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Design Considerations for Antisurge Valve Sizing

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Centrifugal Compressors experience a phenomenon called “Surge” which can be defined as a situation where a flow reversal from the discharge side back into the compressor casing causing mechanical damage.

The reasons are multitude ranging from driver failure, power failure, upset process conditions, start up, shutdown, failure of anti-surge mechanisms, check valve failure to operator error to name a few. The consequences of surge are more mechanical in nature whereby ball bearings, seals, thrust bearing, collar shafts, impellers wear out and sometimes depending on the how powerful are the surge forces, cause fractures to the machinery parts due to excessive vibrations.

The following tutorial explains how to size an anti-surge valve for a single stage VSD system for Concept/Basic Engineering purposes.

General Notes & Assumptions

1. Centrifugal compressors are characterized by “Performance curves” which are a plot of Actual Inlet Volumetric Flow rate [Q] vs. Polytropic head [Hp] for various operating speeds. The operating limits for performance curves are the surge line and the choke flow line, beyond which any compressor operation can cause severe mechanical damage.

2. Below is an image of performance curves characteristics which indicates the surge flow line and choked flow line, both of which extend from the minimum speed Q vs. Hp curve to the maximum speed Q vs. Hp curve. The surge curve is defined as the Surge Limit Line [SLL] and an operating margin is provided [e.g., 10% on flow rate] which is called the surge control line [SCL].

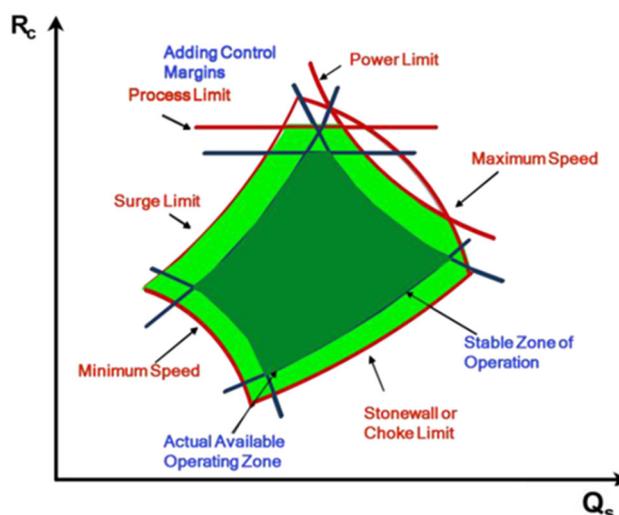


Figure 1. Performance Curves Operating Limits [1]

To ensure process safety & avoid mechanical damage, the anti-surge valve (ASV) must be large enough to recycle flow sufficiently. An undersized valve would fail to provide enough recycle flow to keep the compressor operating point away from SCL and SLL. Whereas over sizing the ASV leads to excess gas recycling that can drive the compressor into the choke flow region. Oversized valves also create difficulties in tuning the controllers due to large controller gain values and limited stroke.

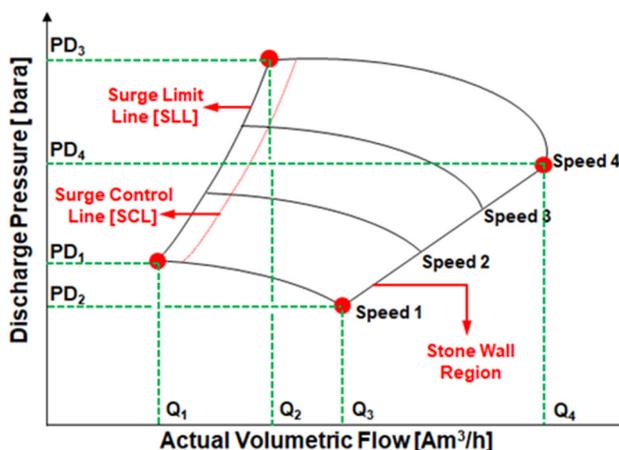


Figure 2. Sizing Criteria for Anti-surge Valve

To size the anti-surge valve (ASV), the philosophy employed should consider, operating the compressor on the right hand side of the SCL while also ensuring the operating point does not cross the choke flow line. Towards this, the recycle flow rates across the ASV can be taken to be 1.8 to 2.2 times the surge flow rate.

5. Traditionally ASVs have linear opening characteristics, though sometimes equal percentage characteristics can be incorporated into the linear trend. Quick opening characteristics are not preferred due to poor throttling characteristics while Equal percentage valves suffer from slow opening during the early travel period.

6. The stroking time of the valve should be ideally less than 2 sec with less than 0.4 sec time delay and no overshoot. The actuator response time must be less than 100 msec and the noise limit is ~85 dBA. The maximum noise level allowed is 110 dBA.

7. Anti-surge valves are Fail-open [FO] type and should provide stable throttling. Fluid velocities should be less than 0.3 Mach to avoid piping damage and valve rattling.

8. The anti-surge valve can be operated pneumatically or by solenoid action. For valve sizes greater than 16", a motor operated valve can be used to effectuate the fast opening requirements.

9. Although the current tutorial provides a methodology to size an ASV which is suitable during Concept/Basic Engineering stage, a compressor dynamic simulation shall be performed with the actual plant layout based on detailed design to verify if the ASV can cater to preventing a surge during start-up & shutdown scenarios.

10. The final ASV size must be verified and arrived in concurrence with the turbomachinery vendor, valve manufacturer, if the ASV can cater to the surge control philosophy employed, slope of the performance curves and polytropic efficiency maps at the choke points.

Anti-Surge Valve Sizing Methodology

To size the anti-surge valve, the ANSI/ISA S75.01 compressible fluid sizing expression is chosen and the flow rates are taken for at least 1.8 to 2.2 times the surge flow rate.

Step 1: Calculate Piping Geometry (Fp)

$$F_p = \left[1 + \frac{\sum K}{890} \left(\frac{C_v}{d^2} \right)^2 \right]^{-1/2} \quad (1)$$

Where,

Fp = Piping geometric Factor [-]

Cv = Valve Coefficient [-]

d = Control Valve Size [inch]

SK = Sum of Pipe Resistance Coefficients [-]

The value of Fp is dependent on the fittings such as reducers, elbows or tees that are directly attached to the inlet & outlet connections of the control valve.

If there are no fittings, Fp is taken to be 1.0. The term SK is the algebraic sum of the velocity head loss coefficients of all the fittings that are attached to the control valve & is estimated as,

$$\sum K = K_1 + K_2 + K_{B1} - K_{B2} \quad (2)$$

Where,

K₁ = Upstream fitting resistance coefficient [-]

K₂ = Downstream resistance coefficient [-]

K_{B1} = Inlet Bernoulli Coefficient [-]

K_{B2} = Outlet Bernoulli Coefficient [-]

Where,

$$K_{B1} = 1 - \left(\frac{d}{D_1} \right)^4 \quad (3)$$

$$K_{B2} = 1 - \left(\frac{d}{D_2} \right)^4 \quad (4)$$

Where,

D₁ = Inlet Pipe Inner Diameter [in]

D₂ = Outlet Pipe Inner Diameter [in]

The most commonly used fitting in control valve installations is the short-length concentric reducer. The expressions are as follows,

$$K_1 = 0.5 \times \left[1 - \left(\frac{d^2}{D_1^2} \right) \right]^2, \text{ for inlet reducer} \quad (5)$$

$$K_2 = 1.0 \times \left[1 - \left(\frac{d^2}{D_2^2} \right) \right]^2, \text{ for outlet reducer} \quad (6)$$

Step 2: Calculate Valve Coefficient (C_v)

To calculate the valve C_v, the following ANSI/ISA expression is used.

$$C_v = \frac{M}{N_g F_p P_1 Y \sqrt{\frac{X \times MW}{T_1 \times Z}}} \quad (7)$$

$$X = \frac{\Delta P}{P_1} \quad (8)$$

$$Y = 1 - \frac{X}{3 \times F_k \times X_T} \quad (9)$$

$$F_k = \frac{k_1}{1.4} \quad (10)$$

If $X > F_k \cdot X_T$, then flow is Critical.

If $X < F_k \cdot X_T$, then flow is Subcritical.

For Critical flow, the value of 'X' is replaced with $F_k \cdot X_T$ and the gas expansion Factor [Y] and valve coefficient [C_v] is to be computed as,

$$Y = 1 - \frac{F_k \times X_T}{3 \times F_k \times X_T} = 0.667 \quad (11)$$

$$C_v = \frac{M}{0.667 \times N_8 \times F_p \times P_1 \sqrt{\frac{F_k \times X_T \times MW}{T_1 \times Z}}} \quad (12)$$

If the control valve inlet and outlet piping is provided with reducers and expanders, then the value of X_T is replaced with X_{TP} as follows,

$$X_{TP} = \frac{X_T}{F_p^2} \times \left[1 + \frac{X_T(K_1 + K_{B1})}{1000} \left(\frac{C_v}{D_1^2} \right)^2 \right]^{-1} \quad (13)$$

Where,

C_v = C_v value at Valve 100% Open [-]

M = Mass Flow Rate [kg/h]

N_8 = Constant [Value = 94.8]

F_p = Piping Geometry Factor [-]

DP = Pressure drop across ASV [bar]

P_1 = Inlet Pressure [bara]

Y = Gas Expansion Factor [-]

X = Pressure Drop Ratio [-]

Z = Gas compressibility Factor [-]

T_1 = Inlet Temperature [°K]

F_k = Gas specific heat to air specific heat ratio

k_1 = Gas specific heat ratio at valve inlet [-]

X_{TP} and X_T = Pressure drop ratio factor [-]

MW = Molecular Weight of gas [kg/kmol]

To estimate the compressor mass flow rate from the suction density [ρ_s] and compressor actual inlet flow rate, it can be estimated as,

$$\rho_s = \frac{P \times MW}{Z \times R \times T} \quad (14)$$

$$M = Q_s \times \rho_s \quad (15)$$

Where,

R = Gas Constant [0.0831447 m³.bar/kmol.K]

Q_s = Compressor Suction Vol flow rate [m³/h]

To arrive at a converged value of F_p , the valve C_v at each iteration, can be computed iteratively by replacing the F_p value in each iteration of the C_v equation. Applying the Sizing method, to the four points shown in Figure 2, the various sizing scenarios are,

- Minimum Speed - Surge Flow [Q_1]
- Minimum Speed - Surge Flow [$Q_1 \cdot 1.8$]
- Minimum Speed - Surge Flow [$Q_1 \cdot 2.2$]
- Maximum Speed - Surge Flow [Q_2]
- Maximum Speed - Surge Flow [$Q_2 \cdot 1.8$]
- Maximum Speed - Surge Flow [$Q_2 \cdot 2.2$]
- Minimum Speed - Choke Flow [Q_3]
- Maximum Speed - Choke Flow [Q_4]

The ASV C_v computed for the surge points would be closer to each other in most cases. Similarly, the ASV C_v at the choke points would also be closer to each other. Therefore, to arrive at conservative results, the higher of the C_v values at the surge points & the lower of the C_v values at the choke points are to be considered to determine a suitable ASV size.

Case Study

68.1 MMscfd of hydrocarbon gas at 11.61 bara [suction flange conditions] and 47.50C is to be compressed to 30.13 bara pressure [discharge flange conditions]. The compressed gas is cooled to 50C via an air cooler. The gas composition is as follows,

Parameter	Mol %
Methane [CH ₄]	94.09
Ethane [C ₂ H ₆]	0.03
Propane [C ₃ H ₈]	0.02
Nitrogen [N ₂]	3.93
Carbon Dioxide [CO ₂]	0.96
Water [H ₂ O]	0.97
Total	100

The compressor performance curves for various operating speeds are as follows,

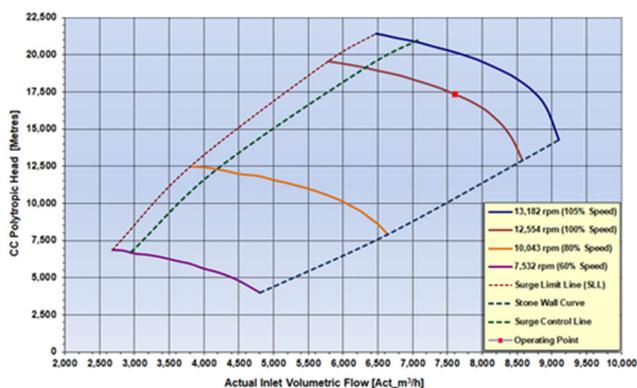


Figure 3. Compressor Performance Curves

The upstream and downstream piping for the anti-surge line is taken as NPS 4", Ref [2] with a thickness of 0.237 inches. The anti-surge valve chosen to be checked is a NPS 4" valve [OD 4.5"] [Single ported, Cage Guided, Globe Style Valve body] with a Cv of 236 and corresponding XT value of 0.69.

The surge control line [SCL] chosen is taken as 10% on the surge flow rate at each speed and the corresponding polytronic head.

The Gas Properties are as follows for the suction and discharge flange conditions,

Table 2. Gas Properties at Flange Conditions

Parameter	Value	Units
Gas MW	16.81	kg/kmol
Suction Pressure	11.61	bara
Suction Temperature	47.5	°C
Discharge Pressure	30.13	bara
Discharge Temperature	143.0	°C
Inlet Z [Z ₁]	0.9810	-
Outlet Z [Z ₂]	0.9848	-
Specific Heat of Gas - Inlet	1.3229	-
Suction Density	7.464	kg/m ³
Discharge Density	14.868	kg/m ³
Actual Volumetric Flow	7,611	Am ³ /h
Inlet Mass Flow	56,809	kg/h

The compressor parameters are as follows,

Table 3. Compressor Parameters

Parameter	Value	Units
Polytronic Head	17,333	m
Polytronic Efficiency	79.71	%
Power Consumed	3,365	kW
Polytronic Head Factor	1.0009	-
Polytronic Exponent	1.3839	-

Performing calculations for all cases, with the approximation that heat losses are negligible, ASV discharge side piping and compressor suction side DP is negligible, the ASV outlet pressure is nearly equal to the compressor inlet pressure, the results are shown below,

Table 4. ASV Sizing Cases – Surge Points

Parameter	Min Surge	Max Surge	Units
Q _s	2,683	6,471	Am ³ /h
r _s	7.46	7.46	kg/m ³
M	20,028	48,295	kg/h
P _D	17.44	36	bara
Discharge DP	0.35	0.35	bar

ASV Inlet P ₁	17.09	35.99	bara
ASV Outlet P ₂	11.61	11.61	bara
ASV DP	5.48	24.38	bar
Cooler Outlet T	323.15	323.15	°K
ASV Inlet Z	0.9732	0.9465	-
ASV Inlet C _p /C _v	1.3348	1.3781	-
X _T	0.69	0.69	-
ASV Outlet F _k	0.9534	0.9843	-
X	0.321	0.677	-
X _{TP}	0.690	0.690	-
Flow Condition	Subcritical	Subcritical	-
C _{v, Min}	113	110	-
C _{v, Min} [Q x 1.8]	203	198	-
C _{v, Max} [Q x 2.2]	248	242	-

Table 5. ASV Sizing Cases – Choke Points

Parameter	Min Choke	Max Choke	Units
Q _s	4,805	9,102	Am ³ /h
r _s	7.46	7.46	kg/m ³
M	35,860	67,932	kg/h
P _D	14.77	25.45	bara
Discharge DP	0.35	0.35	bar
ASV P ₁	14.42	25.10	bara
ASV P ₂	11.61	11.61	bara
ASV DP	2.81	13.49	bar
Cooler T	323.15	323.15	°K
ASV Inlet Z	0.9769	0.9615	-
ASV Inlet C _p /C _v	1.3279	1.3511	-
X _T	0.69	0.69	-
ASV Outlet F _k	0.9485	0.9651	-
X	0.195	0.537	-
X _{TP}	0.690	0.690	-
Flow Condition	Subcritical	Subcritical	-
C _{v, Choke}	286	229	-

From the C_v values calculated, the governing case becomes the Min Speed surge point case.

$$C_{v,min} = 203 \leq C_v = 236 \leq C_v = 248 \quad (39)$$

Hence the selected 4" control valve with a C_v of 236 and X_T of 0.69 is adequately sized to provide anti-surge control.

Transient Study to Verify ASV Sizing

With the ASV size selected, a transient study is performed to check for ASV adequacy. Centrifugal compressors during shutdown experience surging and the ASV must be able to provide sufficient cold recycle flow to keep the operating point away from the SLL as the compressor coasts down.

Normal shutdown [NSD] refers to a planned event where the anti-surge valve is opened first by 100%, prior to a compressor trip. An emergency shutdown [ESD] is an unplanned event, where for example, upon loss of driver power, the ASV opens quickly to recycle flow and prevent the operating point from crossing the SLL during coast down. For this tutorial, the ESD case considered is "Driver trip" where the compressor driver experiences a sudden loss of power.

To simulate the transient case, the air cooler and suction scrubber can be sized with preliminary estimates to cater to maximum speed choke flow case.

Suction Scrubber Volume

Using GPSA K-Value method for suction scrubber sizing, Ref [3], for a flow rate of 67,932 kg/h and 11.61 bara operating pressure, the H × D size is 6.9m × 2.3m with an ellipsoidal head and inside dish depth of 0.25m. The total scrubber volume is 30.1 m³.

Air Cooler Volume

Similarly, the air cooler is sized for maximum speed choke flow case, Ref [4], for a flow rate of 67,932 kg/h & duty of 4,351 kW. The overall heat transfer coefficient [U] is assumed to be 25 W/m².K. The inlet

temperature is 1420C which is cooled to 500C with an air side temperature of 350C. The air cooler geometry is a single tube pass with 3 tube rows & each tube is 9.144m in length. The fan & motor efficiencies are taken as 75% and 95% respectively. With this data, the air cooler has a tube OD of 1" [0.0254m] and total 307 tubes [Tube volume of 1.423 m3].

Compressor Coast down

Coast down time is influenced by a number of factors including fluid resistance, dynamic imbalance, misalignment between shafts, leakage and improper lubrication, skewed bearings, radial or axial rubbing, temperature effects, transfer of system stresses, resonance effect to name a few and therefore in reality, shutdown times can be lower than estimated by the method shown below.

The decay rate of driver speed is governed by the inertia of the system consisting of the compressor, coupling, gearbox and driver, which are counteracted by the torque transferred to the fluid. Neglecting the mechanical losses, the decay rate can be estimated as,

$$N[t] = \frac{1}{\frac{1}{N_0} + \frac{216.000 \times k \times [t - t_0]}{[2\pi]^2 \times J}} \quad (42)$$

Where, 'N₀' is the speed before ESD, 'J' is the total system inertia, 'k' is fan law constant and 't₀' is time at which the ESD is initiated. The total system inertia is taken as 108 kg.m².

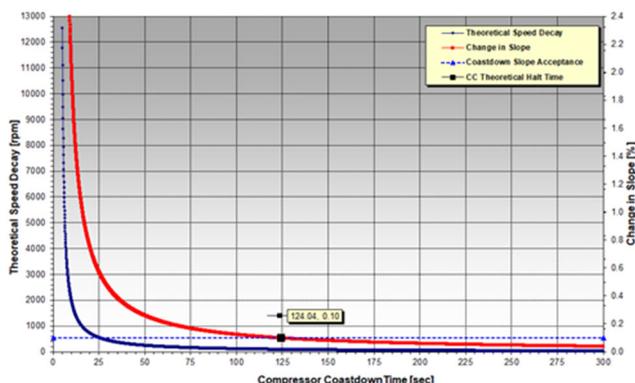


Figure 4. Compressor Coast down Time

From the curve, the compressor is expected to reach a standstill in ~124 sec.

ESD and NSD Analysis

With the equipment volumes, ASV Cv chosen and compressor speed decay rate imposed, an ESD and

NSD analyses is performed to track operating point during coast down.

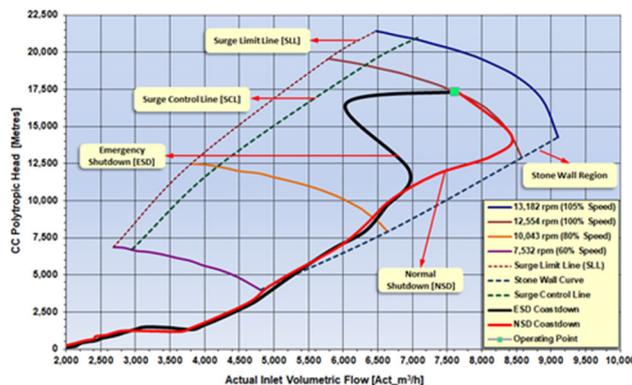


Figure 5. ESD/NSD Operating Point Migration

From the analysis 4" [Cv 236] ASV is sufficient to prevent surge during ESD and NSD.

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About the Author



Vijay Sarathy holds a Master’s Degree in Chemical Engineering from Birla Institute of Technology & Science (BITS), Pilani, India and is a Chartered Engineer from the Institution of Chemical Engineers, UK. His expertise over 14 years of professional experience covers Front End Engineering, Process Dynamic Simulation and Subsea/Onshore pipeline flow assurance in the Oil and Gas industry. Vijay has worked as an Upstream Process Engineer with major conglomerates of General

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Impact of Suction, Evaporating Temperatures & Pressures on Refrigerant Compressor Capacity

Reuben Abraham

A Crucial Parameter to Consider

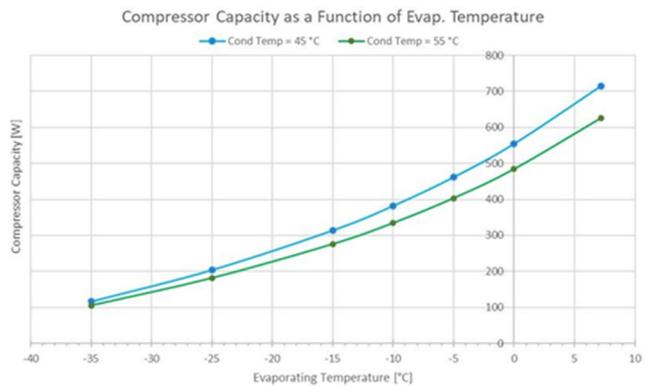
One of the most important aspects of designing a refrigeration system is the selection of an optimum evaporating temperature for the application at hand. This criterion has a domino effect on other parameters such as suction & discharge pressures, current consumption, refrigeration capacity & mass flow rate. Although each parameter is crucial to the system in its own way, this article delves into how the capacity of the compressor is affected by variations in the evaporating & suction temperatures, which in turn is governed by the refrigerant's pressure at saturation.

A Box of Secrets

Datasheets from compressor manufacturers reveal an extensive amount of hidden information if looked upon with a curious eye. One such example is of a German compressor manufacturer, SECOP, whose specification sheets personifies this point. An excerpt from one such sheet for an R290 reciprocating compressor [table below] highlights the capacity variation at two separate condensing temperatures. Table A represents the capacities at a condensing temperature of 45°C & table B represents those at 55°C. The vapour specific volume column was not part of the specification sheet and was added to better explain the concepts below.

Off the bat, the most striking variation that one would notice is the evaporating temperature as a

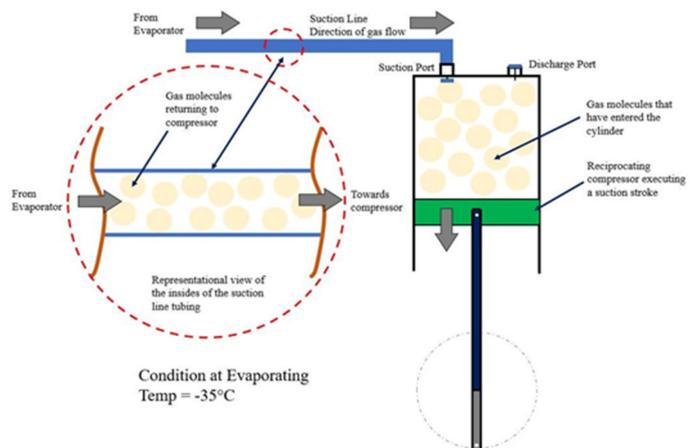
A	Evap	Cap	COP	P in	I	m	Vapour Specific Vol at Sat. Press
	°C	W	W/W	W	Amp	kg/hr	m ³ /kg
Cond Temp = 45°C Return Gas = 32°C	-35	116	1.03	112.8	0.8	1.3	0.3125
	-25	203.7	1.51	135.2	0.83	2.3	0.2160
	-15	313.9	1.98	158.3	0.93	3.57	0.1538
	-10	382.2	2.26	169.2	0.99	4.37	0.1311
	-5	461.8	2.58	179.1	1.04	5.32	0.1122
	0	554.6	2.95	187.7	1.09	6.43	0.0966
	7.2	715.3	3.63	197	1.11	8.4	0.0789
	B	Evap	Cap	COP	P in	I	m
Cond Temp = 55°C Return Gas = 32°C	-35	104.9	0.92	114	0.85	1.3	0.3125
	-25	181.3	1.28	141.9	0.9	2.26	0.2160
	-15	275.7	1.6	171.9	1.02	3.47	0.1538
	-10	334.3	1.79	186.8	1.09	4.23	0.1311
	-5	403.2	2.01	201	1.16	5.14	0.1122
	0	484.1	2.26	214.3	1.22	6.22	0.0966
	7.2	625.8	2.71	230.8	1.26	8.15	0.0789



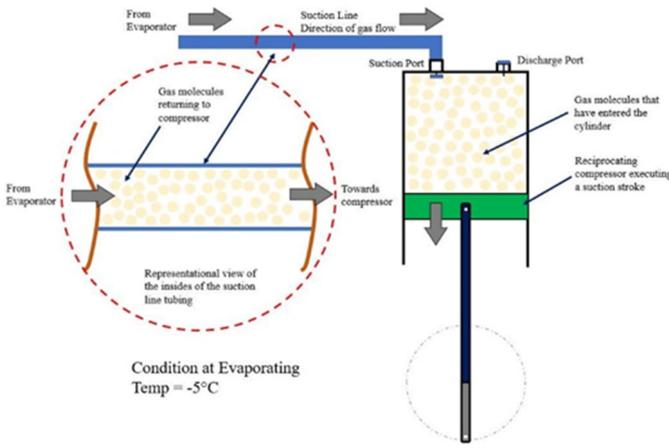
Reiterating the analyses of the graph, the compressor's capacity increases with an increase in evaporating temperature. We will now try to uncover the reason for this trend.

Simplified Representations for the Win

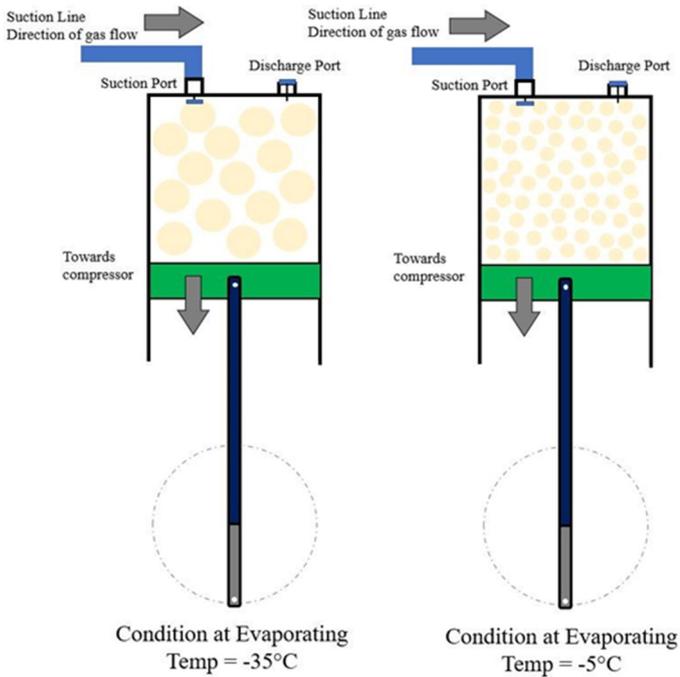
A higher evaporating temperature would mean that the system is functioning at a higher evaporating pressure. Since the entire evaporator & suction line is exposed to this same pressure (assuming no drop), its effect has a profound impact on the molecules of gas in the suction line. The diagram below is a simplified representation of reciprocating compressor arrangement executing a suction stroke at an evaporating temperature of -35°C at its respective saturation pressure [for simplicity, let us assume that only saturated vapour is entering the compressor with no superheat].



Let us look at the same arrangement when the system is functioning at a higher evaporating temperature (-5°C) & higher evaporation pressure (saturation pressure of R290 at -5°C).



The space occupied by the R290 gas molecules at the -35°C configuration is greater than that occupied by the molecules at -5°C [at their corresponding saturation pressures]. The higher overall pressure in the evaporator & suction line causes the molecules to arrange themselves in a closely packed condition with each other due to the compressible nature of gas molecules in general. Let us isolate the compressor representations of the 2 configurations for comparison



Graphing Things Make Them Simpler to Understand

A look at the saturation properties of the refrigerant under consideration (R290) paints a clearer picture of this scenario.

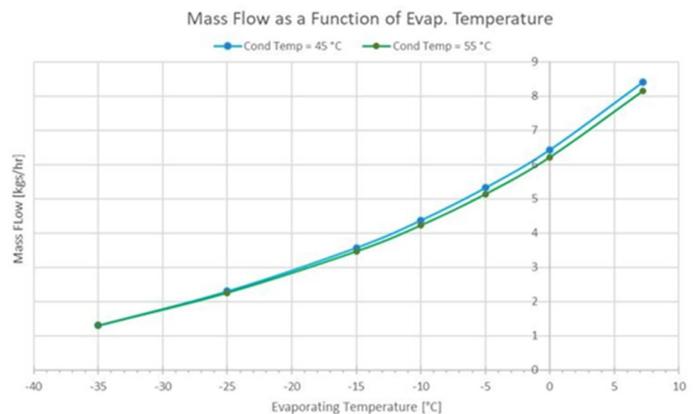
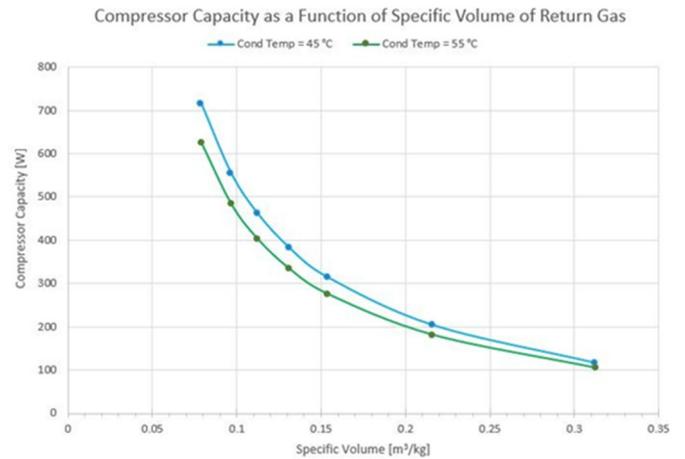
Saturation Properties of R290

Temperature	Pressure	Vapor Density	Vapor Volume	Vapor Internal Energy	Vapor Enthalpy	Vapor Entropy
°C	kPa	kg/m ³	m ³ /kg	kJ/kg	kJ/kg	kJ/kg-K
-35	137.2	3.2	0.3121	491.62	534.45	2.4306
-30	167.8	3.87	0.2586	496.98	540.38	2.4192
-25	203.4	4.63	0.216	502.35	546.28	2.409
-20	244.5	5.5	0.1817	507.71	552.13	2.3999
-15	291.6	6.5	0.1538	513.07	557.93	2.3918
-10	345.3	7.63	0.131	518.41	563.65	2.3846
-5	406	8.91	0.1122	523.73	569.3	2.3781

Source: GenePro – Genetron Properties Software from Honeywell

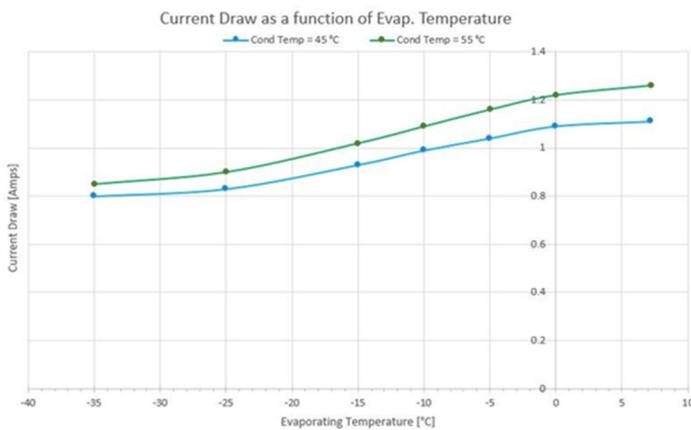
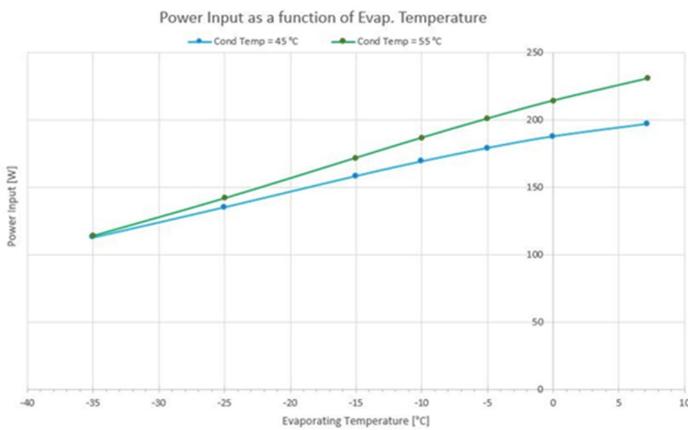
At saturation, the specific vapour volume of gaseous R290 drops from 0.3121 m³/kg to 0.1122 m³/kg when its saturation temperature is raised from -35°C to -5°C corresponding to their respective saturation pressures.

The lower specific volume of the molecules at the -5°C condition causes a very dense refrigerant vapour to enter the compressor in its suction stroke. Hence more mass of refrigerant flows into the compressor during this stroke than during the -35°C case. This is a direct implication that the mass flow rate handled by the compressor at -5°C is much greater than that at -35°C, at the same condensing pressure/temperature. This is easy to visualize with the help of a graph [evident from the spec sheet].



Observe that the mass flow handled by the compressor is higher for the 45°C condensing temperature configuration. This is because the compressor needs to compress the refrigerant to a lower pressure compared to the 55°C configuration before returning for the suction stroke. Thereby executing the suction & discharge strokes faster and hence handling more refrigerant in the process.

The greater mass of molecules handled by the refrigerant in the suction stroke [for the -5°C configuration] signifies that as the compression stroke begins, the compressor will need to do more work compared to the -35°C configuration. This is characterized by the increase in input power of the compressor & the amp draw of the compressor with an increase in evaporating temperature.



Suction gas behaviour at Super-heated Conditions

Beyond saturation i.e. in the superheated state, a rise in suction temperature is associated with an increase in specific volume at a constant pressure. This is because the gas expands as it absorbs heat while the pressure in the suction line is maintained constant. This means that the capacity of the compressor decreases with higher superheat, but its current consumption lowers due to the lower mass of vapor entering the suction stroke. Although superheat

Although superheat is an absolute necessity to ensure only vapour enters the compressor, it has negative implications on refrigeration capacity as a side effect. This variation in the specific volume of the refrigerant at constant pressure, as a comparison to that at varying pressures, needs to be understood to get a thorough grasp of what is happening inside the system. This trend is clear upon close analysis of the super-heated properties of the refrigerant.

Superheated Table (PC Model), Propane (C3H8)									
K	p = 50 kPa (T _{sat} = 216.22 K)			p = 100 kPa (T _{sat} = 230.77 K)			p = 200 kPa (T _{sat} = 247.72 K)		
	m ³ /kg	kJ/kg	kJ/kg-K	m ³ /kg	kJ/kg	kJ/kg-K	m ³ /kg	kJ/kg	kJ/kg-K
T	v	h	s	v	h	s	v	h	s
Sat	0.79617	208.31	5.7715	0.41851	225.72	5.7227	0.21917	245.83	5.6826
240	0.89026	241.87	5.9187	0.43749	239.29	5.7804	-	-	-
260	0.96824	271.64	6.0378	0.47787	269.57	5.9016	0.23242	265.25	5.7591
280	1.04557	302.97	6.1539	0.51755	301.27	6.0190	0.25338	297.76	5.8795
300	1.12246	335.96	6.2677	0.55677	334.53	6.1337	0.27382	331.61	5.9962
320	1.19904	370.68	6.3797	0.59566	369.45	6.2463	0.29391	366.96	6.1103
340	1.27540	407.16	6.4902	0.63433	406.09	6.3574	0.31375	403.93	6.2223
360	1.35160	445.41	6.5995	0.67283	444.47	6.4670	0.33341	442.57	6.3327
380	1.42767	485.45	6.7077	0.71120	484.61	6.5755	0.35294	482.91	6.4418
400	1.50365	527.25	6.8149	0.74947	526.49	6.6829	0.37237	524.96	6.5496
420	1.57956	570.79	6.9211	0.78767	570.10	6.7893	0.39172	568.72	6.6563
440	1.65540	616.06	7.0264	0.82581	615.43	6.8947	0.41101	614.16	6.7620
460	1.73119	663.01	7.1307	0.86389	662.43	6.9991	0.43024	661.27	6.8667
480	1.80694	711.61	7.2341	0.90194	711.08	7.1027	0.44943	710.00	6.9704
500	1.88266	761.83	7.3366	0.93995	761.33	7.2052	0.46859	760.34	7.0731
520	1.95835	813.62	7.4382	0.97793	813.16	7.3068	0.48772	812.23	7.1749
540	2.03402	866.95	7.5388	1.01589	866.52	7.4075	0.50683	865.65	7.2757
560	2.10966	921.77	7.6385	1.05382	921.37	7.5073	0.52591	920.56	7.3755
580	2.18529	978.05	7.7372	1.09174	977.67	7.6060	0.54497	976.91	7.4744
600	2.26090	1035.74	7.8350	1.12964	1035.38	7.7039	0.56402	1034.67	7.5722

A Balancing Act

It is clear, that working at elevated evaporating temperatures & pressures could improve compressor capacity. The evaporating temperature is usually selected by maintaining a dT of about 10° C to 12°C from the target temperature that we need to maintain in the refrigerated space. However, raising the evaporating temperature significantly, in the name of increasing its compressor capacity does not always correlate to improved performance in practice.

If the refrigerant evaporates at a significantly higher temperature than the designed evaporating temperature, the cooling needs of the room may not be met. This is due to the lower delta T between the room & the refrigerant. So, there is a fine balancing act that needs to be made to ensure the best results.

In the case of expansion valves sensing superheat values from downstream of the evaporator, the suction temperature is usually maintained constant. But in certain cases where the load rises greater than designed limits, there will be a rise in suction temperature that correlates to higher discharge temperatures when functioning at elevated evaporating temperatures.

For capillary systems, the effect is more critical. As the higher evaporating temperature

proportionally raises the suction temperature due to a lack of regulating valves in between like the TXV or EEV.

In the case of capillary systems, raising the evaporating temperature & pressure, raises the suction pressures & a proportional rise in discharge pressures are noticed. This may knock the pressures outside the envelope of the compressor.

An interesting trend that I have noticed in my experience with capillary systems is that if the suction pressure is raised slightly from its designed point (by reducing the capillary length), it is possible to achieve a so-called "sweet spot" that helps achieve the initial pull down time faster than the usual configuration. But any further increase may put the system into imbalance. It is a fine range of operation that needs to be discovered.

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About the Author



Reuben Abraham has been at the helm of research, development & design of heating/cooling systems for over 7 years. From designing refrigeration plants, the size of football fields to systems that sit on a countertop, Reuben's experience stretches across industrial, commercial & domestic applications with projects executed for global markets. Although formally trained with a degree in Mechanical engineering, he leverages his curiosity for electronics & coding to develop innovative solutions to pertinent problems.



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Deep Conversion of Bottom Barrel Streams – Residue Fluid Catalytic Cracking (RFCC) Technologies

Marcio Wagner

Introduction

The downstream industry faces a transition period where the focus of the players is changing from transportation fuels to petrochemicals aiming to ensure maximum added value to processed crude oils as well as to allow the growth of low carbon energies in the global energetic matrix.

The growing market of petrochemicals have been lead some refiners to look for a closer integration between refining and petrochemicals assets aiming to reach more adherence with the market demand, improve revenues, and reduce operation costs. In this business environment, flexible refining technologies like Fluid Catalytic Cracking (FCC) reach highlighted position in the strategy of the refiners to reach competitiveness in the market. Recent technology developments like additive manufacturing (3 D printing) can deeply change the transportation fuels demand as well as the growing practice of home office like demonstrated in the current COVID 19 crisis, facing this scenario, the look for alternatives to transportation fuels can be transformed in a survival question to refiners in middle term and the petrochemicals can offer an interesting alternative.

Taking into account the current scenario and the forecasts, is expected a great contribution of FCC units to the economic sustainability of downstream industry, mainly related to the maximization of petrochemicals from bottom barrel streams.

Residue Fluid Catalytic Cracking Technologies

Residue upgrading technologies aim to raise the H/C ratio in the bottom barrel streams through hydrogen addition or carbon rejection. Among the technologies that apply the carbon rejection principle, the Fluid Catalytic Cracking (FCC) is one of the most widely employed and which add more value to the refiners. Figure 1 presents a simplified process scheme of a conventional fluid catalytic cracking process.

The typical feed streams to fluid catalytic cracking units are gas oils from vacuum distillation. However, some variations are found as the use of heavy coker naphtha, coker gas oils and deasphalted oil from solvent deasphalting units, according to the adopted refining scheme.

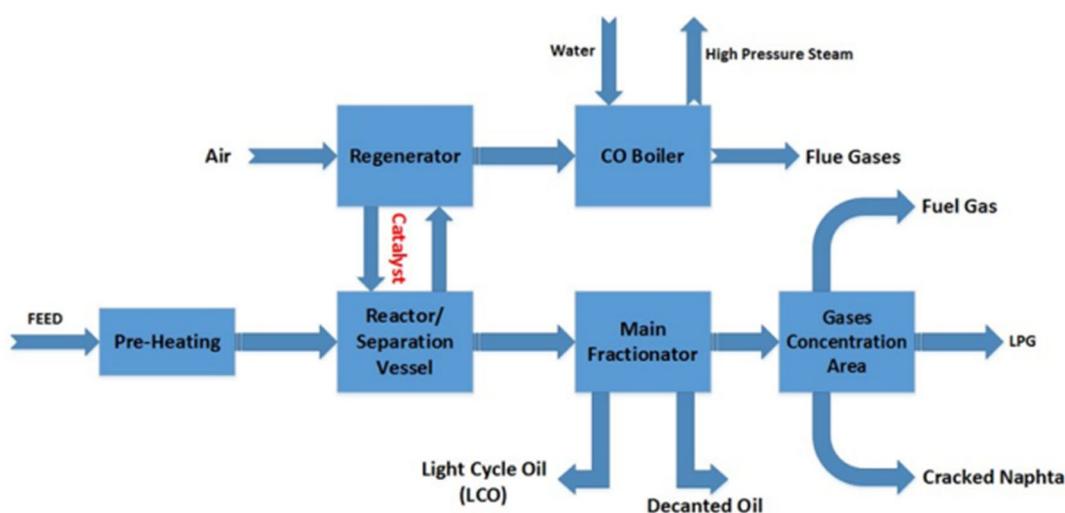


Figure 1 – Schematic Process Flow Diagram for a Typical Conventional Fluid Catalytic Cracking Unit (FCC)

One variation of the fluid catalytic cracking that has been widely applied in the last years is the Residue Fluid Catalytic Cracking (RFCC). In this case, the feed stream to the process is basically the bottom stream from the atmospheric distillation column, called atmospheric residue, that have high carbon residue and higher contaminants content like metals, nitrogen, and sulfur.

Due to the feed stream characteristics, the residue catalytic cracking units require design and optimization changes. The higher levels of residual carbon in the feed stream lead to higher temperatures in the catalyst regeneration step and a lower catalyst circulation rate to keep the reactor in constant temperature, this fact reduces the catalyst/oil ratio that leads to a lower conversion and selectivity. To avoid these effects, the RFCC units normally rely on catalyst coolers, as presented in Figure 2.

Installation of catalyst cooler system raises the process unit profitability through the total conversion enhancement and selectivity to noblest products as propylene and naphtha against gases and coke production, furthermore, helps the refinery thermal balance, once produces high-pressure steam. The use of catalyst cooler is also necessary when the unit is designed to operate under total combustion mode, in this case, the heat release rate is higher due to the total burn of carbon to CO₂, as presented below.



In this case, the temperature of the regeneration vessel can reach values close to 760 oC, leading to higher risks of catalyst damage which is minimized through catalyst cooler installation. The option by the total combustion mode needs to consider the refinery thermal balance, once, in this case, will not the possibility to produce steam in the CO boiler, furthermore, the higher temperatures in the regenerator requires materials with noblest metallurgy, this raises significantly the installation costs of these units.

As pointed earlier, the feed streams characteristics to RFCC units require modifications when compared with the conventional fluid catalytic cracking. The presence of higher content of nitrogen compounds leads to an accelerated process of catalyst deactivation through acid sites neutralization, the presence of metals like nickel, sodium, and vanadium raise the coke deposition on the catalyst and lead to a higher production of hydrogen and gases, besides that, reduces the catalyst lifecycle through the zeolitic matrix degradation. Beyond these factors, heavier feed streams normally have high aromatics content that are refractory to the cracking reactions, leading to a higher coke deposition rate and lower conversion.

Due to this operation conditions, the residue fluid catalytic cracking units presents higher catalyst consumption when compared with the conventional process, this fact raises considerably the operational costs of the RFCC units. However, the most modern units have applied specific catalysts to process residual feed streams, in this case,

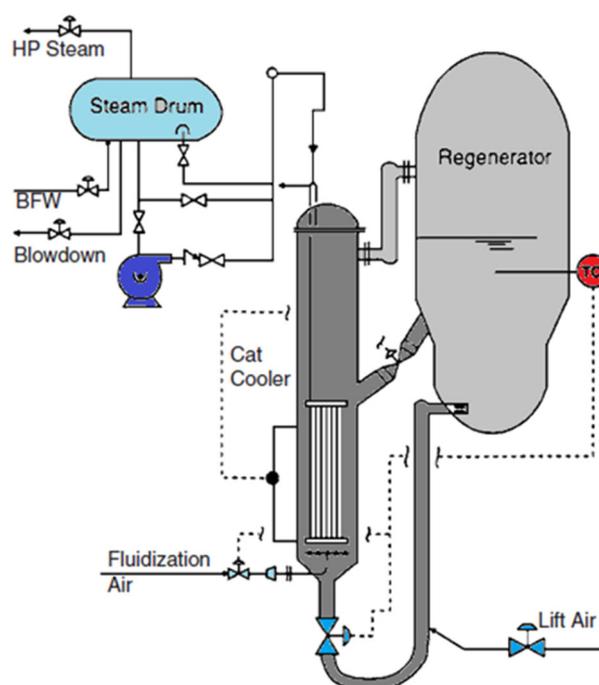


Figure 2 – Catalyst Cooler Process Arrangement for a Typical RFCC Unit
(Handbook of Petroleum Refining Processes, 2004)

the catalyst has a higher porosity aiming to allow a better adaptation to the high aromatics content, furthermore, the catalyst needs to have a higher metals tolerance.

The control of contaminants content in the feed stream or his effects is a fundamental step to the residue fluid catalytic cracking process. Sodium content can be minimized through an adequate crude oil desalting process and the effects of nickel (dehydrogenation reactions) can be reduced by dosage of antimony compounds that act like neutralizing agent of the nickel dehydrogenation activity, reducing the generation of low added value gases, in its turn, the vanadium effects can be controlled through the addition of rare earth to the catalyst, like cerium compounds. The addition of these compounds needs to be deeply studied once raises significantly the catalyst cost.

The use of visbreaking units to treat the feed streams to RFCC units is a process scheme adopted by some refiners, in these cases, the most significant effect in the reduction in the residual carbon, however, due to his higher effectiveness, the tendency in the last decades is to treat the bottom barrels streams in deep hydrotreating or hydrocracking units before to pump for RFCC units, with this processing scheme it's possible to achieve lower contaminants content, mainly metals, leading to a higher catalyst lifecycle. Furthermore, the hydroprocessing has the advantage of the reduction of the sulfur content in the unit intermediate streams, minimizing the necessity or severity of posterior treatments, a clear disadvantage of this refining scheme is the high hydrogen consumption that raises significantly the operational costs.

Like to the conventional FCC units, the main operational variables to RFCC units are the reaction temperature, normally considered in the highest point in the reactor (also called riser), feed stream temperature, feed stream quality, feed stream flow rate and catalyst quality. It's relevant to quote that the conventional FCC units can process atmospheric residue as the feed stream, however, it's necessary to control the contaminants content, mainly metals, which requires processing lighter crudes with higher costs that raise the operational costs and reduces the flexibility of the refiner in relation of the crude oil supplier.

The other steps of the residue fluid catalytic cracking (RFCC) are essentially the same to the conventional fluid catalytic cracking (FCC), Figure 3 shows a process scheme for a typical gases recovery section of fluid catalytic cracking units.

Some of the most relevant residue fluid catalytic cracking technologies available commercially are the R2R™ by Axens Company, the INDMAX™ process licensed by McDermott company and the RxPro™ process developed by the UOP Company.

Due to the current tendency of reduction in the transportation fuel consumption, especially by the developed countries, the fluid catalytic cracking units (FCC) and residue catalytic cracking (RFCC) have been optimized to maximize the yields of petrochemical intermediates, mainly propylene, against the transportation fuels intermediates (naphtha and LCO). Figure 4 shows a block diagram to the PetroFCC™ technology developed by UOP Company aiming to maximize the

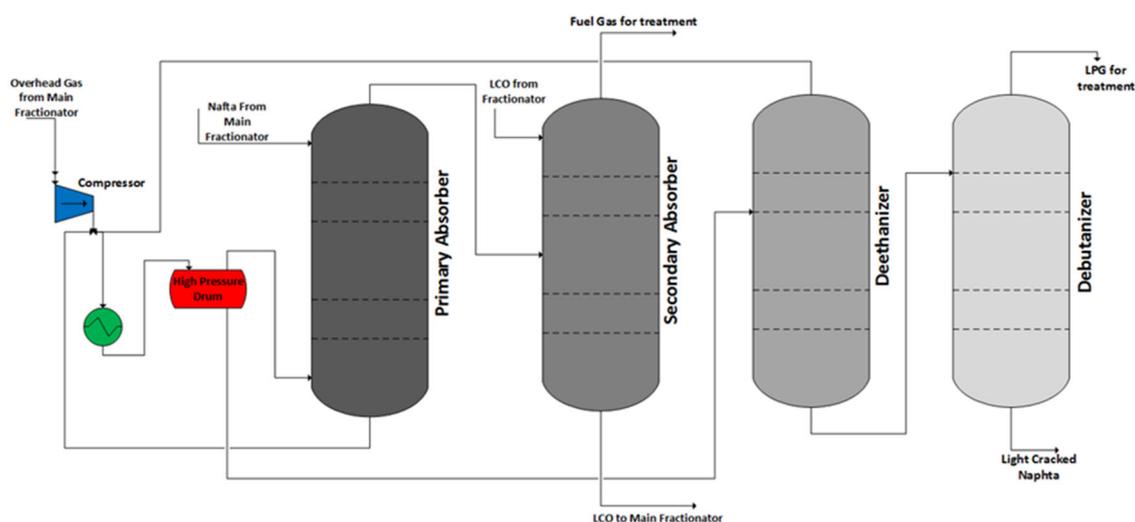


Figure 3 – Process Flow Diagram for a Typical Gases Separation Section of Fluid Catalytic Cracking Units (FCC/RFCC)

petrochemical intermediates production, other available technologies are the HS-FCC™ process developed by Axens Company and the Maxofin™ technology by KBR Company.

The fluid catalytic cracking units have a key role in the current scenario of the downstream industry, once allow a closer integration between refining and petrochemical processes in view of the aforementioned tendency of reduction in the transportation fuels demand, making that the petrochemical sector be responsible to sustain the crude oil demand in the next decades.

Conclusion

Beyond the tendency of reduction in transportation fuels demand, the necessity to meet environmental regulations like IMO 2020 requires a strong reduction of contaminants content in residual streams to produce commercial bunker. In the first moment, there is a tendency of the bunker market be partially supplied by diesel or bottom barrel streams with low sulfur content leading to a raising in the diesel prices and a devaluation of the high sulfur content fuel oil. This scenario can pressure the refiners with low bottom barrel conversion capacity to carry out capital investments to improve the production of high added value derivatives. In this sense, the residue fluid catalytic cracking technologies (RFCC) can be attractive alternatives, once allow a better balance between the flexibility in relation of the quality of processed crude oil (heavier and cheaper crude oils), high yields of petrochemical intermediates and production of low contaminants content derivatives which contributes to enhance the refining margin.

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Dr. Marcio Wagner da Silva is Process Engineer and Project Manager focusing on Crude Oil Refining Industry based in São José dos Campos, Brazil. Bachelor in Chemical Engineering from University of Maringa (UEM), Brazil and PhD. in Chemical Engineering from University of Campinas (UNICAMP), Brazil. Has extensive experience in research, design and construction to oil and gas industry including developing and coordinating projects to operational improvements and debottlenecking to bottom barrel units, moreover Dr. Marcio Wagner have MBA in Project Management from Federal University of Rio de Janeiro (UFRJ) and is certified in Business from Getulio Vargas Foundation (FGV).

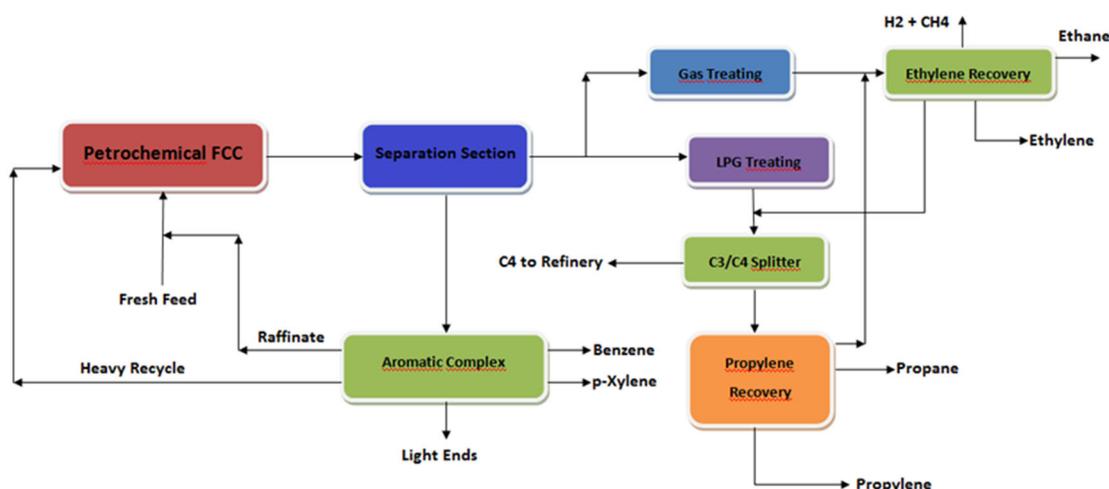


Figure 4 – PetroFCC™ Process Technology by UOP Company.

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Batch Reactor Temperature Control

Joe Bonem

A Pragmatic Theoretical Basis

A previous article entitled “REACTOR PROBLEM SOLVING PART I” dealt with temperature control in a continuous stirred tank reactor. Temperature control in a batch reactor has some similarities, but is radically different in most aspects. The similarities are associated with the fact that the basic heat transfer and mixing relationships apply to batch reactors as well as continuous reactors. In addition, the same guidelines associated with the heat of reaction are valid. That is – essentially all reactions have a heat of reaction and 85 – 90 % of reactions are exothermic. However, there are some major differences such as:

- The most significant heat removal problem normally occurs at startup of the reactor when a reactant is introduced to the reactor. This occurs because the initial reaction is rapid and the reactant being introduced often reacts as it enters the reactor.
- Since unsteady state is involved, the thermal capacity of the reactor must be included. This includes both the coolant in a jacketed reactor and the reactor metal as well as the reactor contents.
- The heat of reaction can sometimes be used to get the reaction to the desired temperature as part of a hold period.

These differences make the estimation of the temperature of a reactor over time more complicated. This complication along with time pressures often lead to viewing these calculations as too complicated. This viewpoint often causes the engineer to

resort to rule of thumb and/or guessing what the reactor temperature response over time will be. This paper illustrates how using a fundamental and simplified approach can overcome this tendency.

Because of the preponderance of exothermic reactors, the thrust of this article is on these reactors. While this illustration is totally fictitious, it mirrors a similar situation that occurred in real life. The basic concept for considering unsteady state operations is equation (1) shown below:

$$AD = I - O - RD \quad (1)$$

Where:

AD = Rate of accumulation. This could be accumulation of anything - level, heat, or reactant.

I = Inflow of material or heat.

O = Outflow of material or heat.

RD = Removal/addition of heat, or formation/destruction of material by reaction.

This relationship can be used as a building block for all unsteady state considerations from something as simple as the change of level in an accumulator to the development of complicated dynamic models. This article emphasizes using this approach for the development of simple, but accurate dynamic models associated with removing heat from a batch reactor.

If it is considered desirable to build a dynamic model of the unsteady state process, all physical

components of the system must be considered. For example, with steady state, the heat capacity of the process vessel is correctly never considered. However, for unsteady state heat balances, the heat capacity of the vessel (walls and contents) must be considered and will often be a moderating influence. This moderating influence will cause the rate of temperature change to be less than calculated if the heat capacity of the vessel is ignored. The dynamic model should be kept simple by use of the perfectly mixed vessel or plug flow assumptions where appropriate, use of lumped parameter constants (such as overall heat transfer coefficients), and an assumption of uniform metal temperature. There are 2 possibilities for the assumption of uniform metal temperatures. They are:

- Ignoring the thermal capacity of the reactor metal. This is generally the most conservative approach.
- Including the thermal capacity of all the reactor metal. This will give a less rapid estimated rise in temperature. However, since it uses all the reactor metal it does not take into account that the reactor is only partially filled.

The problem discussed in this paper gives an example of both cases. It shows how an incorrect assumption of the thermal capacity creates a potential for overheating the reacting mixture.

Batch Reactor Design Example

A batch reaction was carried out successfully in the laboratory using a small reactor cooled in an ice bath. The reaction between component A and component B was determined to be exothermic with a heat of reaction equal to 200 BTU/lb of component B added. It was also determined that

100% of Component B reacted instantaneously as it entered the reactor. Experimental results indicated that in order for the product to have the desired quality the component B should be added quickly. The minimum rate of addition was 0.76 lbs/hr of component B/ lb of component A. It was also determined from laboratory data that at a temperature above 180°F that an undesirable secondary reaction took place.

It was desirable to scaleup using an existing reactor with a water-cooled jacket. It was also desired to produce the product with a batch size of at least 1400 lbs/batch. The existing reactor is a vertical vessel 3 feet in diameter and 5 feet tangent to tangent with a water-cooled jacket on the straight side. Other physical and chemical bases are as follows:

Reaction:	A+B	C	
	A	B	C
Molecular Weight	25	40	70
Density lbs/gal	8	7	7
Molar Density mols/gal	.32	.156	.1
Lbs	524	944	1468
Reactor Wall thickness, inches	0.5		
Dished Head Vol, ft ³	3.5		
Dished Head wt., lbs	200		
Cooling Water Temperature	90°F		
Initial Reactor Temperature	100°F		
Jacket Heat Transfer Coefficient to Liquid	BTU/Ft ² -°F-hr		
	50		

Specific Heat BTU/lb-°F

Contents	0.6
Water	1.0
Metal	0.12

The design question to be considered is will this reactor be adequate. With the physical and chemical parameters shown and the flow rate of component B set at an hourly rate of 0.76lbs/lb of Component A will the temperature limit of 180°F be exceeded? The calculation approach is shown as follows:

If initially, the heat generated is greater than the heat removal capability, then there is a risk of the reactor temperature getting above 180°F. This can be stated mathematically as follows:

$$Q_g > (Q_r)_{\max} \quad (2)$$

Where:

Q_g = The rate of heat generation, BTU/hr.

$(Q_r)_{\max}$ = The maximum rate of heat removal, BTU/hr.

$$Q_g = \Delta HR * R \quad (3)$$

$$(Q_r)_{\max} = U * A * (TR - TC) \quad (4)$$

Where:

ΔHR = The heat of reaction, BTU/lb.

R = Rate of reaction. For the case of an instantaneous reaction, it is the rate of reactant addition, lbs/hr.

U = The heat transfer coefficient, BTU/hr-°F-ft².

A = The heat transfer area, ft².

TR = The temperature of the reactor, °F.

TC = The minimum temperature of the coolant, °F.

Note that equation (4) has been simplified from the traditional heat transfer equation that involves use of a logarithmic relationship. This is valid for this particular case since the coolant is flowing through the reactor jacket at a high rate with minimal increase in temperature. The temperature to be used is the minimum coolant temperature. It is assumed that the control system will react to provide the minimum coolant temperature possible. In addition, because the reactor vessel is well mixed, the reactor temperature throughout the vessel is constant at any point in time.

For a batch reactor equations (3) and (4) can be combined and a heat accumulation term can be added to take into account the unsteady state nature. In addition, any heat accumulation will cause an increase in the temperature of the reactor. The equations below represent this situation.

$$AC = \Delta HR * R - U * A * (TR - TC) \quad (5)$$

$$AC = W * CP * dT/d\theta \quad (6)$$

Where:

AC = The heat accumulation, BTU/hr.

W = The weight of material (metal, water, reactants), lbs.

CP = The average specific heat of material, BTU/lb-°F.

$dT/d\theta$ = The rate of temperature rise, °F/hr.

An examination of equation (6) will allow introduction to a term called "heat capacity". Heat capacity is the weight of a material multiplied by the specific heat or $W * CP$. When the heat accumulation is divided by this term, the rate of temperature rise can be calculated.

Since the reaction of the reactant being pressured into the reactor is instantaneous, the heat generated ($\Delta HR * R$), depends only on the rate of reactant addition. Equation (5) and (6) can be combined to give equation (7) as shown below:

$$W * C_p * \frac{DT}{d\theta} = \Delta HR * R - U * A * (TR - TC) \quad (7)$$

When considering equation (7), it will be recognized that as the reactor temperature increases, the amount of heat that is removed from the reactor also increases. This means that the reactor temperature will increase to a maximum and then begin to decrease unless the coolant rate is reduced. This is expressed mathematically by a derivation from equation (7) as shown in equation (8).

$$\frac{DT}{d\theta} = (\Delta HR * R - U * A * (TR - TC)) / (W * C_p) \quad (8)$$

In equation (8), the total weight of the contents of the reactor and the area covered by the liquid is a function of time of addition of Component B as discussed as follows:

Weight of reactor contents – This will be the initial fill of Component A plus a time dependent amount of Component B.

Weight of Coolant – Since the assumption is that the water rate is so high that the coolant temperature does not change this can be ignored.

Weight of metal – The total weight of the metal can be easily calculated. However, to be theoretically correct, calculations must include the weight of the metal in contact with both the gas and liquid. The heat transfer between the reactor wall and liquid is different than the heat transfer between reactor wall and gas. The split between the weight of the reactor in contact with the gas phase and liquid phase is a time dependent function. The example given

in this paper shows how these two assumptions can be used to develop the boundary conditions for the impact of this variable.

Area (A) – This depends on the amount of straight side of the vessel that is covered with liquid.

The desirable design result is that component B is added at the specific rate (400 lbs/hr in this case) and the maximum temperature will remain below 180°F. As this maximum temperature is reached, the control system will begin to reduce the coolant rate and control the temperature at the desired level. Equation (8) along with a spreadsheet can be used to evaluate whether the reactor will perform as desired.

If it is assumed that the **total** weight of the metal in contact with the reactor liquid and vapor is at the temperature of the liquid, Figure 1 describes the temperature vs time profile. As shown in this figure, the reactor temperature peaks at about 175°F somewhat below the maximum temperature specification of 180°F.

One of the key guidelines in engineering is “Check Your Assumptions”. The critical assumption in this calculation is that all of the metal reached the temperature of the reactor contents. It is unlikely that this will occur since the heat transfer coefficient of vapor in contact with the metal will be very low. There is also a second assumption that can be made. It can be assumed that the total weight of the metal is constant at the temperature of the coolant. In this case, only the content of the liquid in the reactor is included in the reactor heat capacity. Figure 2 illustrates both of these assumptions and is summarized below:

- If the total metal heat capacity is included, the maximum reactor temperature is 175°F.
- If the metal heat capacity is not included, the maximum reactor temperature is 187°F. Since this violates the maximum temperature criteria of 180°F, the reactor would likely not perform as desired.

Since one of the boundary conditions appears to produce acceptable results and one does not, the

most conservative answer is to conclude that the reactor is not acceptable. A review of the maximum temperature or addition rate criteria might allow use of the reactor. For example, decreasing the addition rate from 0.76 to 0.66 would result in a maximum temperature of 179°F even at the most conservative conditions. Thus, this careful study of the reactor design, might indicate that further research was desirable.

Figure 1
Estimated Reactor Temperature
including Metal Heat Capacity
B Component Rate =400 lbs/hr

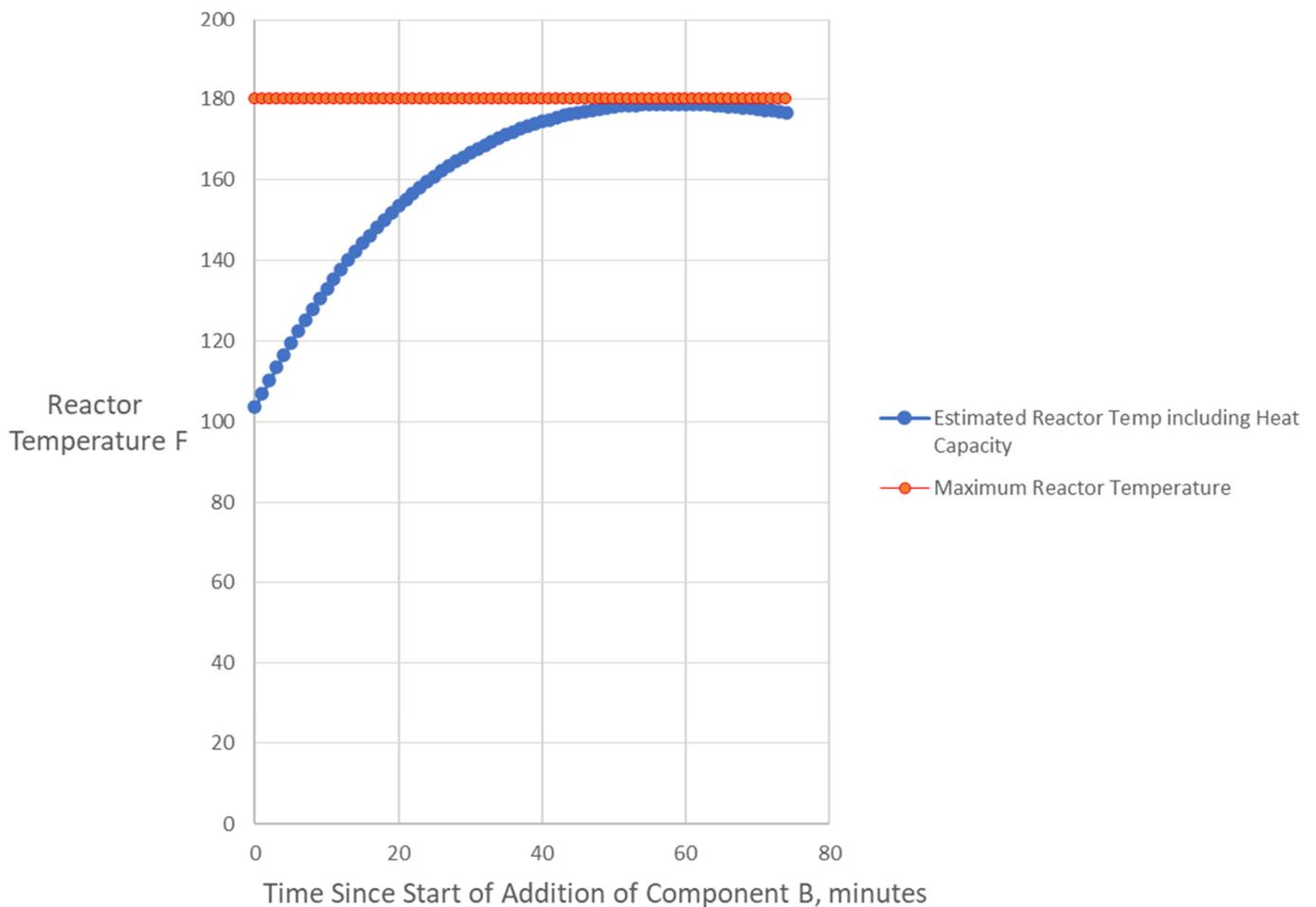
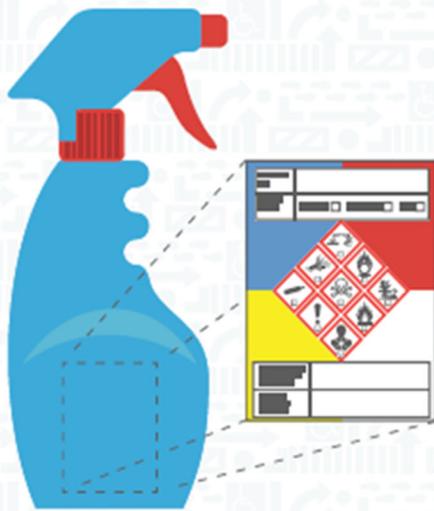
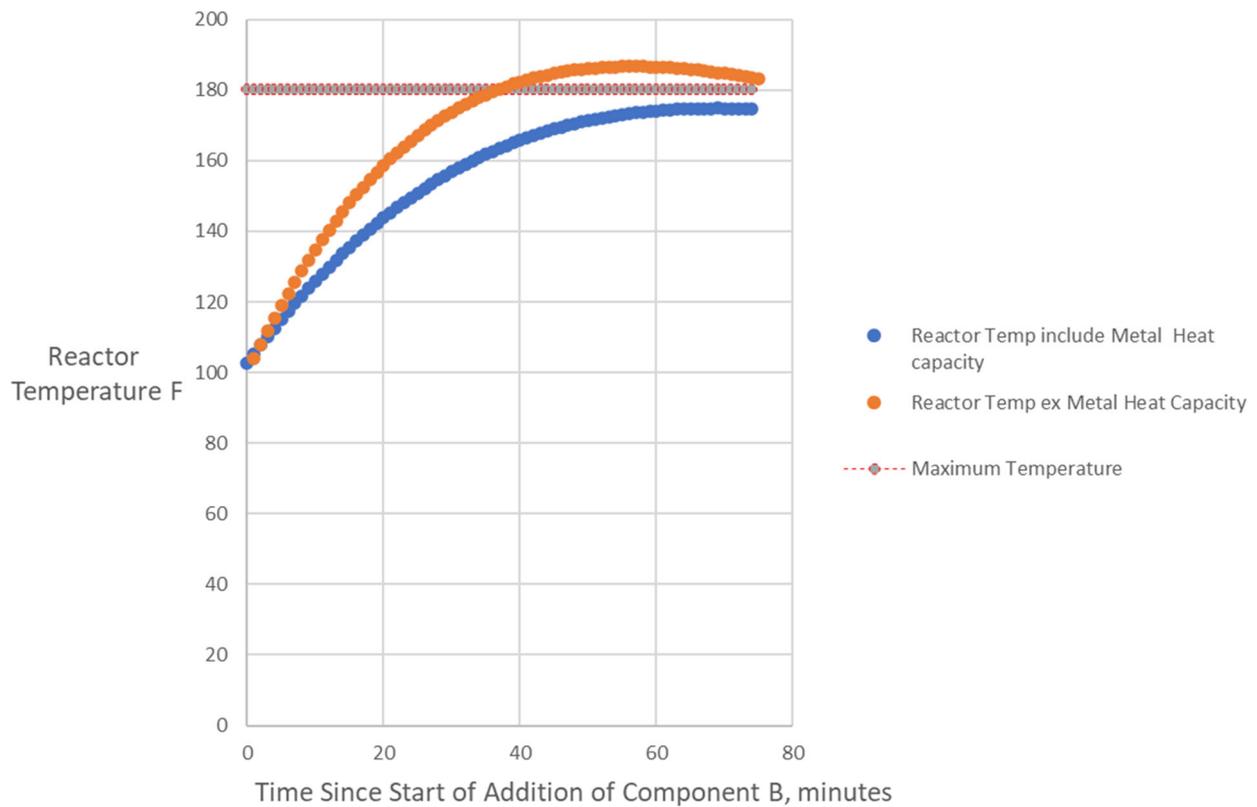


Figure 2
 Estimated Reactor Temperature
 with/without
 including metal heat capacity
 B Component Rate = 400lbs/hr



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How to... FIXED VALVE TRAY | Part 4

How to design and optimize Fixed Valve Trays

Dr.-Ing. Volker Engel

Tower trays and internals are the heart of all distillation columns. Their design is an essential part of a process engineer's task and determines the process reliability and economy.

This article is the 4th part of a series on different kinds of trays and internals.

Fixed Valve trays were first published and patented by Nutter in 1967. After some improvements in scale and development of NC-machinery, they gained technical relevance in the 1990s.

Therefore they are the most recent, technical relevant development in contact elements of trays.

On a distillation tray vapor enters liquid and forms a two-phase regime (bubbling, froth, spray). The tray types differ mainly in the way the vapor enters the liquid.

The combination of "Fixed" and "Valve" is somehow contradicting. In principle the naming describes a "static covered hole". The gas passes the panel openings and is directed by the cover to the vertical outlet area (so-called curtain area). Therefore the gas enters the liquid in horizontal direction. This feature reduces entrainment and increases the capacity of the tray type compared to sieve trays.

Some Fixed Valve types emphasize this feature by doming the Fixed Valve cover (see Fig. 1):



Fig. 1: Domed cover

This shape achieves an outlet vector of the gas flow of less than 90° to the vertical and helps to reduce entrainment.

Another feature is the orientation of the Fixed Valve elements. To fix the cover to the tray panel, there are normally two "legs". Both are oriented in flow direction: one is located on the upstream side of the element, one on the downstream side. At most of the Fixed Valves in the market, the width of the upstream leg is larger than the downstream one. There is a resulting pushing effect of the gas outlet to the liquid flow (see Fig. 2):

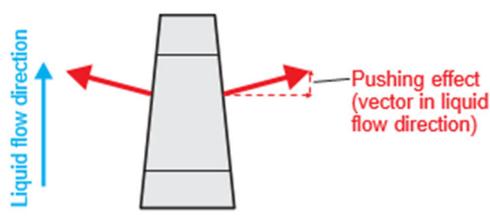


Fig. 2: Pushing effect

These two features increase gas and liquid capacity. Therefore the Fixed Valves are often used for revamps.

To support this pushing effect, there are sometimes additional, special Push Valves on the tray. The gas outlet direction of these Push Valves is only in flow direction of the liquid. The combination of Fixed Valves with push valves and special downcomers (truncated, sprouts, multi-chordal,

sloped, ...) are often called high performance trays. Suppliers have special brand names for these designs (e.g. "SuperFrac", "PlusTray", ...).

The production of Fixed Valves is quite easy: A punching tool cuts and bends the element at a time. The cover is part of the panel material. The lift height can be controlled by the punching machine. Important: The length of the element depends on the lift height (see Fig. 3).

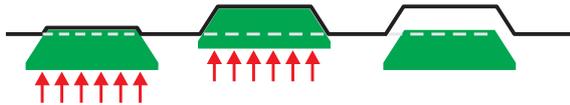


Fig. 3: Punching process of Fixed Valves

The lift of a Fixed Valve is limited by the punching tool as well as by the mechanical properties and material thickness of the panel material. The small scale valves (often called "Mini-valves") with domed covers are normally limited to 2mm material thickness.

The pitch of the Fixed Valves is also limited by the punching tool. The minimum spacing is defined by the individual tool dimensions.

The tray spacing of Fixed Valve trays is normally about 450-500mm. It can be less, but due to inspection and maintenance reasons this is a typical value.

The costs of Fixed Valve trays are close to sieve trays (and significant less than float valve trays).

Most of the suppliers keep their knowhow of Fixed Valve private. Therefore there are only few publications, data and models available.

There are two Fixed Valve classes: The Round Shape (with brand names like VG0, VG10, ...) and the Trapezoid Shape (with brand names like LVG, MVG, MMVG, R-MV, ...).

Within each class there are different sizes and different lift heights. To get an overview, the following list shows the most common types.

The VG0 valve is one of the standard Fixed Valves in the market (see Fig. 4). Its cover is slightly domed and has a diameter of 29mm.

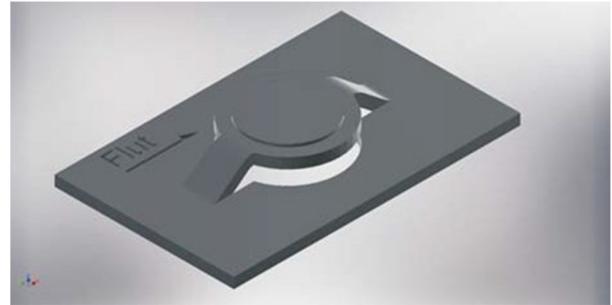


Fig. 4: Round Fixed Valve

For heavy duty applications (e.g. fouling service), there is a large variant (called e.g. VG10, see Fig. 5) with a cover diameter of 35mm. It is often built in 3mm material thickness, therefore the cover has no doming.

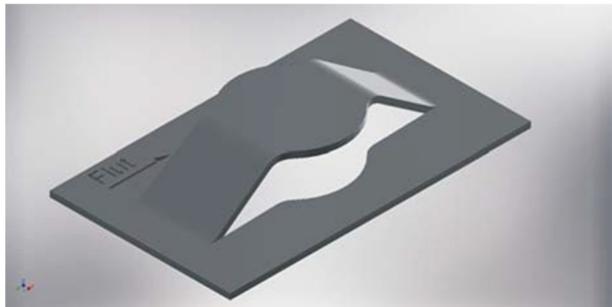


Fig. 5: Large, round Fixed Valve

At the trapezoid shaped Fixed Valves, the so-called MVG (length about 35mm, upstream width 18mm, downstream width 15mm) is the most common type (see Fig. 6).

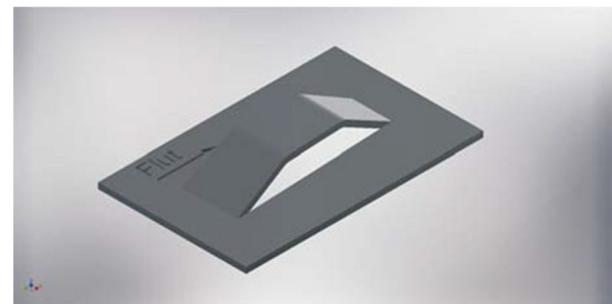


Fig. 6: Trapezoid Fixed Valve

For heavy duty applications there is a trapezoid

of about 44mm (upstream width 32mm, downstream width 25mm).

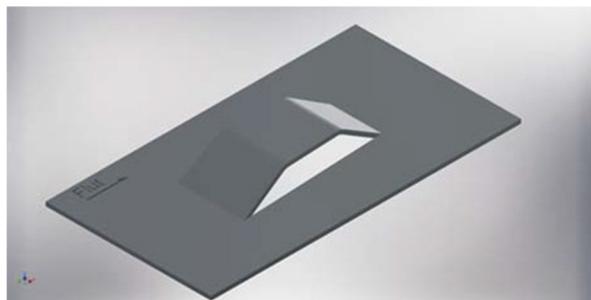


Fig. 7: Large, trapezoid Fixed Valve

In the first patent of Nutter the dimension of the Fixed Valve element was significantly larger than the newer ones (see Fig. 8). It is called LVG and has a length of about 116mm. As it hardly can be manufactured on standard NC-punching machines, it is quite expensive and rare.

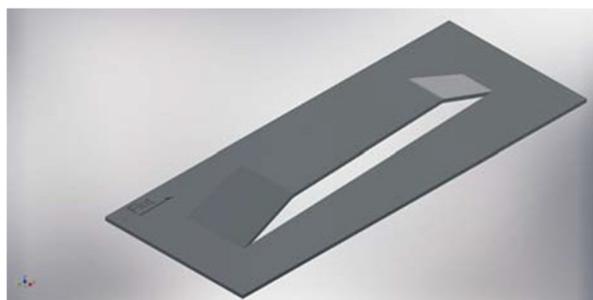


Fig. 8: Long version of trapezoid Fixed Valve

Contrary, there is a small version (brand name e.g. MMVG, see Fig. 9) with a length of about 34mm (upstream width 13mm, downstream width 10mm).

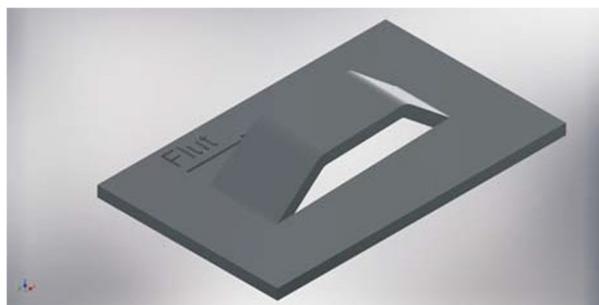


Fig. 9: Small size, trapezoid Fixed Valve

Another mini Fixed Valve is called R-MV and has a domed cover (see Fig. 10).

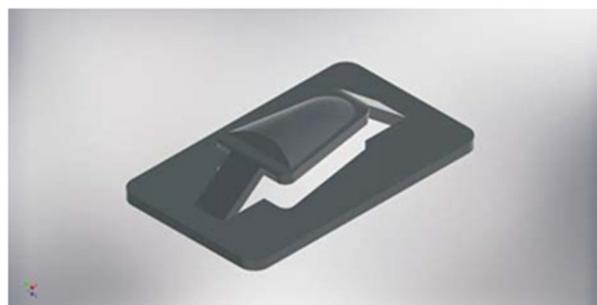


Fig. 10: Mini Fixed Valve R-MV

Another type for heavy duty is the so-called ProValve (see Fig. 11). Its cover is not formed from the panel material, but is mounted as an additional part. The opening in the panel has a diameter of 39mm.

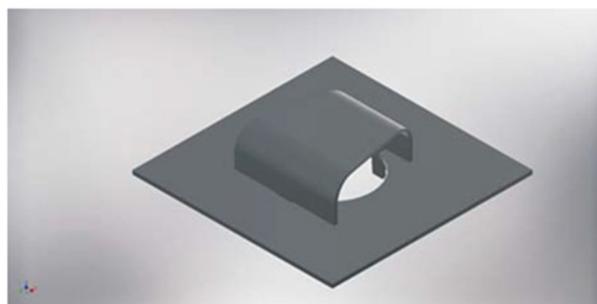


Fig. 11: Fixed Valve with extra cover

The Operation Area of a fixed valve tray is defined by different limits. In Fig. 12, a qualitative operation diagram is shown. Please note, that the position and shape of all curves depend on the physical data, the tray and downcomer geometry and the gas/liquid load. Each curve can be limiting! The first step in analyzing a design is – of course calculating all relevant parameters. For a Fixed Valve tray design there are 9 main parameters shown as curves in Fig. 12. These parameters are discussed in this article. There are some additional effects you will have to look at: entrainment, head loss at downcomer exit (clearance), flow regime, downcomer residence time, efficiency, sealing, construction issues, statics, ...

The Operation diagram of Fixed Valves is similar to this for sieve trays. The limiting curves are almost of the same type. Only the Blowing- curve of sieve trays is normally not part of an operation

diagram of Fixed Valves. The effect of Blowing at sieve trays describes the separation of the two-phase layer from the tray panel and its lift off. This effect has not this relevance for Fixed Valve trays – this is due to the different gas entry direction of a Fixed Valve tray.

The Operation Point (Op in Fig. 12) of the design case (as well as the minimum and maximum load) has to stay inside all limiting curves.

For stable operation and good efficiency there is a useful operation area with narrower limits (e.g. 80%-FFCF and 85%-FFJF curves).

Please note, that all free suppliers' software only show a limited number of these parameters and therefore are not save to use for design, rating and troubleshooting of trays. For safe design you should be able to calculate all parameters! (e.g. software TRAYHEART OF WELCHEM)

In the following sections, all 9 main parameter curves of Fig. 10 are described. Each suggested action for preventing a certain effect may result in fertilizing another. The main task for designing trays is to balance these different and contradicting effects.

System Flood FFSF

1

There is a system limit set by the superficial vapor velocity in the tower. When the vapor velocity exceeds the settling velocity of liquid droplets („Stokes Law Criterion“), vapor lifts and takes much of the liquid with it. A well known model was published by STUPIN AND KISTER 2003.

This flooding effect cannot be reduced by use of other tray types or by increasing tray spacing.

The only way is to enlarge the vapor cross section area (e.g. enlarging tower diameter or reduce downcomer area).

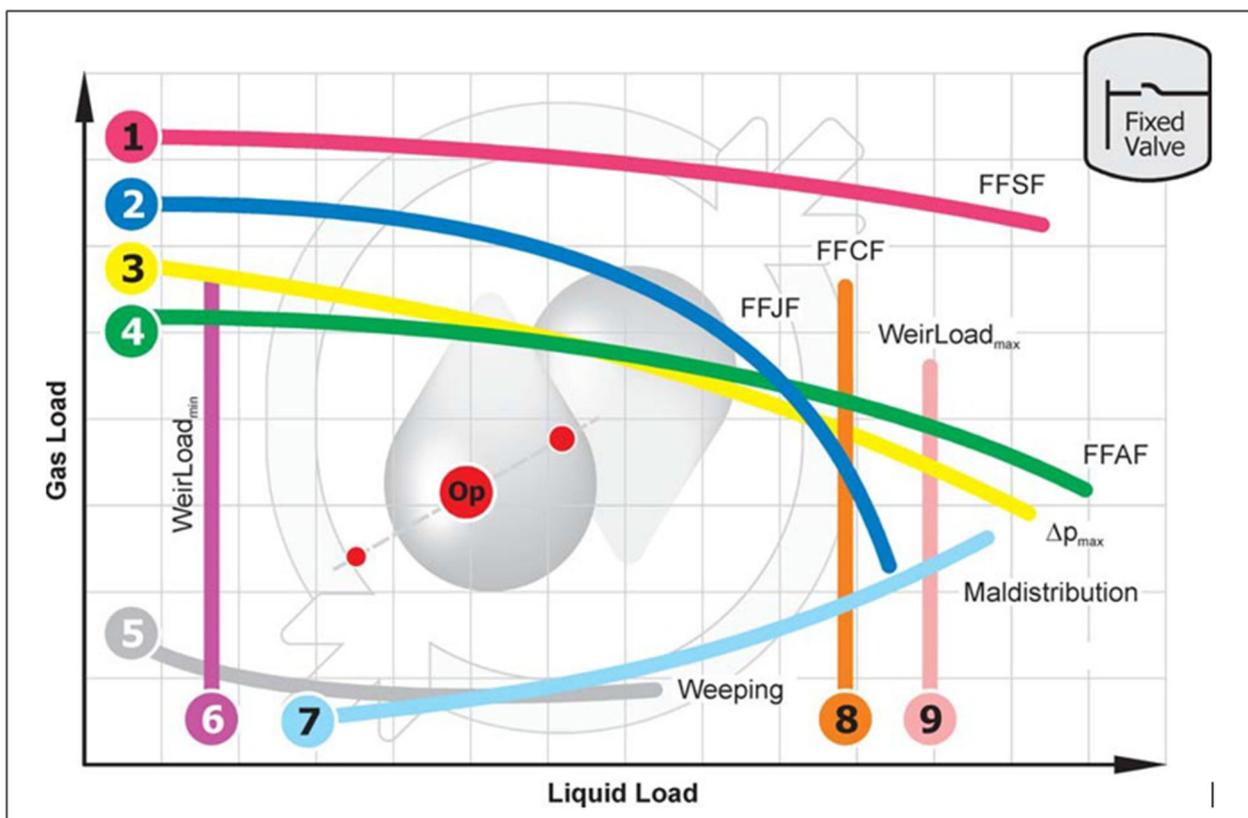


Fig. 12: Qualitative Operation Diagram for Fixed Valve trays

2

Jet Flood FFJF

There are several definitions in literature for the so-called Jet Flood. Similar definitions are Entrainment Flood, Massive Entrainment, Two-Phase Flood or Priming. For practical understanding, Jet Flood describes any liquid carried to the tray above by the gas stream. This leads to a shortcut recycling of the liquid with loss of tray efficiency, additional pressure drop and additional downcomer load. For good tray performance, the Jet Flood value should be less than 75-80%.

You can reduce Jet Flood by

- a. lowering the gas velocity (higher open area, i.e. more valve elements)
- b. enlarging the tray spacing
- c. lowering the froth height on the tray deck (by reducing weir height or weir crest height)
- d. enlarging the active area by sloping the downcomers

3

Pressure Drop

In most design cases there is specified a maximum allowable pressure drop of the tower.

You have to ensure that the pressure drop per tray does not exceed a certain value. This leads to a limiting curve within the operation diagram.

To reduce the pressure drop of a design, you can

- a. lower the gas velocity by enlarging the number of valve elements
- b. use small size Fixed Valves (at same or higher open area)
- c. lower the froth height on the tray deck (by reducing weir height or weir crest height)
- d. enlarge the active area (with place for more

valve units) by reducing the downcomer area or sloping the downcomers

Aerated Downcomer Backup FFAF

4

This limiting effect is also known as Downcomer Backup Flood. It describes the (aerated) backup of the downcomer due to pressure drop effects. It is important to not mix this up with the Choke-Flood-effects (ref. to 8).

The level of the liquid in the downcomer is the result of (i) head loss at the clearance, (ii) the liquid height on the outlet deck, (iii) an inlet weir (if present) and (iv) the pressure drop of the tray itself. All these effects can be expressed by "hot liquid height". This resulting level in the downcomer has to compensate these effects! Taking into account the aeration of the liquid in the downcomer, the level has to be less than tray spacing plus weir height.

To reduce a high Aerated Downcomer Backup value you have to

- a. reduce the pressure drop of the tray (ref. to 3)
- b. reduce the head loss of the clearance (use higher clearance height or radius lips or recessed seal pans in case of insufficient sealing)
- c. avoid inlet weirs

Please note, that it is no option to enlarge the downcomer area to reduce this flooding effect!

5

Weeping

Like on sieve trays, there is an operation limit by weeping. As the gas outlet area ("curtain area") for standard size Fixed Valves is significantly larger (expressed by the hydraulic diameter of the

openings) than those of sieve holes, the weeping limit is similar or worse than that of sieve trays.

To minimize weeping, you can

- reduce the number of elements (to achieve a higher gas velocity)
- use small-size ("mini") Fixed Valve types

6 Minimum Weir Load

The uniform thickness of the two-phase layer is essential for the successful operation of a tray. To achieve this uniform flow, the tray panels have to be in level and the outlet weir has to be installed accurately.

To compensate small tolerances, the weir crest should be higher than 3mm and the weir load more than 9 m³/m/h. In case of low weir loads you will normally have to consider gasketing the tray to avoid any leakage and loss of liquid.

To ensure these minimum values, you can use

- notched weirs
- blocked weirs

Gas Maldistribution

7

In all types of trays the liquid must have a driving force to flow from the inlet to the outlet. As long as there is no gas driven flow, the hydraulic gradient is the main reason for liquid flow.

Because the valve units are obstacles in the liquid flow pass, the hydraulic gradient has to be considered for valve trays, too.

Why might the hydraulic gradient be a problem? At a high hydraulic gradient, the tray will not work properly (see Fig. 13): At the tray inlet the liquid "closes" the valves. The gas will use less liquid affected valves for passage. This leads to a gas maldistribution and a bad efficiency of the tray. Furthermore, if the liquid head of rows with high gradient gets too high, weeping occurs!

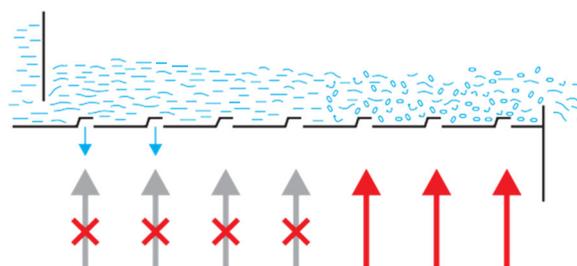


Fig. 13: Gas Maldistribution

To reduce gas maldistribution you have to

- reduce the number of valve rows (e.g. by switching to a design with more flow passes)
- cascade the active area

Choke Flood FFCF

8

The maximum liquid throughput of a downcomer is limited by the liquid velocity and the effect of overload (so-called Choke Flood). The maximum allowable liquid velocity in the downcomer depends on the density ratio of gas to liquid, the tray spacing and the system factor. (The system factor describes the difficulty of phase separation. For common applications it is 1.0.) The most popular downcomer choke flooding calculation was published by GLITSCH 1993.

Another effect of Choke Flood at center and off-center downcomers is initiated by the mutual interference of the two liquid flows into the downcomer.

To prevent downcomer Choke Flood you have to

- enlarge the downcomer area
- implement more flow passes (with in sum an overall higher downcomer area)
- enlarge the tray spacing (if limiting)
- install anti-jump baffles for center / off-center downcomers

The maximum liquid flow handled by a downcomer can also be limited by the weir.

If the weir crest exceeds 37mm or the weir load 120 m³/m/h, the liquid will not enter the downcomer properly.

To prevent overload of the weir, you have to extend the weir length by

- a. larger downcomers with longer weirs (or multichordal downcomers)
- b. more flow passes
- c. swept back weirs at the side downcomers

Conclusion

There are multiple limiting effects that have to be considered at the design and operation of Fixed Valve trays. Fixed valves are comparatively new contact elements. As they can handle higher loads than sieve or float valve trays, they are often used for revamps but also for new towers. Together with special downcomer features and in combination with push valves, a classical Fixed Valve can be enhanced to a high performance tray.

About the Author

Volker Engel studied process engineering at the Technical University of Munich and did his Ph.D. thesis on packed columns with Prof. Johann G. Stichlmair. Since 1998 he has been the managing director of WelChem Process Technology GmbH and head of the TrayHeart software. TrayHeart has developed into a state-of-the-art design tool for trays and internals in process technology.

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Failure of Cat Cracker Catalytic Cracking Unit

Norm Lieberman

In the late 1980's, an explosion at a very large Catalytic Cracking Unit (i.e., the FCU) in Norco, Louisiana, blew the large catalytic cracker fractionator over. It landed on the control room, and killed all six of the operators inside.

What Happened

An overhead carbon steel vapor line from the cracking unit depropanizer 6" line ruptured. A vapor cloud of propane-propylene detonated, and blew the main fractionator (perhaps 16' I.D. x 100' T-T) off of its foundation, and killed six operators in the control room.

A Litany of Errors

The depropanizer overhead vapor consisted of a mixture of:

- A few percent ethane.
- A few percent H₂S.
- A few percent butanes.
- About 40% propane.
- About 50% propylene.
- Saturated with water.

The overhead vapors were at a pressure of roughly 280 psig and 130°F. All of the above is typical for this service. What was not typical was that the water drawn from the reflux drum boot was acidic. Typically, the NH₃ content of the Cracker Depropanizer Reflux Drum boot would be sufficient to result in a pH of 8 – 10. Why was this not the case here (the boot pH, prior to the addition of the neutralizer, was about 5 – 6 pH), is not known.

The operating company had found that the overhead condenser's carbon steel tubes to be subject to high rates of corrosion. Thus, the tubes were changed to alloy. But, not the carbon steel vapor line upstream of the condenser. To control corrosion, a water injection point was installed upstream of the condenser. Water was injected through a "quill". That is a piece of tubing, cut at an angle, and inserted just into the piping, upstream of an elbow (see Figure 1) in the vapor line, feeding the condenser.

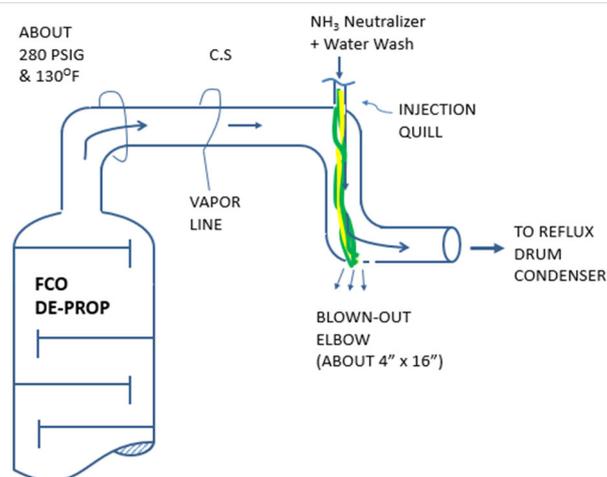


Figure 1—Fluid Catalytic Cracker Unit Vapor Line Failure Due To Quill Location

NH₃ was added to this wash water to control the reflux drum water boot pH at about 7.5 – 8.0. The water was clarified Mississippi River water, without de-aeration. After several years of operation, the elbow blew-out and the catastrophe I described above, followed.

What Went Wrong

I inspected the failed elbow in the investigation lab

and reviewed all of the internal documents in my role as an expert witness for the plaintiffs. Here is a tabulation of the concerns that I documented, in order of importance:

1. A “Bete” or “Spraying Systems” spray or mist nozzle, should have been used. It was obvious from my inspection of the elbow failure pattern, that the outer periphery of the water jet had absorbed much of the acidic components in the vapor. A groove had been eaten through the pipe wall exactly around the outer impingement area of the spray. Inside and outside of this ½” – 1” groove, the pipe was at its original thickness.
2. The “quill” (i.e., the cut-off tubing), was pointing directly down at the failed elbow, which was about 30” below. The quill should have been in a horizontal run of line.
3. If the company felt the need to change the condenser tubes from carbon steel to alloy tubes, why not the carbon steel piping upstream of the condenser as well?
4. Why was the Catalytic Crack Depropanizer Overhead vapors acidic? Typically, towers in this service have a pH of 8 – 10 in the reflux drum water boot? The company should have investigated this many years ago, at the start of operations, when they first observed the low pH problems of the water in the boot.
5. The wash water should have been de-aerated water, not filtered river water from the Mississippi, which contains corrosive dissolved oxygen.
6. The company had been advised – in writing – by their chemical treatment provider – of a potential problem in this service, a year or so before the failure.
7. The company had a piping contractor design and install the quill. It did not receive a review from their local engineering organization, which

I believe would have most likely specified a properly designed dispersion nozzle, and not a quill.

In Retrospect

Whenever I speak to a company engineer at one of my Refinery Troubleshooting Seminars, I always ask them, “Do you know what happened back in 1988, at the Catalytic Cracker Explosion in Norco, Louisiana, when all those operators were killed?” Almost always, they do not. So, I then say, “Those who do not study history, are condemned to repeat it.”

Nitrogen in FCU Feed

Unlike a crude tower, the overhead vapors of FCU fractionators and downstream towers, typically have enough NH₃, due to the higher nitrogen content of the FCU feed, to cause the water in the reflux drum boot to be naturally basic. In crude towers (virgin service), we typically have to add NH₃ or amine to the overhead, to prevent corrosion of the carbon steel piping, due to the HCl in the overhead vapors that originate from MgCl₂ in the desalter effluent.

About the Author



Norm Lieberman is a Process Engineer working as an independent consultant, mainly troubleshooting refinery process problems. Over 19,000 engineers and operators have attended his 800 plus in-house troubleshooting seminars since 1983. He has authored eleven books describing his process plant troubleshooting techniques and experiences.



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