_3 = Q_4 \tag{1}$$

$$Q_2 = m_{air} \cdot c_{air} \cdot (\Delta t)_{air} \tag{2}$$

$$Q_4 = K \cdot A \cdot \Delta t_{med} \tag{3}$$

where $Q_1$ is the heat delivered by the hot fluid (water); $Q_2$ is the heat received by the cold fluid (air); $Q_3$ is the lost heat; $Q_4$ is the transmitted heat; $m_{air}$ - the air flow rate, $kg \cdot s^{-1}$; $c_{air}$ - specific heat of air to the average temperature, $J \cdot kg^{-1} \cdot K^{-1}$; $(\Delta t)_{air}$ - the temperature differences for air at the heat exchanger extremities, $^\circ C$; $\Delta t_{med}$ - the average temperature difference between air and water, $^\circ C$. The heat transfer area $A$ was calculated with the relationship $A = \pi \cdot d_{med} \cdot H_1$, where $d_{med}$ and $H_1$ are the average diameter and the height of the inner pipe ($A = 0.05 m^2$). The unit for $Q_1$-$Q_4$ is J $s^{-1}$(W).

The partial heat transfer coefficient for air $\alpha$ was determined from the values of total heat transfer coefficient $K$, taking into account the thermal resistance to conduction of the pipe wall ($2.6 \cdot 10^{-6}\ m^2 \cdot K \cdot W^{-1}$) and the partial heat transfer coefficient for water ($\alpha_w = 920 W \cdot m^{-2} \cdot K^{-1}$) analytically calculated in the case of heat transfer in laminar flow through vertical pipes, when the forced movement direction is reversed to the free convection (GrxPr $>5 \times 10^5$) [14,15], Gr and Pr being Grashof and Prandtl numbers.

For each analyzed case (absence, respectively, the presence of annular baffles), the increasing of air flow rate (therefore, hydrodynamic regime intensification) leads to the increasing of the heat received by the air. Also, the heat received is greater the more increases the number of annular baffles ($n$) placed in the inner tube (decreases their step, $l_1=H_1/n$).

For the situation of transient flow regime, it was found that the dependence between the heat received by the air and the value of Reynolds number is linear, the established equations are presented in Table 1.

**Table 1.** Equations $Q_2$=f($Re$) in the presence of annular baffles in transient regime

| Step, mm | Eq. $Q_2$=a$Re$ + b | $R^2$ |
|---|---|---|
| - | $Q_2$=0.0028$Re$ + 8.4 | 0.9906 |
| 200 | $Q_2$=0.0046$Re$ + 12.7 | 0.9982 |
| 100 | $Q_2$=0.0048$Re$ + 17.9 | 0.9927 |
| 50 | $Q_2$=0.0054$Re$ + 19.5 | 0.9913 |

The favorable effect of the presence of annular baffles on the air path is also evidenced by the values' evolution for partial ($\alpha$) and total ($K$) heat transfer coefficients. In Fig. 1 is comparatively shown the dependence $\alpha$=f($Re$) for air flow over annular baffles (placed at different steps), respectively, in their absence.

In the absence of annular baffles were analytically calculated the values of partial heat transfer coefficient for air $\alpha$, using eq. (4) for transient flow regime and eq. (5) for turbulent regime. It was admitted $\varepsilon_l$=1.15 for the ratio $H_1/d_i$=23.1, where $d_i$ is the inner diameter of the inner tube [15].

$$\frac{Nu}{\mathrm{Pr}^{0.43}\cdot\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_p}\right)^{0.25}}=-9.214+5.964\cdot10^{-3}\cdot\mathrm{Re}-1.786\cdot10^{-7}\cdot\mathrm{Re}^2 \quad (4)$$

$$Nu=0.018\cdot\varepsilon_l\cdot\mathrm{Re}^{0.8} \quad (5)$$

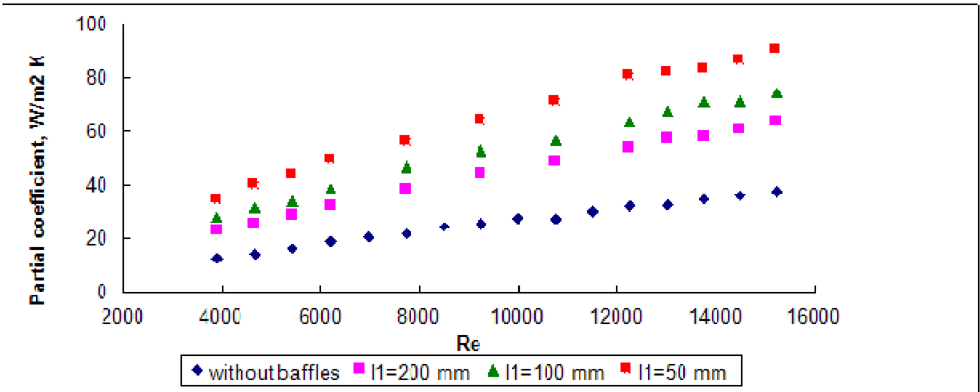


**Figure 1.** The variation of partial heat transfer coefficient for air *vs.* hydrodynamic regime in the presence of annular baffles

In Fig. 2 are comparatively presented the values of partial coefficient $\alpha$ analytically calculated and those established from experimental measurements. Lower values obtained from experimental measurements may be due to the existence of the lost heat.

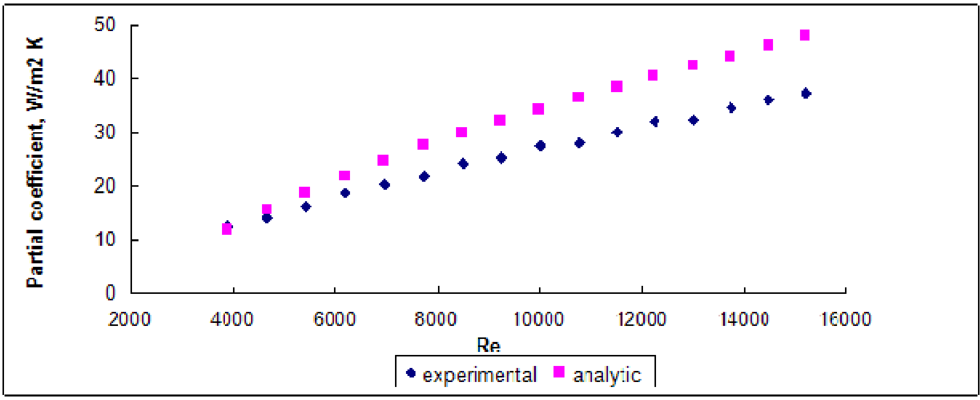


**Figure 2.** Analytical and experimental values for *α* in the absence of annular baffles

From the values of partial heat transfer coefficient experimentally determined were calculated the corresponding values of Nusselt number (*Nu*), both in the presence and absence of annular baffles. In the conditions of turbulent regime, from the *Nu*=f(*Re*) dependence and admitting the value 0.8 for the exponent of Reynolds number in the relation $Nu=CRe^m$, the values of *C* constant were calculated.

The constant *C* is influenced by the geometrical dimensions of the baffles and, respectively, the ratio $l = l_1/\delta$. From Table 2 it is observed that the step reduction leads to the heat transfer intensification to the same hydrodynamic regime. From the measurements performed with a number of 3; 4; 6; 8 and 12 baffles, respectively, $20 \le l \le 80$, it was obtained that the constant *C* can be expressed in relation to the step, through the relationship $C=0.23(l_1/\delta)^{-0.5}$.

**Table 2.** *Nu*=f(*Re*) equations as a function of the annular baffles step

| Step, mm | Re>$10^4$ | Re<$10^4$ | |
|---|---|---|---|
| | **Eq.** $Nu = C \cdot \mathrm{Re}^{0.8}$ | **Eq.** $Nu = a + b \cdot \mathrm{Re} + c \cdot \mathrm{Re}^2$ | $R^2$ |
| 200 | $Nu = 0.027 \cdot \mathrm{Re}^{0.8}$ | $Nu = 6.9 + 3.8 \cdot 10^{-3} \cdot \mathrm{Re} + 0.2 \cdot 10^{-8} \cdot \mathrm{Re}^2$ | 0.9996 |
| 100 | $Nu = 0.034 \cdot \mathrm{Re}^{0.8}$ | $Nu = 6.9 + 3.8 \cdot 10^{-3} \cdot \mathrm{Re} + 0.2 \cdot 10^{-8} \cdot \mathrm{Re}^2$ | 0.9964 |
| 50 | $Nu = 0.051 \cdot \mathrm{Re}^{0.8}$ | $Nu = 5.7 + 7.6 \cdot 10^{-3} \cdot \mathrm{Re} - 18 \cdot 10^{-8} \cdot \mathrm{Re}^2$ | 0.9978 |

In Fig. 3 are shown the values of Nu number calculated from the experimental values of the partial heat transfer coefficient, respectively, analytically calculated with equations derived above (Table 2), depending on the values of Re number ($Re>10^4$). In the presence of annular baffles there is a great overlap between the two sets of values.

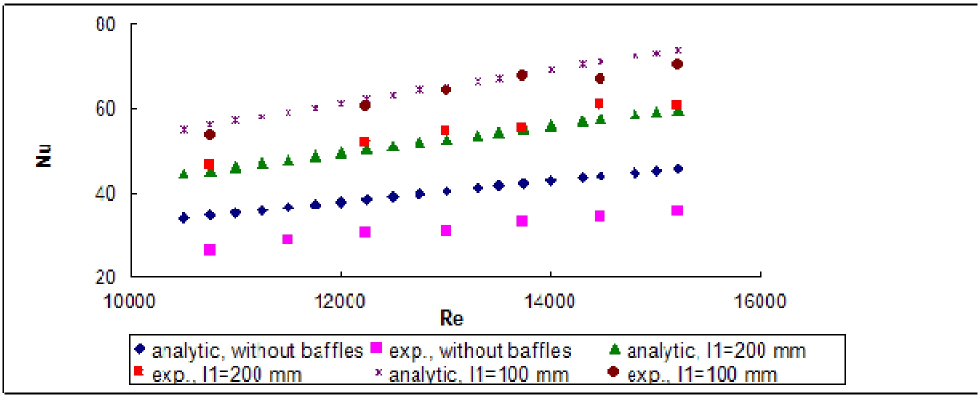


**Figure 3.** Analytical and experimental values of Nusselt number in the presence or absence of annular baffles, $Re>10^4$

Fig. 4 shows the dependence $Nu$=f($Re$) calculated from experimental values, for air flow in transient regime. The corresponding equations are given in Table 2.

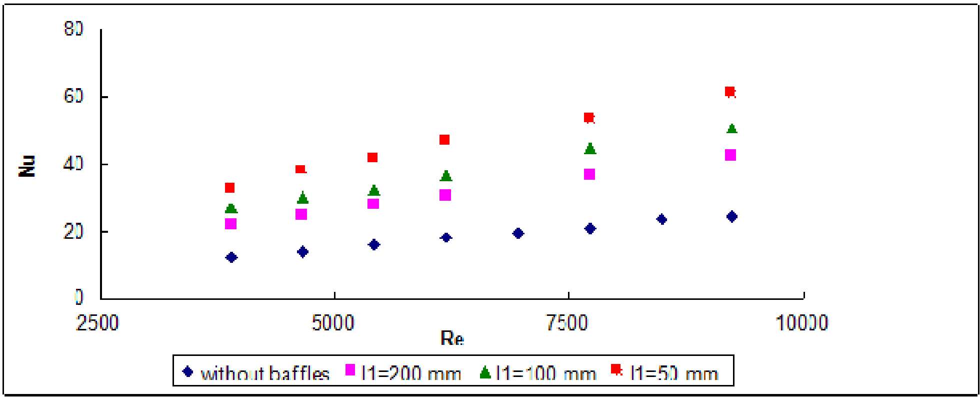


**Figure 4.** Experimental values of Nusselt number in the presence of annular baffles, $Re<10^4$

Heat transfer intensification reflected by the ratio $\alpha_n/\alpha_0$ (*n* being the number of annular baffles, *0* the situation without baffles) is more effective the greater the number of turbulence promoters (decreases the step), regardless of flow regime. However, at the same *l* value, enhancing heat transfer is more pronounced when operating in transient flow regime, Fig. 5.

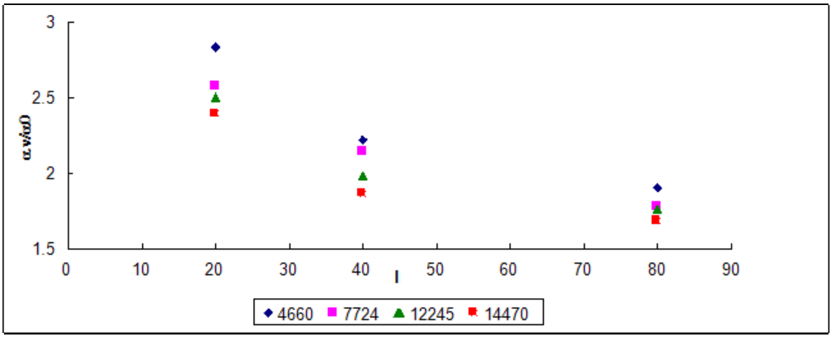


**Figure 5.** The dependence $\alpha_n/\alpha_0$ =f($l$) at different *Re* values

And if they were used copper spires with wire thickness $\delta$=2.5 mm as turbulence promoters, having the same winding step $w_s$ (25 mm) but different lengths $L$ (125 and 250 mm) and, respectively, the same length (250 mm) but different winding steps (25, 50 and 75 mm), it was found a benefic effect in terms of heat transfer to air. This fact is evidenced by the values evolution of air partial coefficient $\alpha$ depending on the hydrodynamic regime, Fig. 6a and 6b.

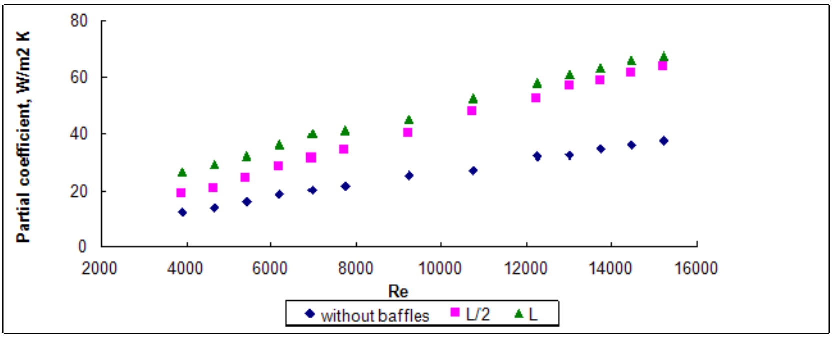


**Fig. 6a**

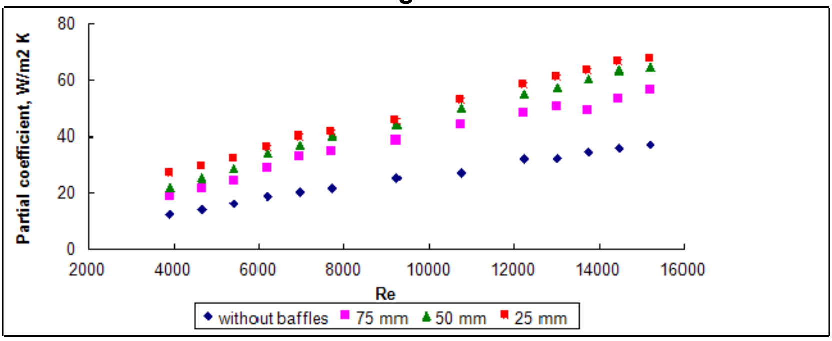


**Fig. 6b**

**Figures 6a, b.** The dependence $\alpha$=f(*Re*) for spires with different:
a) lengths; b) winding steps

The percentage increase in total heat transfer coefficient *K,* in each of the analyzed cases is shown in Fig. 7.

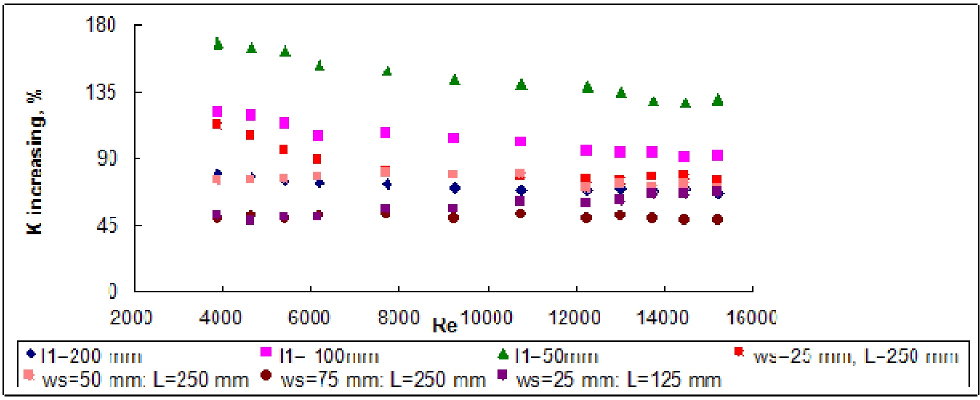


**Figure 7.** The increasing of the total heat transfer coefficient in the presence of turbulence promoters

## CONCLUSIONS

Based on the information from literature on the role of turbulence promoters on the intensification of heat transfer with fluids flowing in transient and turbulent regime, a double pipe heat exchanger was designed and made, with annular or spiral baffles inside the inner pipe that caused a more efficient "mixing" of the hot fluid.

In all the analyzed cases, the air rate flow increasing leads to the decrease of the difference between the air temperature at the top and bottom of the inner pipe.

The heat received by the air $Q_2$, the partial coefficient $\alpha$ and, respectively, the total heat transfer coefficient *K* increase with the air rate flow increasing and are higher in the presence of turbulence promoters.

In the case of annular baffles, the heat transfer to the air is more effective since their number is higher (total heat transfer coefficient increases between 60 and 160 %).

And the presence of spiral baffles on the air path has a benefic effect in terms of heat transfer intensification to it. The intensification of heat transfer in this case is influenced by both the length and the winding step of the spires used. However, in the experiments conducted under these conditions, the use of spires led to more modest results than those obtained when using annular baffles.

## EXPERIMENTAL SECTION

The heat exchanger used is made of pipes with the following dimensions: the copper inner pipe-diameter $d = 28x1\,mm$, height $H_1 = 600\,mm$, the outer pipe-diameter $D = 49x1.5\,mm$, height $H = 580\,mm$. The inner tube is equipped with an additional part of 450 mm length to stabilize air flow. The turbulence promoters of the inner pipe were rings with dimension $25x2\,mm$ placed at different distances (steps) or spires with different winding step and lengths. Both types of promoters were made of copper wire with a diameter $\delta = 2.5\,mm$.

The heat exchanger is vertically positioned. The air enters on the base of the inner pipe where is ensured its flow regulation and measurement. The heat required for the air warming is delivered by the hot water circulating through the annulus and is transmitted through the cylindrical surface of tubular wall. The air temperature was measured with thermometers placed at the top and bottom of the pipe.

The hot water from the annulus is provided by a thermostat and moves upward, being recycled with a constant rate flow of 0.145 $m^3 \cdot h^{-1}$. Its temperature was measured both at the entrance (constant value 65ºC) and exit of the annulus, the water cooling has not been greater than 1ºC. Experiments were performed at air flow rates from 5 to 20 $m^3 \cdot h^{-1}$, providing both a turbulent and transient flow.

Table 3 presents a set of experimental measurements in the case of annular baffles ($l_1 = 100\,mm$).

**Table 3.** Experimental data for annular baffles ($l_1 = 100\,mm$)

| Air flow rate, $m^3\,h^{-1}$ | Air temperature, °C | | Water temperature, °C | | $(\Delta t)_{air}$, °C | $\Delta t_{med}$, °C |
|---|---|---|---|---|---|---|
| | bottom | top | bottom | top | | |
| 5 | 26.5 | 50.0 | 65 | 64.5 | 23.5 | 24.9 |
| 6 | 28.0 | 49.5 | | | 21.5 | 24.7 |
| 7 | 29.5 | 49.0 | | | 19.5 | 24.5 |
| 8 | 30.0 | 49.0 | | | 19.0 | 24.3 |
| 10 | 31.0 | 49.0 | | | 18.0 | 23.9 |
| 12 | 32.5 | 49.0 | | | 16.5 | 23.3 |
| 14 | 34.5 | 49.0 | | | 14.5 | 22.5 |
| 16 | 35.0 | 49.0 | | | 14.0 | 22.3 |
| 17 | 36.0 | 49.5 | | | 13.5 | 21.5 |
| 18 | 37.0 | 50.0 | | | 13.0 | 20.8 |
| 19 | 38.0 | 50.0 | | | 12.0 | 20.4 |
| 20 | 39.0 | 50.5 | | | 11.5 | 19.7 |

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