<u>Guidelines for Heat Exchanger Bid Evaluation</u>

by

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Biography

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INTRODUCTION

In executing the responsibilities as a Process Engineer, Production Engineer, or Maintenance Engineer, it is common for a Chemical Engineer to have to perform a heat exchanger bid analysis. This would include generating the required performance data, selecting the candidate suppliers or heat exchanger types, and evaluating the subsequent quotations. To the novice engineer who has not had the opportunity to perform this seemingly innocent task before, it seems like a simple, straight-forward procedure.....until the quotations are received. After reviewing the initial quotations the question that is inevitably asked is , “How can five manufactures, all using what seems as the same process data, arrive at 5 different sizes ranging from 10ft2 to 65ft2 ?

The answer to this question can usually be found by looking at the process information supplied (or lack there of), the service and/or process fluids physical properties, and the quoted operating conditions of the unit. This paper will concentrate on the last two areas to illustrate how a change in either group can effect the theoretical operation of a heat exchanger, and hence the proper size for the application.

This paper will use simple a heat transfer process to show the effects of varying the process fluids thermal conductivity, specific heat, and viscosity has on the unit selection. We also look at the effects of altering the process fluid allowable pressure drop, the fouling factor and the service fluid inlet/outlet temperatures.

EFFECTS OF PROCESS FLUID PHYSICAL PROPERTIES

In appendix A is a shell and tube heat exchanger specification sheet for a unit designed for cooling a 20% sulfuric acid solution using 60 degree F cooling tower water. In this selection the manufacturer was not given the physical properties by the client, as is typical with well known process fluids such as sulfuric acid. The wealth of information made available by computer fluid databases, currently used by many heat exchanger manufactures, is a tool that could be utilized by the project engineer. There are situations when the process fluid properties are different than the “standard” published values. In these cases the fluid properties must be indicated on the heat exchanger specification sheet. In all cases, by not specifying the physical properties completely in the original inquiry, the project engineer may find varying results that will have to be scrutinized to verify the correctness of the selection . This usually involves unnecessary time and financial resources.

As an example of what can happen with differences in the process fluid physical properties, let’s look at additive effects on our base selection with changes in the thermal conductivity(k), specific heat(Cp), and viscosity(v).

In the base case the average value for k is .326 Btu/(ft*h*F). Along with the other conditions listed on the specification sheet, we use these values to calculate a overall heat transfer coefficient, U, of 154 Btu/(ft2*h*F). By changing k to an average value of .395 Btu/(ft*h*F), we calculate a U value of 163. This would represent a decrease in the required heat transfer area(HTA) of almost 6%. The comparison is summarized as follows:

	k Value	**U value**
Base Case	.326	154
Case 1	..395	163
	Increase in Efficiency	5.8%

Now we will take Case 1 and modify the specific heat(Cp) of the process fluid. The influence of Cp is most evident in the calculated heat duty. And a change in the heat duty will have a direct effect on the temperature change in the service fluid, and hence the LMTD. Any change in the LMTD is directly proportional to the required heat transfer area(HTA). In the base case, the original value used for Cp was .84 Btu/(lb*F). Using this value the calculated heat load, based on the process conditions is 1,714,674 Btu/hr, the calculated LMTD was 21.4 F, and the required HTA was 520 ft2. By changing Cp for the process fluid to .75 Btu/(lb*F) the new heat load is calculated to be 1,427,731 Btu/hr and the new LMTD is 22.42 F. The new calculated HTA is 393 ft2. This represents a reduction in area from the base case of 25%. The comparison is summarized as follows:

	K	Cp	LMTD	Duty	HTA
Base Case	.326	.84	21.4	1,714,674	520
Case 2	.40	.75	22.4	1,427,731	392
				Decrease in Required HTA	**25%**

Now let's take the base case and also change the process fluid viscosity(v). It is important to note that there should be two values given for viscosity. The reason for this is that most fluid viscosity's change with temperature, and the change in viscosity in the boundary layer of the fluid changes the velocity distribution profile, and hence the calculated U value. The values used for v in the base case were 1.076 and 1.354. The resulting U value was 154 Btu/(ft2*h*F). By changing the values for v to .80 and 1.00 when calculate a new U value for Case 3 of 163. This represents a further reduction in the required HTA when compared to the base case. The new results are summarized as follows:

	K	Cp	v	LMTD	Duty	U	HTA
Base Case	.326	.84	1.07/1.35	21.4	1,714,674	154	520
Case 3	.40	.75	.80/1.00	22.4	1,427,731	165	375
					Decrease in Required HTA		**28%**

What we have done is reduced the original(and probably the most correct) required HTA by 28% by simply using different values for the process fluid properties. The conclusion of this exercise is that all physical properties of the process fluid should be specified in order to avoid discrepancies in the required HTA. This will help eliminate the additional time and effort needed to sort through the data.

EFFECTS OF PROCESS AND SERVICE LIMITATIONS

Changes in the process and service limitations can have a much greater effect on the heat exchanger selection than changes in the process physical properties. A slightly higher allowable pressure drop or service flow rate can allow for a manufacture to use a smaller diameter unit or smaller HTA to meet the process conditions. This is not necessarily a bad thing. If a manufacturer can avoid a temperature cross by increasing the service flow rates, or use a smaller diameter unit by increasing the allowable pressure drop by 10%, then he should offer these alternatives. The problem arises when the process and service limitations have not been defined properly, or not at all. The project engineer is unintentionally limiting the heat exchanger selection and/or does not understand the impact of changes on the other equipment used in the process. You know your plants better than any supplier. Therefor, it is your responsibility to completely specify the Process and service limitations. **We will not try and discuss all of the limitations that should be specified, but will concentrate on process fluid allowable pressure drop and the service fluid flow rate. We will begin with process pressure drop.**

According to Bernoulli's equation, there is a relationship between fluid flow pressure drop and velocity. As the velocity is increased the pressure drop also increases by approximately a factor of the square. As we increase the velocity the flow becomes more turbulent, as can be seen by an increase in the Reynolds Number (a ratio of inertial forces to viscous forces), increasing the process side film coefficient(h) and the overall U value, and lowering the required HTA. Once the fluid enters than turbulent region, it will remain turbulent through out the length of the heat exchanger.

In the base case the project engineer specified a maximum allowable pressure drop of 5 psi. Based on the standard design available from the manufacturer, the calculated pressure drop was 1.83 psi. The calculated film coefficient was 643 Btu/(ft2*h*F) and the U value was 154 Btu/(ft2*h*F). By increasing the allowable pressure drop to 7.5 psi, we can generate a Case 4 which has a pressure drop of 5.2 psi, h value of 891 Btu/(ft2*h*F), and a U value of 170 Btu/(ft2*h*F). We have reduced the required HTA 10%.

In Case 5 we will specify an allowable pressure drop of 10 psi. The new calculated pressure drop is 10.01 psi, the h value is 1,124 Btu/(ft2*h*F), and the U value is 180 Btu/(ft2*h*F). We have reduced the required heat transfer area by 15% when compared to the base case. These results are summarized as follows:

	Allow Pr Drop	**Calc Pr Drop**	**h**	**U**	**HTA**
Base Case	5	1.83	643	154	520
Case 4	7.5	5.20	891	170	471
Case 5	10.0	10.01	1124	180	445

The specified service fluid flow rate, inlet temperature, and outlet temperature should be dependent on the facilities available at each plant site. It is normally advantages to specify the maximum allowable service fluid flow rate and/or temperature change in your inquiry. Specifying the maximum values will allow for the manufacturer of the heat exchanger to optimize the unit selection. As an example we will make some changes in our base case and compare the effects.

In our base case the maximum allowable service flow rate was specified at 114,129 lb/hr. This resulted is a temperature increase of 15 F based on the calculated heat duty. Now let's increase the water flow rate by 50% to a value of 171,174 lb/hr for Case 6. This change has two immediate effects. First, it increases the LMTD by 15%. Secondly, it increases the velocity in the same diameter unit, resulting in an increase of 16% in the overall U value. Keep in mind that in the example that we have been using, the shell diameter has been dictated by the process fluid conditions, not the service fluid conditions. **These two increases result in a total reduction in the required heat transfer area of approximately 33%.**

Now let's increase the water flowrate by 300% to a value of 342,305 lb/hr for Case 7. It increases the LMTD by 28% and the overall U value by 26%. **These two increases result in a total reduction in the required heat transfer area of approximately 61%.** We summarize these results as follows;

	Flow Rate	**Temp Ch.**	**U**	**LMTD**	**HTA**
Base Case	114,129	15	154	21.4	520
Case 6	171,174	10	180	24.7	392
Case 7	342,305	5	194	27.5	320

The important point to remember in review of the above examples is that the required HTA is greatly effected by the process and service limitations that you specify in your inquiry. A responsible manufacturer will design the unit to operate within these parameters. It is the responsibility of the project engineer to supply the manufacturer with complete and correct information.

EFFECT OF FOULING FACTOR

The purpose of a fouling factor in the selection of a heat exchanger is to take into account scaling that will occur during operation of the unit. The scale builds up on the on the service and process sides of the unit and acts as an insulator. The scale adds other layer to the overall resistance of the unit, reducing its effectiveness. In most applications the question is not "Will fouling occur ?" but "How quickly will fouling occur ?".

If the process is low scaling then it would be appropriate to use a low fouling factor. If the process will scale quickly then it would be wise to use a high fouling factor. The fouling factor is going to determine how often the system will have to be shut down for cleaning and how will this effect the economics, production rates, and/or quality of the process.

The manufacturer can give you a recommendation on the fouling factors based on known industry standards, but they should not be considered the final authority. **Only the project engineer is in the position to correctly determine the proper fouling factor to use for the application and must always include the value on the specification sheet. In order to reduce the cost of a unit a manufacturer will sometimes reduce the fouling factor in order to decrease the size. This is very common and it should not be accepted by a project engineer without a good understanding on its impact on the unit selection.**

It is not uncommon for the fouling factor to account for 40-60% of the required HTA in any given heat exchanger selection. The percentage of excess area resulting from the use of the fouling factor can be calculated using the "Clean" and "Dirty" U values listed on a specification sheet. The Percent Fouling can be calculated by using formula

$$\text{Percent Fouling} = (1-(U_{dirty}/U_{clean}))*100$$

If both U values are not available the Percent Fouling can be closely estimated by using the formula

$$\text{Percent Fouling} = 100*\text{Total Fouling Factor}*U$$

It is normal for a Project Engineer to specify a separate fouling factors for the service and process sides of the unit. The manufacturer must combine these two values into one, correcting for either the tube inside or outside diameter. **Now let's look at the effect of changing the fouling factors in our base selection.**

In our base selection we used s fouling factor of .001 of both the service and process sides of the unit. Correcting this for the tube outside diameter we calculate a total fouling factor of .0024. Taking into the account this fouling factor in the clean U value, we reduce the dirty U value to 154 Btu/(ft2*h*F). This represents 37% excess surface area. For Case 8 we will reduce the process side fouling to .0005, reducing the overall fouling factor to .0017. This results in a dirty U value of 173 BTU/(ft2*h*f). This small change increased the U value by approximately 40%. The result can be summarized as follows;

	Overall Fouling	U	HTA	% Excess Area
Base Case	.0024	154	520	37%
Case 8	.0017	173	462	29%

<u>SUMMARY</u>

The objective of this exercise was not to cover all of the variables involved in the proper selection of heat exchangers. There are many excellent books and industry information available on this subject. It would not be possible to cover even the most minimum amount of required information in this format. The purpose of this paper is to alert the Project engineer who is responsible for acquiring a heat exchanger, that there are many ways that the proper unit selection can be effected. By listing in detail the fluid physical properties, process and service limitations, and the fouling factors, a large number of these variables can be eliminated.