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Thermo-hydraulic design of a finned tube forced-draft air cooler for methanol cooling.

Diseño térmico-hidráulico de un enfriador por aire de tiro forzado con tubos aleteados para el enfriamiento de metanol.

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Abstract.
Air-cooled heat exchangers are one of the most efficient and widely used equipment for heat exchange. The present paper deals with the thermo-hydraulic design of a finned tube, forced-draft air-cooled heat exchanger in order to cool down 28,000 kg/h of a liquid methanol stream from 90 ºC to 45 ºC using a well-known design methodology. Several important design parameters were determined such as the air flowrate (55.44 kg/s), the required number of tubes (277), the heat exchanger width (2.78 m) and the overall heat transfer coefficient (280.95 W/m2.K). The calculated pressure drop of the methanol stream (559.73 Pa) was below the maximum allowable value established by the process, while the fin efficiency had a value of 0.853, which can be considered adequate. The driver power for the fan had a value of 7.21 kW, and the designed air cooler could be satisfactorily used for the requested heat exchange service since the value of the parameter ratio between the projected area of the fan and the face area is higher than the value established by the design methodology.

Keywords.
Air cooler, fan driver power, fin efficiency, pressure drop, thermo-hydraulic design.

Resumen.
Los intercambiadores de calor enfriados por aire son uno de los equipos más eficientes y ampliamente usados para el intercambio de calor. El presente artículo trata acerca del diseño térmico-hidráulico de un intercambiador de calor enfriado por aire de tiro forzado con tubos aleteados con el fin de enfriar 28 000 kg/h de una corriente de metanol líquido desde 90 ºC hasta 45 ºC usando una metodología de diseño bien conocida. Varios parámetros importantes de diseño fueron determinados tales como el caudal de aire (55,44 kg/s), el número requerido de tubos (277), el ancho del intercambiador de calor (2.78 m) y el coeficiente global de transferencia de calor (280.95 W/m^2.K). La caída de presión calculada de la corriente de metanol (559,73 Pa) estuvo por debajo del valor máximo permisible establecido por el proceso, mientras que la eficiencia de la aleta tuvo un valor de 0,853, el cual puede considerarse adecuado. La potencia del motor del ventilador tuvo un valor de 7,21 kW, y el enfriador por aire diseñado puede ser usado satisfactoriamente para el servicio de intercambio de calor requerido debido a que el parámetro relación entre el área proyectada del ventilador y el área frontal es superior que el valor establecido por la metodología de diseño.

Palabras clave.
Enfriador por aire, potencia del motor del ventilador, eficiencia de la aleta, caída de presión, diseño térmico-hidráulico.

1. Introduction

One of the most prevalent solutions in the heat exchanger industry is the use of an air-cooled heat exchanger (ACHE) involving air-cooling operation at environment temperature. ACHEs (also known as air coolers) are used in the cooling system and use ambient air for the cooling medium. Air is a resource that can be obtained without cost consideration in the surrounding environment [1].

Air coolers are heat exchangers composed of one or more banks of finned tubes that are utilized to transfer heat from a hot stream to atmospheric air, usually using fans to promote the air flow. Taking into account environmental concerns such as probable water shortage difficulties and regions where the available water needs expensive treatments, air coolers can be a significant choice in relation to conventional water coolers [2].

The operating principle of an ACHE is the following: hot process fluid enters one end of the ACHE and flows through tubes, while ambient air flows over and between the tubes, which typically have externally finned surfaces. The heat is transferred to the air, which cools the process fluid, and the heated air is discharged into the atmosphere. While this is a fundamentally simple concept, maintaining optimum ACHE performance takes persistence on the part of the end user [3].

Among the numerous important benefits that presents the air cooler are that there is no need of using a second working fluid to achieve the heat exchange (that is, it is not required

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to have a cooling medium due to using air); allows to save energy costs and to make a heat exchanger less in dimensions; no fouling o corrosion in the tube outside surface [4], no contamination and lower maintenance cost [1].

Environmental concerns such as shortage of water, blow-down disposal and thermal pollution tend to favor the ACHE. Although the capital cost of ACHE is generally high, the operating cost is usually considerably lower than that required by a water-cooled heat exchanger [5].

The main disadvantages associated with the use of ACHE are their low temperature performance due to the low air-side heat transfer coefficient, while its performance is affected by the ambient air temperature, particularly on summer season [1].

To meet these disadvantages, a widespread suggested solution for these heat exchangers is the use of extended surfaces or fins in the air cooler tube bundles, such as wavy fins, slit fins, louvered fins and fins with vortex generators [1] flat fins and serrated fins [6]. The use of fins has enhanced the air-side heat transfer coefficient with relative success.

Air coolers are widely used in power, air conditioning, refrigeration [6] chemical and petrochemical processes [1]. There are thousands of these exchangers in use today, performing the cooling and/or condensing of fluids ranging from engine jacket water to process steam to highly viscous tar [3].

The ACHEs are classified mainly as *forced draft* or *induced draft* [5]. The forced draft ACHE (Fig. 1) is the most economical and most common style air cooler where axial fans used to force air across the fin tube bundle are mounted below the bundle and therefore the mechanical sections are not exposed to the hot exhaust air flow. Also, another advantage of this arrangement is the fact that provides direct access to bundle for replacement. The induced draft ACHE involves axial fans to pull air across the fin tube bundle and it is the second most economical arrangement. Contrary to the previous arrangement, the fans are situated above the bundle thus offering greater control of the process fluid and bundle protection due to the additional structure. Fans used in ACHEs are of the axial-flow type [7].

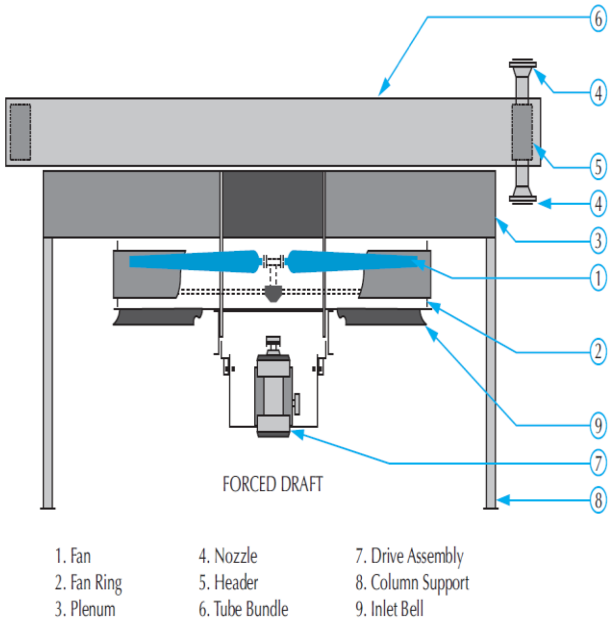

Fig 1. Forced draft air cooler configuration.
Source: Adapted from [8]

The main advantages of forced draft air coolers are [8]:

1. Probably lower horsepower requirements if the effluent air is very hot. (Horsepower varies inversely with the absolute temperature).
2. Better accessibility of fans and upper bearings for maintenance.
3. Better accessibility of bundles for replacement.
4. Accommodates higher process inlet temperatures.

While their main disadvantages are:

1. Less uniform distribution of air over the bundle.
2. Increased probability of hot air recirculation, resulting from low discharge velocity from the bundles, high intake velocity to the fan ring, and no stack.
3. Low natural draft capability of fan failure.
4. Complete exposure of the finned tubes to sun, rain, and hail, which results in poor process control and stability.

The construction of one or more tube bundles, served by one or more fans, complete with a plenum, structure, and auxiliary devices, is called a *bay*. When large flow rates are handled, it is common for the unit to be divided in several bays to facilitate transport and construction. Fig. 2 shows some air cooler configurations. Generally, the design basis consists in using two fan bays [5].

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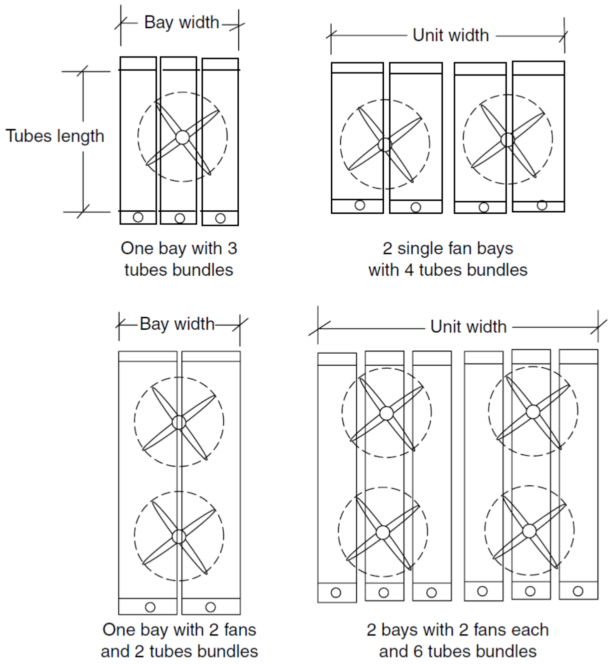

Fig 2. Air cooler configurations.
Source: Adapted from [7]

The heart of the ACHE is the tube bundle. A tube bundle is an assembly of tubes, headers, side frames, and tube supports. Usually, the airside tube surface has extended surface in the form of helical fins to compensate for the low heat-transfer rate of air at atmospheric conditions and low velocities. Because tube-side process pressures normally exceed 1 atm, the optimal tube is usually round and can be of any metal suitable for the process fluid given its corrosivity, pressure, and temperature [3].

The published papers that addressed the design and optimization of air coolers are based on two main approaches: meta-heuristic methods and mathematical programming [2].

The conventional design procedure of ACHEs is similar to that used for shell and tube heat exchangers. By following design guidelines, the first step of the methodology consists of finding a preliminary design for the unit using an approximate overall heat transfer coefficient. Then, by using a trial-error the initial design is iteratively modified in order to obtain an satisfactory design that satisfy specific tolerance criteria. Even though this method can be proficiently used to rate existing units or to obtain new designs via simulation, it is not well suited for optimization especially when many optimization variables should be considered.

Certainly, the application of this design methodology for a rigorous optimization of ACHEs may be laborious and time consuming if discrete decisions (type of the flow regime, type of the finned tube, number of tube rows, number of tube per row, number of passes, fins per unit length and the mean fin thickness) and operating conditions (velocities, pressure drops, overall transfer coefficient) are measured as optimization parameters [5].

More parameters are taken into account in the thermal design of ACHEs compared to shell and tube exchangers. ACHEs are susceptible to a wide diversity of constantly changing climatic conditions which cause problems of control not encountered with shell and tube exchangers. Designers must achieve an economic balance between the cost of electrical power for the fans and the initial capital expenditure for the equipment. A decision must be made as to what ambient air temperature should be used for design. Air flowrate and exhaust temperature is initially unknown and can be varied in the design stage by varying the number of tube rows and thus modifying the face area [8].

There are several studies where an air cooled heat exchanger is designed or evaluated. In this sense, [4] analyzed the efficiency of different fin types and the use of tube inserts, as well as changes in the heat transfer coefficients of air and natural gas, depending on the changes in the types of fins and tube inserts, and their geometry, for a force draught air-cooled heat exchanger. For that, the application software Xchanger Suite was used.

Also, [9] proposed a modeling approach for air coolers in Concentrated Solar Power plants, and also an optimization-based algorithm to define the optimal operation of the units under fouling and their optimal cleaning scheduling.

Likewise, [10] investigated the use of global sensitivity analysis (GSA) and harmony search (HS) algorithm for design optimization of air cooled heat exchangers from the economic perspective. In this study, GSA was performed to study the influence of the design parameters and to identify the non-influential parameters, in order to reduce the size of the optimization problem. Then the HS is applied to optimize the influential parameters.

Other authors [11] established a three-dimensional simplified model to be used to simulate a temperature flow field of a dry air cooler-finned tube based on Fluent software, while the cooling effect of the equipment on high-temperature and high-pressure natural gas at compressor outlet was also studied. In this study, by comparing and analyzing the relationships among natural gas mass flow, inlet air temperature, inlet natural gas temperature, and outlet natural gas temperature for an air cooler, the formula of temperature drop of dry air cooler to natural gas is fitted, and the fitting formula is verified by field operation data.

Similarly, [6] proposed a strategy by multi-objective genetic algorithm combined with Kriging response surface for optimum design of wavy fin tube in an air cooler, while minimum pressure loss factor, maximum heat transfer factor and overall performance factor were the three objective functions considered.

Carvalho et al. [2] presented the optimization of air coolers considering the fan performance, where the selection of the fan is a design variable and the corresponding air flow rate is the result of a mechanical energy balance. This approach permitted an accurate solution, represented by the air cooler tube bundle and bays as well as fans, without the need of any rounding up procedure to identify a suitable fan to be applied after the optimization, as it is used implicitly in all previous approaches.

In [12] a design procedure for a carbon dioxide (CO_2) finned-tube gas cooler was developed and validated experimentally. With focus on the heat transfer characteristics of the air side, the developed model was corroborated with an experimental setup using water as working fluid. Investigations of the effects of fan frequency, water inlet temperature, and water mass flow on the overall heat transfer coefficient were conducted, while deviations between the model and the test results were extracted according to the fan frequency.

In [13] the mathematical modeling of a finned-tube air-cooled CO_2 gas cooler by means of distributed method was described, where the updated correlations of heat transfer coefficients and pressure drops for both refrigerant and air sides were utilized. An efficient solving method was also proposed in the simulation. The model was validated with the test results from published literature, and was then utilized to explore the potential for improved designs of gas coolers to achieve the minimum approach temperature and maximum system operating efficiency.

Manassaldi et al. [5] presented a disjunctive mathematical model for the optimal design of air cooled heat exchangers. The model involved seven discrete decisions which were related to the selection of the type of the finned tube, number of tube rows, number of tube per row, number of passes, fins per unit length, mean fin thickness and the type of the flow regime. Each discrete decision was modeled using disjunctions, Boolean variables and logical propositions, while the main continuous decisions are: fan diameter, bundle width, tube length, pressure drops and velocities in both sides of the ACHE, heat transfer area and fan power consumption.

Hu et al. [14] selected and validated a simulation model using experiment data, and then investigated the heat transfer performance and factors influencing CO_2 air coolers.

In a recent study, [1] suggested the optimal amount of water sprayed of the system to solve the low cooling performance of ACHEs and the equipment's corrosion problem simultaneously. The optimal amount of water sprayed was derived with the developed experimental results and CFD model, where the CFD model calculated the cooling effect and water evaporation ratio according to the ambient air temperature, and the amount of water sprayed. As a result, the optimal amount of water sprayed that can solve cooling performance problems and corrosion problems at the same time was presented, and water mist system was applied to commercial processes.

In [15] a numerical analysis of localized solidification within an air-cooled heat exchanger utilizing molten salt, employing a three-dimensional, pressure-based Newtonian solver coupled with a realizable k-ε turbulence model and an enthalpy-porosity approach was presented. In this study, the operational limits of the working fluid (air) that trigger solidification within the molten salt air-cooled heat exchanger tube bundle were established. A two-step method, involving steady-state and transient analyses, was employed to evaluate the effect of air inlet temperature, pressure, air velocity, and initial molten salt temperatures on the air outlet temperature, overall heat transfer coefficient, and effectiveness.

In [16] a theoretical study on the performance of air cooler to guide the optimal design of air cooler was conducted. The distributed parameter model of the air cooler was first established along with some sub-models calculating the structural parameters, the physical properties of the wet air, the heat transfer coefficient and the flow resistance on the refrigerant side and the air side. Then, the proposed models were integrated to calculate the performance of the air cooler under the effect of different structural parameters, such as fin spacing, fin thickness, tube spacing, outer tube diameter and finned tube materials.

Finally, in [17] a new approach for the globally optimal design of air coolers was presented. First, by coupling the air cooler geometric options with commercially available fans, the traditional mismatch emerging from designing the air cooler first and selecting a commercial fan later was removed. Second, the method departed from the traditional LMTD and ε-NTU methods, adopting a model composed of a differential–algebraic system of equations (DAE system) for the air cooler simulation, which was discretized to consider properties variable with temperature. The resultant optimization problem was solved using Set Trimming and Smart Enumeration, which can identify the global optimum through the simulation of only a small fraction of the search space.

The book written by [18] outlined the advantages and disadvantages of ACHEs, explained and illustrated what ACHEs are, discussed the thermal design of ACHEs as well as the optimization of the thermal design, and examined several special applications of this equipment.

In a certain chemical processing plant it's desired to cool down a liquid methanol stream from 90 ºC to 45 ºC, and for that a forced draft air cooler is proposed due to the scarcity of water supply and the availability of space. In this context, the present work deals with the thermo-hydraulic design of a finned tube forced-draft air-cooled heat exchanger, using a well-known calculation methodology, where several important design parameters are determined such as the

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ratio between the projected area of the fan and the face area of the bay, the overall heat transfer coefficient, the air flowrate, the pressure drop of both streams, and the fan driver power, among others.

2. Materials and methods.

2.1. Definition of the design problem.

It's desired to design a forced draft air cooler to cool down 28,000 kg/h of a methanol stream from 90 ºC to 45 ºC, using air at an inlet temperature of 25 ºC and pressure of 1 atm. The internal and external diameters of the tubes are 20.5 mm and 25.4 mm, respectively, while they will be provided with aluminum fins 15 mm in height, 0.380 mm in thickness, and will have 394 fins per meter. The maximum admissible tube length is 6 m. The arrangement of the tubes will be of triangular pattern with 60.32 mm of pitch. The maximum allowable pressure drop for methanol is 800 Pa, and an internal fouling resistance of 0.0003 K.m^2/W must be used for this fluid. Finally, the value of the parameter S_F is 52.23 mm (Fig. 3), while two fans will be used with 1.8 m diameter. According to the design calculation methodology [7], the ratio between the projected area of the fan and the face area of the bay (that is, the parameter ϕ) should be at a minimum 40 % to have a uniform distribution, and that will be the constraint selected in this study to design the forced draft air cooler.

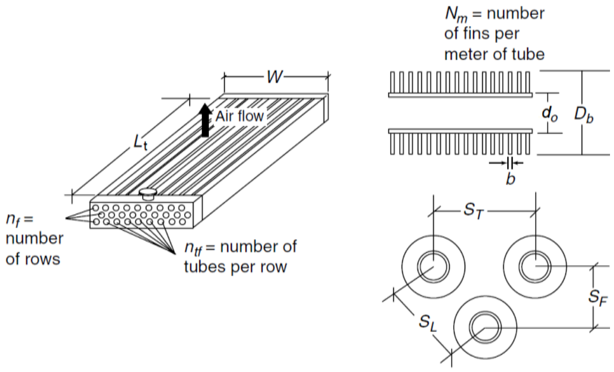

Fig 3. Air cooler nomenclature.
Source: Adapted from [7].

2.2. Design methodology.

To design the forced draft air cooler several correlations, tables and data primarily published in [7,8,19] were used, where firstly the preliminary design is carried out, then the detailed design is realized in order to determine the ratio between the projected area of the fan and the face area of the bay, which must be higher than 40%. Likewise, several important parameters are determined such as the heat transferred, the required air mass flowrate, the fin efficiency, the overall heat transfer coefficient, and the pressure drop for both fluids, among others. Finally, the fan driver power is calculated.

2.3. Preliminary design

Step 1. Definition of the initial data available:

- Methanol mass flowrate (m_h).
- Inlet temperature of methanol (T_1).
- Outlet temperature of methanol (T_2).
- Inlet temperature of air (t_1).
- Pressure of air (P_a).
- Internal diameter of tubes (d_i).
- External diameter of tubes (d_0).
- Fin thickness (b).
- Fin height (H_f).
- Tube pitch (S_L).
- Parameter S_F
- Number of fins per meter of tube (N_m).
- Tube length (L_t).
- Maximum allowable pressure drop for methanol [$\Delta p_{(m)h}$].
- Fouling resistance for methanol (R_h).
- Thermal conductivity of aluminum (k_f).
- Number of fans (n_F).
- Diameter of a fan (D_F).

Step 2. Average temperature of methanol ($\bar{T}$):

$$\bar{T} = \frac{T_1 + T_2}{2} \tag{1}$$

Step 3. Definition of the physical properties for methanol (at the average temperature determined in the previous step) and air (at the inlet temperature):

Table 1 presents the physical properties that must be defined for both the methanol and air streams. The physical properties of methanol are defined at the average temperature determined in the previous step ($\bar{T}$), while for air they are defined at the inlet temperature (25 ºC).

Table 1. Physical properties to be defined for both streams.

Property	Methanol	Air	Units
Density[1]	ρ_h	-	kg/m^3
Viscosity	μ_h	μ_c	Pa.s
Thermal conductivity	k_h	k_c	W/m.K
Heat capacity	Cp_h	Cp_c	J/Kg.K

[1]The density of air will be calculated in a posterior step.
Source: Own elaboration.

Step 4. Heat transferred (Q):
Using data of the hot stream (methanol):

$$Q = \frac{m_h}{3{,}600} \cdot Cp_h \cdot (T_1 - T_2) \tag{2}$$

Step 5. Assumption of the overall heat transfer coefficient (U^a):
A value for the overall heat transfer coefficient must be assumed, taking into account values suggested by [7] depending on the heat exchange application.

Step 6. Parameter Z:

$$Z = \frac{(T_1 - T_2)}{(T_1 - t_1)} \tag{3}$$

Step 7. Selection of the number of tube rows (n_f) and face velocity (v_f):
The tube rows and face velocity must be selected depending on the calculated value of the parameter Z, by employing the values of Table 2.

Table 2. Design recommendations as a function of Z parameter.

Parameter Z	Number of tube rows (n_f)	Face velocity (v_f) [m/s]
0.4	4	3.3
0.5	5	3
0.6	6	2.8
0.8-1	8-10	2-2.4

Source: Adapted from [8].

Step 8. Assumption of the air outlet temperature (t_2).

Step 9. Preliminary air mass flowrate (m_c):

$$m_c = \frac{Q}{Cp_c \cdot (t_2 - t_1)} \tag{4}$$

Step 10. Air inlet density (ρ_c):

$$\rho_c = \frac{M_c \cdot P_a}{R \cdot t_1} \tag{5}$$

Where M_c = 29 kg/kmol; P_a = 1 atm; R = 0.082 atm.m^3/kmol.K; and t_1 = 298 K.

Step 11. Face area to get the value of the face velocity (a_v):

$$a_v = \frac{m_c}{v_f \cdot \rho_c} \tag{6}$$

Step 12. Heat exchanger width (W^a):

$$W^a = \frac{v}{L_t} \tag{7}$$

Step 13. Log mean temperature difference ($LMTD$):

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln\frac{(T_1 - t_2)}{(T_2 - t_1)}} \tag{8}$$

Step 14. Heat transfer area (A):

$$A = \frac{Q}{U^a \cdot LMTD} \tag{9}$$

Step 15. Required number of tubes (N):

$$N = \frac{A}{\pi \cdot d_0 \cdot L_t} \tag{10}$$

Where d_0 is given in meters.

Step 16. Number of tubes per row (n_{tf}):

$$n_{tf} = \frac{N}{n_f} \tag{11}$$

Step 17. Parameter S_T.

Step 18. Calculated heat exchanger width (W^c):

$$W^c = S_T \cdot n_{tf} \tag{12}$$

Step 19. Verification of the value of W^a calculated in step 12 with the value of W^c determined in the previous step.
If the value of the calculated heat exchanger width (W^c) differs from the value of W^a, the value of W^c will be maintained, and the parameters air flowrate, air outlet temperature, and $LMTD$ will be corrected considering the result of W^c through the following equations:

Air flowrate (m_c):

$$m_c = v_f \cdot \rho_c \cdot W^c \cdot L_t \tag{13}$$

Air outlet temperature (t_2):

$$t_2 = t_1 + \frac{Q}{m_c \cdot CP_c} \tag{14}$$

Corrected log mean temperature difference ($LMTD$):

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln\frac{(T_1 - t_2)}{(T_2 - t_1)}} \tag{8}$$

Step 20. Compilation and definition of the design parameters determined in the preliminary design.

The following design parameters must be compiled and defined for the heat exchange process:

- Required number of tubes (N).
- Number of tubes per row (n_{tf}).
- Number of tube rows (n_f)
- Calculated heat exchanger width (W^c).
- Air flowrate (m_c).
- Outlet temperature of air (t_2).
- Log mean temperature difference ($LMTD$).

2.4. Detailed design

Step 21. Adoption of the number of passes for the hot fluid (n).

Step 22. Parameters R and S:

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)} \tag{15}$$

$$S = \frac{(t_2 - t_1)}{(T_1 - t_1)} \tag{16}$$

Step 23. Parameter P_X:

$$P_X = \frac{1 - \left(\frac{R \cdot S - 1}{S - 1}\right)^{1/Ns}}{R - \left(\frac{R \cdot S - 1}{S - 1}\right)^{1/Ns}} \tag{17}$$

Where N_S is the number of shell passes, and is equal to 1 for this design approach.

Step 24. $LMTD$ correction factor (F_t):

$$F_t = \frac{\sqrt{R^2+1}}{R-1} \cdot \frac{ln\left[\frac{(1-P_X)}{(1-R\cdot P_X)}\right]}{ln\left[\frac{\left(\frac{2}{P_X}\right)-1-R+\sqrt{R^2+1}}{\left(\frac{2}{P_X}\right)-1-R-\sqrt{R^2+1}}\right]} \quad (18)$$

If R = 1 the following equations must be employed:

$$P_X = \frac{S}{(N_S - N_S \cdot S + S)} \quad (18a)$$

Then:

$$F_t = \frac{\frac{P_X\cdot\sqrt{R^2+1}}{1-P_X}}{ln\left[\frac{\left(\frac{2}{P_X}\right)-1-R+\sqrt{R^2+1}}{\left(\frac{2}{P_X}\right)-1-R-\sqrt{R^2+1}}\right]} \quad (18b)$$

Step 25. Effective mean temperature difference (ΔT):

$$\Delta T = LMTD \cdot F_t \quad (19)$$

Step 26. Flow area for the tube-side fluid (a_t):

$$a_t = \frac{N\cdot\left(\frac{\pi\cdot d_i^2}{4}\right)}{n} \quad 20)$$

Step 27. Velocity of methanol in the tubes (v_t):

$$v_t = \frac{\frac{m_h}{3,600}}{\rho_h \cdot a_t} \quad (21)$$

Where m_h is given in kg/h.

Step 28. Reynolds number of methanol (Re_h):

$$Re_h = \frac{d_i \cdot v_t \cdot \rho_h}{\mu_h} \quad (22)$$

Step 29. Prandtl number of methanol (Pr_h):

$$Pr_h = \frac{Cp_h \cdot \mu_h}{k_h} \quad (23)$$

Step 30. Internal film coefficient for methanol (h_i):

- For laminar regime (Re_h < 2,100):

$$h_i = 1.86 \cdot \left(\frac{k_h}{d_i}\right) \cdot \left[Re_h \cdot Pr_h \cdot \left(\frac{d_i}{L_t}\right)\right]^{0.33} \quad (24)$$

- For transition regime (2,100 ≤ Re_h ≤ 10,000):

$$h_i = 0.116 \cdot Cp_h \cdot \rho_h \cdot v_t \cdot \left(\frac{Re_h^{0.66} - 125}{Re_h}\right) \cdot \left[1 + \left(\frac{d_i}{L_t}\right)^{0.66}\right] \cdot Pr_h^{-0.66} \quad (25)$$

- For turbulent regime (Re_h > 10,000):

$$h_i = 0.023 \cdot \left(\frac{k_h}{d_i}\right) \cdot Re_h^{0.8} \cdot Pr_h^{0.33} \quad (26)$$

Step 31. Internal film coefficient referred to the external area (h_{i0}):

$$h_{i0} = h_i \cdot \frac{d_i}{d_0} \quad (27)$$

Step 32. Plain tube area per meter of tube (A_0):

$$A_0 = \pi \cdot d_0 \quad (28)$$

Where d_0 is given in meters.

Step 33. Bare tube surface per meter of tube (A_D):

$$A_D = A_0 \cdot (1 - b \cdot N_m) \quad (29)$$

Where b is given in meters.

Step 34. Fin diameter (D_b):

$$D_b = d_0 + 2 \cdot H_f \quad (30)$$

Where H_f and d_0 are given in meters

Step 35. Fin surface (A_f):

$$A_f = \frac{\pi \cdot 2 \cdot N_m \cdot (D_b^2 - d_0^2)}{4} \quad (31)$$

Where all the diameters are given in meters.

Step 36. Projected perimeter (P_p):

$$P_p = 2 \cdot (D_b - d_0) \cdot N_m + 2 \cdot (1 - b \cdot N_m) \quad (32)$$

Where all the diameters and b are given in meters.

Step 37. Equivalent diameter (D_{eq}):

$$D_{eq} = \frac{2 \cdot (A_f + A_D)}{\pi \cdot P_p} \quad (33)$$

Step 38. Airflow area (a_S):

$$a_S = W^c \cdot L_t - n_{tf} \cdot L_t \cdot [d_0 + N_m \cdot (D_b - d_0) \cdot b] \quad (34)$$

Where all the diameters and b are given in meters.

Step 39. Reynolds number for air (Re_c):

$$Re_c = \frac{D_{eq} \cdot m_c}{a_s \cdot \mu_c} \quad (35)$$

Step 40. Parameter j:

$$j = 0.0959 \cdot (Re_c)^{0.718} \quad (36)$$

Step 41. Prandtl number for air (Pr_c):

$$Pr_c = \frac{Cp_c \cdot \mu_c}{k_c} \quad (37)$$

Step 42. Convection film coefficient for tube and fins (h_f):
Since a fouling factor on the air side is not considered:

$$h_f = j \cdot \left(\frac{k_c}{D_{eq}}\right) \cdot Pr_c^{0.33} \quad (38)$$

Step 43. Parameter m:

$$m = \sqrt{\frac{2 \cdot h_f}{k_f \cdot b}} \quad (39)$$

Where b is given in meters.

Step 44. Parameter H:

$$H = \frac{D_b - d_0}{2} \quad (40)$$

Where d_0 is given in meters.

Step 45. Parameter Y:

$$Y = \left(H + \frac{b}{2}\right) \cdot \left(1 + 0.35 \cdot ln \frac{D_b}{d_0}\right) \quad (41)$$

Where b and d_0 are given in meters.

Step 46. Fin efficiency (Ω):

$$\Omega = \frac{\tanh m \cdot Y}{m \cdot Y} \quad (42)$$

Step 47. Film coefficient corrected for fin efficiency, and considering as reference area the external surface of the plain tube (h_{fo}):

$$h_{fo} = (\Omega \cdot A_f + A_D) \cdot \frac{h_f}{A_0} \quad (43)$$

Step 48. Fouling resistance of tube-side fluid corrected for the internal/external diameters ratio (R_{hi}):

$$R_{hi} = R_h \cdot \frac{d_0}{d_i} \quad (44)$$

Step 49. Overall heat transfer coefficient (U):

$$\mathrm{U} = \frac{1}{\frac{1}{h_{fo}} + \frac{1}{h_{io}} + R_{hi}} \quad (45)$$

Step 50. Projected area of the fans (a_F):

$$a_F = n_F \cdot \frac{\pi \cdot D_F^2}{4} \quad (46)$$

Step 51. Ratio between the projected area of the fan and the face area (ϕ):

$$\phi = \frac{a_F}{a_v} \cdot 100 \quad (47)$$

2.5. Pressure drop

Step 52. Friction factor for the tube side fluid (f_t):

- Laminar regime (Re_h< 2,100):

$$f_t = \frac{16}{Re_h} \quad (48)$$

- Turbulent regime (Re_h≥ 2,100):

$$f_t = 0.0035 + \frac{0.264}{Re_h^{0.42}} \quad (49)$$

Step 53. Pressure drop 1 (Δp_1):

$$\Delta p_1 = 4 \cdot f_t \cdot n \cdot \left(\frac{L_t}{d_i}\right) \cdot \frac{\rho_h \cdot v_t^2}{2} \quad (50)$$

Step 54. Pressure drop 2 (Δp_2):

$$\Delta p_2 = 4 \cdot n \cdot \frac{\rho_h \cdot v_t^2}{2} \quad (51)$$

Step 55. Total pressure drop for the tube side fluid (Δp_t):

$$\Delta p_t = \Delta p_1 + \Delta p_2 \quad (52)$$

Step 56. Net free volume (N_{FV}):

(53)

$$N_{FV} = W^c \cdot L_t \cdot S_F - n_{tf} \cdot \pi \cdot L_t \cdot \frac{d_o^2}{4} - N_m \cdot n_{tf} \cdot \pi \cdot L_t \cdot b \cdot \left(\frac{D_b^2 - d_o^2}{4}\right)$$

Where S_F, d_o, b and D_b are given in meters.

Step 57. Equivalent diameter for friction (D'_{eq}):

$$D'_{eq} = \frac{4 \cdot N_{FV}}{L_t \cdot n_{tf} \cdot (A_f + A_D)} \quad (54)$$

Step 58. Reynolds number of air for pressure drop calculations (Re'_c):

$$Re'_c = \frac{D'_{eq} \cdot m_c}{a_s \cdot \mu_c} \quad (55)$$

Step 59. Friction factor for air (f_a):

$$f_a = 1.276 \cdot Re'^{-0.14}_c \quad (56)$$

Step 60. Average temperature of air through the air cooler ($\bar{t}$):

$$\bar{t} = \frac{t_1 + t_2}{2} \quad (57)$$

Step 61. Average air density in the tube bundle at $\bar{t}$ ($\bar{\rho_c}$):

$$\bar{\rho_c} = \frac{M_c \cdot P_a}{R \cdot (\bar{t} + 273)} \quad (58)$$

Step 62. Pressure drop of air through the bundle (Δp_a):

$$\Delta p_a = \frac{f_a \cdot \left(\frac{m_c}{a_s}\right)^2 \cdot n_f \cdot S_F}{2 \cdot \rho_c \cdot D'_{eq}} \cdot \left(\frac{D'_{eq}}{S_T}\right)^{0.4} \cdot \left(\frac{S_L}{S_T}\right)^{0.6} \quad (59)$$

Where ρ_c is the air inlet density calculated in Step 10.

2.6. Fan driver power

Step 63. Air velocity at the fan (v_F):

$$v_F = \frac{m_c}{2 \cdot \bar{\rho_c} \cdot \frac{\pi \cdot D_F^2}{4}} \quad (60)$$

Step 64. Pressure developed by the fan (P_F):

$$P_F = \Delta p_a + \frac{\bar{\rho_c} \cdot v_F^2}{2} \quad (61)$$

Step 65. Volumetric flowrate in the fans (q_F):

$$q_F = \frac{m_c}{2 \cdot \bar{\rho_c}} \quad (62)$$

Step 66 . Fan driver power (P_D):

$$P_D = \frac{q_F \cdot P_F}{\eta_F \cdot \eta_d} \quad (63)$$

Where η_F is 0.75 and η_d is 0.95, according to [7].

3. Results.

3.1. Preliminary design

Step 1. Definition of the initial data available:

- Methanol mass flowrate (m_h): 28,000 kg/h.
- Inlet temperature of methanol (T_1): 90 ºC.
- Outlet temperature of methanol (T_2): 45 ºC.
- Inlet temperature of air (t_1): 25 ºC.
- Pressure of air (P_a): 1 atm.
- Internal diameter of tubes (d_i): 20.5 mm = 0.0205 m.
- External diameter of tubes (d_0): 25.4 mm = 0.0254 m.
- Fin thickness (b): 0.380 mm = 0.00038 m.
- Fin height (H_f): 15 mm = 0.015 m.
- Tube pitch (S_L): 60.32 mm = 0.06032 m.
- Parameter S_F: 52.23 mm = 0.05223 m.
- Number of fins per meter of tube (N_m): 394.
- Tube length (L_t): 6 m
- Maximum allowable pressure drop for methanol [$\Delta p_{(m)h}$]: 800 Pa.
- Fouling resistance for methanol (R_h): 0.0003 K.m^2/W.
- Thermal conductivity of aluminum (k_f): 200 W/m.K [7].
- Number of fans (n_F): 2.
- Diameter of a fan (D_F): 1.8 m.

Step 2. Average temperature of methanol ($\bar{T}$):

$$\bar{T} = \frac{T_1 + T_2}{2} = \frac{90 + 45}{2} = 67.5\,ºC \quad (1)$$

Step 3. Definition of the physical properties for methanol and air:

Table 3 presents the values of the physical properties to be defined for both streams, according to data reported by [20]. The physical properties for methanol are established at 67.5 ºC; while for air they are defined at 25 ºC.

Table 3. Physical properties to be defined for both streams.

Property	Methanol	Air	Units
Density	746.73	-	kg/m^3
Viscosity	0.000336	0.000018	Pa.s
Thermal conductivity	0.1880	0.0260	W/m.K
Heat capacity	2,835.79	1,011	J/Kg.K

Source: Own elaboration.

Step 4. Heat transferred (Q):

Using data of the hot stream (methanol):

$$Q = \frac{m_h}{3{,}600} \cdot Cp_h \cdot (T_1 - T_2) = 992{,}526.5\,W \quad (2)$$

Step 5. Assumption of the overall heat transfer coefficient (U^a):

As suggested by [7], a value for the overall heat transfer coefficient of 260 W/m^2.K was assumed for this type of heat exchange application.

Step 6. Parameter Z:

$$Z = \frac{(T_1 - T_2)}{(T_1 - t_1)} = \frac{(90 - 45)}{(90 - 25)} = 0.692 \qquad (3)$$

Step 7. Selection of the number of tube rows (n_f) and face velocity (v_f):
According to Table 1 [8] for a value for the parameter Z of 0.692, the parameters number of tube rows and face velocity will have values of 6 and 2.8 m/s, respectively.

Step 8. Assumption of the air outlet temperature (t_2):
A value for the air outlet temperature of 50 ºC will be assumed.
Table 4 displays the results of the parameters determined in steps 9-18.

Table 4. Parameters determined in steps 9-18.

Step	Parameter	Symbol	Value	Units
9	Preliminary air flowrate	m_c	39.27	kg/s
10	Air inlet density	ρ_c	1.187	kg/m^3
11	Face area	a_v	12.02	m^2
12	Heat exchanger width	W^a	2.00	m
13	Log mean temperature difference	$LMTD$	28.85	ºC
14	Heat transfer area	A	132.32	m^2
15	Required number of tubes	N	277	-
16	Number of tubes per row	n_{tf}	46.17	-
17	Parameter S_T[1]	S_T	0.06032	m
18	Calculated heat exchanger width	W^c	2.78	m

[1]As suggested by [7], the value of the parameter S_T is equal to the tube pitch (S_L) = 0.06032 m, for the proposed geometry.
Source: Own elaboration.

Step 19. Verification of the value of W^a calculated in step 12 with the value of W^c determined in the previous step.
The value of the calculated heat exchanger width (W^c = 2.78 m) slightly differs from the value of W^a determined in step 12 (2.00 m) for a face velocity of 2.8 m/s. In this case, we will keep this geometry and correct the air flowrate, the air outlet temperature and $LMTD$. Consequently:

Air flowrate:

$$m_c = v_f \cdot \rho_c \cdot W^c \cdot L_t \qquad (13)$$
$$m_c = 2.8 \cdot 1.187 \cdot 2.78 \cdot 6 = 55.44\ kg/s$$

Air outlet temperature:

$$t_2 = t_1 + \frac{Q}{m_c \cdot CP_c} = 25 + \frac{992{,}526.5}{55.44 \cdot 1{,}011} \qquad (14)$$
$$t_2 = 42.71\ ºC$$

Corrected log mean temperature difference ($LMTD$):

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln\frac{(T_1 - t_2)}{(T_2 - t_1)}} \qquad (8)$$

$$LMTD = \frac{(90 - 42.71) - (45 - 25)}{ln\frac{(90 - 42.71)}{(45 - 25)}} = 31.71\ ºC$$

Step 20. Compilation and definition of the design parameters determined in the preliminary design.
The proposed air cooler will present the following preliminary design parameters:

- Required number of tubes (N) = 277.
- Number of tubes per row (n_{tf}) = 46.17.
- Number of tube rows (n_f) = 6.
- Calculated heat exchanger width (W^c) = 2.78 m.
- Air flowrate (m_c) = 55.44 kg/s.
- Outlet temperature of air (t_2) = 42.71 ºC.
- Log mean temperature difference ($LMTD$) = 31.71 ºC.

3.2. Detailed design:

Step 21. Assumption of the number of passes for the hot fluid (n).
We shall assume two passes for the hot fluid.

Table 5 presents the values of the parameters determined in steps 22-29.

Table 5. Parameters determined in steps 22-29.

Step	Parameter	Symbol	Value	Units
22	Parameter R	R	2.541	-
	Parameter S	S	0.272	-
23	Parameter P_X[1]	P_X	0.272	-
24	LMTD correction factor[2]	F_t	0.843	-
25	Effective mean temperature difference	ΔT	26.73	ºC
26	Flow area for the tube-side fluid	a_t	0.0457	m^2
27	Velocity of methanol in the tubes	v_t	0.228	m/s
28	Reynolds number of methanol[3]	Re_h	10,388	-
29	Prandtl number of methanol	Pr_h	5.068	-

[1] Since $R \neq 1$, equation (17) will be used to determine P_X.
[2] Since $R \neq 1$, equation (18) will be used to determine F_t.

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[3] The methanol flows under turbulent regime since $Re_h > 10{,}000$.
Source: Own elaboration.

Step 30. Internal film coefficient for methanol (h_i):
Since the methanol flows under turbulent regime, equation (26) will be used to determine the internal film coefficient. Thus:

$$h_i = 0.023 \cdot \left(\frac{k_h}{d_i}\right) \cdot Re_h^{0.8} \cdot Pr_h^{0.33} \tag{26}$$

$$h_i = 0.023 \cdot \left(\frac{0.1880}{0.0205}\right) \cdot 10{,}388^{0.8} \cdot 5.068^{0.33}$$

$$h_i = 588.59\ W/m^2.K$$

Table 6 exhibits the results of the parameters determined in steps 31-51.

Table 6. Parameters determined in steps 31-51.

Step	Parameter	Symbol	Value	Units
31	Internal film coefficient referred to the external area	h_{io}	475.04	W/m².K
32	Plain tube area per meter of tube	A_0	0.0797	m²/m
33	Bare tube surface per meter of tube	A_D	0.0677	m²/m
34	Fin diameter	D_b	0.0554	m
35	Fin surface	A_f	1.503	m²/m
36	Projected perimeter	P_p	25.34	m
37	Equivalent diameter	D_{eq}	0.0394	m
38	Airflow area	a_s	8.399	m²
39	Reynolds number for air	Re_c	14,448	-
40	Parameter j	j	93.02	-
41	Prandtl number for air	Pr_c	0.700	-
42	Convection film coefficient for tube and fins	h_f	54.56	W/m².K
43	Parameter m	m	37.89	-
44	Parameter H	H	0.0150	-
45	Parameter Y	Y	0.0193	-
46	Fin efficiency	Ω	0.853	-
47	Film coefficient corrected for fin efficiency	h_{fo}	924	W/m².K
48	Fouling resistance of tube-side fluid corrected for the internal/external diameters ratio	R_{hi}	0.000372	K.m²/W
49	Overall heat transfer coefficient	U	280.95	W/m².K
50	Projected area of the fans	a_F	5.087	m²
51	Ratio of the projected area of the fan/face area	ϕ	42.32	%

Source: Own elaboration.

3.3. Pressure drop

Table 7 exposes the results of the parameters determined in steps 52-62.

Table 7. Results of the parameters determined in steps 52-62.

Step	Parameter	Symbol	Value	Units
52	Friction factor for methanol[1]	f_t	0.0089	-
53	Pressure drop 1	Δp_1	404.46	Pa
54	Pressure drop 2	Δp_2	155.27	Pa
55	Total pressure drop for methanol	Δp_t	559.73	Pa
56	Net free volume	N_{FV}	0.6518	m³
57	Equivalent diameter for friction	D'_{eq}	0.0060	m
58	Reynolds number of air for pressure drop calculations	Re'_c	2,200	-
59	Friction factor for air	f_a	0.4327	-
60	Average temperature of air through the air cooler	$\bar{t}$	33.85	ºC
61	Average air density in the tube bundle at $\bar{t}$	$\bar{\rho_c}$	1.153	kg/m³
62	Pressure drop of air through the bundle	Δp_a	164.79	Pa

[1] Equation (49) was employed to determine the friction factor for methanol since $Re_h \geq 2{,}100$.
Source: Own elaboration.

3.4. Fan driver power

Table 8 shows the results of the parameters included in steps 63-66.

Table 8. Results of the parameters included in steps 63-66.

Step	Parameter	Symbol	Value	Units
63	Air velocity at the fan	v_F	9.45	m/s
64	Pressure developed by the fan	P_F	216.27	Pa

65	Volumetric flowrate in the fans	q_F	24.04	m^3/s
66	Fan driver power	P_D	7,297.03 ~ 7.3	W kW

Source: Own elaboration.

4. Discussion.

The preliminary designed air cooler will have 277 tubes, a number of tube rows of 6 and a number of tubes per row of 46.17 (five rows with 46 tubes and one row with 47 tubes). The width has a value of 2.78 m, which can be considered adequate to contain the two fans with a diameter of 1.8 m. It will be necessary an air flowrate of 55.44 kg/s, while the outlet temperature of air will be 42.71 ºC and the LMTD is 31.71 ºC.

In [7] an ACHE was designed to cool down 180,000 kg of heavy-gas oil from 76 ºC to 66 ºC. The preliminary design of this air cooler yielded the following results: number of tubes: 262; four tube rows; a face velocity of 3.3 m/s; a number of tubes per row of 65.5 (two rows with 65 tubes and two rows with 66 tubes); an air flowrate of 105.5 kg/s; an LMTD of 35.2 ºC; an air outlet temperature of 41.5 ºC and a width of 3.94 m to contain two fans with a diameter of 2.5 m.

According to Table 5, the effective mean temperature difference has a value of 26.73 ºC, while the tube-side fluid (methanol) will flow under turbulent regime since its Reynolds number is higher than 10,000, thus the equation (26) is used to determine the internal film coefficient, which has a value 588.59 $W/m^2.K$. The Reynolds number for air (14,448) is 1.39 times higher than the Reynolds number of methanol (10,388), which is due primarily to the higher value of the mass flowrate (55.44 kg/s) and the lower value of the viscosity (0.000018 Pa.s) of air as compared with the values of these parameters for methanol (mass flowrate of 7.78 kg/s and viscosity of 0.000336 Pa.s). Then, the internal film coefficient referred to the external area has a value of 475.04 $W/m^2.K$ (Table 6).

The value of the convection film coefficient for tube and fins is 54.56 $W/m^2.K$ (Table 6) which can be considered typically low as reported by the literature [19]. The fin efficiency has a value of 0.853, and can be classified as adequate [7], while the film coefficient corrected for fin efficiency is 924 $W/m^2.K$, which has a high value mainly due to the high value for the fin efficiency and the rather low value of the parameter plain tube area per meter of tube (0.0797 m^2/m). The calculated overall heat transfer coefficient has a value of 280.95 $W/m^2.K$, and reasonably agrees with the assumed value for this parameter in Step 5 (260 $W/m^2.K$), indicating the feasibility and viability of the results obtained for the overall heat transfer coefficient through this design methodology. Finally, the ratio of the projected area of the fan with the face area is 42.32%, which is higher than the value established by the process (40%) to have a uniform distribution, thus suggesting that the designed air cooler is feasible to use for this heat exchanger service.

In [7] the tube-side fluid flows under transition regime, since the Reynolds number for this fluid has a value of 4,511, thus the internal film coefficient referred to the external area has a value of 405 $W/m^2.K$. The Reynolds number for the air has a value of 14,784, the convection film coefficient for tube and fins is 58.1 $W/m^2.K$, the fin efficiency is 0.83 and the film coefficient corrected for fin efficiency is 1,028 $W/m^2.K$. Likewise, the calculated overall heat transfer coefficient is 264 $W/m^2.K$, which is very close to the value assumed of 250 $W/m^2.K$. Finally, the value of the ratio of the projected area of the fan with the face area is 46%, which is above than the 40% requested, thus indicating that the designer air cooler is viable to employ.

The pressure drop for methanol is 559.73 Pa, which is below the maximum allowable value established by the process for this stream (800 Pa), while the pressure drop of air through the bundle is 164.79 Pa, i.e. the pressure drop of methanol is 3.39 times higher than the pressure drop of air (Table 7).

In [7] the pressure drop for the tube side fluid (heavy-gas oil) has a value of 46,996 Pa, which can be considered high, although is below the value set by the process for this stream (50,000 Pa), while the pressure drop for air is 164.7 Pa.

As shown in Table 8, the air velocity at the fan, which depends on the adopted fan diameter (the smaller this diameter, the higher will be the velocity for the same volumetric flow), is 9.45 m/s, the pressure developed by the fan, which is the sum of the velocity head and the frictional pressure drop through the bundle, is 216.27 Pa, and the volumetric flowrate in the fans is 24.04 m^3/s. Finally, the fan driver power is around 7.3 kW.

In [7] the air velocity at the fan is 9.21 m/s, the pressure developed by the fan is 214.2 Pa, and the volumetric flowrate in the fans is 45.2 m^3/s, thus requiring a fan driver power of 13.6 kW.

5. Conclusions.

A finned tube forced-draft air-cooled heat exchanger was designed using a well-known calculation methodology, in order to cool down a liquid methanol stream. Several parameters were determined such as the heat exchanged (992,526.5 W), the air flowrate (55.44 kg/s), the required number of tubes (277), the heat exchanger width (2.78 m) and the overall heat transfer coefficient (280.95 $W/m^2.K$). The methanol stream will flow under turbulent regime inside the tubes of the designed ACHE, while the pressure drop of this stream is 559.73 Pa, which is below the maximum allowable value established by the process (800 Pa). The fin efficiency has a value of 0.853, which can be considered acceptable, and the value for the fan driver power is 7.3 kW. The designed forced draft ACHE could be

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satisfactorily used for the heat exchange application since the value of the parameter ratio between the projected area of the fan and the face area (42.32%) is higher than the value established by the design methodology (40%).

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8. Appendix.

Nomenclature.

a_F	Projected area of the fans	m^2
a_S	Airflow area	m^2
a_t	Flow area for the tube-side fluid	m^2
a_v	Face area	m^2
A	Heat transfer area	m^2
A_D	Bare tube surface per meter of tube	m^2/m
A_f	Fin surface	m^2/m
A_0	Plain tube area per meter of tube	m^2/m
b	Fin thickness	m
Cp	Heat capacity	J/Kg.K
d_i	Internal diameter of tubes	m
d_0	External diameter of tubes	m
D_b	Fin diameter	m
D_{eq}	Equivalent diameter	m
D'_{eq}	Equivalent diameter for friction	m
D_F	Diameter of a fan	m
f	Friction factor	-
F_t	LMTD correction factor	-
h_f	Convection film coefficient for tube and fins	$W/m^2.K$
h_{fo}	Film coefficient corrected for fin efficiency	$W/m^2.K$
h_i	Internal film coefficient	$W/m^2.K$
h_{i0}	Internal film coefficient referred to the external area	$W/m^2.K$
H	Parameter	-
H_f	Fin height	m
j	Parameter	-
k	Thermal conductivity	W/m.K
k_f	Thermal conductivity of fin material	W/m.K
L_t	Tube length	m
$LMTD$	Log mean temperature difference	ºC
m	Mass flowrate	kg/h
m	Parameter	-
n_f	Number of tube rows	-
n_F	Number of fans	-
n_{tf}	Number of tubes per row	-
N	Required number of tubes	-
N_{FV}	Net free volume	m^3
N_m	Number of fins per meter of tube	-
P_a	Pressure of air	atm
P_D	Fan driver power	kW
P_F	Pressure developed by the fan	Pa
P_p	Projected perimeter	m
P_X	Parameter	-
Pr	Prandtl number	-
Δp_1	Pressure drop 1 for the tube-side fluid	Pa
Δp_2	Pressure drop 2 for the tube-side fluid	Pa
Δp_a	Pressure drop of air through the bundle	Pa
Δp_t	Total pressure drop for the tube-side fluid	Pa
$\Delta p_{(m)h}$	Maximum allowable pressure drop for methanol	Pa
q_F	Volumetric flow in the fans	m^3/s
Q	Heat transferred	W
R	Fouling resistance	$K.m^2/W$
R	Ideal Gas Constant	$atm.m^3/kmol.K$
R	Thermal parameter	-
R_{hi}	Fouling resistance of tube-side fluid corrected for the internal/ external diameters ratio	$K.m^2/W$
Re	Reynolds number	-

Re'_c	Reynolds number of air for pressure drop calculations	-
S	Thermal parameter	-
S_F	Parameter	m
S_L	Tube pitch	m
S_T	Parameter	m
t	Temperature of cold fluid	ºC
$\bar{t}$	Average temperature of cold fluid through the air cooler	ºC
T	Temperature of hot fluid	ºC
$\bar{T}$	Average temperature of hot fluid	ºC
ΔT	Effective mean temperature difference	ºC
U^a	Overall heat transfer coefficient assumed	W/m^2.K
U	Overall heat transfer coefficient	W/m^2.K
v_f	Face velocity	m/s
v_F	Air velocity at the fan	m/s
v_t	Velocity of methanol in the tubes	m/s
W^a	Heat exchanger width (assumed)	m
W^c	Heat exchanger width (calculated)	m
Y	Parameter	-
Z	Parameter	-

Greek symbols

ρ	Density	kg/m^3
$\bar{\rho}$	Average density	
μ	Viscosity	Pa.s
Ω	Fin efficiency	-
ϕ	Ratio of the projected area of the fan/face area	%

Subscripts

1	Inlet
2	Outlet
a	Air
c	Cold fluid
h	Hot fluid
t	Tube-side fluid

7.- References.

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