Fire Heat Flux to Unwetted Vessel for Depressuring Calculations

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25-Nov-2012

Introduction

Depressuring calculation is one the subject that has been greatly developed over the past decade. The use of software has extremely improved the accuracy and speed of the calculations too. However, providing the proper inputs to the software in one hand and utilizing the available software capabilities to produce the meaningful results on the other hand has ended up with lots of lessons learned. One of the lessons learned was that using "Fire API521" heat input model (originally developed for the wetted vessels) in Hysys for an unwetted vessel results in zero heat flux as the actual wetted area of this type of vessel is zero. Then, engineers started exploring how to define this parameter in the depressuring calculations.

They noticed that API-521, 5th edition, section 5.15.1.2.2 states that "the recent calculation indicates that heat flux of the fire is in the range of approximately 80 to 100kW/m^2". This is based on the test results published in the Annex 1, Table A.1 of the same standard for wetted vessels. The results of the same test have been used by API committee to introduce $q = 70.7\ F\ A^{0.82}$ (q in kW, where there is no draining facility) for calculation of heat fluxes to the wetted vessels.

Therefore, in absence of any standard guideline for the depressuring, many used this range (typically 100kW/m^2) as the heat input for depressuring calculations of unwetted (gas filled) vessels. This note reviews the adequacy of this range for the depressuring calculations and recommends the alternative method of defining the fire heat flux to the unwetted vessels.

Fire Heat Flux

As discussed in the note "The Effect of Different Parameters on Depressuring Calculation Results", the following equation is usually used in Hysys depressuring utility to define the heat input to the gas filled vessels[1]:

$$q = C_1 + C_2\ time + C_3\ (C_4 - Vessel\ Temp) + C_5\ LiqVol_{(time = t)} / LiqVol_{(time = 0)} \qquad (1)$$

Where

$C_1 = Q\ A$

$Q = 80\text{-}100\ kW/m^2$

$C_2 = C_3 = C_4 = C_5 = 0$

Using equation (1) with above coefficients seems to resolve the problem of setting fire heat flux, however, it introduces new questions. For example:

- This value seems to be high at the first glance compared to the wetted vessel heat flux. According to the above equation ($q = 100\ A$), the gas inside an unwetted vessel with the surface area of 1 m^2 can absorb 100kW while the wetted vessel's heat absorption formula $q = 43.2\ F\ A^{0.82}$ (where there is a proper draining facility) predicts that liquid absorbs 43.2kW per each square meter of wetted surface (assuming F=1.0). The higher heat transfer to the gas through the unwetted surface totally contradicts the heat transfer fundamentals.
- Considering the constant heat flux of 100kW/m^2 will result in highly unreasonable final temperature (sometimes beyond 2000°C) in Hysys depressuring results. This is because the vessel gas content is depleted as depressuring continues while heat input remains constant causing the gas temperature to increase drastically. Therefore, using the constant heat flux method which does not take the vessel internal and external heat transfer coefficient into account may not be correct.

Heat Transfer Rate

From the heat transfer viewpoint, the fire heat is absorbed by the vessel metal through the radiation and convection in the first step. Then, depending on the metal surface temperature, part of the absorbed heat is radiated back to the surroundings and the rest of the heat is accumulated in the system (vessel metal + vessel fluid) by time causing temperature to increase. The amount of the heat transferred from the fire to the vessel body is the function of many parameters which is out of the scope of this note but looking at heat transfer background of this phenomenon will help to understand the difference between the wetted and unwetted vessels exposed to fire:

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[1] Compared to Hysys "Fire API521", for fire depressuring calculations of wetted vessels:
$q = C_1\ C_3\ (Wetted\ Area)^\wedge C_2$
Where q is in kW, $C_1 = 43.2$, $C_2 = 0.82$ and $C_3 = 1$

- **External Heat Transfer Mechanism**

  The external free/force convection is the same for both wetted and unwetted vessel as it is related to the fire condition not to vessel fluid. The external heat radiation to the wetted vessels should be a bit higher than the unwetted vessels as the wall temperature is lower in the wetted vessel due to high internal heat transfer coefficient which keeps the wetted wall cool. Despite the difference between the wall temperatures, since the flame temperature is the same for both cases and much higher than wall temperatures, therefore the radiation heat transfer to the wetted and unwetted vessel will be almost identical.

  This means that what is transferred from the fire to the vessel metal will be almost constant irrespective of the vessel type. Therefore, the API statement that 80-100 kW/m^2 is transferred from the fire to the vessel should be correct.

- **Internal Heat Transfer Mechanism**

  For the wetted vessel, the internal heat transfer mechanism is boiling which provides extremely high heat transfer coefficient. For the unwetted vessels, the internal heat transfer mechanism is mainly free convection with some radiations at high metal temperatures. But in general the rate of heat transfer through free convection and radiation is lower than the wetted vessel.

  This means that almost all the heat absorbed by metal during fire will be transferred to liquid whereas only part of it will be absorbed by gas in unwetted vessels. The rest of the heat input which is not absorbed by the fluid is used to heat up the metal, reducing its strength and leading to an earlier failure.

Therefore, the wall temperature of unwetted vessels will be much higher than the wetted ones as the heat absorption rate is much higher than the heat dissipation rate. Furthermore, the overall heat input to the gas inside the unwetted vessel should be lower than 100kW/m^2 as the heat transfer resistance between the gas and metal is more than liquid and metal.

Based on the above paragraph, the heat transfer resistance inside the unwetted vessel is much higher than the external resistance, therefore, neglecting the radiation from the vessel wall to the gas, the overall heat transfer rate from the fire to the gas is mainly controlled by the vessel internal free convection heat transfer which can be calculated through equation (2):

$$q_i = h\,A\,\Delta T \qquad (2)$$

In order to further simplify the solution of this equation, with the following assumptions are made:

1) The first minutes of the fire during which the vessel wall temperature sharply increases is ignored.
2) The maximum wall temperature is 593°C (1100°F) during depressuring. This is not very realistic assumption as the flame temperature is much higher than 593°C, driving the system towards the higher temperatures. However, considering the fact that depressuring is started on the onset of fire, and just for the purpose of this calculation credit can be taken for the fire water and depressuring systems to prevent very high metal temperatures. Furthermore, there is high likely that vessel won't withstand the temperature higher than 593°C for a long time; therefore there is no point in protecting a vessel which has already failed.

For such conditions (isothermal surface), the free convection heat transfer coefficient can be calculated using the following equations:

$$Nu = C\left(Gr\,Pr\right)^m \qquad (3)$$

Where,

$$Gr = \frac{g\rho^2\beta\Delta T x^3}{\mu^2}\ ,\quad Pr = \frac{\mu C_p}{k}\ ,\quad Nu = \frac{h\,x}{k}\ ,\quad \Delta T = T_w - T_g$$

Therefore, the internal free convection heat transfer coefficient can be calculated by equation (4).

$$h = \frac{C\,k}{x}\left(\frac{g\rho^2\beta\Delta T x^3}{\mu^2}\right)^m\left(\frac{\mu C_p}{k}\right)^m \qquad (4)$$

Replacing $\beta = \frac{1}{T_{av}}$ in equation (4) where $T_{av} = \frac{T_w + T_g}{2}$:

$$h = C\left(\frac{\rho^2 g C_p\, k^{\left(\frac{1}{m}-1\right)}}{\mu\, x^{\left(\frac{1}{m}-3\right)}}\left(\frac{\Delta T}{T_{av}}\right)\right)^m \qquad (5)$$

NOMENCLATURE	
A	Vessel surface area exposed to fire, m^2
C	Constant
Cp	Gas heat capacity, J/kg K
D	Vessel diameter, m
L	Vessel length, m
g	Gravity acceleration, 9.81m/s^2
Gr	Grashof number, dimensionless
h	Heat transfer coefficient, J/s m^2 K
k	Gas thermal conductivity, J/s m K
m	Constant
Nu	Nusselt number, dimensionless
P_r	Prandtl number, dimensionless
q	Fire heat input, kW
q_i	Internal heat transfer rate, kW
Q	Heat flux, kW/m2
T_g	Vessel gas temperature, K
T_w	Vessel wall temperature, K
T_{av}	Gas average temperature, K
x	Characteristic dimension, m
β	Cubical expansion coefficient, 1/K
μ	Gas dynamic Viscosity, kg/m s
ρ	Gas density, kg/m^3

Equation (5) clearly shows that systems with the higher gas density (higher pressure, higher molecular weight, lower temperature) have higher heat transfer coefficient. My past experience in heat transfer calculations also reveals that for most process vessels, flow on the inner surface of the vessel will be turbulent (10^9 < Gr Pr < 10^{13}) whereas the laminar flow (10^4 < Gr Pr < 10^9) is observed in very low pressure (basically atmospheric) systems.

Table 1 – Equation 5 Coefficients

Vessel type	Gr Pr	C	m	x
Vertical	10^4-10^9 10^9-10^{13}	0.59 0.10	1/4 1/3	L
Horizontal	10^4-10^9 10^9-10^{13}	0.53 0.13	1/4 1/3	D

Case Study

Table 2 shows the fire heat flux to a horizontal vessel with total surface area of 20m^2, containing Methane at initial temperature and pressure of 0°C and 100bar. It assumes that the wall temperature is at 593°C to calculate the internal free convection heat transfer coefficient when the gas temperature increases from the initial to very close to the wall temperature.

Table 2 - Internal Heat Transfer Calculation for Methane at Different Temperatures

T_g	**°C**	0	100	200	300	400	500
T_{av}	**°C**	297	347	397	447	497	547
ρ at T_{av}	**kg/m^3**	33.555	30.675	28.279	26.250	24.507	22.990
Cp at T_{av}	**J/kg °C**	3254.5	3421.2	3589.5	3754.6	3913.4	4063.4
μ at T_{av}	**kg/m s**	1.97E-05	2.07E-05	2.18E-05	2.28E-05	2.39E-05	2.49E-05
k at T_{av}	**J/m s °C**	8.38E-02	9.29E-02	1.02E-01	1.12E-01	1.22E-01	1.32E-01
Gr	**-**	3.70E+12	1.71E+13	9.71E+12	5.29E+12	2.60E+12	9.48E+11
Pr	**-**	0.765	0.764	0.764	0.765	0.766	0.767
Gr Pr	**-**	2.83E+12	1.30E+13	7.42E+12	4.05E+12	1.99E+12	7.27E+11
h	**W/m^2 °C**	308.1	284.0	259.1	231.7	199.0	154.1
Q	**kW/m^2**	**182.74**	**140.03**	**101.85**	**67.90**	**38.42**	**14.34**

According to the results, the internal heat flux from the wall at 593°C to the Methane at 100 bar and low temperatures can be as high as double of API fire flux range. For this particular system, this means that the entire heat from the fire (100kW/m^2) is initially transferred to the gas inside the vessel before the heat flux drops due to the reduction in heat transfer coefficient and more importantly the differential temperature between the vessel wall and the gas. The heat flux calculated in the above example is almost the highest value possible among all other hydrocarbon gases. Table 3 summarizes the effects of the pressure on the free convection heat transfer coefficient from the vessel wall at 593°C to the different gases for comparison.

Table 3 - Internal Heat Transfer Coefficient for Different Gases

Composition	Pressure (bar)	Temp. (°C)	h (kW/m^2 °C)	Heat flux (kW/m^2)
Methane	100	0	**0.308**	182.7
Methane	10	0	**0.065**	43.2
Methane	1	0	**0.011**	6.6
Ethane	10	0	**0.080**	76.8
Air	100	0	**0.173**	102.9
Air	1	0	**0.006**	3.6

Another case study for the same vessel at the conditions mentioned above indicates that if it is assumed that the wall temperature during the depressuring is always 200°C above the gas temperature (when the gas temperate increases from 0°C to 500°C - not constant), the heat flux from the vessel wall to Methane will be in the range of 40 to 50kW/m^2 which is almost the half of the heat flux from the fire to the vessel.

Conclusion

80-100 kW/m^2 is the total heat input to the unwetted system (metal and content). However, depending on the gas heat transfer features, all or part of this heat is transferred to the gas inside the vessel. Therefore, considering fire heat flux of 80 to 100kW/m^2 for unwetted vessel depressuring calculation during the fire may result in unrealistic size for depressuring system depending on the system conditions. This value is certainly conservative for relatively low pressure systems but it may be fairly valid for the high pressure systems. Therefore, it should be verified on case by case basis.

Based on the above observations, there are two methods to build a realistic Hysys model for the unwetted vessels with respect to the fire heat input:

1. Using the "Fire" model (equation (1)) with the following settings along with the heat loss model of "None" to simulate the net heat input to the gas which is important from the depressuring point of view.

$C_3 = h\ A$ (kW/°C)

$C_4 = 593°C$

$C_1 = C_2 = C_5 = 0$

This is nothing but reproducing equation (2) by using available tools in Hysys. This method produces sensible results as the gas temperature is limited to the reasonable plate temperature and the heat input to the gas is the function of time as the gas temperature ("Vessel Temp" in equation 1) is changing by time.

By use of the maximum heat transfer coefficient from Table 3 (Methane at 100bar and 0°C), C_3 for the above example can be set at:

$C_3 = 0.308$ kW/m^2°C x 20 = 6.0 kW/°C.

Using this value is conservative because:

- The heat transfer coefficient decreases as the system pressure reduces during depressuring. Refer to Table 3 where heat transfer coefficient decreases from 0.308 kW/m^2°C at 100bar to 0.065kW/m^2°C at 10bar.
- The heat transfer coefficient decreases as the gas temperature increases during fire depressuring. Refer to Table 2 where heat transfer coefficient decreases from 0.308 kW/m^2°C at the gas temperature of 0°C to 0.154kW/m^2°C at 500°C.

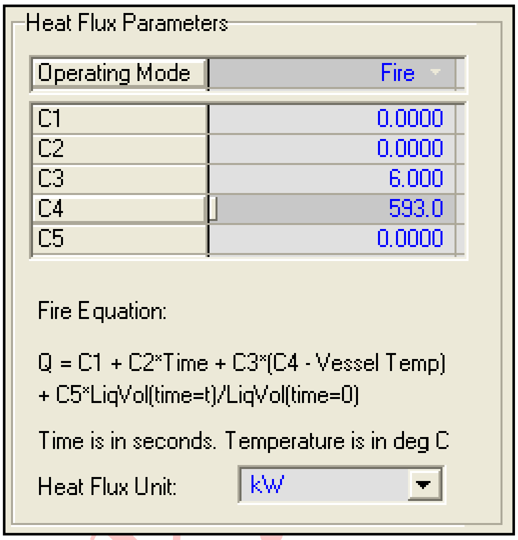

Figure 1 – Hysys Fire Heat Flux Equation

2. Using the "Fire" model (equation (1)) with the following settings along with the "Detailed" heat loss model to include the vessel metal mass and other ambient conditions which will help to have the realistic heat input to the gas. Figure 2 shows some of the inputs to Hysys "Detailed" heat loss model.

$C_1 = 100\ A$ (kW)

$C_2 = C_3 = C_4 = C_5 = 0$

This approach will produce the realistic results for depressuring rate but the problem of the unrealistic final temperature will remain unresolved.

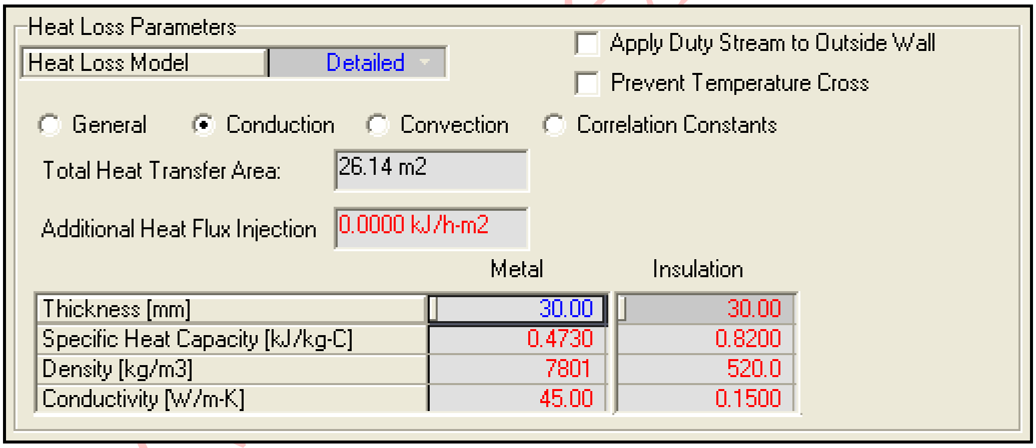

Figure 2 – Hysys Depressuring Detailed Heat Loss Model Parameters

Contact

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