FLOWNEX®
SIMULATION ENVIRONMENT

CASE STUDY

A BASIC IMMERSION FIRETUBE FLOWNEX MODEL

This case study demonstrates the implementation of a basic immersion fire-tube model in Flownex and presents natural draft and forced draft examples.

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Challenge:

The main challenge is to model an immersion fire-tube in Flownex. Immersion fire-tubes are widely used in industry, most commonly in indirect heating applications where either gas or oil burners are used as a heat source.

Benefits:

Flownex allows the user to model combustion, heat transfer and fluid flow processes in an elegant and easy to understand way.

Solution:

Using Flownex's compound component and scripting capabilities, a simple immersion fire-tube model has been developed and is presented in this case study. Furthermore, examples of natural draft and forced draft design cases are presented.

"Flownex has the unique ability to simultaneously model combustion, heat transfer and fluid mechanics problems. This capability makes Flownex the ideal tool to design and analyze immersion fire-tube heat transfer processes."

Hannes van der Walt
Principal Thermal Engineer
Gasco Pty Ltd

Introduction

Indirect heating processes have been widely used in the oil and gas and several other industries for many decades. Advantages of indirect heating include:

- The relatively low cost of the equipment.
- Separation of the high pressure process fluid from the heating medium via simple pressure piping.
- Relatively high efficiencies.
- Low maintenance and running cost.
- Reduced heat loss.
- Long operational life.

"It would be much more challenging to design and analyze an entire immersion fire-tube-based heater system at the level of detail presented without Flownex as an engineering design and analysis tool."

The heating medium may be water, water-glycol, salt, steam and air for indirect heaters, or any fluid that needs to be heated directly. Water is possibly the most common medium due to its low cost.

The Immersion Fire-tube

Heating processes may be best explained in terms of a heat balance diagram. Energy is supplied by combusting a fuel and is referred to as the *gross heat input* when specified in terms of higher heating value (HHV) as is typical in natural gas burner applications. The largest loss of heat is likely to be the hot flue gas that leaves the process via an exhaust stack.

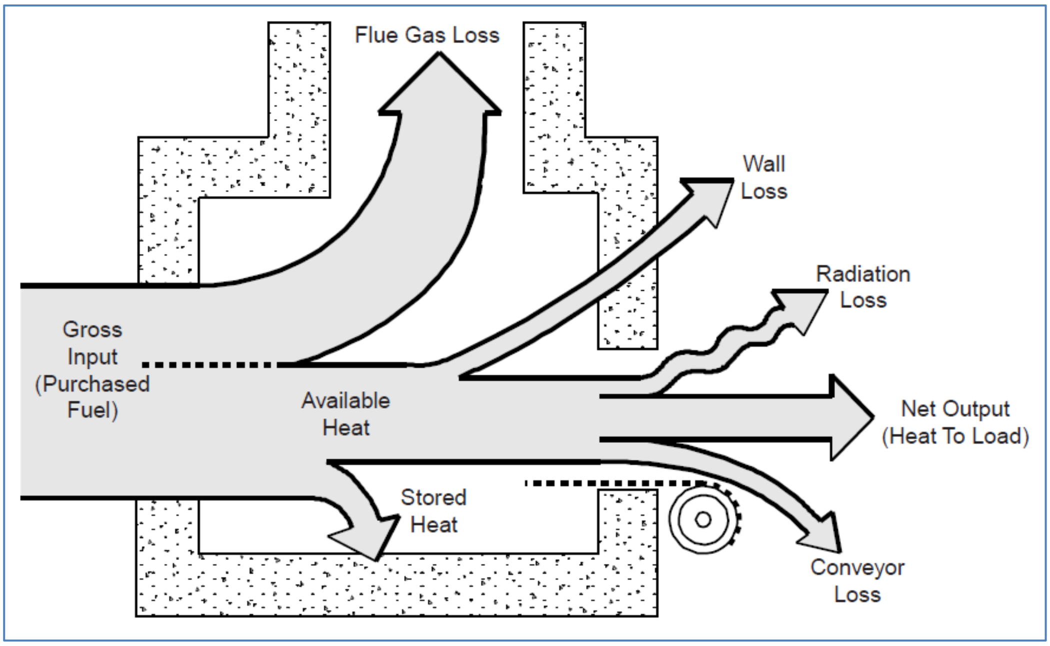

Figure 1: Sankey heat balance diagram (from Eclipse Engineering Guide).

Smaller amounts of heat may be lost through insulation, radiation and other effects. The difference between the *gross heat input* and the sum of all the losses results is the *net output* (also known as the *heat to load*, *heat to process* or *heat duty*).

Immersion fire-tubes, as the name suggests, are tubes or pipes fully immersed in a fluid with a burner firing into one end. The combustion gases flow through the fire-tube and leave at the other end, normally into an exhaust stack. The immersion tube aims to transfer as much heat as possible to the fluid within the boundaries of inevitable practical constraints. A typical example is shown in the following figure.

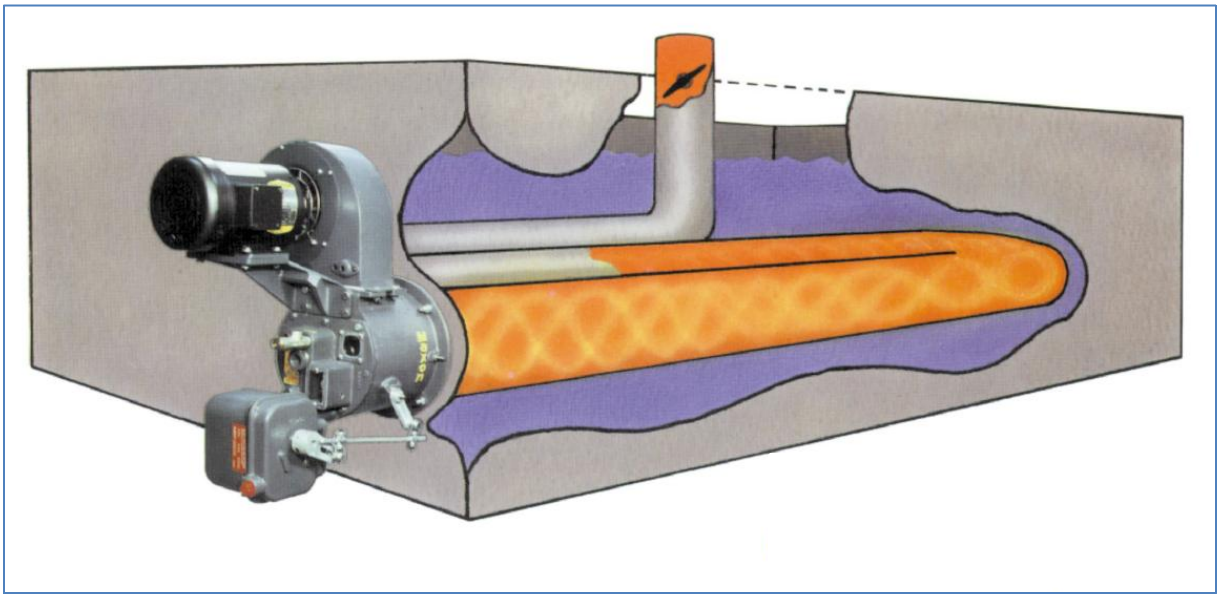
Figure 2: Typical burner and immersion fire-tube application heating a liquid (from Maxon Series "67" Tube-O-Flame Bulletin 2200).

In the example above, it is shown that the immersion fire-tube is a 3-pass unit. However, different designs are common in industry, each with its own advantages and disadvantages. Some common immersion fire-tube layouts are shown in the following figure.

The U-tube design is obviously the simplest and is probably therefore also the most common. The burner-end straight leg is called the *radiant section* since it is subjected to the direct luminous radiation of the burner flame. The return leg or legs are not subjected to direct luminous radiation and mainly receives heat via convection and some gas radiation. This part of the immersion fire-tube is known as the *convection section*. The W-tube design is possibly the second-most commonplace. Essentially it has a similar *radiant section* to the ordinary U-tube design, but it has 3 times the *convection section* length and area. The Trident tube design does not appear to be popular in industry. Therefore, the Flownex immersion fire-tube model presented is limited to U-tube and W-tube designs.

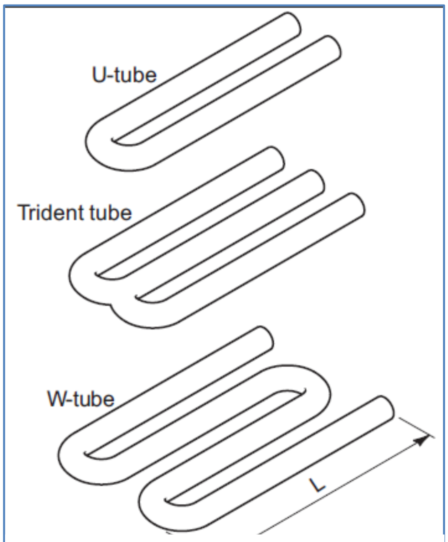

Figure 3: Typical immersion fire-tube layouts.

Immersion Fire-tube Design

The thermal design considerations of an immersion fire-tube include the following:

- Burner design (natural draft or forced draft operation).
- Thermal efficiency.
- Heated medium (most commonly water) temperature.
- Size (length and diameter).
- Burner heat release density (burner duty divided by fire-tube cross sectional area).
- Firetube outer wetted surface heat flux (heat transferred to heated fluid divided by fire-tube total outer wetted surface area).
- Turn-down operation.
- Firetube flue gas pressure loss (especially important for natural draft designs).
- Flue gas oxygen content (combustion design – dictated by legislation).
- Flue gas temperature (directly related to thermal efficiency).
- Flue gas water vapor and SO_x acid gas dew point temperatures (important for corrosion considerations).
- Materials of construction (cost, corrosion resistance).
- Corrosion allowance (cost, mass).

Mechanical Design

Fire-tube designs typically consist of Schedule 40 or lighter pipe (6 mm wall thickness is the recommended minimum to allow for some corrosion). The first straight pass (radiant section) length should be at least 10 pipe diameters, however in practice it is often significantly longer as it depends on the flame length of the selected burner. As a rule of thumb, the flame length should not be longer than 80% of the radiant section, however this is rarely, if ever, a problem.

Natural draft applications are typically based on U-tube designs due to the limited amount of natural draft available. Forced draft applications are normally based on W-tube designs to take advantage of the increased efficiency resulting from the longer fire-tube length. Since forced draft designs rely on a fan to supply the combustion air flow, fire-tube pressure loss is not normally an issue.

Empirical Fire-tube Thermal Efficiency Correlation

Several factors influence fire-tube thermal efficiency, however, it was found that the most significant factors are fire-tube length and burner duty. Other factors have a much smaller influence. A simple empirical efficiency correlation based on higher heating value (HHV) was developed by the AGA Testing Laboratories and published in 1944 in Research Bulletin No. 24 "Research Fundamentals of Immersion Tube Heating with Gas":

$$\eta_{HHV} = 20\ log_{10}\left(\frac{L_e^2}{\dot{Q}_b}\right) + 71 \qquad \text{Equation 1}$$

where η_{HHV} is the thermal efficiency of the immersion fire-tube based on HHV [%].

L_e is the effective immersion fire-tube length [ft]. The effective length is the physical centerline length plus 1.1 ft additional for each return bend. Field tests revealed that return bends slightly increase the fire-tube thermal efficiency due to the effects of improved heat transfer.

$\dot{Q}_b$ is the burner heat release based on HHV [10^3 BTU/hr].

At first glance the equation seems to suggest a minimum assumed efficiency of 71%, but as will be shown below, the constant of 71 is simply a unit-dependent constant which could also be written inside the brackets of Equation 1. The above equation accepts the burner duty $\dot{Q}_b$ in 1000 BTU/hr units, however if the burner duty was supplied in BTU/hr instead, Equation 1 would be written as:

$$\eta_{HHV} = 20\ log_{10}\left(\frac{L_e^2}{\dot{Q}_b}\right) + 131 \qquad \text{Equation 2}$$

where $\dot{Q}_b$ is the burner heat release based on HHV [BTU/hr].

Therefore, for engineers in the modern era, this equation could also be written in SI units as follows:

$$\eta_{HHV} = 20\ log_{10}\left(\frac{L_e^2}{\dot{Q}_b}\right) + C_u \qquad \text{Equation 3}$$

where η_{HHV} is the thermal efficiency of the immersion fire-tube based on HHV [%]

L_e is the immersion fire-tube effective length [m]. The effective length is the physical centerline length plus 0.335 m additional for each return bend.

$\dot{Q}_b$ is the burner heat release based on HHV [W or kW]

C_u is a unit-dependent constant, where

$C_u = 81$ if the burner heat release based on HHV is specified in kW

$C_u = 141$ if the burner heat release based on HHV is specified in W

Figure 4 below was published in the *Eclipse Combustion Engineering Guide (1986) Tech Notes Section 3 Sheet L-1 Immersion Tube Sizing* and was generated from Equation 1.

The correlation given by Equation 1 (or Equation 3) is quite simple, easy to implement in a model and produces reasonable results, however, it appears to have one limitation. For very long immersion fire-tubes, Equation 1 will predict efficiencies larger than 100%. This is the case when the ratio is:

$$20\, log_{10}\left(\frac{L_e^2}{\dot{Q}_b}\right) \geq 29 \qquad \text{Equation 4}$$

This equation may be simplified to show that efficiencies of 100% and larger will be calculated when:

$$L_e \geq 5.309\sqrt{\dot{Q}_b} \qquad \text{Equation 5}$$

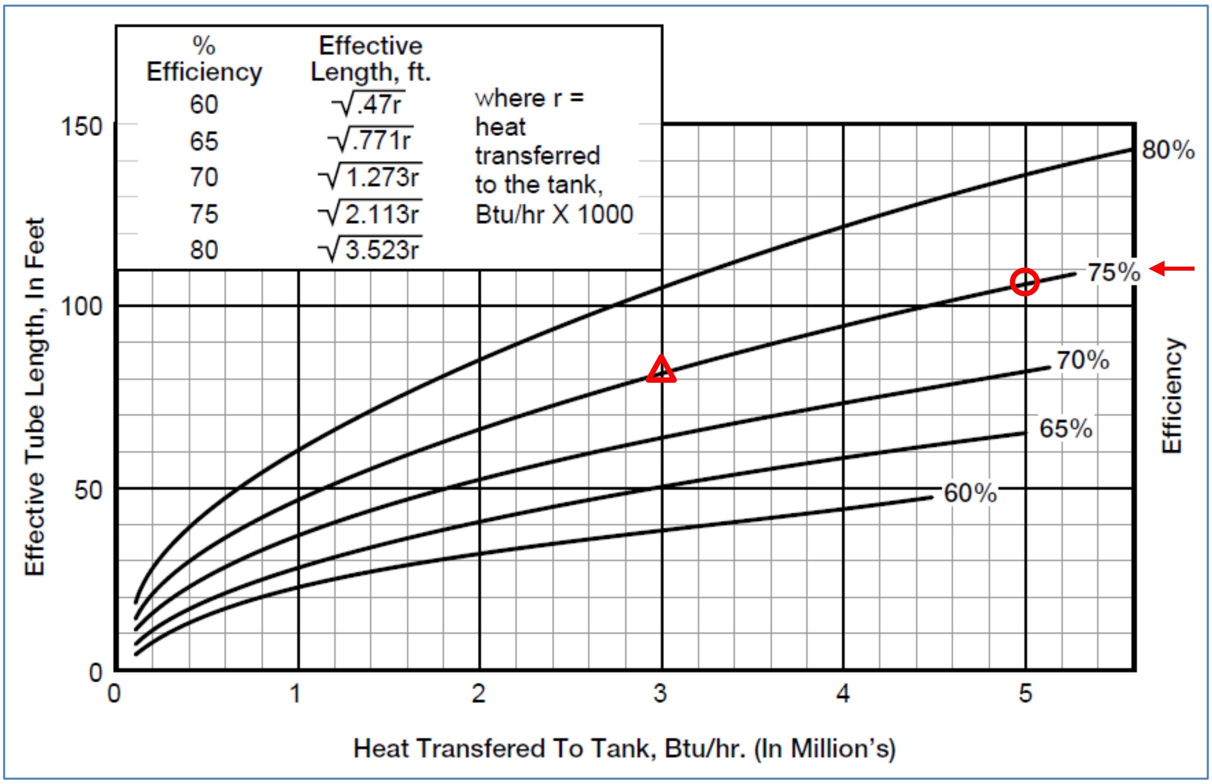

Figure 4: Fire-tube thermal efficiency as a function of effective fire-tube length and heat transfer rate.

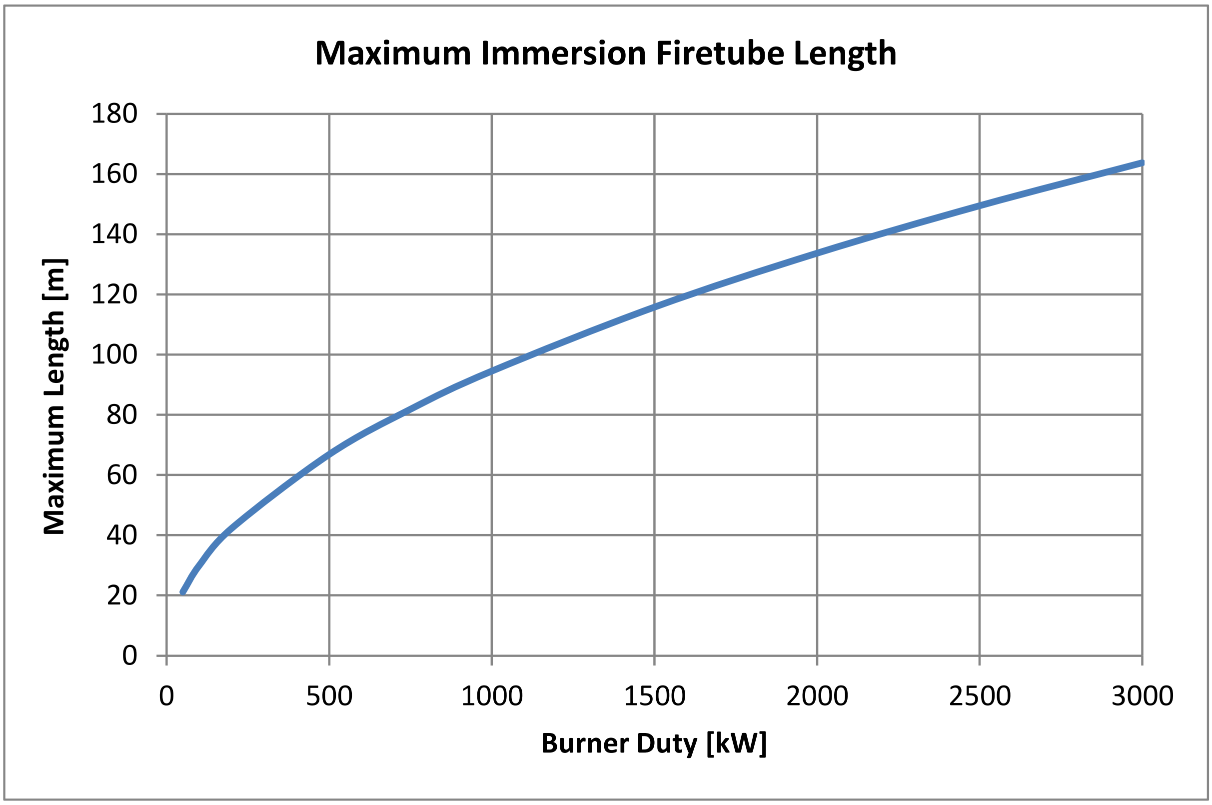

Figure 5: The maximum "100% Efficient" immersion fire-tube length as a function of burner duty.

As shown above, very long immersion fire-tubes are required to calculate efficiencies of 100% or above, and therefore the upper limit is not normally an issue. Furthermore, efficiencies are normally kept below 85% in an effort to avoid condensation of water and sulfur in the fire-tube, especially in turn-down operation. Condensation may cause corrosion problems and will be discussed in more detail below. API 12K recommends minimum stack temperatures of approximately 120°C for sulfur-free fuels and 150°C to 200°C for fuels containing sulfur. The exhaust stack flue gas temperature should remain comfortably above the SO_x acid dew point temperature at all times. A margin of at least 20°C during the worst operating case is often recommended.

Burner Heat Release Density

Natural draft immersion fire-tubes have relatively large diameters to ensure that the amount of natural draft created by the exhaust stack is sufficient to drive the flow of atmospheric air into, and flue gases through the fire-tube and stack. The recommended fire-tube diameter is typically expressed in terms of a heat flux - known as the *burner heat release density* :

$$\dot{q}_b = \dot{Q}_b / A_c \qquad \text{Equation 6}$$

where $\dot{q}_b$ is the *burner heat release density* [W/m^2 or kW/m^2]

A_c is the fire-tube internal cross sectional area at the burner-end [m^2]

This serves as a rule of thumb to ensure burners function properly under the conditions of limited available natural draft.

- API 12K recommends a maximum *burner heat release density* of 15000 BTU/hr.in^2 (6814 kW/m^2) for natural draft burners.
- For forced draft burners, much higher *burner heat release densities* are achieved. For example, the following table is from Eclipse's ImmersoJet Design Guide 330:

Select burner model

Choose a burner model with a maximum capacity greater than the gross burner input calculated previously. Refer to Figure 3.1.

Figure 3.1 **Capacity Guide**

Model	Tube Size in.	Tube Size mm	Low-Pressure Packaged Blower Btu/hr.	Low-Pressure Packaged Blower kW	High-Pressure Packaged Blower Btu/hr.	High-Pressure Packaged Blower kW	Remote Blower Btu/hr.	Remote Blower kW	Limited Capacity Zinc Phosphate Btu/hr.	Limited Capacity Zinc Phosphate kW	Limited Capacity Iron Phosphate Btu/hr.	Limited Capacity Iron Phosphate kW
2" IJv2	2	50	190,000	55	235,000	69	370,000	108	110,000	32	220,000	64
3" IJv2	3	80	440,000	129	550,000	161	850,000	249	250,000	73	500,000	146
4" IJv2	4	100	830,000	243	1,000,000	293	1,800,000	527	440,000	129	880,000	258
6" IJv2	6	150	2,000,000	586	2,500,000	732	3,600,000	1054	1,000,000	293	2,000,000	586
8" IJv1	8	200	n/a	n/a	n/a	n/a	8,000,000	2344	1,800,000	527	3,600,000	1055

Eclipse ImmersoJet v2.20 Design Guide 330, 10/02

Figure 6: Eclipse ImmersoJet Burner Capacity Guide.

Using the DN250 (8") nominal pipe diameter listed and the 2344 kW burner duty achievable for the Remote Blower option, a *burner heat release density* of 72600 kW/m^2 is calculated.

- Eclipse Engineering Guide, Tech Notes Section 3 Sheet L-1 provides the following table relating to maximum *burner heat release density* values for natural and forced draft applications:

Table 1: Maximum Burner Heat Release Densities (Eclipse Engineering Guide, Tech Notes Section 3 Sheet L-1).

Burner System Type	Max Burner Heat Release Density [BTU/hr.in^2]	Max Burner Heat Release Density [kW/m^2]
Atmospheric, natural draft, 7' (2.1 m) high stack	7000 - 8000	3180 - 3634
Atmospheric with eductor, 0.2" w.c. (50 Pa) draft	15000 - 18000	6814 - 8177
Atmospheric with eductor, 0.4" w.c. (100 Pa) draft	21000 - 25000	9540 - 11360
Packaged forced draft, low pressure fan	15000 - 35000	6814 - 15900
Sealed nozzle-mix, high pressure blower	30000 - 85000	13628 - 38612
Small bore nozzle-mix	80000 - 180000	36340 - 81767

Fire-tube Heat Flux

Another important parameter is the *fire-tube heat flux* :

$$\dot{q}_{ft} = \dot{Q}_p / A_s \qquad \text{Equation 7}$$

where $\dot{q}_{ft}$ is the *fire-tube heat flux* over its outside wetted surface area [kW/m^2]

$\dot{Q}_p$ is the heat transfer to the process fluid [kW]

A_s is the fire-tube outside wetted surface area [m^2]

This parameter is considered important to prevent unwanted boiling of- and/or thermal damage to the heated fluid in direct contact with the fire-tube outside surface. As a result, *fire-tube heat flux* recommended limits depend on the application. The following guidelines are found in the open literature:

- API 12K recommends a *fire-tube heat flux* upper limit of 12000 BTU/hr.ft^2 (37.8 kW/m^2) for water-glycol heating applications. This limit does not apply for pure water heating applications.
- The GPSA Engineering Data Book presents the following recommended average *fire-tube heat flux* ranges for different applications:

Table 2: GPSA Engineering Data Book Recommended Average Fire-tube Heat Flux.

Heating Application	Fire-tube Heat Flux kW/m^2	Heating Application	Fire-tube Heat Flux kW/m^2
Water	32-41	Molten Salt	47-57
50% Ethylene Glycol	25-32	TEG Reboiler	19-25
Low Pressure Steam	47-57	Amine Reboiler	21-32
Hot Oil	19-25		

- The above figures are likely to be quite conservative. For example, pure water heaters are known to have successfully operated at average fire-tube heat fluxes as high as 66 kW/m^2.

Fire-tube First Turn-Around Temperature

The flue gas temperature at the end of the fire-tube radiant section, i.e. at the first turn-around U-bend depends on numerous factors, most of which are beyond the scope of this discussion. The Petroleum Technology Alliance Canada (PTAC) produced an excellent report in August 2005 titled "*Improved Immersion Fire-Tube Heater Efficiency Project*" in which immersion fire-tube performance was analyzed in depth. Several burners were bench-tested in detail albeit these were relatively small units. Nevertheless, that study showed that the flue gas temperature at the first turn-around typically varied between 400°C and 800°C depending on the burner duty. For the burners tested, the higher temperatures occurred during maximum burner duty whereas the lower temperatures were achieved during 4:1 turn-down operation. Since burners are often selected to have 15% to 20% excess capacity, a good estimate of the first turn-around temperature at 100% fire-tube duty (80% burner capacity) would be 700°C.

For forced draft units, the mechanisms for heat transfer may be influenced by the significantly higher flue gas velocities. Two main influences may be identified:

1. Convection heat transfer will be higher due to the increased velocities.
2. Radiation heat transfer may also be influenced. Firstly; forced draft systems will likely have different flame shapes (flame length and diameter) which will impact on the direct luminous flame radiation. Secondly; non-luminous gas radiation forms a significant portion of the overall heat transfer. This component relies on high temperature water and carbon dioxide particles radiating to the fire-tube inner surface. With increasing flue gas velocities (typical of forced draft systems), the gas particle residence time, and hence the non-luminous gas

radiation in the radiant fire-tube section, may be reduced, however no information on this topic could be found in the open literature.

Rather than merely guessing the first turn-around temperature, the model presented implements a simple approach for both natural draft and forced draft. The same AGA correlation is applied to the radiant section only and an efficiency is calculated. From this efficiency the radiant section heat transfer rate may be calculated.

Thermal Efficiency Calculation

For natural draft immersion fire-tube heating applications, it is recommended that Equation 1 or Equation 3 be used unaltered. Note, however, that the effective fire-tube length is longer than the physical fire-tube length as explained in the section below Equation 1.

Forced draft immersion fire-tubes are often significantly smaller in diameter than natural draft fire-tubes of similar capacity as the flow through the fire-tube and stack does not rely on natural draft. Therefore, *burner heat release density* values and fire-tube velocities are typically much higher which may also result in improved thermal efficiencies. Some manufacturers of high velocity burners such as Eclipse's ImmersoJet (IJ) report tested immersion fire-tube efficiencies that are higher than those predicted by Equation 1. This may be due to the increased convection heat transfer resulting from significantly higher flue gas velocities in the fire-tube than what was tested by AGA when Equation 1 was developed.

Forced draft burners are available as packaged units with the burner and combustion air fan packaged as a single unit, as well as units requiring external fans as shown in the following figure:

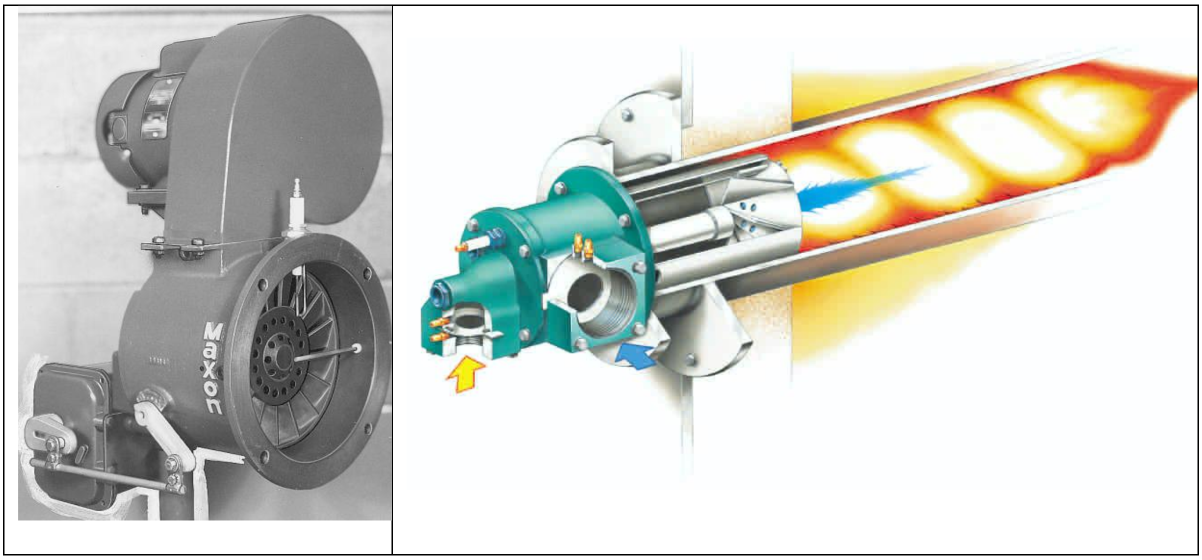

Figure 7: Forced draft burners: Maxon packaged burner (left), and Eclipse burner requiring an external combustion air fan (right).

Figure 8 below was published in the *Eclipse Combustion Design Guide No 330, 10/02 (1997) for ImmersoJet Version 2.2 Series Immersion Burners*. A comparison with Figure 4 (which is based on Equation 1) shows that the Eclipse ImmersoJet series burners offer efficiencies that are approximately 5% higher – compare the red circle and triangle markers in Figure 4, Figure 8, Figure 9 and Figure

10. As shown in Figure 9, the Eclipse ImmersoJet-fitted fire-tube produces efficiencies of up to 5% higher than those predicted by the AGA correlation.

Figure 10 shows the published performance of the Maxon Series "67" burner. Comparison of the triangles between Figure 10 and Figure 4 shows that the Maxon Series "67" curves are simply based on the standard AGA correlation given in Equation 1. Note that this graph is plotted relative to the burner heat release and not in terms of the heat transfer to the heated medium, hence comparison with Figure 4 and Figure 8 will require the burner heat release values in Figure 10 to be multiplied by the efficiency (75%). This graph also provides values for the maximum *burner heat release density* indirectly by specifying required fire-tube diameters at a range of burner duties. It can be shown that for this particular packaged burner, Maxon recommends a maximum *burner heat release density* between approximately 12000 and 16500 kW/m^2. Similarly, for Eclipse TFB series burners (Eclipse Tube Firing Burners Design Guide 310), maximum *burner heat release densities* between 10700 and 14000 kW/m^2 are recommended. Eclipse states that exceeding these maximum burner heat release density values may result in burner pulsation or other operational problems.

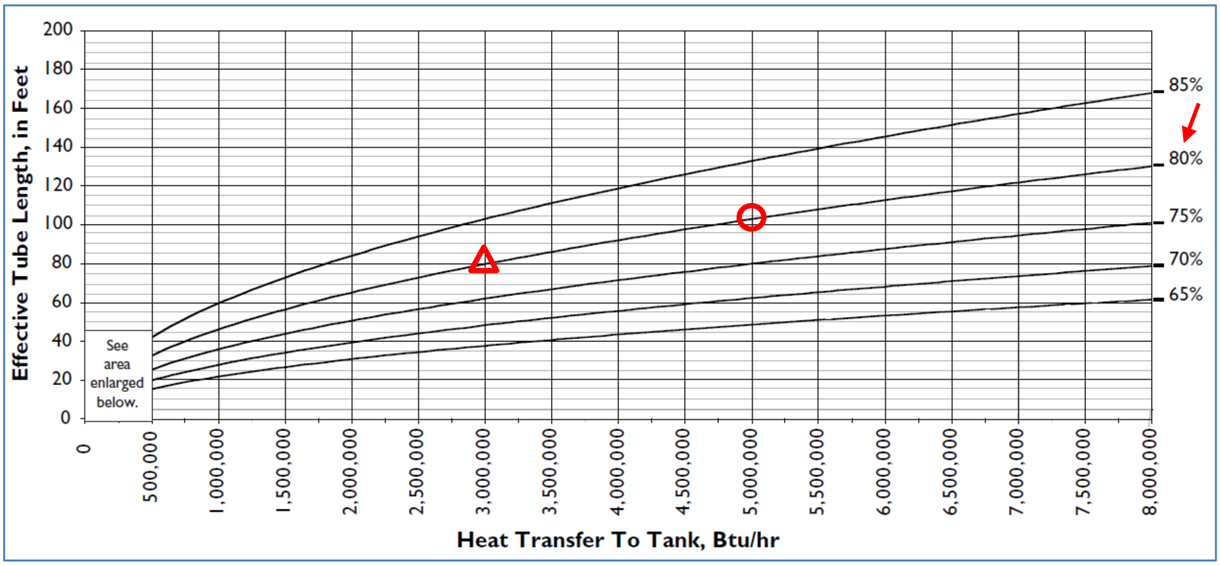

Figure 8: Eclipse ImmersoJet-fitted fire-tube efficiencies.

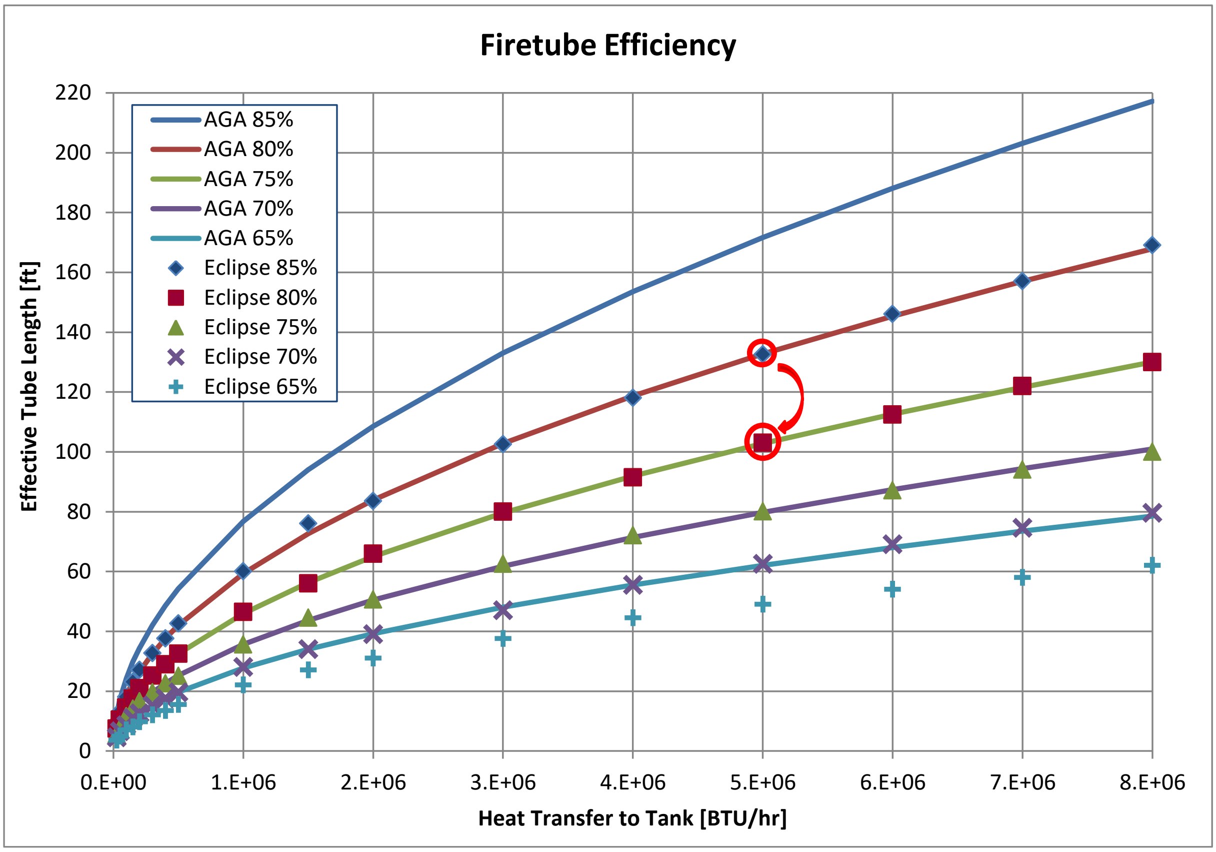

Figure 9: Eclipse ImmersoJet-fitted fire-tube efficiencies vs. the AGA correlation (Equation 1).

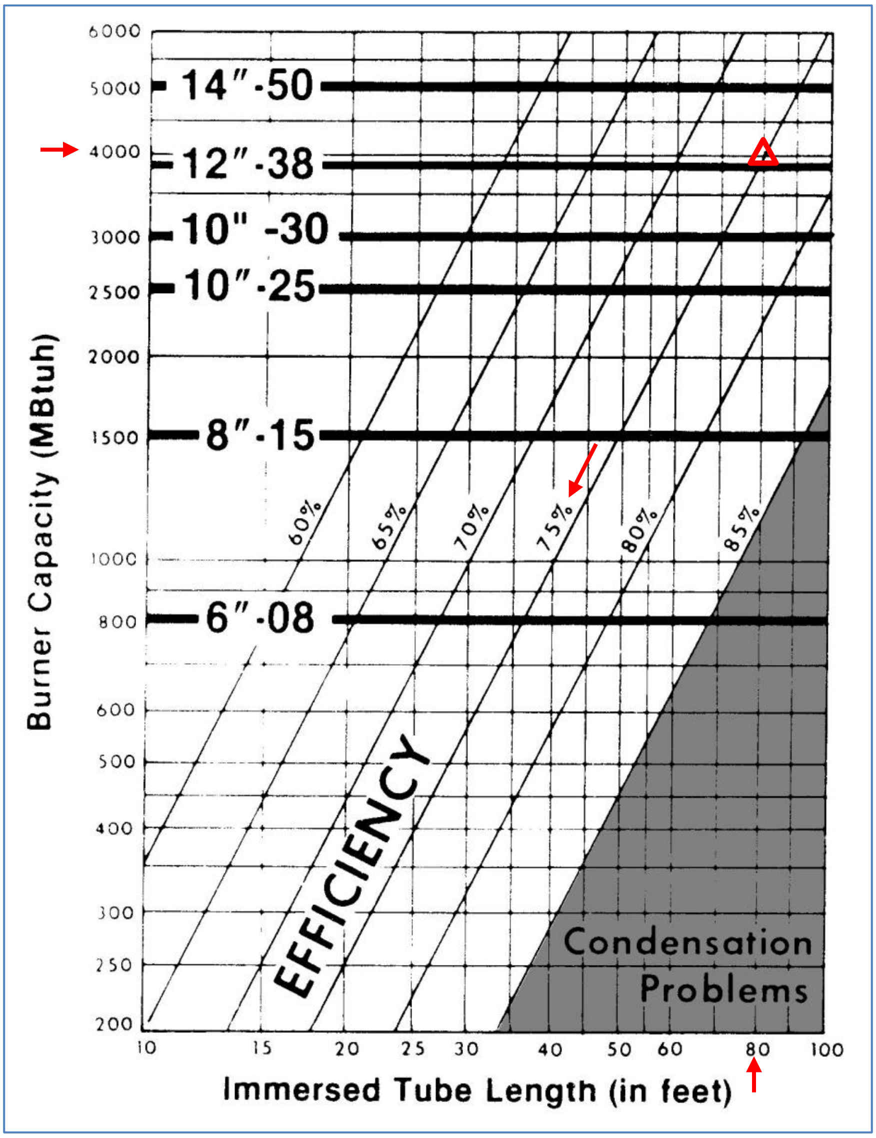

Figure 10: Maxon Series "67" TUBE-O-FLAME gas burner performance (from Maxon Bulletin 2200).

In this case study, the AGA correlation is therefore adapted slightly to accommodate specific burner manufacturer efficiency increases for forced draft burners:

$$\eta_{HHV} = 20\, log_{10}\left(\frac{L_e^2}{\dot{Q}_b}\right) + C_u + C_{fd} \qquad \text{Equation 8}$$

where

C_u is the unit-dependent constant introduced in Equation 3

C_{fd} is the forced draft efficiency adder, typically 3% to 5%

Flownex Immersion Fire-tube Model

A basic immersion fire-tube model has been implemented in Flownex as a compound component which may be added to the library for reuse in future projects. This case study presents a simple Flownex network which utilizes the fire-tube compound component together with previously developed gas composition and property utility scripts, a burner compound component and a simple natural draft stack compound component. The implementation is shown in Figure 11 below.

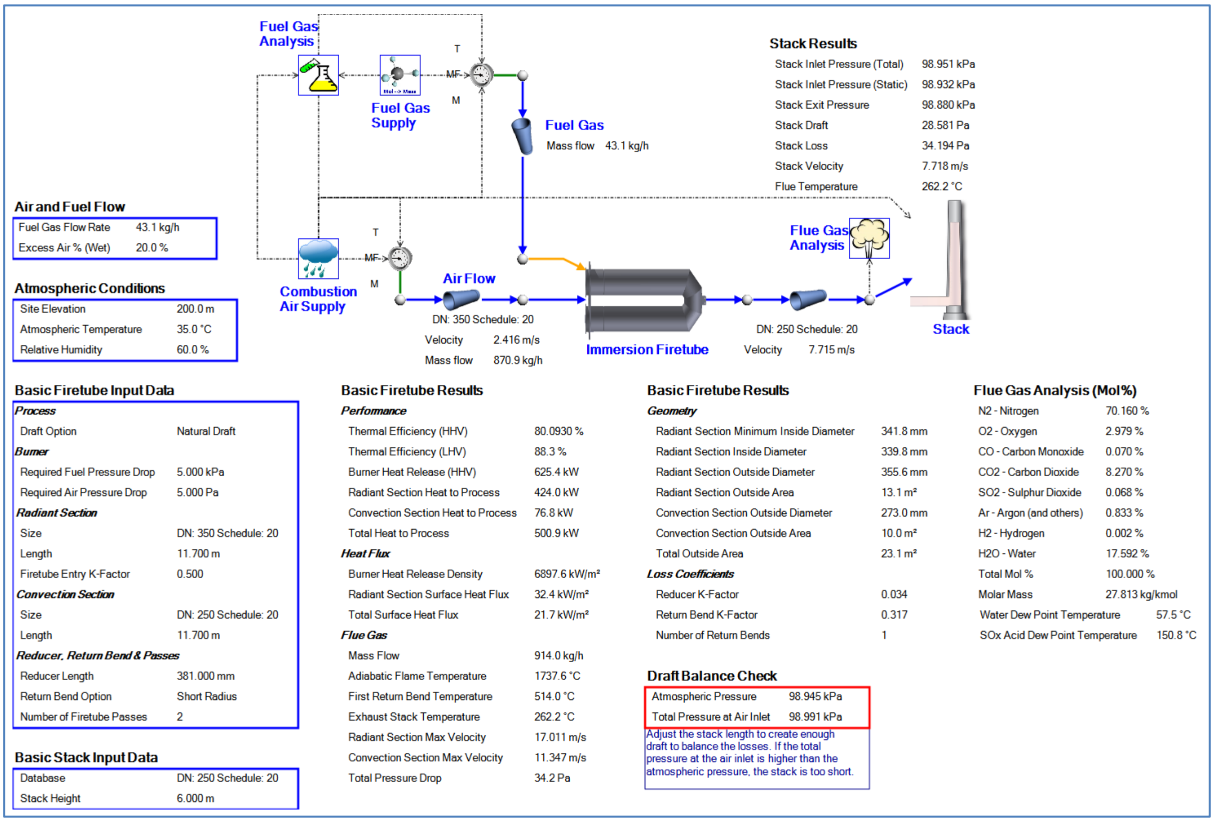

Figure 11: A basic immersion fire-tube model implemented in Flownex.

Combustion fuel may be specified in mol% directly in the *Fuel Gas Supply* script whilst ambient air conditions are specified in the *Combustion Air Supply* script. Not only does the basic immersion fire-tube model implement Equation 8, it also implements a basic burner model to perform the combustion process associated with a fire-tube. As shown above, the flue gas is then ducted to a stack which adds a natural draft component to the flue outlet. Finally, the fire-tube performance is shown in terms of thermal efficiencies, heat transfer rates, heat fluxes, flue gas flow rates, velocities, pressure losses and temperatures. Other metrics such as calculated surface areas and the recommended minimum fire-tube diameter (at the burner end), a draft balance check as well as a flue gas analysis are also presented.

The above example represents a natural draft design and that option is selected in the Basic Fire-tube Input Data section. As discussed before, natural draft designs typically have much lower fire-tube velocities than forced draft designs due to the limited available draft which is supplied only by the stack. For natural draft applications, the stack height is adjusted until the Draft Balance Check balances. In the example above the Draft Balance Check shows that the required total pressure at the air inlet is still higher than the actual atmospheric air pressure, and hence the 6 m-high stack is still incapable of supplying sufficient draft. This will be discussed in detail later.

Figure 12 shows the model inside the compound immersion fire-tube component. Combustion air flows from the left into the burner (itself being a compound component) whilst fuel gas flows to the burner from the top. The burner model will combust the mixture and deliver high temperature flue gas to the radiant fire-tube (or radiant section). Between the radiant section and the convection section there is the option to add a reducer for cases where the radiant section is of a larger diameter than the convection section. Furthermore, depending on the fire-tube geometry – U or W – there will be one or three 180 degree return bends.

In the interest of accuracy, the fire-tube radiant and convection sections are subdivided into 10 segments each. As the flue gas moves through the fire-tube, its properties are allowed to vary along each segment resulting in changes in temperature, density, viscosity and velocity.

Several scripts are employed to calculate heat transfer and pressure loss coefficients. The Fire-tube Efficiency Script at the top-right implements Equation 8 and also calculates the heat transfer rates apportioned to each fire-tube section. These heat transfer values are then assigned to each fire-tube pipe component using data transfer links which will then remove the relevant amount of heat from the flue gas flowing through it. The same script also calculates areas, diameters and heat fluxes. The left-most two scripts are used to obtain fuel gas heating values from the incoming fuel gas and provide that information to the Fire-tube Efficiency Script. The other scripts are used to calculate pressure loss coefficients used in the fire-tube flue gas pressure loss estimation.

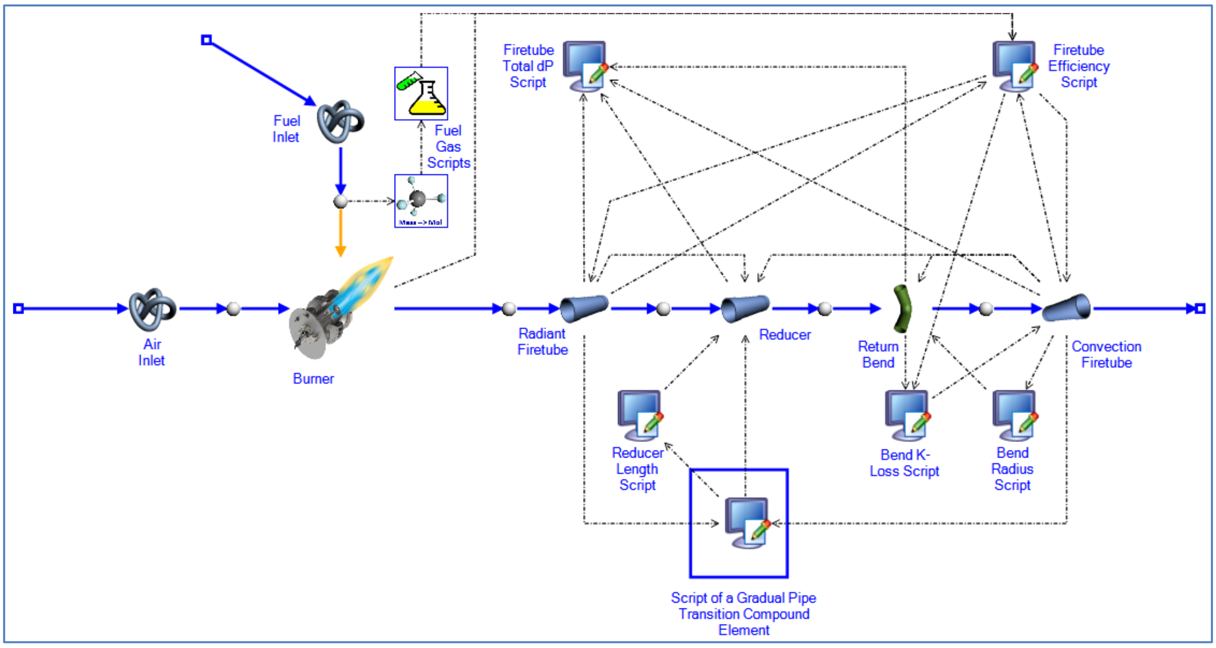

Figure 12: Immersion fire-tube compound component internal elements.

Figure 13 and Figure 14 below shows the immersion fire-tube compound component's input and results property pages. Note that burner heat release density and fire-tube heat flux warning messages are given in the warnings area. API 12K and GPSA recommended fire-tube heat flux values are also listed for convenience.

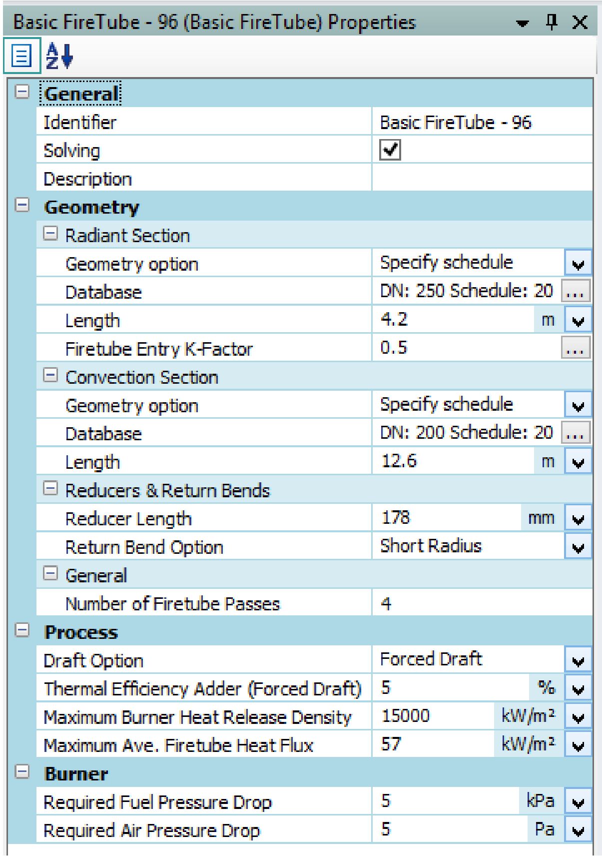

Figure 13: Immersion fire-tube component inputs property page.

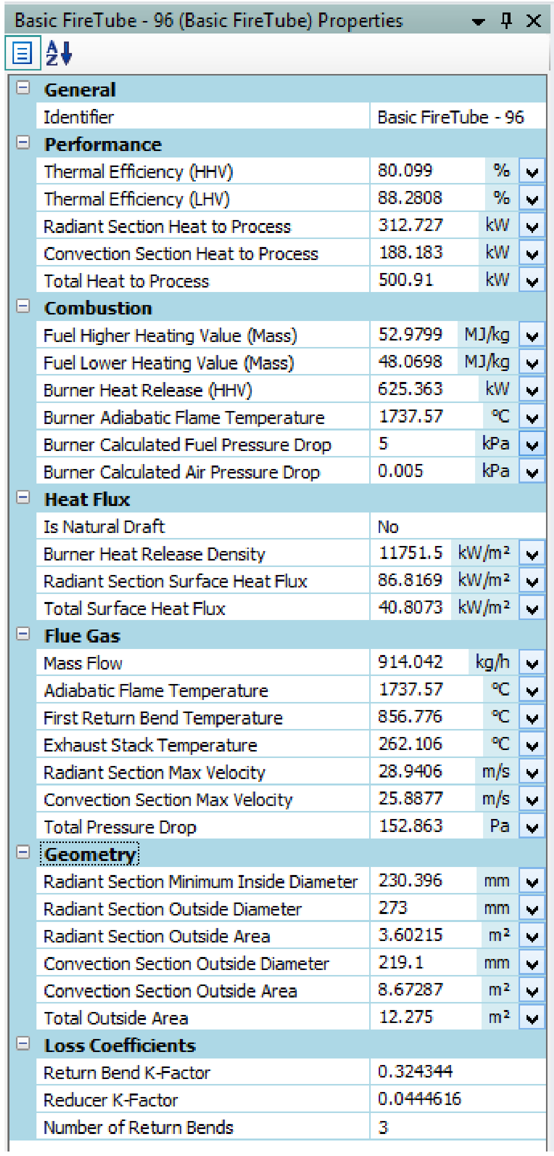

Figure 14: Immersion fire-tube component results property page.

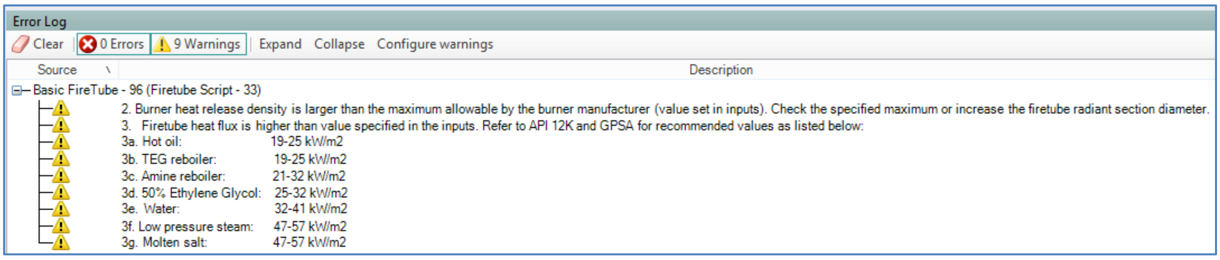

Figure 15: Immersion fire-tube compound component internal elements.

Case Study

As an example, two equivalent immersion fire-tubes are sized, the first one is designed as a natural draft system whilst the second is a forced draft system. Both will aim to achieve the same duty and efficiency. The process requirements are set out as follows:

- Single fire-tube application serving as a molten salt heater.
- Process duty is to be 500 kW.
- Target thermal efficiency is 80% (HHV).
- Maximum practical self-supporting stack height is 6 m.
- Site elevation is 200 m.
- Design atmospheric temperature is 35°C.
- Design atmospheric relative humidity is 60%.

Natural Draft Immersion Fire-tube Design

Natural draft designs normally use a U-shaped fire-tube due to the limited available natural draft produced by the exhaust stack. From Table 2, the recommended average fire-tube surface heat flux upper limit is 47 - 57 kW/m^2 for a molten salt heater application. Following the API 12K recommendation of a maximum burner heat release density of 6814 kW/m^2 as discussed on page 8, the first resulting design is shown in Figure 11 above. There are several problems with this design:

- Using the maximum practical stack height of 6 m, there is a large draft deficit as shown in the Draft Balance Check.
- The main cause of this problem is the relatively high fire-tube velocities of 17 m/s and 11.3 m/s.
- The burner heat release density is at the upper limit.
- The fire-tube, and hence the heater, will be quite long – approximately 12 m – which may become impractical.

The solution is to utilize a larger fire-tube diameter. The redesigned immersion fire-tube design is shown in Figure 16 below. It can be seen that much larger fire-tube diameters are required (DN450 and DN350) to achieve low enough flue gas velocities to reduce the flue gas pressure losses to within the draft capabilities of the exhaust stack. Typical natural draft fire-tube velocities should be approximately 10 m/s or less. The required exhaust stack height has been calculated as 5.4 m which is still quite tall, moreover additional allowances for flame arrestors and possibly an exhaust stack spark arrestor should still be made, hence the exhaust stack length could still be up to 6 m. Furthermore, the burner heat release density and fire-tube surface heat flux values are now comfortably low for the molten salt application.

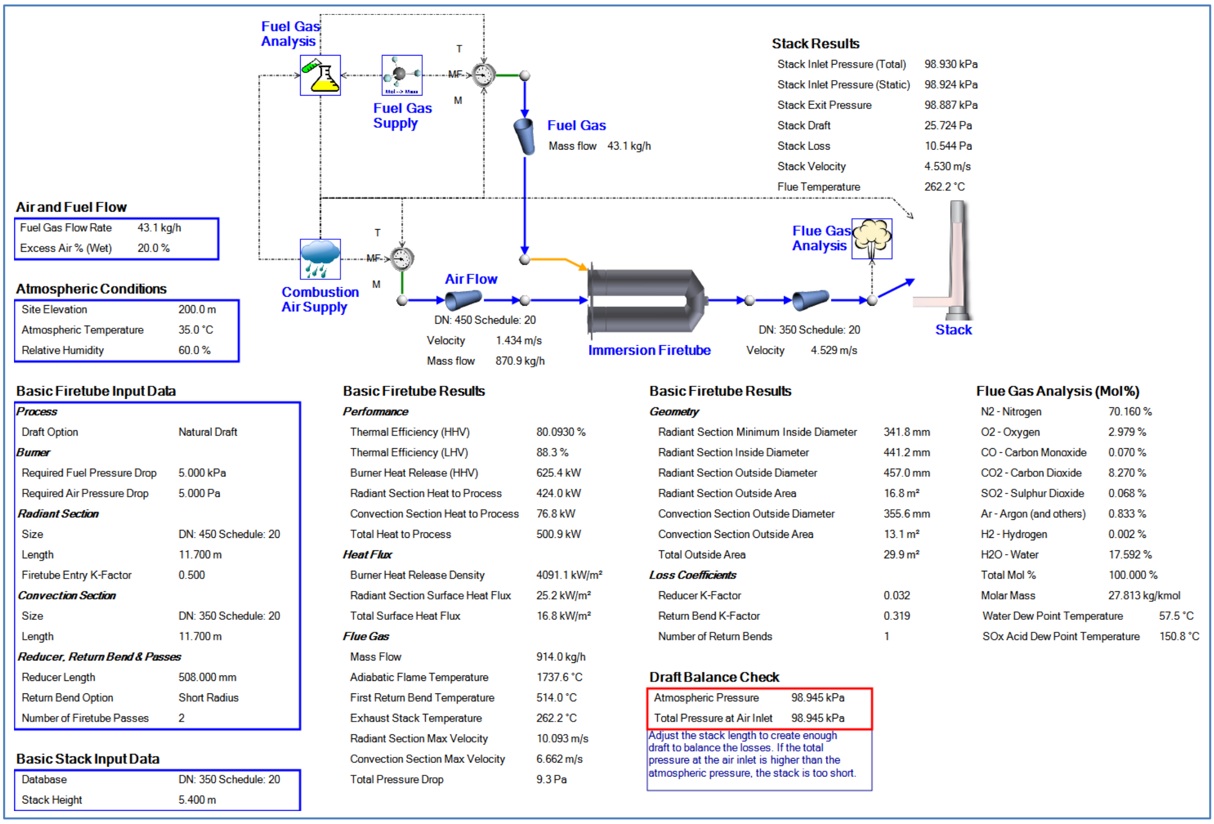

Figure 16: Final natural draft immersion fire-tube design.

The immersion fire-tube is still quite long and the size (DN450) is large in comparison to other 500 kW heater applications. This may be an indication that a natural draft design may not be the most elegant design solution for this particular application. The next section repeats the design process using a forced draft design instead.

Forced Draft Immersion Fire-tube Design

Forced draft fire-tubes often employ W-shaped designs as the combustion air fan is sized to produce enough pressure to overcome the additional pressure losses. For the equivalent forced draft fire-tube design, the same average fire-tube surface heat flux upper limit of 47 - 57 kW/m^2 applies, however much higher burner heat release densities can be used. Assuming a packaged forced draft low pressure fan burner will be used as shown in Table 1, an upper limit for the burner heat release density is taken as 15900 kW/m^2.

As shown in Figure 17 below, for a forced draft design much smaller fire-tube diameters are required – in fact they are almost half the nominal diameters of the natural draft design. The results show that a W-shaped design is used with a 4.2 m radiant section, resulting in a heater of approximately a third of the length of the natural draft equivalent. Also shown is the smaller diameter of the convection fire-tube section in an attempt to keep the flue velocities approximately constant. These velocities are based on the first turn-around temperature estimated for the fire-tube. Since the fire-tube radiant section is fairly short, a comparatively high first turn-around temperature of 857 °C is estimated, resulting in high turn-around velocities.

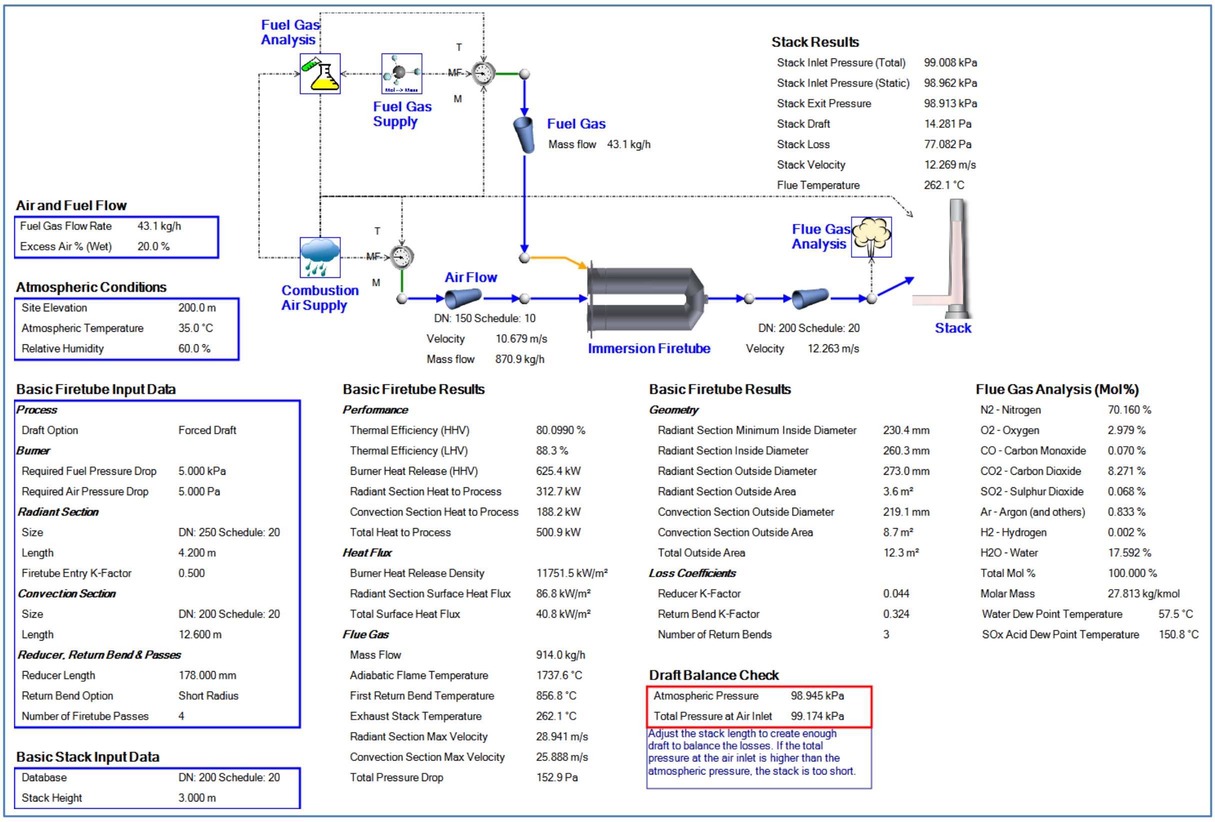

Figure 17: Final forced draft immersion fire-tube design.

Using a more acceptable 3 m stack height, the Draft Balance Check shows a draft deficit of 229 Pa whilst the required combustion air flow rate is 870.9 kg/hr. This information may now be used to select an appropriate combustion air fan. Additional allowances for fan ducting should be made and this Flownex model could easily be extended to include these.

It is interesting to compare the two designs in terms of size and performance as shown in the following table:

Table 3: Natural vs. Forced Draft Immersion Fire-tube Design Comparison.

	Natural Draft Heater	Forced Draft Heater
Heat transfer to process	500 kW	500 kW
Thermal efficiency (HHV)	80%	80%
Radiant section size	DN450	DN250
Convection section size	DN350	DN200
Approximate heater length	12 m	4.5 m
Burner heat release density	4091 kW/m^2	11752 kW/m^2
Fire-tube surface heat flux	16.8 kW/m^2	40.8 kW/m^2
Maximum fire-tube velocity	10.1 m/s	28.9 m/s
Fire-tube flue gas pressure drop	9.3 Pa	152.9 Pa
Stack height	5.4 m	3 m

Even though an average fire-tube surface heat flux of up to 57 kW/m^2 is allowable for the molten salt application, the natural draft design only achieves a fraction of this flux due to draft limitations. This directly contributes to the large size difference between the two designs.

Turn-Down Operation

One of the major design considerations with immersion fire-tubes is the influence of a possible turn-down operation case on the performance of the fire-tube. As shown by Equation 8, a reduction in the burner duty for a specific immersion fire-tube length will result in an increase in the immersion fire-tube thermal efficiency and consequently a lower exhaust stack flue gas temperature. For the example given in Figure 17 above, the duty has been progressively reduced by reducing the fuel flow rate. The results are shown in Figure 18.

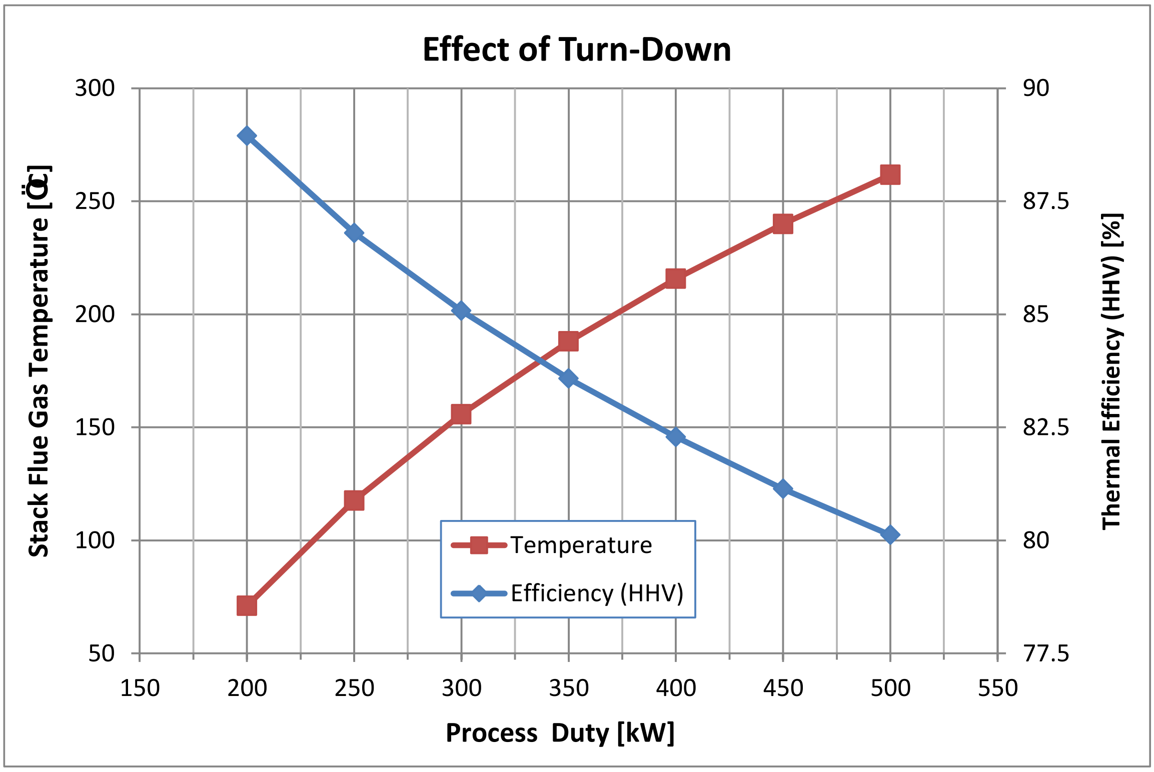

Figure 18: The effect of turn-down of thermal efficiency and exhaust stack temperature.

These results need to be considered in terms of:

- The water dew point temperature; and
- SO_x acid dew point temperature.

As shown in Figure 17, the water and SO_x dew point temperatures are 57.5°C and 150.8°C respectively. The high SO_x dew point temperature is due to the 1.1% (mol%) H_2S present in the fuel gas used for this example.

With efficiencies of up to approximately 87.5% (HHV) for this example, the exhaust stack temperature is above 100°C which is still significantly above the water dew point temperature, so the water dew point is then not a problem. However, the SO_x acid dew point temperature mostly depends on the sulfur content of the fuel gas and is typically between 120°C and 150°C. A typical relationship between H_2S content in fuel gas and the SO_x (SO_3 and H_2SO_4) dew point temperatures are presented in Figure 19 below. Therefore as stated earlier, when sulphur is present in the fuel gas, stack temperatures should at least be 150 - 200°C minimum, and this must be considered at the maximum turn-down (minimum duty) case.

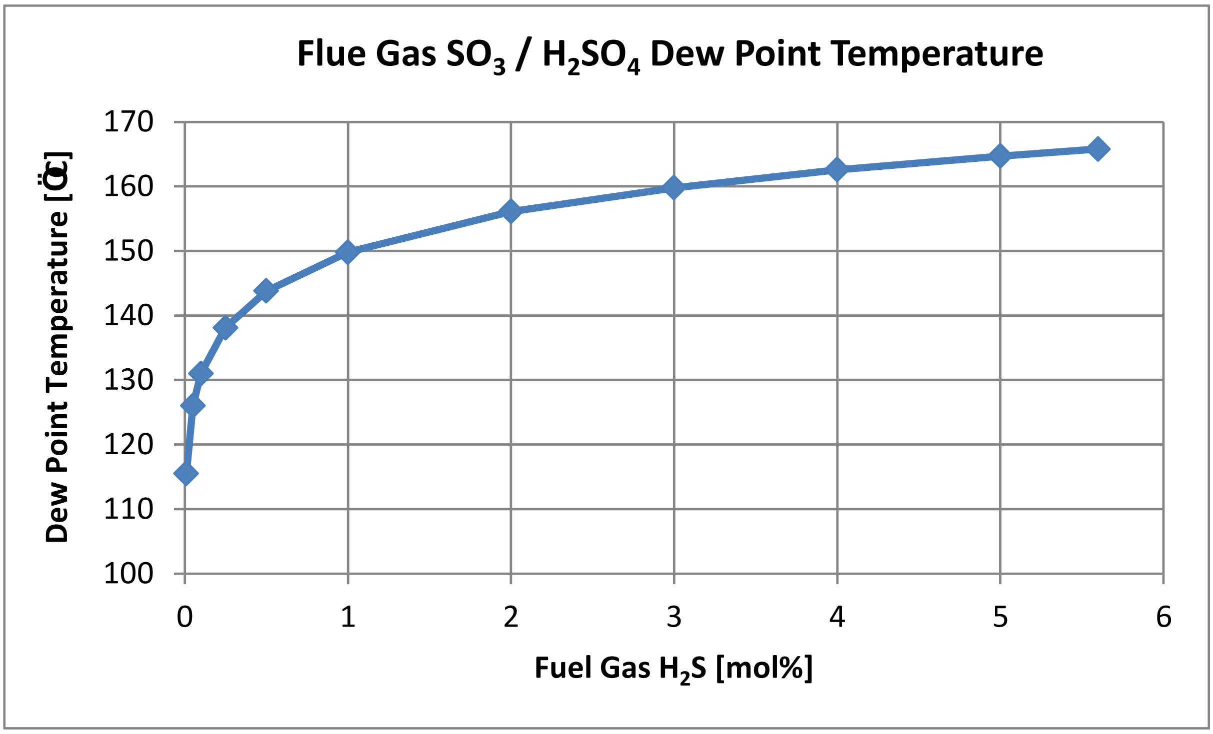

Figure 19: The effect of fuel gas sulfur content on flue gas SO_x dew point temperature.

For the above example the acid dew point temperature is 150.8°C, hence a minimum exhaust stack temperature of 200°C should be targeted which will occur at a duty of approximately 375 kW as shown in Figure 18. Therefore, for this specific high sulfur containing fuel gas, this design can virtually not accommodate any turn-down at all.

There are a few potential solutions if turn-down is required:

- Use sulfur-free or low sulfur fuel. According to the API 12K recommendation discussed earlier, the stack temperature can then be allowed to drop to approximately 120°C.
- Reduce the immersion fire-tube thermal efficiency by reducing its length. This will cause a higher stack temperature during normal operation which may be considered undesirable, but it will also result in higher stack temperatures during turn-down, enabling the heater to turn down further.
- Employ an on-off burner control strategy instead of a modulated burner control system. This approach is possibly the simplest and most sensible and is discussed in more detail below.

Burner control systems typically operate in two modes: modulating and on-off. For modulating burners the flow rate of fuel gas to the burner is modulated by an upstream flow control valve depending on the required heat load. On-off burners are simpler and rely on the thermal inertia of the heated fluid. The burner simply operates between maximum duty (fully on) and no duty (off) where the burner is off and only the pilot burner remains on. Since on-off burner systems do not modulate the burner duty, the exhaust flue gas temperature will be at the maximum value when the burner is on and hence the stack temperature remains high. Therefore, in cases where the turn-down duty causes the exhaust flue gas temperature to fall below dew point values, an on-off burner system may offer a simple solution.

Summary

A simple immersion fire-tube model has been developed and implemented in Flownex as a compound component. In this case study, a natural draft and a forced draft heater were designed to meet the same process requirements. A detailed analysis and comparison of the two designs have been presented. The complete combustion and heat transfer process have been modeled and the effects of turn-down operation and water and SO_x dew point temperatures have been discussed and possible problem areas highlighted. Solutions to some of the problems have also been offered.

Flownex has the unique ability to simultaneously model combustion, heat transfer and fluid mechanics problems. This capability makes Flownex the ideal tool to design and analyze immersion fire-tube heat transfer processes. It would be much more challenging to design and analyze an entire immersion fire-tube-based heater system at the level of detail presented without Flownex as an engineering design and analysis tool.