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Impact of triangular tube pitch on air-cooler external fouling

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ABSTRACT

Transverse pitch is an important parameter of air-cooler tube bundle design, with tighter pitch meaning more compact bundles, but also increased airside pressure drop at the same airflow rate. Tighter pitch also means a higher rate of external fouling, over and above the inherent higher pressure drop.

This work presents a study of 120+ airflow measurements across 16+ unique units, including before- and after external cleaning measurements. The measurements are used to compare the tightness of the bundle with the degree of fouling. Various fin heights and diameters are included in this study. The effects of variations in these parameters are past the scope of this study (due to lack of data) and can be investigated in future research.

This study found a non-linear increase in fouling as tightness increases. While cleaning brings great improvement, the relative pressure drop remains much higher than the design values.

Using HTRI *XAce* to perform thermal ratings on four real air-coolers reveals that the performance increases negligibly as pitch increases. The most impactful benefit identified was a decrease in size. Therefore, based on the increased rate of fouling accumulated with tight pitches, the question is asked: do the advantages of tighter pitches outweigh the costs?

INTRODUCTION

External fouling of the finned surface is a ubiquitous problem impacting practically all air-cooled heat exchangers (ACHEs) [1,2]. This can (and must) be alleviated by regular cleaning. However, as will be shown, tube bundles can be designed with a lower propensity for fouling. Finned tube bundles are designed with fairly standard dimensions. Parameters relevant to fouling include fin height, fin density, fin type, tube outer diameter (OD) and transverse tube pitch (simply called pitch in this paper). Of these parameters, this paper considers pitch as the most impactful, since it commonly differs between units (where other parameters are more consistent for the same tube OD), and pitch will be shown to have a significant impact on fouling.

This paper considers over 120 airflow measurements conducted on more than 16 unique units to investigate the effect of the pitch on the degree of external fouling of the tube bundles. In order to compare the tightness of bundles with different tube diameters, the dimensionless number denoted as 2-D tightness will be presented, along with fouling factor (FF), representing the ratio of the adapted static pressure (SP) to the design static pressure. These two parameters are presented as a fair, universal measure of the impact of compactness on the degree of fouling.

Note on Airflow Measurements

The airflow measurements in this paper were all performed by Elbrons B.V. over the course of more than two years. These measurements were not performed for research purposes, but rather to either (i) assess the cleanliness of a bundle or (ii) assess the performance of a fan. The methodology is the same and applied as consistently as practically possible, regardless of the goal. Measurements were performed in general accordance with the ASME PTC 30 [3] standard and are considered to be as accurate as is possible under real-world conditions in industrial plants. However, the challenging conditions under which many of the measurements were performed does mean that some inaccuracies exist due to reasons such as

- Ambient wind
- Performance of adjacent fans
- Structures, winterization and louvers.

Regardless of these difficulties, the measurements are generally considered accurate, especially in the aggregated form presented in this study.

DEFINING PARAMETERS

2-D Tightness

The first parameter (independent variable) is the 2-D Tightness (T_{2D}), which is a design choice and used to characterize the compactness of a tube bundle. This is calculated as

$$T_{2D} = \text{blocked area} \div \text{pitch area} \quad (1)$$

$$T_{2D} = \frac{1}{2}\pi\left(\frac{OD_f}{2}\right)^2 \div \frac{\sqrt{3}}{4}P_t^2 \quad (2)$$

where T_{2D} is the 2-D tightness, OD_f is the fin OD and P_t is the transverse pitch. This is illustrated in Figure 1.

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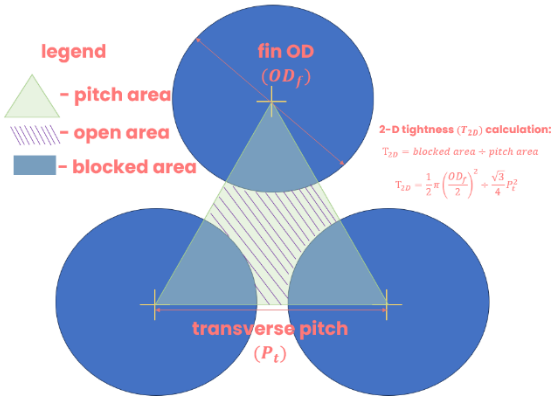

Figure 1: Pitch tightness

Table 1 shows the tube diameters, pitches and corresponding fouling factor (FF) for the units measured.

Table 1: Bundle designs seen in study

Tube OD (mm)	Tube OD (inch)	Pitch (mm)	Pitch (inch)	T_{2D}	Fin tip clearance* (mm)
25.4	1"	60.33	2 3/8	0.81	3.2
25.4	1"	63.5	2.5	0.73	6.35
25.4	1"	66.7	2 5/8	0.66	9.5
31.75	1.25"	64.9	2 5/9	0.75	1.4
31.75	1.25"	69.85	2 3/4	0.87	6.35
38.1	1.5"	76.2	3	0.76	6.35
38.1	1.5"	84.7	3 1/3	0.61	14.8

*For fin height of 15.875 mm

Fouling Factor

The fouling factor (FF) is the independent variable and calculated as the ratio of the adapted SP and the design SP using

$$FF = SP_a \div SP_d \tag{3}$$

where SP_a is adapted SP (described below) and SP_d is design SP.

Adapted Static Pressure

Airflow measured in the field is often less than the design airflow of the fan. Of course pressure drop over the bundle, measured as static pressure (SP_m), also decreases as airflow decreases. The decrease in airflow could be due to increased pressure drop over the bundle (due to fouling), or due to lack of fan performance unrelated to fouling. In order to compare the level of fouling in a bundle, it is necessary to account changes in airflow due to fan performance parameters, in order to isolate the effects of fouling of the bundle from other causes of underperformance. To this end, the fan laws are used to relate changes in airflow to theoretical changes in SP. The fan laws state that SP increases to the square of airflow [4] in the form

$$SP = \text{fn}(\text{airflow}^2). \tag{4}$$

Therefore, SP_a is obtained by increasing SP_m to the square of the ratios of the measured and design airflow using

$$SP_a = SP_m \times \left(\frac{\text{airflow}_d}{\text{airflow}_m}\right)^2. \tag{5}$$

This value therefore represents the expected SP if measured airflow were the same as design airflow. This allows reduced SP due to fan underperformance to be separated from increased SP due to fouling.

EFFECT OF PITCH ON FOULING

Using the two dimensionless numbers (T_{2D} and FF), the relation between tightness and fouling can be investigated. Plotting the variables produces Figure 2. The averages and raw data are plotted to give the reader a fuller understanding of the basis for the mean-average fouling factor.

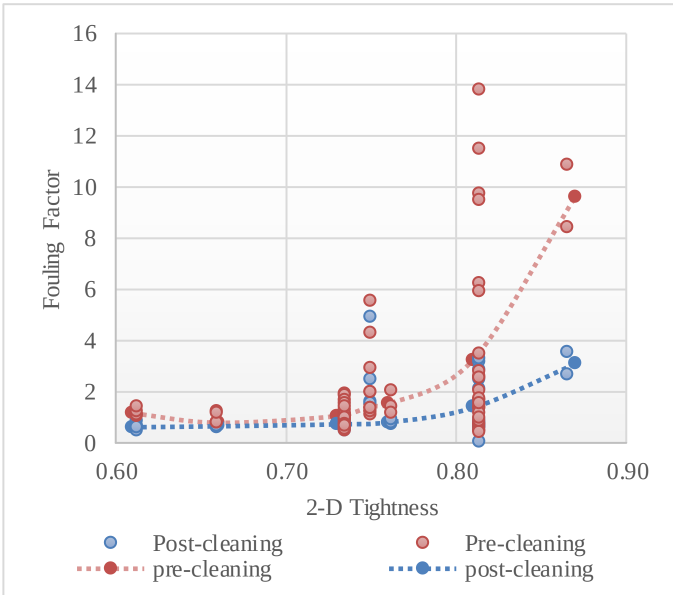

Figure 2: Impact of tightness on fouling

Here, it is clear that up to a tightness of 0.73, the fouling factor before- and after cleaning remains relatively constant around one. This means that for these more open bundles, the pressure drop is, on average, close to the design pressure drop. Cleaning also has a limited impact on airside pressure drop. Note that this does not imply that cleaning is not beneficial, only that static pressure is not decreased by cleaning. From a tightness of 0.76 (which is a 1.5” tube with a 3” pitch) and up, the FF starts to decrease more after cleaning. Here, the pre-cleaning FF is 1.5, meaning that the adapted static pressure is 50 % higher than design.

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This significant increase in back-pressure throttles fan airflow, leading to high loss of cooling duty. For tightness factors higher than 0.76, the degree of fouling both before and after cleaning explodes to levels more than 3 times design. This is obviously detrimental for fan performance and cooling duty.

MECHANISMS OF FOULING

The most reasonable explanation for this extreme acceleration of fouling is that the accumulation of particles can "bridge" the gap between tubes, which totally restricts the airflow. When this is not possible, fouling mostly accumulates between fins on the same tube, but not between the adjacent tubes. Figure 3 shows an example of external fouling build-up on a tight bundle, while Figure 4 shows the same for an open bundle.

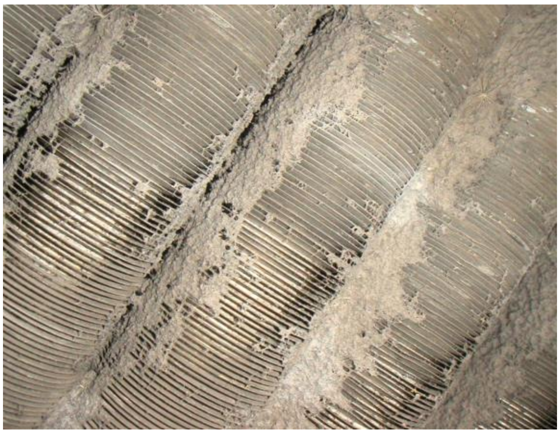

Figure 3: Fouling on tight bundle (T_{2D} = 0.81)

Figure 4: Fouling on open bundle (T_{2D} = 0.66)

Figure 5: Adjacent dirty and cleaned bundles

From the images, it is clear that fouling accumulates more easily on the tight bundle, with the debris "bridging" the distance between the first two rows whole on the wider pitch, the fouling accumulates around the OD of the second row of fins. Although the second type of fouling is also problematic, it does not have the same degree of negative effect as the first type. This is because it has a very limited resistance to airflow, which is a greater source of loss of duty [1].

Other Parameters Affecting Fouling

Another potential problem drastically affecting the external fouling of the finned tube bundle in practice is the use of the fins designed to increase turbulence on the airside, which in turn enhances heat transfer. In the practice that serrated fins have been observed to cause excessive airside pressure drop and are difficult to clean as the fouling accumulates in the "pocket" or "inserts" of the fin. This makes the bundle very difficult to clean using conventional cleaning methods. An example of such a fin is shown in Figure 6. Other modifications such as dimpled ("groovy") fins exist, but are not included in this study and therefore no comment can be made here about the effects of various types of modifications.

Figure 6: Serrated fin

Cleaning Methods

All the units covered in this study as post-cleaning were cleaned using the ELBLAST™ dry-

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cleaning method. This method uses bicarbonate of soda and low pressure air to clean the fins from below the bundle. The method is well proven to be effective and fast. Similar methods are also often used in industry.

STUDYING IMPROVED PERFORMANCE USING HTRI XACE SOFTWARE

In order to investigate the effect of pitch on air-cooler performance, four real ACHEs were considered.

For each unit, HTRI ratings were performed with pitches corresponding to every T_{2D} seen in this study. This provides ratings over a wide range of pitches (which may or may not be practically manufactured) in order to investigate how various aspects of the performance will be impacted. To ensure a broad coverage of units, the four chosen units had the design parameters shown in Table 2. The fin tip clearances for all units were 6.35 mm.

Table 2: Test case ACHE data

Unit ID	Service	Tube OD	Pitch	FF
3003	Gas Cooler	1.5 in	3 in	0.76
		38.1 mm	76.2 mm	
3004	Gas Cooler	1.25 in	2.75 in	0.75
		31.75 mm	69.86 mm	
3005	Gas Cooler	1 in	2.5 in	0.73
		25.4 mm	63.5 mm	
3010	Liquid Cooler	1 in	2.5 in	0.73
		25.4 mm	63.5 mm	

The HTRI simulation mode was used to calculate the outlet temperature (and corresponding heat duty) of these units. The following parameters were chosen to be studied as measures of unit design:

- Heat duty to evaluate overall heat exchange performance
- Static pressure to study the effect on design pressure drop, which will affect fan selection
- Bundle width to investigate the effect on unit size, impacting plot space and construction cost.

Of course, these three parameters measure different things and therefore have different units. In order to compare the relative effect of tightness on each of these outcomes, the trends are normalized around the average. These normalized values can be plotted on the same axis system (shown in Figure 7), which allows a direct comparison of the relative impact tightness has on each parameter. Note that both axes are dimensionless. The results are very similar for each of the four exchangers evaluated and therefore only the results of unit 3003 will be shown to avoid repetition.

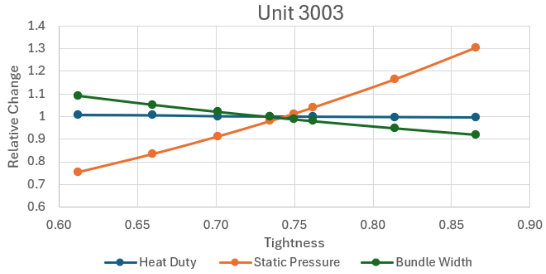

Figure 7: Impact of tightness on ACHE performance

The basic design data for unit 3003 is shown in Table 3.

Table 3: Basic design data of Unit 3003

Process Fluid	Flow rate (kg/h)	Temp In/Out (˚ C)	Air Flowrate (m^3/s)	LMTD (˚ C)
Mixed refrigerant	1126662.7	64.8	1129.7	9.47
		42.8		

Figure 7 clearly shows that T_{2D} has a negligible impact on heat duty, meaning that designing ACHEs with tight bundles has no influence on cooling performance. Bundle width does vary significantly with different pitches. For reference, the smallest pitch has a width of 10 % lower than the average pitch and the largest pitch has a width of 10 % higher than the average. The largest impact of tighter pitch is seen in static pressure, where the largest and smallest pitches cause -30 % and +30 % variation in static pressure.

In summary, pitch has a notable impact on bundle width, which in turn will impact plot space requirements as well as capital expenditure (CAPEX). However, it has a much stronger inverse impact on design static pressure (regardless of increased rate of fouling described earlier) which increases operational expense (OPEX).

Impact of Increased Static Pressure on Fan Selection

Increasing pressure loss through the bundle increases the required fan static pressure capacity, since the fan will need to overcome this higher pressure drop to maintain the same airflow. For a reference point, design static pressures the fans in this study ranged from 115 Pa to 221 Pa at the respective design airflow rates. This has a significant impact on the fan selection, especially since American Petroleum Institute (API) 661 [5] mandates that a fan selection be performed with an extra margin of 21 % on the design static pressure. Also, many fan manufacturers have selection software which is optimistic about the performance of the fan, meaning that the SP capacity indicated on the fan curve will possibly not be achieved on site. Increasing fan SP capacity can be achieved by either (i) increasing the solidity ratio or (ii) increasing the tip speed [6,7].

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Increasing Fan Solidity Ratio

The solidity ratio of a fan is the ratio of the total fan ring area to the bladed area. Therefore, increasing the solidity ratio can be done by either increasing the number of blades, or the width of the blades. Both of these options increase the weight and cost of a fan. Increasing the solidity ratio can also, in some cases, decrease the static efficiency of the fan. Therefore, this is a viable but unfavorable option to increase the SP capacity of a fan.

Increasing Tip Speed

An extremely effective and surefire way to increase SP capacity is to increase the fan revolutions per minute (RPM) and therefore tip speed. This has many advantages, whereby it increases SP capacity along with fan efficiency. However, the maximum API 661tip speed of 61 m/s (12000 ft/min) must be adhered to. Also, increasing tip speed increases fan noise. Currently, plants have to adhere to strict environmental legislation regarding noise pollution and this might therefore be a big limitation to tip speed.

This means that a unit with a high SP requirement on a plant with noise restrictions might limit the choices of applicable fans, since only low tip speeds will be achievable to remain within noise limits.

Higher Energy Use

Increasing the SP requirement for a fan might cause higher energy consumption than would otherwise be incurred. This is because fan transfers energy to the air in the form of pressure and airflow. However, only airflow has an impact on the duty of the air-cooler. Therefore, increasing static pressure (which is necessary for tighter bundles) increases the amount of electrical power required, but does not increase the cooling duty. Of course, the increased electrical power required will also necessitate larger electrical motors.

Describing this more mathematically, the energy consumption of the electrical motor ($\mathrm{W_{mtr}}$) is calculated using

$$\mathrm{W_{mtr}} = \mathrm{Airflow} \times \mathrm{SP_m} \times \eta_{mtr} \times \eta_{drive} \times \eta_{fan} \quad (6)$$

with η_{mtr}, η_{drive} and η_{fan} denoting motor, drive and fan efficiency respectively. Clearly, this means that increasing SP at the same airflow will increase motor power consumption. This will incur higher OPEX and increase the carbon footprint of the fan, without the benefit of extra cooling, because the extra energy goes into producing pressure and not airflow. Larger electrical motors will increase CAPEX as well. It should, however, be noted that fans often have low static efficiency under low SP conditions and the fan power consumption should be considered case by case.

CONCLUSION

This work has shown that decreasing tube pitch increases the design static pressure of the tube bundle. Over and above this, it increases the rate at which fouling accumulates and the difficulty by which it is cleaned. These effects cause higher OPEX due to more frequent cleaning and higher electricity consumption and (often severe) loss of heat duty due to reduced conductivity and reduced airflow. This comes with the moderate benefit of smaller bundles.

There also seems to be a point of diminishing returns regarding rate of fouling, where using a larger pitch does not further reduce the rate of fouling. The hope is that this work will contribute to a better understanding of the impact of pitch on fouling and enable ACHE users and manufacturers to select optimal pitch sizes for finned tube bundles.

Recommendations

Based on this study, the following is recommended:

- Avoid designing or specifying very tight bundles ($\mathrm{T}_{2D} > 0.76$), such as 1” tubes with a 2 3/8 “ pitch,
- In terms of tip clearance, this can be seen as avoiding clearances of 3.2 mm or smaller,
- For existing units with these bundles, ensure that the fan selection allows for extra SP capacity,
- Existing units with tight pitches will require more frequent and more extensive external cleaning.

The only possible application for using tight pitch bundles would be very clean environments with high plot space restrictions, such as an offshore oil platform.

NOMENCLATURE

FF Fouling Factor , dimensionless
SP Static Pressure , Pascal (Pa)
η efficiency , dimensionless
P Pitch , millimeter (mm) / inch (”)
T_{2D} Tightness , dimensionless
W Power , Watt

Subscript

Subscripts and superscripts should be identified under a separate second-level heading.

a adapted
d design
f fin
m measured
mtr motor
t transverse

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