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Electrification of a small furnace

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The electrification of industrial furnaces may be desirable to lower carbon dioxide emission, provided that electricity is produced using "green" sources. Electrical heating elements are available in a variety of shapes, materials, and sizes, which can be used to design an electrical furnace where the tubes are not necessarily arranged in 1 or 2 rows close to the wall as in a traditional fuel furnace.
This work presents some design solutions for a small cylindrical furnace, using vertical tubes for the process fluid and vertical Tubothal® cylindrical heaters of the same length. The basic layout recalls that of a hive: the tubes are placed at the corners of a hexagon, with the electrical heater at the center: replicating this hexagonal pattern, tubes and heaters take up all the section of the furnace. Radiant heat is mutually exchanged between each element (tube, heating device and refractory lining) and all the other elements in sight, being a function of the absolute temperature, and the geometry and the location of each couple of elements. Depending on the flow arrangement of the process fluid, each tube may be at different temperature, and the same is true for each heater, depending on their different location inside the furnace. Accordingly, the problem involves a great number of variables and heat transfer equations, and a Matlab code was arranged to determine temperatures and heat exchanged by each element. The feasibility of the proposed arrangements was assessed by comparing temperatures and power outputs of Tubothal® heating elements with the manufacturer data; it is interesting to notice that, for all tested configurations, refractory lining was found to provide around 50% of the heat globally supplied to the process fluid.

1. Introduction

Climate changes show that we need to lower, as soon as possible, the carbon impact of human activities: to achieve the EU's objective of climate neutrality by 2050, increasingly restrictive limits to the use of fossil fuels are being imposed. This scenario forces the process industry to undertake measures to reduce its emissions, by improving equipment efficiency, achieving better heat integration in the process, adopting techniques for CO_2 capture and storage, and using cleaner energy sources, such as hydrogen or electricity. The conversion of heating consumptions of industrial furnaces, mostly based on the combustion of heavy liquid fuels, into electrical ones may represent an important step in the energy transition route, if electricity is produced by renewable sources (wind power, solar, etc.), which are increasingly available.
The electrification of a furnace implies the use of resistive radiant heaters, which are available in a variety of shapes, including rectangular panels, half-cylinders, and tubes. In traditional fuel furnaces, only 1 or 2 rows of tubes close to the walls can be provided, since additional tubes may prevent radiant heat to reach the tubes behind them. On the contrary, the use of electrical heaters allows devising design solutions where tubes and heating elements can be alternated in the whole radiant section of the furnace, with the aim to maximize radiant heat exchange obtaining a compact geometry.
In the present work, a study case of a small cylindrical furnace, provided only with an electrical radiant section, is presented. The design is based on the use of commercial cylindrical radiant heating elements of almost the same dimensions (length and diameter) as the tubes. A sort of "hive" structure is devised, with vertical tubes at the corners of a hexagon and a vertical heating element at the center. Some different arrangements are obtained replicating this hexagonal pattern, varying the number of heating elements, and the results are discussed, comparing the obtained values of temperature and heat power of the heating elements with the manufacturer data.

2. Design of the electric furnace

The case study is a cylindrical furnace used to heat the charge to a reformer: in particular, 36000 kg/h of a hydrocarbon vapors and hydrogen mixture, at a pressure of 2750 kPa, should be heated from 280 to 350 °C, requiring a duty of 2000 kW, which was increased to 2020 kW, assuming 1% heat losses through the walls.
The average properties of the mixture are listed in Table 1.

Table 1: Average properties of the mixture

Molecular weight (kg/kmol)	Density (kg/m^3)	Heat capacity (J/kg°C)	Thermal conductivity (W/m°C)	Viscosity (mPa·s)
56	31.5	2857	0.057	0.014

The process fluid flows in vertical standard stainless steel 6" tubes, schedule 40, 20 ft long (length 6.096 m, outer diameter 168.3 mm, and thickness 7.11 mm). Preliminary estimates indicate that 2-passes should be used for the process fluid and that 24 tubes will be required. The heat transfer coefficient of the process fluid was estimated with the usual equations (Sinnot and Towler, 2020) and resulted in h_{PF} = 1036 W/m^{2}K.

2.1 Heating elements

Electrical heating converts electricity into heat, by the Joule effect; performances and operating life of the heating elements depend on the used material, which should be ductile, should exhibit a high melting point and resistivity, and good stress, oxidation, and corrosion resistance. In the present work reference is made to Tubothal® elements (see Figure 1a), high-power, long-life cartridge heaters, for vertical or horizontal mounting, coming in diameters ranging from 68 to 170 mm and lengths from 0.5 to 6 m (Kanthal, 2020a). They are made in Kanthal APM™ alloy (72.2% Fe, 22% Cr, 5.8 Al), which can operate continuously at 1425 °C.
The maximum design power outputs for all standard element diameters at different furnaces temperatures is shown in Figure 1b: for each element diameter, the power output increases linearly with the length, and decreases as the furnace temperature is increased.

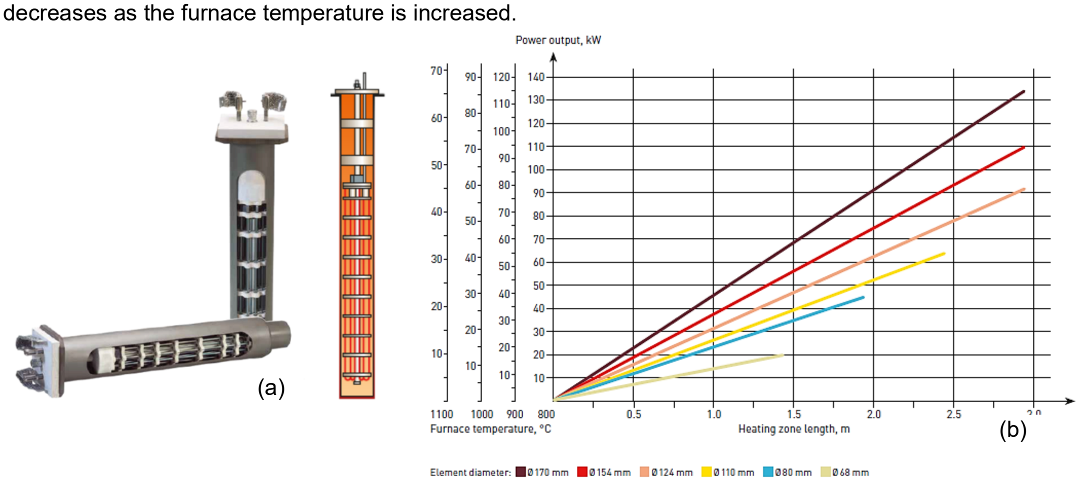

Figure 1: a) Tubothal® heating elements; b) Maximum design power outputs for all standard element diameters at different furnace temperatures (Kanthal, 2020b).

For the study case, Tubothal® elements with length 6 m and diameter 170 mm were selected: their size is very similar to that of the tubes (diameter 168.3 mm, length 6.096 m). Based on the graph of Figure 1b, one of such Tubothal® will give a maximum power output of around 180 kW for a furnace temperature of 1000 °C: accordingly, around 11 heating elements will be needed to provide the desired total duty of 2020 kW.

2.2 Furnace geometry

A cylindrical furnace was assumed, consisting only of the radiant section, provided with vertical tubes, with a tube pitch 420 mm (i.e. 2.5 of outer diameter). The use of vertical cylindrical heaters, approximately the same size as the tubes, suggested to adopt a layout based on a "hive" structure, consisting of 6 tubes, placed at the corners of a hexagon, with the heater in the center (Figure 2a). The hive pattern is replicated to fill the entire

section of the furnace, ensuring that each tube will receive the heat radiated by more than one heating element (Figure 2b): the black lines show the 2 passes of the process fluid.
The minimum distance between the center of tubes or heaters and the refractory lining was also set at 420 mm, giving a diameter of the furnace equal to 3.36 m.

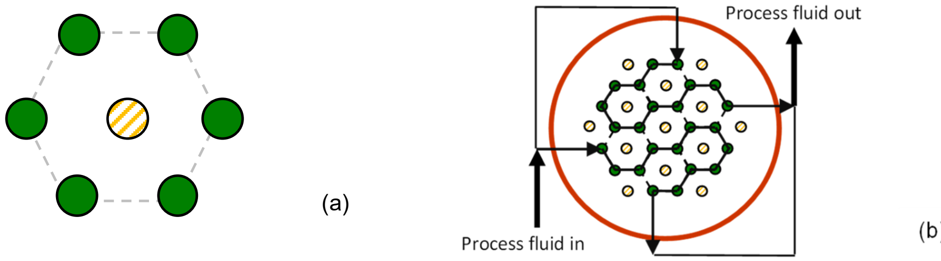

Figure 2: a) Basic "hive" cell: Tubothal® heating elements: yellow; process fluid tubes: green; b) Arrangement of heating elements, process fluid tubes and fluid passes.

2.3 Radiant heat exchange

Heat is assumed to be transferred entirely by a radiant mechanism in the furnace: convection contribution is regarded as negligible since air is almost static inside the equipment.
Radiant heat is mutually exchanged between all the elements in the furnace, i.e. Tubothal® heaters, process fluid tubes, and refractory lining; only the contribution of the refractory walls is considered, neglecting that of floor and ceiling. All the elements are assumed to present opaque and diffuse gray cylindrical surfaces; then, the heat exchanged between any couple of elements i and j, can be then expressed as (Howell et al., 2010):

$$Q_{ij} = A_i \cdot f(F_{ij}, \varepsilon_i, \varepsilon_j) \cdot \sigma \cdot (T_i^4 - T_j^4) \quad (1)$$

In Eq (1) ε_i and ε_j are the emissivity of i and j elements, respectively: for Tubothal®, ε_{HE}, is 0.7, in fully oxidized condition (Kanthal, 2020a); for process tubes ε_{PT} = 0.8 and for refractory lining ε_{RL} = 0.94 (Howell et al.2010). F_{ij} are the view factors, which account for the geometry and the relative position of the elements. For each couple formed by Tubothal® heaters (HE) and process fluid tubes (PT) (heater-heater, tube-tube, and heater-tube) they are first estimated as for heat exchange between two indefinite cylinders (Howell et al., 2010):

$$F_{ij} = \frac{\sqrt{h^2-4} - h + 2 \cdot arcsin\left(\frac{2}{h}\right)}{2 \cdot \pi} \quad (2)$$

Then, the corresponding view factor for finite cylinders is obtained by multiplying the value calculated from Eq(2) by an adjustment factor, lower than unity (Jiang et al., 2020).
Due to the symmetrical arrangement of process tubes and Tubothal® heating elements, the distances between their axis are 0.42, 0.727, or 1.111 m: the corresponding view factors are listed in Table 2.

Table 2: Reciprocal view factors of process tubes and Tubothal® heating elements

Distance (m)	F_{ij} (i = j = HE)	F_{ij} (i = j = PT)	F_{ij} (i = HE; j = PT)	F_{ij} (i = PT; j = HE)
0.42	-	0.0628	0.0632	0.0628
0.727	0.0347	0.0347	-	-
1.111	-	0.0217	0.0219	0.0217

Both Tubothal® heating elements and process fluid tubes are completely enclosed inside the furnace wall: accordingly, for each element, the view factor relevant to the heat exchange with the refractory lining is the complement to 1 of the sum of the view factors of such element with all the other elements inside the furnace.
Concerning Figure 3 a, the view factors with the refractory lining are as follows:

- 0, for the heating element III placed at the center of the furnace, and for the tubes C, located in the central hexagon, which have no view of the furnace lining;
- 0.3162, for the heating elements I inside the outer hexagons and 0.6536, for those II in the zone between the hexagons and the furnace wall;
- 0.3542, for the inner process tubes B, and 0.5361, for the outer A ones.

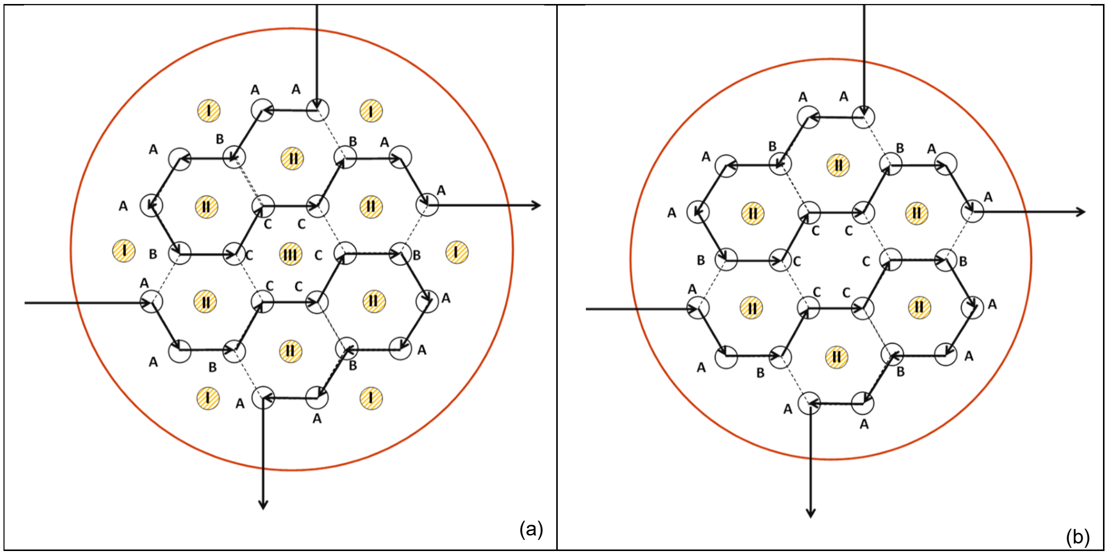

Figure 3: Arrangement of heating elements (locations I, II, and II) and tubes (locations A, B, and C) in the furnace; a) scheme 1; b) scheme 3.

The heat Q_{ij} exchanged between each couple of elements in the furnace was calculated with the equations listed in Table 3 (Howell et al., 2010).

Table 3: Radiant heat exchange equations (Howell et al., 2010)

Radiant heat exchange equations	
$Q_{PT_i-PT_j} = \dfrac{A_{PT} \cdot \sigma \cdot F_{PT_i-PT_j} \cdot \left(T_{TP_i}^4 - T_{PT_j}^4\right)}{\dfrac{2}{\varepsilon_{PT}} - 1}$	$Q_{HE_i-HE_j} = \dfrac{A_{HE} \cdot \sigma \cdot F_{HE_i-HE_j} \cdot \left(T_{HE_i}^4 - T_{HE_j}^4\right)}{\dfrac{2}{\varepsilon_{HE}} - 1}$
$Q_{PT_i-HE_j} = \dfrac{A_{PT} \cdot \sigma \cdot F_{PT_i-HE_j} \cdot \left(T_{TP_i}^4 - T_{HE_j}^4\right)}{\dfrac{1}{\varepsilon_{PT}} + \dfrac{1}{\varepsilon_{HE}} - 1}$	$Q_{HE_i-PT_j} = \dfrac{A_{HE} \cdot \sigma \cdot F_{HE_i-PT_j} \cdot \left(T_{HE_i}^4 - T_{PT_j}^4\right)}{\dfrac{1}{\varepsilon_{HE}} + \dfrac{1}{\varepsilon_{PT}} - 1}$
$Q_{PT_i-RL} = \dfrac{\sigma \cdot \left(T_{PT_i}^4 - T_{RL}^4\right)}{\dfrac{1-\varepsilon_{PT}}{\varepsilon_{PT} \cdot A_{PT}} + \dfrac{1}{F_{T_i-RL} \cdot A_{PT}} + \dfrac{1-\varepsilon_{RL}}{\varepsilon_{RL} \cdot A_{RL}}}$	$Q_{HE_i-RL} = \dfrac{\sigma \cdot \left(T_{HE_i}^4 - T_{RL}^4\right)}{\dfrac{1-\varepsilon_{HE}}{\varepsilon_{HE} \cdot A_{HE}} + \dfrac{1}{F_{HE_i-RL} \cdot A_{HE}} + \dfrac{1-\varepsilon_{RL}}{\varepsilon_{RL} \cdot A_{RL}}}$

3. Simulation of the electrical furnace

To solve the system of radiant heat exchange equations, listed in Table 3, the temperature of each element (process fluid tube, heater, refractory lining) should be determined, as well as the heat exchanged between each couple of elements. The input data are the heat to be supplied to the process fluid, and its initial and final temperature; all physical properties, the geometry of the system, and the view factors are also known.
The temperature of the process fluid varies as it flows from one tube to the subsequent one of each pass (see Figure 2b) and can be calculated from heat balance. The average temperature of each i tube depends on the total heat received by the tube, ΣQ_{TPi}, and on the average process fluid temperature inside that tube, T_{PFi}:

$$\boldsymbol{T_{PTi} = T_{PFi} + \frac{\sum Q_{TPi}}{A_{TPi} \cdot h_{PF}}} \tag{3}$$

where A_{TPi} the surface of the tube and h_{PF} is the heat transfer coefficient of the process fluid.
Since each tube and heater is at a different temperature, due to the path of the process fluid and the different position inside the furnace, each couple of elements in the furnace (24 tubes, 13 heaters, and the furnace wall) will exchange radiant heat, giving rise to a very large number of equations and variables, which are not easily solved, unless under very rough simplifications, such as assuming that all tubes and all heating elements are at the same temperature, T_{PT} and T_{HE}, respectively. In all other cases, a code should be written, providing

reasonable initial values for the variables and suitable criteria to iterate, adjusting these values to minimize the difference between the heat provided to the process fluid in the furnace and the required duty.

3.1 Scheme 1

Regarding the furnace arrangement of Figure 3a, the first very simplified assumption made was that all the tubes were at the same average temperature, T_{PT} = 340 °C. Assuming also that all heating elements are at the same temperature, T_{HE}, the system of equations in Table 3 is easily solved obtaining the temperature of the heaters, T_{HE} = 923 °C, and of the refractory lining, T_{RL} = 707 °C.
Then, the temperature of each tube, ranging from 358 and 432 °C, depending on the sequence in the two-passes of the process fluid, was calculated from eq(3) and the system of equations was solved again, still assuming all heating elements to be at the same temperature, obtaining T_{HE} = 920 °C and T_{RL} = 702 °C.
Finally, also the assumption that all Tubothal® are at the same temperature was removed: in this case, a Matlab code was written, and the temperatures of each tube and heater were calculated. The results were as follows: temperatures of the tubes in the range 369 - 429 °C; temperature of the refractory lining 681 °C, temperatures of the Tubothal® heaters in the range 881 – 913 °C, except the central one (III, see Figure 3a), whose temperature was much higher (1164 °C). It should be noticed that the refractory cannot irradiate the six C tubes placed in the most inner hexagon (Figure 3a), and this explains the higher temperature needed for the central heater III. In fact, the refractory lining gives a substantial contribution to the overall heat transfer, providing around 43% of the heat received by the inner B tubes, and 62% of that received by the outer A ones.
The power supplied by the heating elements was in the range 134 - 149 kW, except the central III one (213 kW): such values are remarkably lower than that of about 275 kW, which is the maximum power output for a 6 m long and 0.17 m diameter Tubothal®, at the lowest furnace temperatures reported in Figure 1b (800 °C).

3.2 Scheme 2

The results obtained from Scheme 1 configuration showed that the number of heating elements can be reduced: accordingly, the six I Tubothal® heaters located outside the hexagon structure (Figure 3a) were removed, and the calculations were repeated, with 7 heaters instead of 13. By removing the most external Tubothal® heaters, the configuration becomes more compact and a smaller inner diameter of the furnace (i.e. 3.062 m) was assumed. With this new geometry, tube-refractory lining and Tubothal® – refractory lining view factors change and become higher than in the previous Scheme 1 configuration: the values 0.3856 for the II heating elements, 0.4604 for the inner B process tubes, and 0.5989 for the outer A ones are obtained.
The results of the simulation show that, as expected, the temperature of the heaters is higher than in Scheme 1, ranging from 958 to 963 °C (but is 1514 °C for the central III Tubothal®), while that of the refractory lining is lower (602 °C), since the external I Tubothal® heaters have been removed. The contribution of the refractory lining to the heat received by the tubes is unchanged for the inner B tubes (43%), while decreases for the outer A ones (52%); the temperatures of the tubes are like Scheme 1 (363 – 431 °C).
The power supplied by the heating elements is in the range 192 - 196 kW, except for the central III one (855 kW). This value is clearly unacceptable; moreover, the temperature of this Tubothal® exceeds the maximum allowable temperature for Kanthal alloy: therefore Scheme 2 configuration was discarded.

3.3 Scheme 3

This configuration (Scheme 3), shown in Figure 3 b, is like Scheme 2, but also the central heating element III has been removed: this allows radiant heat to be exchanged also between the C tubes located in the central hexagon, with a view factor of 0.0297 (such exchange was previously obstructed by the central heater III). Consequently, also the Tubothal® – refractory lining and some tube-refractory lining view factors change: 0.4204 for the II heating elements and 0.6206 for the outer A tubes (that of inner B tubes do not change).
The results are as follows: temperatures of the tubes in the range 364 - 429 °C, temperatures of the heaters in the range 1076 - 1081 °C, with power outputs in the range 334 - 339 kW. The temperature of the refractory lining is 676 °C; its contribution to the heat exchange remained 43% for the inner B tubes, while increased to 67% for the outer A ones.

3.4 Discussion

To assess the feasibility of the tested configurations, the obtained results were compared with the characteristics of the selected Tubothal® heaters. For a correct operation, their temperature should be above 600°C, but their operating life greatly increases as their temperature decreases: moving from 1300 to 1125 °C will increase Tubothal® life by a factor of 5 (Kanthal 2020a), show that. In the tested configurations, the temperature of the heating elements is below 1100 °C, ensuring operation without problems for a long time.

Then, the power output capability of the heaters should be compared with the required one. The graph of Figure 1b shows that a Tubothal® with a diameter 170 mm can supply a maximum design power output of about 46 kW/m when the furnace temperature is 800 °C. This value decreases when the temperature of the furnace increases, being 38 kW/m at 900 °C, 28 kW/m at 1000 °C, and 21 kW/m at 1100 °C, with an almost linear trend. The "furnace temperature" is to be intended as the refractory lining temperature, which, in the configurations under exam, was found around 680 °C. The expected maximum design power output at such furnace temperature is estimated as 56 kW/m, by linearly interpolating the above data. Accordingly, a 6 m long Tubothal® heater should be designed at a maximum power output of approximately 336 kW. The calculated operating conditions of the heaters are on the safe side for Scheme 1 configuration (134-213 kW), while may be critical for Scheme 3 (334-339 kW), and more detailed information about the actual maximum design power output at temperatures around 700 °C should be needed to assess its feasibility.
Finally, an additional configuration (Scheme 4) with 12 heating elements, like Scheme 1, but removing the central III Tubothal® heater, was also tested, obtaining temperatures of 714 °C for the refractory lining and of 924-942 °C for the heating elements, with power outputs in the range 166-171 kW, well below the corresponding maximum design power output, estimated as 319 kW. Accordingly, this configuration can be adopted if the operating conditions of Scheme 3 result too much demanding.

4. Conclusions

The study case demonstrates that substituting a small fuel furnace with an electrical one is technically feasible, using standard heating elements and without the need for drastic changes in the furnace structure. The use of a "hive" geometry, with vertical tubes placed around a central cylindrical Tubothal® heater, may offer some advantages, in terms of radiant heat exchange efficiency and compact layout. In the study case, a duty of 2 MW was obtained with 6/12 Tubothal® heaters, 170 mm in diameter and 6 m long, in a cylindrical radiant section about 3.4/3 m in diameter. The refractory lining gives a remarkable contribution (more than 50%, as an average) to the overall heat received by the process fluid: this allows to reduce the number of needed heaters or to adopt less demanding operating conditions. Both furnace and heating elements temperatures are reasonably low, ensuring high design power outputs and long life to the heaters.
The technical results are therefore encouraging and the effective replacement of fuel furnaces with electrical ones will mainly depend on its economic feasibility.

Nomenclature

A – surface, m^2
F_{ij} – view factor (element i with respect to j)
h – ratio H/R, -
h_{PF} – process fluid heat transfer coefficient, $W/(m^2K)$
H – distance of the centers of the cylinders, m
Q_{ij} – heat exchanged from element i to element j, W/m^2
R – radius of the cylinder, m
T - temperature, K

ε – emissivity, -
σ - Stefan Boltzman constant $5.67 \cdot 10^{-8}$, $W/(m^2K^4)$

Pedices
HE – Tubothal® heater, -
PF – process fluid, -
PT – process tubes, -
RL – refractory lining, -

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