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Reciprocating Compressor Optimization

Kelsy Holmes
Pablo Castillo
Bruce McCain



Biographies

- **Kelsy Holmes**, Staff Engineer, Oxy Oil and Gas Corporation (OOGC).
- **Pablo Castillo**, Project Manager, OOGC. PMP.
- **Bruce McCain**, Engineering Consultant, OOGC. PE in Texas; Chair, API Std 688 (Vib and Puls Control); TAMU Turbo Advisory Committee.



Recip Compressor Optimization Abstract

Reciprocating compressor optimization does not mean the same thing to all operators. Maximizing efficiency, gas throughput, frame horsepower, full load amps and/or motor winding temperatures, rod loads, temperatures, pressures, and torque, are just a few examples of parameters that may be included in an optimization effort.

This Case Study explores a field analysis performed on multiple reciprocating compressor units that have been operating in parallel for 30 + years at an upstream gas processing facility. The goal of the study was to determine the actual compressor operating characteristics by obtaining field measurements of static and dynamic torque as well as in-cylinder pressure measurements on all active cylinders simultaneously. Operating changes were made to the compressor configuration (single acting to double acting, loading and unloading) with all dynamic data being measured and compared. This data was then used to determine actual operational limits, and opportunities for throughput increases. Comparisons of the field analysis results to historical performance data and manufacturer modeling were made and issues identified, along with some unexpected findings.



Problem Statement

A detailed review of compressor performance models yielded discrepancies with clearances, flows, pressures as compared to actual conditions. Clearances and valves were assumed to be as original, but had changed over the course of 30 years. The model is for single machine operation, rather than multiple with common suction and discharge headers. Measured rather than calculated data with fewer assumptions needed to determine options for optimization.



Description

- 8 throw, 3 stg
- 1st stg, 36.5" x 4
- 2nd stg, 29" x 2
- 3rd stg, 18" x 2
- 327 rpm
- 15" stroke
- 6500 HP synch
- 43 MW, CO₂



Configuration

Unit		1 st Stage	2 nd Stage	3 rd Stage
E4-301-1		Throws 1-3 SAHE Throw 4 D/A HE Pocket Unloaders on Throws 3 & 4	Throws 5 & 7 D/A	Throws 6 & 8 D/A
E4-301-2		Throws 1-3 SAHE Throw 4 D/A HE Pocket Unloaders on Throws 2, 3 & 4	Throws 5 & 7 D/A	Throws 6 & 8 D/A
E4-301-3	Tested	Throws 1-3 SAHE Throw 4 D/A HE Pocket Unloaders on Throws 2, 3 & 4	Throws 5 & 7 D/A	Throws 6 & 8 D/A
E4-301-4	Tested	Throws 1 – 4 SAHE HE Pocket Unloaders on Throws 3 & 4	Throws 5 & 7 D/A	Throws 6 & 8 D/A

DA: Double Acting, work occurs on both sides of piston

SA: Single Acting, work occurs on one side of piston

SAHE: Single Acting, Head End



Concerns

- OEM Torsional Analysis indicated marginal TNF separation margin and crankshaft stress safety factor
- Assumptions made during OEM performance analysis differed from actual conditions
 - Process conditions differed from actual
 - Cylinder clearance based on original configuration
 - SA/DA and loading schedule varies due to common suction/discharge headers



Test Procedure

- Strain gage telemetry system installed on motor shaft to measure static and dynamic torque during all testing configurations
- Current loop installed to monitor motor current
- In-cylinder pressure measurements taken on all active cylinder ends simultaneously
- Measured data recorded and used to determine machinery response at all test cases



Strain Gage Telemetry System

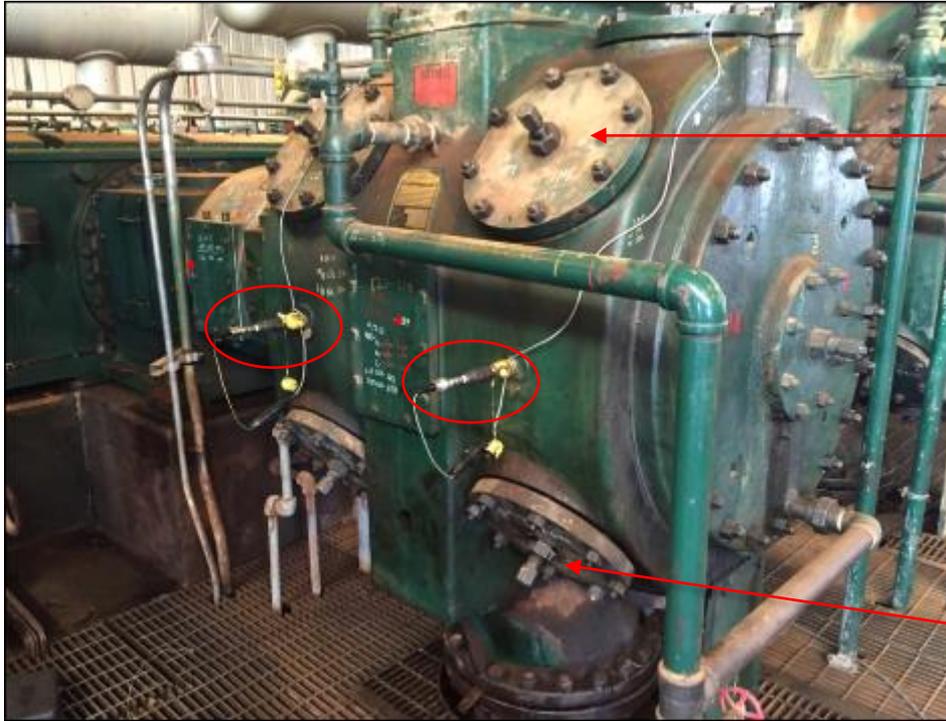
Motor



Barring Wheel

Pressure Transducers

Crank End,
CE



Suction
valve

Head End,
HE

Discharge
valve

Test Cases

Test Case	Time	Date	Unit	Description
1	7:16 PM	4/26/2016	4	Unloaded, As Found, #4 cyl DA
2	7:25 PM	4/26/2016	4	Loaded, As Found, #4 cyl DA
3	3:57 PM	4/27/2016	4	Unloaded, New 1D Valves, #4 cyl DA
4	4:15 PM	4/27/2016	4	Loaded, New 1D Valves, #4 cyl DA
5	5:05 PM	4/28/2016	3	Unloaded, New 1D Valves, #4 cyl DA
6	5:15 PM	4/28/2016	3	Loaded, New 1D Valves, #4 cyl DA
7	10:27 AM	4/28/2016	4	Unloaded, New 1D Valves, #4 cyl SAHE
8	12:00 PM	4/28/2016	4	Loaded, New 1D Valves, #4 cyl SAHE



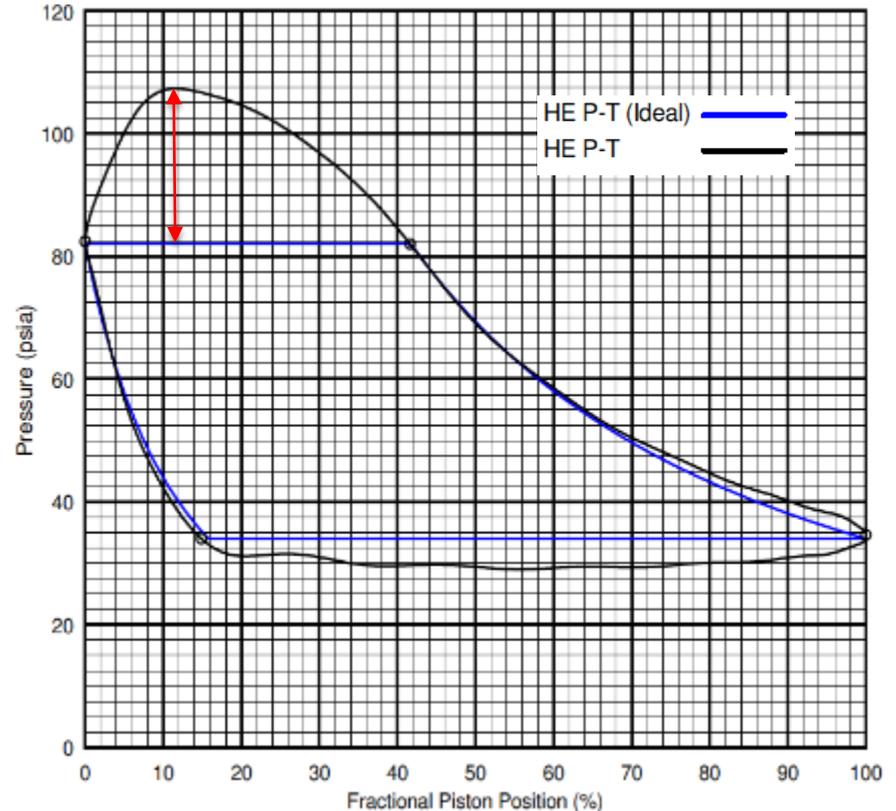
Torsional Measurements

- Field measurements basically supported calculated
 - Measured TNF separation margins > predicted
 - Measured dynamic torque in motor shaft 12% less than predicted
- Torsional measurements didn't add much to cost of field project due to compressor configuration time and increased overall confidence in calculated results



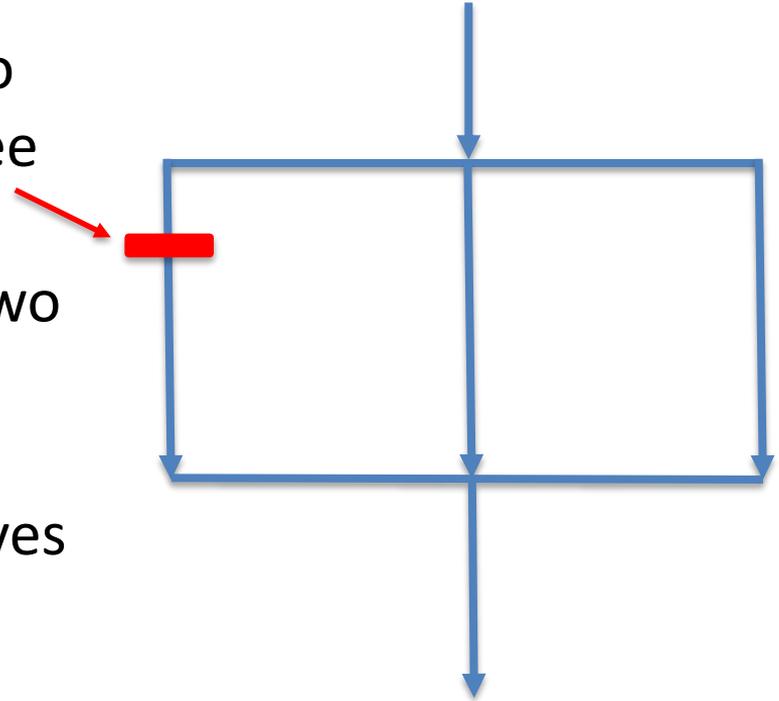
PV Card Analysis

- Static pressure at discharge flange about 82 psia with peak cyl pressure about 107 psia
- Some over-pressure expected, though this appears excessive
- Investigation revealed each 1st stg cyl had 1 blanked discharge valve per active end



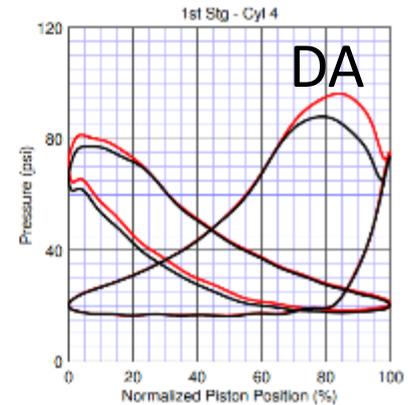
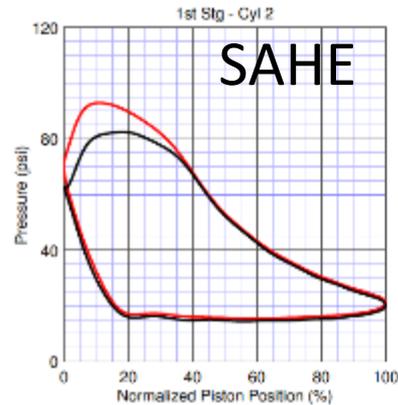
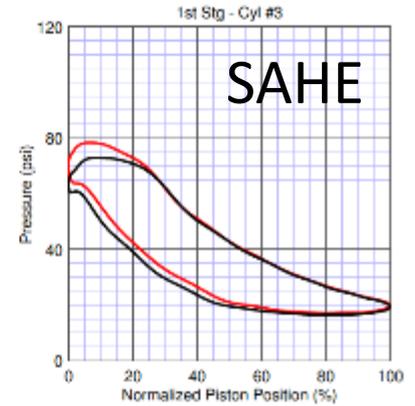
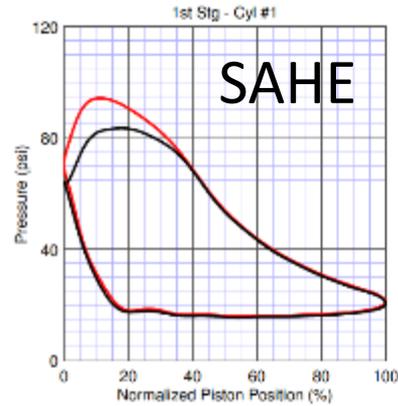
Discharge Blanks

- A blanked discharge is analogous to adding a blind flange to one of three outlet pipes leaving a header.
- All flow is required to go through two rather than three pipes.
- Result is higher pressure drop and velocity, increased hp, wear on valves
- No technical reason to do this



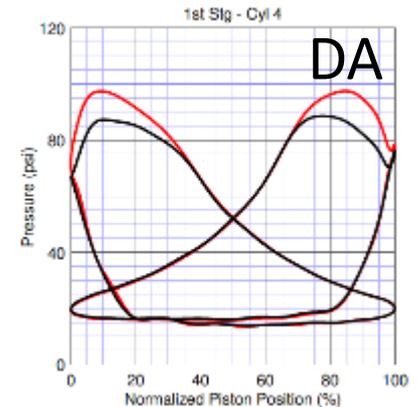
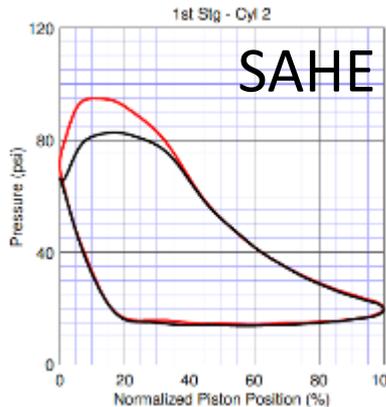
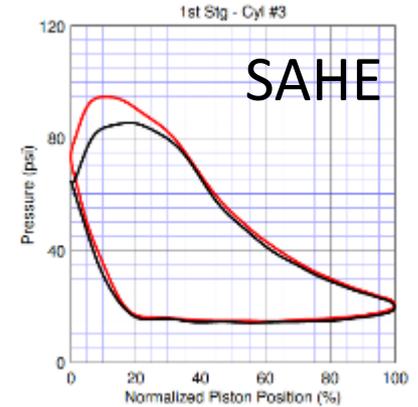
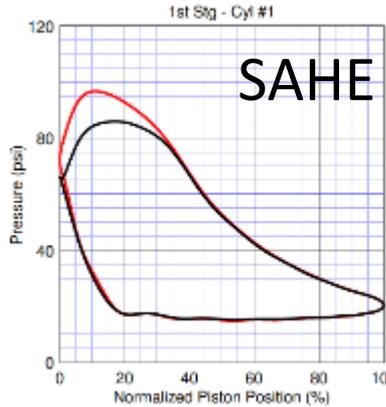
Discharge Blanks, Case 1, 3, Unloaded

- Discharge blanks replaced with valves
- Comparison of 1st stg PV plots with (red) and without (black) discharge valve blanks
- 279 hp difference (unloaded), based on crankshaft strain gage data



Discharge Blanks, Case 2, 4, Loaded

- Discharge blanks replaced with valves
- Comparison of 1st stg PV plots with (red) and without (black) discharge valve blanks
- 380 hp difference (loaded), based on crankshaft strain gage data



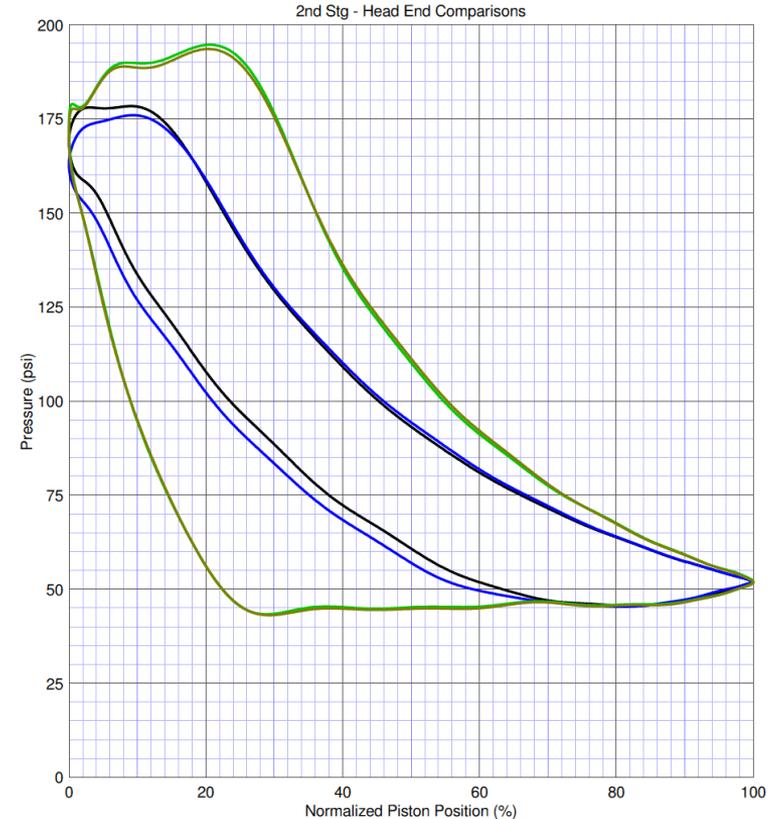
Discharge Blanks

- One discharge blank was installed in each 1st stage, active end of each machine
- SAHE had one blank on HE
- DA had one blank on HE, one on CE
- 5 total blanks on each machine
- 30 total blanks total on 6 compressors
- Not accounted for in model



Clearance Variance, Case 4, 6, Loaded

- Overlay of 4 loaded 2nd stg HE PV plots clearly show additional fixed clearance on 2 cylinders (blue, black) not accounted for in original model
- Additional fixed clearance was installed in the past which reduced capacity 2 mmscfd



Findings

- Measured TNF separation margins > calculated
- Measured dynamic torque in motor shaft 12% less than predicted
- Measureable clearance differences on “identical” cylinders
- Two 2nd stg cylinders had fixed clearance installed not accounted for on model
- All 1st stg cylinders had blanked discharge valves on active ends



Cost Savings

- Removal of the discharge blanks and installation of valves resulted in 380 hp reduction per machine when fully loaded
- $6 * 380 \text{ hp} = 2280 \text{ hp}$
- 99% uptime
- $2280 * .745 * 8760 * .04 * .99 = \$589\text{K}/\text{year}$
- Cost of field study < \$30K
- Cost of new valves approx. \$45K



Lessons Learned

- Be aware of “camouflage” issues.
- Actual operating configuration must be accounted for, not just serial number based configuration when modeling
- Common suction/discharge can cause significant discrepancies between calculated and actual flows due to loading differences
- Multiple maintenance related issues discovered including unloader indicator pin



Lessons Learned (cont)

- Parallel compressor operation requires all machines to be measured at the same time for most accurate results
- As a result of this project, we have acquired enough pressure transducers to instrument HE/CE on multiple parallel machines
 - We've done 5 parallel 6 throw machines at once: more to come on that one...



Summary

Optimizing existing reciprocating compressor trains may require a higher degree of accuracy than modeling alone can attain due to deviations in many factors over time. Measured data increases accuracy by eliminating many assumptions.

