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Centrifugal Compressor Settle Out Conditions

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Centrifugal Compressors are a preferred choice in gas transportation industry, mainly due to their ability to cater to varying loads. In the event of a compressor shutdown as a planned event, i.e., normal shutdown (NSD), the anti-surge valve is opened to recycle gas from the discharge back to the suction (thereby moving the operating point away from the surge line) and the compressor is tripped via the driver (electric motor or Gas turbine / Steam Turbine).

In the case of an unplanned event, i.e., emergency shutdown such as power failure, the compressor trips first followed by the anti-surge valve opening. In doing so, the gas content in the suction side & discharge side mix.

Therefore, settle out conditions is explained as the equilibrium pressure and temperature reached in the compressor piping and equipment volume following a compressor shutdown.

IMPORTANCE OF SETTLE OUT CONDITIONS

The necessity to estimate settle out conditions,

1. Settle Out Pressure (SOP) & Settle Out temperature (SOT) determines the design pressure of the suction scrubber & piping.

2. The suction scrubber pressure safety valve's (PSV) set pressure as well as the dry gas sealing pressures are decided by the settle out pressure.

3. When the compressor reaches settle out conditions, process gas is locked inside the piping and equipment and grips the compressor rotor from rotating effectively when restarted. Hence depressurizing is done by routing the locked gas to a flare, via the vent valve to reduce the pressure and achieve effective re-start.

ESTIMATING SETTLE OUT CONDITIONS

Although there are many process simulations tools that can be used to conduct a transient study to determine settle out conditions, hand calculations based on first principles of thermodynamics can also be easily employed. In order to do so, the gas compressor system can be reduced with the assumptions as follows, with the philosophy of using a lumped parameter model, in which an energy balance is made across the total volume of the compressor loop taking into account, the compressor deceleration rate.

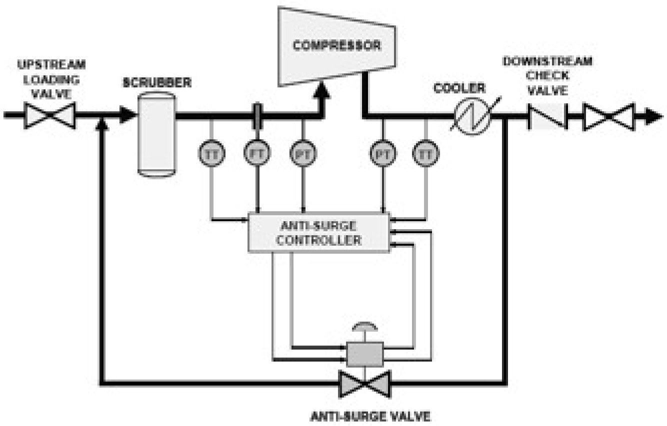

Figure 1. Schematic of Compression System

The assumptions made for this module are,

1. The compressor loop system is a closed loop & no gas has escaped the system.

2. The rate of closure of the suction & discharge block valve in addition to the check valve on the discharge side is neglected.

3. The air cooler is assumed to be running at constant duty before and after the compressor is shut down. If the cooler failure occurs due a power trip, then heat rejection (QCooler = 0) is considered to stop instantaneously.

4. The piping is considered to be adiabatic & no heat escapes from the equipment & piping.
5. The suction scrubber, if considered to have accumulated liquids, then this volume is subtracted from the equipment volumes.
6. The time delay between the fully closed position & fully open position of the Anti-surge valve (ASV) and check valve is not considered.
7. When the driver coasts down after a trip, some amount of residual work is done on the gas.
8. Compressor shutdown times are also influenced by the 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 above method.

CALCULATION METHODOLOGY

The lumped parameter methodology applied to the compressor loop can be depicted as,

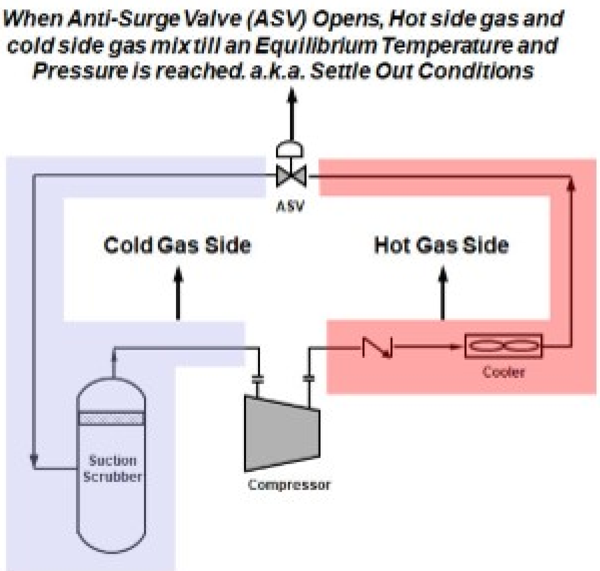

Figure 2. Gas Compressor Loop

Based on the assumptions made, the Settle Out Temperature (SOT) can be estimated as,

$$T = SOT = \frac{(m_S c_{p,S} T_S + m_D c_{p,D} T_D) - Q_{Cooler} + (m_S + m_D)(H_P(t))}{m_S c_{p,S} + m_D c_{p,D}} \quad (1)$$

Where,

$$H_p(t) = A(Q)^2 + B(QN) + CN^2 \quad (2)$$

$$N = N(t) = \frac{1}{\frac{1}{N_0} + \frac{216000k(t - t_0)}{(2\pi)^2 J}} \quad (3)$$

Where,

$H_P(t)$ = Rate of change of polytropic head as the compressor coasts down [kJ/kg/s]

$N(t)$ = Rate of compressor speed decay [rpm/s]

m_s = Suction side gas mass [kg]

m_D = Discharge side gas mass [kg]

T_s = Suction temperature before shutdown [K]

T_D = Discharge temp before shutdown [K]

$C_{p,s}$ = Suction Side Heat Capacity [kJ/kg.K]

$C_{p,D}$ = Discharge Side Heat Capacity [kJ/kg.K]

Q_{Cooler} = Cooler Duty [kJ/s]

k = Fan Power Law Constant

J = Total Inertia of Compressor System [kg.m^2]

The Settle Out Pressure (SOP) can be estimated

$$SOP = \frac{m \times Z_{avg} \times R \times SOT}{MW \times (V_1 + V_2)} \quad (4)$$

Where,

m = Total gas mass [kg]

Z_{avg} = Average Compressibility Factor [-]

R = Gas Constant [m^3.bar/kmol.K]

MW = Gas Molecular weight [kg/kmol]

SOT = Settle Out Temperature [K]

V_1 = Suction side volume [m^3]

V_2 = Discharge Side Volume [m^3]

CASE STUDY

A validation case study is made for a Tank Vapour compressor in a Gas Compression Plant. Suction pressure exists at 1.05 bara, 540C with a discharge pressure of 5.5 bara, 1280C. The coast down period is calculated initially followed by performing settle out calculations. An assumption is made, that the air cooler continues to operate after shutdown. The compressor maps used is

Table 1. Compressor Performance Curves

H_p	Q	Q/N	Hp/N^2
[kJ/kg]	**[Am^3/s]**	**[(Am^3/h)/rpm]**	**[kJ/(rpm^2)]**
136.2	3.0778	0.000322	1.493E-06
133.9	3.4278	0.000359	1.468E-06
130.5	3.6806	0.000385	1.431E-06
126.6	3.8472	0.000403	1.388E-06
123.6	3.9583	0.000414	1.355E-06
115.8	4.1111	0.000430	1.269E-06
109.6	4.1806	0.000438	1.201E-06
100.0	4.2500	0.000445	1.096E-06

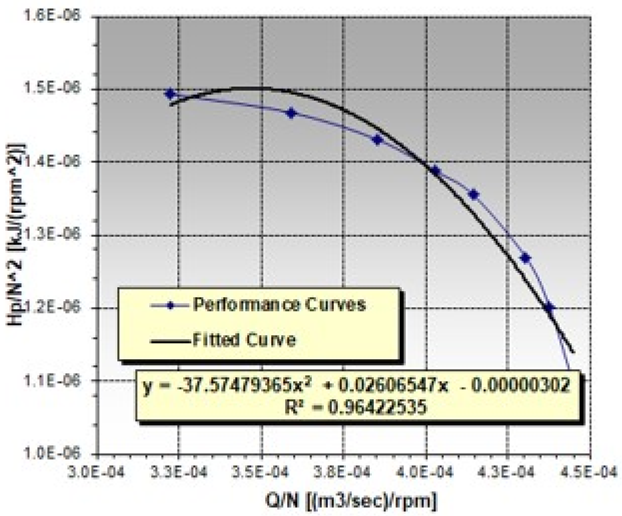

Figure 3. Compressor Performance Curves

Performing calculations as shown in previous sections in MS-Excel based on Table 2 & 3,

Table 2. Compressor Coast down Input Data

Compressor Design Details		
Compressor Inertia	376	$kg.m^2$
Gear Box Inertia	38	$Kg.m^2$
EM Inertia	150.6	$kg.m^2$
Total Inertia (J)	380.6	$kg.m^2$
EM /GT Speed	1493	rpm
Operating Speed	9551	rpm
Gear Ratio (GR)	6.40	-
Fan law constant (k)	8.38E-05	$N.m.min^2$
Fan law Constants (*k*)		
% Speed	**Speed [rpm]**	**k [$N.m.min^2$]**
105	10029	7.57E-05
100	9551	7.00E-05
95	9073	6.68E-05
90	8596	6.42E-05
80	7641	6.03E-05
70	4776	1.66E-04
Avg. Fan Law constant (k)		8.38E-05

It is to be noted, with the Q vs. H_p curve at 9551 rpm, Fan laws were used to derive the compressor curves for other speeds, from 70% to 105%.


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Table 3. Settle Out Conditions Calculations

Suction Piping Data		
Piping Volume	74.55	m^3
Gas Mass Density	1.66	kg/m^3
Mass Specific Heat	*1.83*	kJ/ kg.K
Gas Temperature	54.1	^0C
Comp. Factor (Z_1)	0.9875	-
Suction KO Drum %Vol. Liq	20.0	%
Gas Mass- Suction Side	*98.82*	kg
Discharge Piping Data		
Piping Volume	7.87	m^3
Gas Mass Density	*7.53*	kg/m^3
Mass Specific Heat	*2.16*	kJ/ kg.K
Gas Temperature	128.3	^0C
Comp. Factor (Z_2)	0.9622	-
Discharge KO Drum % Vol. Liq	10.0	%
Gas Mass – Discharge Side	*11.22*	kg
Cooler Data		
Cooler Duty	1432	kW
Cooler Outlet Specific Heat	2.03	kJ/ kg.K

Using the estimated coast down time value of 115 sec for the case studied, the settle out pressure (SOP) & Settle Out Temperature (SOT) is calculated as 0.81 bara, 55.70C & a Settle Out Time of 175 sec. The transient plots of the SOP & SOT based on HYSYS simulations of the case study is as follows,

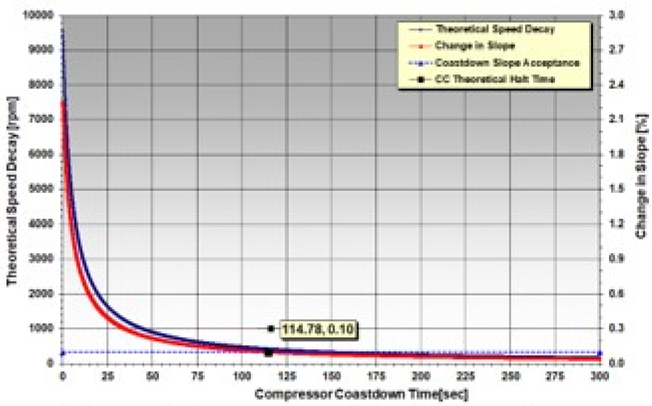

Figure 4. Compressor Coast down Time

The calculated Settle out temperature (SOT) Trend compared with HYSYS 2006.5 is shown as follows,

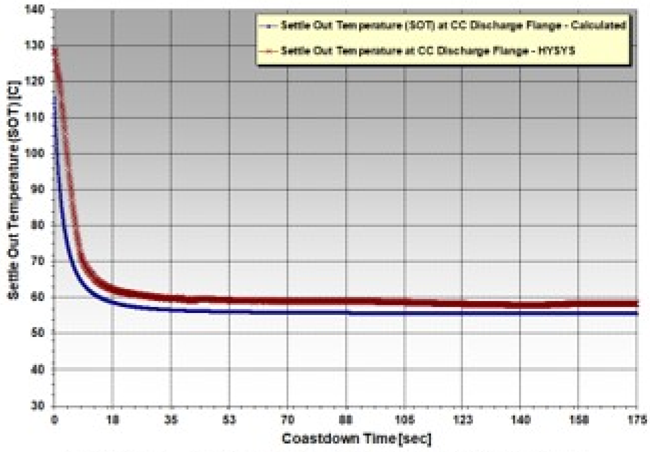

Figure 5. Settle Out Temperature Trend

A comparison made between HYSYS Simulations & the methodology presented shows,

Table 4. HYSYS vs. Calculated Results

Parameter	HY-SYS	Calculat-ed	% Error
SOT [^{0}C]	**58.4**	**55.7**	**-4.8**
SOP [bara]	**0.53**	**0.81**	**+34.6**
Settle Out Time [s]	**167**	**175**	**+4.8**

The SOT & Settle Out Time shows an error margin of < ±5%. Whereas for SOP, between the HYSYS predicted value of 0.34 bara and calculated value of 0.81 bara, represents ~35% error. The author attributes the error in SOP partly to the suction & discharge valve closure time in HYSYS when some vapours were discharged & the remaining for the reasons explained in the next section.

EFFECT OF ASSUMPTIONS ON RESULTS

1. Approximation of compressor curves to Fan Laws – Fan laws are more applicable to fluids with low compressibility, smaller pressure ratios & constant density. Use of these laws would distort the Compressor manufacturer's data thereby causing a difference in calculations. Since the overlap area is significant, the performance curve used in the calculations is assumed to be same throughout the period of coast down. Figure 6 shows the shift in the compressor performance curves.

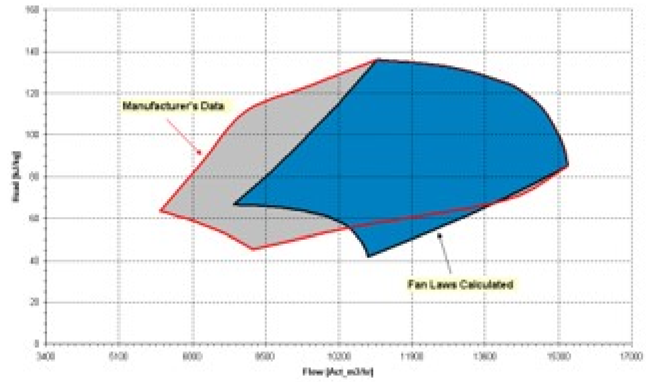

Figure 6. Comparison of Performance Curves between Fan Laws Generated & Vendor Data

2. Equilibrium conditions during Settle Out – During coast down, equilibrium conditions are not reached in the compressor plant piping since the system is dynamic with the gas moving & this is tracked in HYSYS 2006.5. However the calculations methodology considers complete equilibrium being reached at every time step. This causes a difference in the final settle out temperature (SOT) & settle out pressure (SOP).
3. Average Mass Specific Heat Capacity – The calculations methodology considers a constant averaged mass specific heat in the suction & discharge as well as cooler volumes. However, in commercial solvers such as HYSYS 2006.5, the mass heat capacity is computed at every time step which affects the final SOP & SOT.
4. Density & Z Variations – In the calculations made, density and compressibility factor (Z) was assumed to be constant, whereas HYSYS provides density & 'Z' corrections with change in temperature & pressure at every time step.

DESIGN STANDARDS (API 521/NORSOK)

1. In designing suction side of compressor piping & equipment, providing a design margin between settle out pressure and design pressure prevents unnecessary flaring. As per API 521, "Pressure relieving and Depressuring Systems", 5th Edition, Jan 2007, "Design Pressure shall be a minimum of 1.05 times the settle out pressure at maximum pressure drop, calculated assuming the suction side is operated at normal operating pressure and compressor discharge pressure is set to the maximum achievable".
2. As per NORSOK P-001, "The maximum as the settle out pressure occurring at coincident PAHH" (High-High Pressure Alarm) "on both suction side and discharge side, adding a 10% margin for determining design pressure or PSV set pressure". Therefore, NORSOK P-001 standard provides a more conservative estimate of settle out pressure since it takes into account the highest possible suction & discharge pressures.

ANNEXURE A: SETTLE OUT CONDITIONS DERVIATION

The settle out conditions is calculated by considering the suction & discharge volumes as, Suction side gas mass

$$m_S = \left[\left(V_{SuctionSide} - \%V_{SuctionScrubberLiquidVolume}\right)\times \rho_S\right] \quad (1)$$

Discharge side gas mass

$$m_D = \left[\left(V_{DischargeSide} - \%V_{DischargeScrubberLiquidVolume}\right)\times \rho_D\right] \quad (2)$$

Performing heat balance over the closed loop system,

$$E_{In} = E_{Out} \quad (3)$$

$$\text{Or, } Q_{Suction} + Q_{Discharge} + Q_{CC} = Q_{Cooler} \quad (4)$$

Taking that the energy reaching the gas through the compressor is acting only on the mass of gas enclosed & calculating on a per second basis,

$$m_S c_{p,S}(T - T_S) + m_D c_{p,D}(T - T_D) + mH_P = Q_{Cooler} \quad (5)$$

Taking $m = (m_S + m_D)$ & rearranging Eq. (5)

$$T = SOT = \frac{(m_S c_{p,S} T_S + m_D c_{p,D} T_D) - Q_{Cooler} + (m_S + m_D)(H_P(t))}{m_S c_{p,S} + m_D c_{p,D}} \quad (6)$$

The mass specific heat for the cooler in Eq. (6) is taken to be an average value between the upstream & downstream flow. The polytropic head, $H_P(t)$ is treated as a function of time & is calculated by fitting the performance curves (Q vs. H_p).

$$\frac{H_p}{N^2} = A\left(\frac{Q}{N}\right)^2 + B\left(\frac{Q}{N}\right) + C \quad (7)$$

A graph is plotted between $\left(\frac{Q}{N}\right)$ (along x-axis) & $\frac{H_p}{N^2}$ (along y-axis) to obtain the constants A, B & C, followed by rewriting Eq. (E.7) as,

$$H_p(t) = A(Q)^2 + B(QN) + CN^2 \tag{8}$$

In Eq. (8), the compressor speed (N) is calculated as shown in Eq. (9)

$$N = N(t) = \frac{1}{\frac{1}{N_0} + \frac{216000k(t-t_0)}{(2\pi)^2 J}} \tag{9}$$

The volumetric flow calculated using Fan Laws assuming $k_1 = k_2$ during coast down is,

$$\frac{Q_t}{Q_{t+1}} = \frac{N_t}{N_{t+1}} \tag{10}$$

$$\text{Or,}\quad Q = Q_{t+1} = \frac{N_{t+1} \times Q_t}{N_t} \tag{11}$$

It is to be noted that, the value of *'Q'* flowing into the compressor is approximated to value of *'m'* in Eq. (5) (which is constant) since the density lies between suction & discharge density. The settle out pressure is calculated using Ideal Gas equation as,

$$P = SOP = \frac{n \times \left[\frac{(Z_1 + Z_2)}{2}\right] \times R \times SOT}{V_{Total}} \tag{12}$$

$$\text{Or,}\quad SOP = \frac{m \times Z_{avg} \times R \times SOT}{MW \times (V_1 + V_2)} \tag{13}$$

ANNEXURE B: COMPRESSOR COAST DOWN DERIVATION

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

$$T = -(2\pi)J\left(\frac{dN}{dt}\right) \text{ [N-m]} \tag{1}$$

Where,
J = System Inertia (Compressor + gearbox + driver) [kg-m^2],
where,

$$J = J_C + \left[J_M \big/ (Gear\ Ratio)^2 \right]$$

N = Compressor Rotor speed [rpm] or [min^{-1}]
The speed decay rate as well as the system inertia determines the compressor torque. Therefore, the power transferred to the gas, is

$$P = (2\pi NT)\, N \cdot m / \min \tag{2}$$

Substituting Eq. (1) in Eq. (2), the power transferred during (ESD),

$$P = 2\pi N \times \left[-(2\pi)J\left(\frac{dN}{dt}\right)\right] \tag{3}$$

Applying fan power law as an approximation in which 'k' is relatively unvarying for a given curve,

$$P \propto N^3 \Rightarrow P = kN^3;\ k = \frac{60P}{N^3}\left(N \cdot m \cdot \min^2\right) \Rightarrow P = \frac{kN^3}{60} \tag{4}$$

Substituting Eq. (4) in Eq. (3),

$$\frac{kN^3}{60} = 2\pi N \times \left[-(2\pi)J\left(\frac{dN}{dt}\right)\right] \tag{5}$$

Integrating Eq. (6), and also multiplying by (60^2) to convert sec^2 (rev/s) to min^2 (rev/min)

$$\int_{N=N_0}^{N=N(t)} \frac{dN}{N^2} = \frac{k \times 60}{-(2\pi)^2 J} \int_{t_0}^{t=t} dt \tag{7}$$

$$\left[\frac{N^{-2+1}}{-2+1}\right]_{N_0}^{N(t)} = \frac{k \times 60}{-(2\pi)^2 J} \times (t - t_0) \Rightarrow \left[\frac{1}{N}\right]_{N_0}^{N(t)} = \frac{60k(t-t_0)}{(2\pi)^2 J} \tag{8}$$

$$\frac{1}{N(t)} - \frac{1}{N_0} = \frac{60k(t-t_0)}{(2\pi)^2 J} \Rightarrow \frac{1}{N(t)} = \frac{1}{N_0} + \frac{60k(t-t_0)}{(2\pi)^2 J} \Rightarrow N(t) = \frac{1}{\frac{1}{N_0} + \frac{60k(t-t_0)}{(2\pi)^2 J}} \tag{9}$$

Where, N_0 is the compressor speed before ESD. The 2nd denominator term exists with units N.m.min/kg.m^2 & is converted to min^{-1} which gives,

$$N(t) = \frac{1}{\frac{1}{N_0} + \frac{216000k(t - t_0)}{(2\pi)^2 J}} \quad (10)$$

REFERENCES

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