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Kinder Morgan Compressor Equations

and

WinFlow Detailed Station Calculations (DSC)

Show of Hands…

Who Uses Detailed Station Calculations?

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Tennessee Gas Pipeline (El Paso) has been using DSC since the 80's.

1. This was in an era when Companies had dedicated Compressor Services departments that could test machines, develop accurate equations and generate and update coefficients.

2. Planning could easily incorporate these unit parameters in WinFlow models.

3. Today all TGP stations are modeled with DSC.

KMI's acquisition of El Paso resulted in a re-evaluation of the approach for compressor station modeling

1. Evaluation team included members of KMI pipelines NGPL, KMTP, MEP, TGP, SNG, EPNG, CIG

2. Existing capacity models demanded that TGP be able to continue with DSC modeling to ensure firm deliveries can be met (e.g., changing to block horsepower would impact capacity)

3. Modifications were made to equations to accommodate NGPL's station modeling in a stand-alone application

4. Other KMI assets will begin using DSC on an as-needed basis and will adopt the new compressor equations

5. Long term plan is to migrate turbine/centrifugal stations to Gregg Engineering's C5 equation in NextGen

Good News!

The equations that you see today are in the public domain and can be used by your company

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Equations Evaluated by the Team

1. Reciprocating Compressor Throughput
2. Reciprocating Compressor Required Horsepower
3. Reciprocating Engine Allowed Horsepower
4. Reciprocating Engine Fuel
5. Gregg Engineering's Turbine/Centrifugal C5 Tables
6. Turbine Allowed Horsepower
7. Turbine Part Load Fuel
8. Centrifugal Compressor Throughput
9. Generic Driver and Compressor

Reciprocating Compressor Throughput

$$Q = SV * N\left(\frac{\circ_s}{\circ_b}\right)\left(\frac{1{,}440}{1{,}728*1{,}000{,}000}\right)\left(E_0 + E_1R_c + E_2R_c{}^2 - m\left(R_c{}^{\frac{1}{n}} - 1\right)\right)Tf$$

$Q =$	Throughput (mmscfd)	$E_0, E_1, E_2 =$	Volumetric Efficiency Coefficients
$SV =$	Swept Volume (in^3)	$R_c =$	Compression Ratio
$N =$	Compressor Speed (rpm)	$m =$	Clearance Ratio (CV/SV)
$\circ_S =$	Suction Density (lb$_m$/ft^3)	$T_f =$	Throughput Factor
$\circ_b =$	Base Density (lb$_m$/ft^3) @60F, 14.73 psia	$n =$	Polytropic Coefficient

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Reciprocating Compressor Throughput (cont'd)

Fundamental form of throughput equation:

$$Q = SV * N\left(\frac{\circ_s}{\circ_b}\right)\left(\frac{1{,}440}{1{,}728*1{,}000{,}000}\right)EV$$

Many different forms of EV (volumetric efficiency) used by compressor manufacturers, pipeline companies, academics

Volumetric Efficiency = The ratio of the volume of fluid actually displaced by the piston to its swept volume.

Reciprocating Compressor Throughput (cont'd)

Ariel Compressors Volumetric Efficiency
Typical Industrial Form

Volumetric Efficiency

$$VE_S = 100 - R_C - \%CL\left[\left(\frac{Z_s}{Z_d}\right)\left(\frac{P_d}{P_s}\right)^{\frac{1}{K}} - 1\right]$$

Where:
- VE_S = Volumetric efficiency, %
- %CL = Fixed clearance, %
- Z_S = Compressibility factor @ P_S & T_S
- Z_D = Compressibility factor @ P_D & T_D
- P_D = Discharge pressure, psia
- P_S = Suction pressure, psia
- K = Adiabatic exponent, k-value

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%CL = m in KMI equations and "k" is "n" in KMI equations

Reciprocating Compressor Throughput (cont'd)

KMI Volumetric Efficiency

$$EV = \left(E_0 + E_1 R_c + E_2 R_c{}^2 - m\left(R_c{}^{\frac{1}{n}} - 1 \right) \right) Tf$$

Allows user to curve fit to non-linear historical data.
Setting E_0 to 100, E_1 to -1 and E_2 to 0 allows equation to match Ariel equation.

Tf is not so much part of the volumetric efficiency but is another tuning factor that the modeler can adjust with an online model or historical data. TGP uses 1.0 and I'm not aware of cases where a different value is used.

Reciprocating Compressor Required Horsepower

$$HP_{req} = HP_c + HP_x + HP_p$$

$HP_{req} =$ Horsepower Required from Driver

$HP_c =$ Compressor Horsepower (includes pulsation loading and valve losses)

$HP_x =$ Auxiliary Load (e.g., oil pumps, engine driven fan loads)

$HP_p =$ Parasitic Horsepower (deactivated ends)

Reciprocating Compressor Required Horsepower (cont'd)

$$HP_c = \frac{SV * N * P_S}{396{,}000}\frac{n}{n-1}\left(\left(R_c^{\frac{n-1}{n}} - 1\right) - m\left(1 + R_c - R_c^{\frac{1}{n}} - R_c^{\frac{n-1}{n}}\right)\right)(1 + WF + LF)\left(\frac{1}{EM}\right)$$

$HP_c =$ Horsepower

$SV =$ Swept Volume (in^3)

$N =$ Compressor Speed (rpm)

$P_S =$ Suction Pressure (psia)

$n =$ Polytropic Exponent (~1.3)
(no heat transfer, natural gas)

$m =$ Compressor Clearance Ratio (CV/SV)

$R_c =$ Compression Ratio

$WF =$ Waste K Factor (Valve Losses)

$LF =$ Pulsation Loading

$EM =$ Mechanical Efficiency (~0.95)

Reciprocating Compressor Required Horsepower (cont'd)

$$WF = \frac{K}{100\left(R_c - 1\right)}$$

$$K = \frac{N - N_{\min}}{N_{\max} - N_{\min}}\left(W_{\max} - W_{\min}\right) + W_{\min}$$

$$W_{\min} = v_0 + v_1 R_c + v_2 R_c^{\ 2}$$

$$W_{\max} = u_0 + u_1 R_c + u_2 R_c^{\ 2}$$

$$LF = L_0 + L_1 R_c + L_2 R_c^{\ 2} + L_3 R_c^{\ 3} + L_4 R_c^{\ 4}$$

If field test data is not available for the unit then all coefficients can be set to 0 and a single value of "K" used.

Most TGP units use all of these parameters. However, going forward it's unknown if these parameters will be developed on new installations.

For the existing fleet it would not add value to simplify these relationships to a single K value.

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Reciprocating Compressor Required Horsepower (cont'd)

Parasitic horsepower represents losses resulting from deactivated ends. Field test data is necessary to develop these coefficients.

$$HP_p = A_{total}\left(\frac{N - N_{\min}}{N_{\max} - N_{\min}}\left(HP_{\max} - HP_{\min}\right) + HP_{\min}\right)$$

$$HP_{\min} = p_0 + p_1 P_s + p_2 P_s^2$$

$$HP_{\max} = q_0 + q_1 P_s + q_2 P_s^2$$

A_{total} is the number of deactivated ends
P_s is the suction pressure

Reciprocating Compressor Allowed Horsepower and Fuel (cont'd)

Allowed horsepower is obtained from linear interpolation of horsepower between known ambient temperatures. Most recips are rated at single max horsepower across a range of temperatures. Some older recips are "ambient up-ratable".

Max HP @ 100°F (hp)	4,500
Max HP @ 80°F (hp)	4,860
Max HP @ 60°F (hp)	5,220
Max HP @ 40°F (hp)	5,220
Max HP @ 20°F (hp)	5,220

$$2 = \frac{\left(\frac{HP_{Dev}}{N}\right)}{\left(\frac{HP_{Rated}}{N_{Rated}}\right)} x100$$

$$F = \Bigg(F_0 + F_1 HP_{Rated} \left(\frac{2}{100}\right) + F_2 HP^2_{Rated} \left(\frac{2}{100}\right)^2 \Bigg(\frac{N * HP_{Rated}}{N_{Rated}} \left(\frac{2}{100}\right) \Bigg) \Bigg) \left(\frac{24}{LHV * 1,000}\right) \Bigg)$$

LHV = Fuel Lower Dry Heating Value (btu/cf) (default =1000)

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Gregg Engineering C5 in NextGen Centrifugal Data from Vendor

RPM	Q	Head	Eta	SQ	Pressure	Power	SQ/Pres	Ratio	Pwr/Pres	Phi	Psi	Work Fac	Poly Head	Poly Psi	Poly Eta	T2
9,500	8,136	16,884	0.814	861	1,358	16,499	0.9302	1.466	17.818	0.042	1.074	0.66	17058	1.085	0.8224	113.5
9,500	9,438	16,535	0.8432	999	1,348	18,092	1.079	1.456	19.538	0.0487	1.052	0.624	16672	1.061	0.8502	110.9
9,500	10,739	15,863	0.8619	1,137	1,328	19,322	1.2278	1.435	20.866	0.0554	1.009	0.586	15972	1.016	0.8679	107.9
9,500	12,041	14,911	0.8698	1,275	1,301	20,178	1.3766	1.405	21.791	0.0622	0.949	0.545	15002	0.955	0.8751	104.4
9,500	13,342	13,725	0.8664	1,413	1,268	20,663	1.5254	1.369	22.314	0.0689	0.873	0.504	13805	0.878	0.8714	100.6
9,500	14,644	12,346	0.8504	1,550	1,230	20,783	1.6742	1.328	22.443	0.0756	0.786	0.462	12420	0.79	0.8556	96.6
9,500	15,945	10,706	0.8138	1,688	1,186	20,506	1.823	1.281	22.145	0.0823	0.681	0.419	10780	0.686	0.8194	92.1
9,500	17,247	8,877	0.7553	1,826	1,138	19,817	1.9718	1.229	21.401	0.089	0.565	0.374	8950	0.57	0.7615	87.4
9,500	18,548	6,983	0.6761	1,964	1,090	18,730	2.1206	1.178	20.227	0.0958	0.444	0.329	7051	0.449	0.6826	82.4
9,500	19,850	5,065	0.5696	2,102	1,043	17,256	2.2694	1.127	18.635	0.1025	0.322	0.283	5122	0.326	0.576	77.5

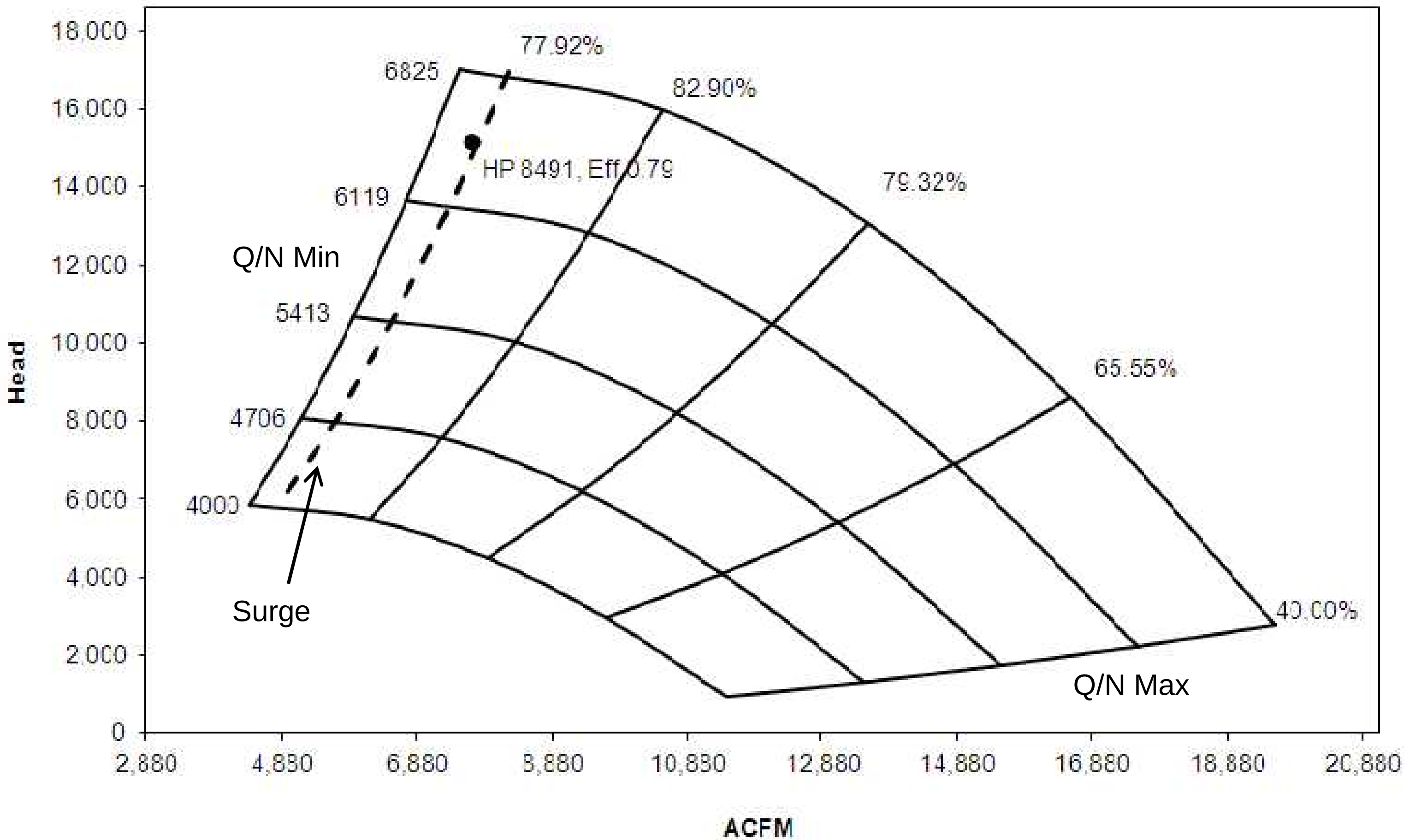
Portion of a table representing data for a Solar C65 compressor with a D-2 impeller. Complete table has values from speeds of 9000, 8000, 7000, 6000, 5000 and 3990.

The rest of the table is truncated. Just showing the parameters involved in the centrifugal compressor.

Not going to go into the details of what these parameters represent, just showing the concept.

Typical Centrifugal Compressor – Fan Law

"Under certain simplifying conditions, operating points of a compressor at different speeds can be compared". *(Kurz - PSIG 0408, 2004)*



Centrifugal Compressor Throughput

$$\frac{H}{N^2} = \left(\frac{(\circledR D)^2}{144(3{,}600)g} \right) \left(A_0 + A_1 \left(\frac{Q_s}{N} \right) + A_2 \left(\frac{Q_s}{N} \right)^2 + A_3 \left(\frac{Q_s}{N} \right)^3 \right)$$

$$H = R * T_s z_s \left(\frac{n}{n-1} \right) \left(R_c^{\left(\frac{n-1}{n} \right)} - 1 \right)$$

$H =$ Head

$N =$ Compressor Speed (rpm)

$D =$ Impeller Diameter (in)

$g =$ Gravitational Constant (32.2 lbm-ft/lbf/s2)

$A_0 - A_3 =$ Pressure Coefficient

$Q_s =$ Flow at Suction Conditions (ACFM)

$R =$ Gas Constant (1545 ft-ftlb/lbmR)

$R_c =$ Compression Ratio

$n =$ Adiabatic Exponent

$T_s =$ Temperature at Suction Conditions

$z_s =$ Compressibility at Suction Conditions

$N =$ Compressor Speed

$E_0 - E_3 =$ Compressor Efficiency Coefficients

Gregg Engineering C5 in NextGen

T1 DEG. F	PWRM HP	NPTO RPM	SFCM BTU/HPH	T1 DEG. F	PWRM HP	NPTO RPM	SFCM BTU/HPH
-20	8,959	6,789	8,196	60	7,721	7,179	10,999
-20	10,751	8,184	10,070	60	9,265	7,671	9,931
-20	12,543	8,578	9,251	60	10,810	8,077	9,129
-20	14,334	8,921	8,572	60	12,354	8,422	8,454
-20	16,126	9,194	7,973	60	13,898	8,720	7,864
-20	17,918	9,451	7,435	60	15,442	9,051	7,517
0	8,806	6,815	8,335	80	7,168	7,009	11,202
0	10,567	8,119	9,975	80	8,602	7,427	10,103
0	12,328	8,505	9,121	80	10,035	7,855	9,268
0	14,090	8,841	8,446	80	11,469	8,205	8,595
0	15,851	9,132	7,857	80	12,903	8,511	8,023
0	17,612	9,399	7,382	80	14,336	8,850	7,681
20	8,538	7,516	10,884	100	6,552	6,860	11,572
20	10,245	8,004	9,903	100	7,863	7,214	10,407
20	11,953	8,399	9,069	100	9,173	7,578	9,528
20	13,660	8,731	8,379	100	10,484	7,951	8,822
20	15,368	9,026	7,802	100	11,794	8,263	8,249
20	17,075	9,325	7,382	100	13,105	8,610	7,930
40	8,188	7,375	10,869	120	5,873	6,738	12,192
40	9,826	7,859	9,875	120	7,048	7,034	10,919
40	11,464	8,260	9,065	120	8,222	7,335	9,967
40	13,101	8,599	8,373	120	9,397	7,637	9,218
40	14,739	8,889	7,788	120	10,571	7,971	8,601
40	16,376	9,218	7,421	120	11,746	8,323	8,283

Tabular data from Solar for a Mars 100-16000S showing ambient temperature, nominal horsepower, optimum power turbine speed and heat rate at sea level.

NextGen will interpolate between data points instead of solving an equation

Turbine Max Allowed Horsepower

Turbine allowed horsepower is derived from on-site test data as a function of ambient temperature and speed. The elevation affect is accounted for in the data regression to obtain the coefficients.

We have not found a need to model the turbine in any more detail, such as at off-optimum power turbine speeds. We only need to know what the maximum available power is based on ambient temp.

$$HP_{avail} = \left(H_0 + H_1 T_A + H_2 T_A^2 + H_3 T_A^3 + H_4 N + H_5 N^2 + H_6 N^3 + H_7 T_A N\right) \text{«}_{Turb}$$

$HP_{avail} =$ Maximum horsepower available from turbine

$H_0 - H_7 =$ Coefficients developed from test data.

$T_a =$ Ambient Temperature (F)

$N =$ Axial Flow Compressor Speed (rpm) (not power turbine speed)

$\text{«}_{turb} =$ Planning tuning factor to allow for turbine degradation over time.

Turbine Fuel

$$Fuel = F_{PL}\left(F_0 + F_1T_A + F_2T_A^2 + F_3T_A^3 + F_4N + F_5N^2 + F_6N^3 + F_7T_AN\right)HP_{Dev}\,24 / LHV$$

$$F_{PL} = \left[C_0 + C_1T_A + C_2T_A^2 + C_3T_A^3 + C_4\left(\frac{100\,HP_{Dev}}{Hp_{Allowed}}\right) + C_5\left(\frac{100\,HP_{Dev}}{Hp_{Allowed}}\right)^2 + C_6\left(\frac{100\,HP_{Dev}}{Hp_{Allowed}}\right)^3 + C_7T_A\left(\frac{100\,HP_{Dev}}{Hp_{Allowed}}\right)\right]$$

$T_A =$ Ambient Temperature

$N =$ Compressor Speed

$LHV =$ Fuel Lower Dry Heating Value (btu/cf) (default =1000)

$HP_{Dev} =$ Horsepower at Turbine Shaft

$HP_{Allowed} =$ Max Allowed HP at Ambient Temp

$F_{PL} =$ Part Load Fuel Factor

$F_0 - F_7 =$ Fuel Coefficients

$C_0 - C_7 =$ Part Load Fuel Coefficients

As with the available horsepower equation the fuel coefficients are based on site test data and so there is no elevation correction factor.

Turbine Fuel (cont'd)

$$HP_{Dev} = \frac{Q_S \,{}^{\circ}{}_S H}{33{,}000 \,\text{«}_{Hydr} \,\text{«}_{Mech}}$$

$$H = RT_s z_s \left(\frac{n}{n-1}\right)\left(R_c^{\left(\frac{n-1}{n}\right)} - 1\right)$$

$$\text{«}_{Hydr} = E_0 + E_1\left(\frac{Q_s}{N}\right) + E_2\left(\frac{Q_s}{N}\right)^2 + E_3\left(\frac{Q_s}{N}\right)^3$$

$Q_s =$ Compressor Throughput

${}^{\circ}{}_s =$ Suction Density

$HP_{Dev} =$ Horsepower at Turbine Shaft

$H =$ Head

$\text{«}_{Hydr} =$ Compressor Efficiency from Fan Curve

$\text{«}_{Mech} =$ ~99% based on type of bearing in centrifugal (overhung, 2 bearing or mag bearings)

$R =$ Gas Constant

$R_c =$ Compression Ratio

$n =$ Adiabatic Exponent

$T_s =$ Temperature at Suction Conditions

$z_s =$ Compressibility at Suction Conditions

$N =$ Compressor Speed

$E_0 - E_3 =$ Compressor Efficiency Coefficients

Generic Unit - Can Represent Turbine, Recip or Electric Motor and Recip or Centrifugal Compressor

$$HP_{avail} = \left(H_0 + H_1 T_A + H_2 T_A^2 + H_3 T_A^3 + H_4 N + H_5 N^2 + H_6 N^3 + H_7 T_A N\right) ¢_{Elev}$$

$$¢_{Elev} = 14.73 * e^{\frac{-elev}{27,000}} \approx -3.215 * e^{-5} Ft_{Elev} + 1$$

Gregg Engineering will use the second form of elevation correction for computational simplicity

$HP_{avail} =$ Maximum horsepower available from turbine

$H_0 - H_7 =$ Coefficients developed from test data.

$T_a =$ Ambient Temperature (F)

$N =$ Compressor Speed (rpm) (not turbine speed)

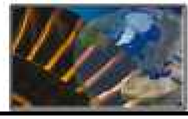
$¢_{Elev} =$ Elevation correction factor

$Ft_{Elev} =$ Feet above sea level (ft)

Coefficients can be generated from the Solar 2 program for the Solar fleet of units.
To simulate an electric or recip H_0 is rated horsepower and elevation term is 1.

Solar 2 Program to Model Turbines

Solar Turbines
A Caterpillar Company



MARS 100-16000S
CS/MD 122F MATCH
Std. Natural Gas Fuel
FASTX REV TMG-2S REV. 1.
SOLAR2.xls Rev. 25-Mar-10 expires: 5-Dec-13
Run on: 5-Nov-13 7:47:55 AM
Kevin Frank 713-895-2397

GS's | MD's | SELECT MECH DRIVE RATING | CHANGE ENGR. UNITS | USER PREFERENCES | SAVE TEXT FOR E-MAIL

Evaporative Cooler
Rel Humid. % 60

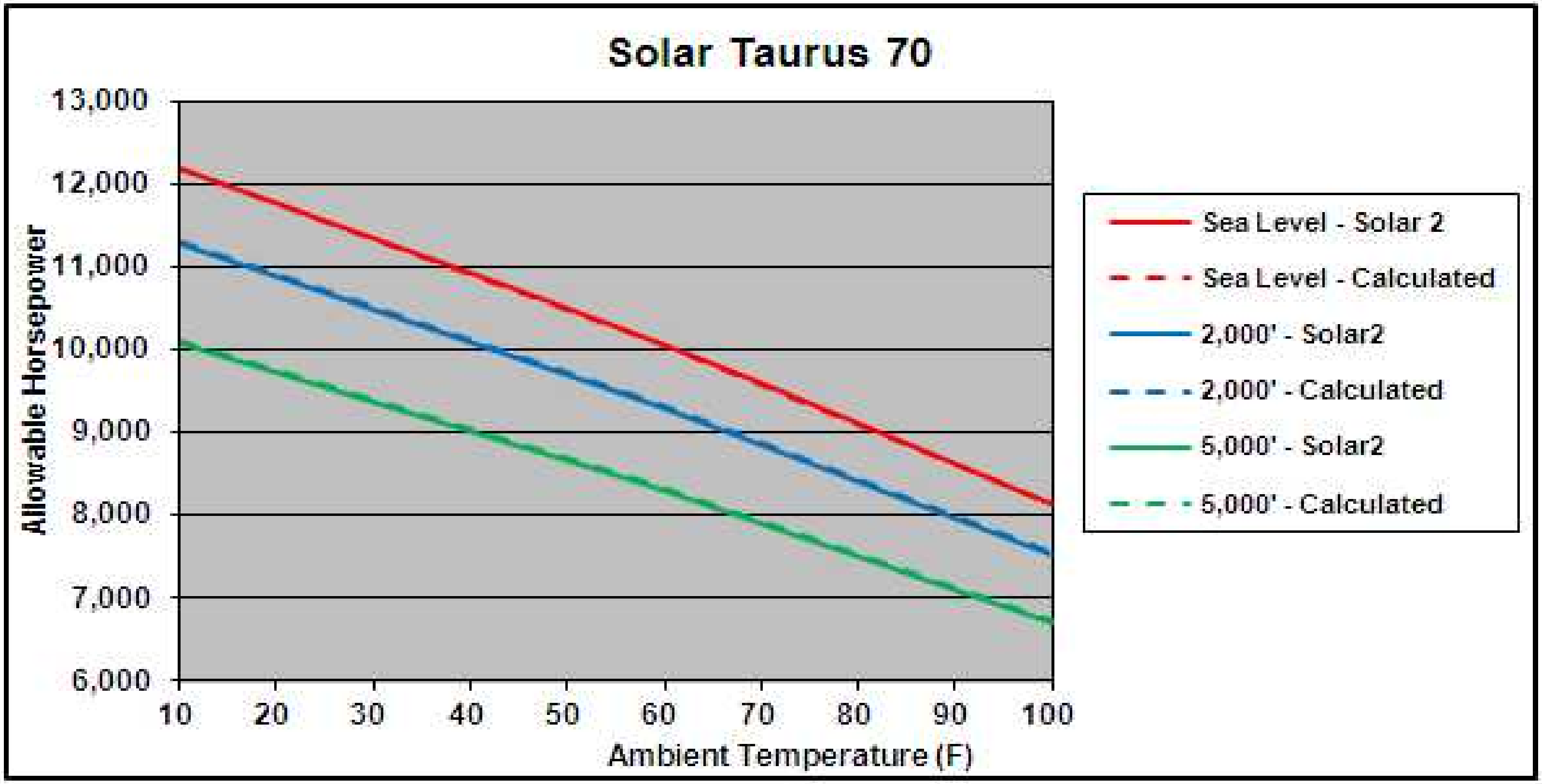
EXPECTED PERFORMANCE, NOT GUARANTEED

SITE ELEVATION:	0	feet					
BAROMETRIC PRESSURE:	29.92	"Hg					
INLET DUCT LOSS:	4	"H2O					
EXHAUST DUCT LOSS:	4	"H2O					
MINIMUM EXHAUST STACK TEMP (T9):	325	°F					
AMBIENT AIR TEMPERATURE (T1):	59	59	59	59	59	59	°F
PART POWER (hp), % LOAD, or 0 for MAX:	0	0	0	0	0	0	hp
(0 = opt) POWER TURBINE SPEED, NPT,%:	0	0	0	0	0	0	% of 9500 RPM
ENGINE INLET AIR TEMPERATURE (T1):	59	59	59	59	59	59	°F
Nominal OUTPUT POWER:	15437	15437	15437	15437	15437	15437	hp
FUEL FLOW (LHV):	118.2	118.2	118.2	118.2	118.2	118.2	mmBTU/hr
EXHAUST GAS TEMPERATURE (T7):	910	910	910	910	910	910	°F
EXHAUST GAS FLOW:	339524	339524	339524	339524	339524	339524	lb/hr
OPERATING POWER TURBINE SPEED:	8963	8963	8963	8963	8963	8963	RPM
OPTIMUM POWER TURBINE SPEED:	8963	8963	8963	8963	8963	8963	RPM
Nominal THERMAL EFFICIENCY:	33.2	33.2	33.2	33.2	33.2	33.2	%
Nominal HEAT RATE:	7656	7656	7656	7656	7656	7656	BTU/hp-hr
PCD PRESSURE:	247.9	247.9	247.9	247.9	247.9	247.9	PsiG
EXHAUST HEAT (from T7 to T9):	51.7	51.7	51.7	51.7	51.7	51.7	mmBTU/hr

ADD EXH ANAL | ADD 1 DIGIT | DROP 1 DIGIT

Generate table of values from these inputs and calculated power output

Sample Output from Solar 2



Coefficients generated from Solar 2 data (solid lines)
Equation results match well (dash lines)

Generic Unit - Can Represent Turbine, Recip or Electric Motor and Recip or Centrifugal Compressor (cont'd)

$$Q = \frac{HP_{Dev}\,\text{«}_{Mech}\,Tf\left(\frac{\circ_{S}}{\circ_{B}}\right)}{3.0303\,P_S\left(\frac{n}{n-1}\right)\left(R_c^{\left(\frac{n-1}{n}-1\right)}\right)\left(1+\frac{K}{R_c-1}\right)}$$

$HP_{Dev} =$ Horsepower available from driver (turbine, motor or recip engine)

$P_s =$ Suction pressure

$T_f =$ Throughput Factor

$R_c =$ Compression Ratio

$\text{«}_{Mech} =$ Mechanical Efficiency (~0.95)

$K =$ Recip waste K factor

$n =$ Polytropic exponent

$\circ_s =$ Suction density

$\circ_B =$ Base density

Tf and K can be adjusted to provide a desired overall unit/station efficiency.

Adjust these terms to get ~ 80% to 83% depending on expected efficiency of proposed unit.

The generic horsepower and throughput equations on this slide and the previous slide are used for quick analysis in the absence of specific unit paramters and are not recommended for capacity determinations.

Examples Illustrating the Benefit of Using Detailed Station Calculations

Why Use Detailed Station Calculations?

1. Ensure Capacity Isn't Oversold

2. Ensure Max Capacity Is Identified

3. Identify Possible Compressor/Station Operating Gaps
   a. Expansion scenarios or change in station operations may require compressors to operate outside of design range
   b. Block horsepower won't capture this

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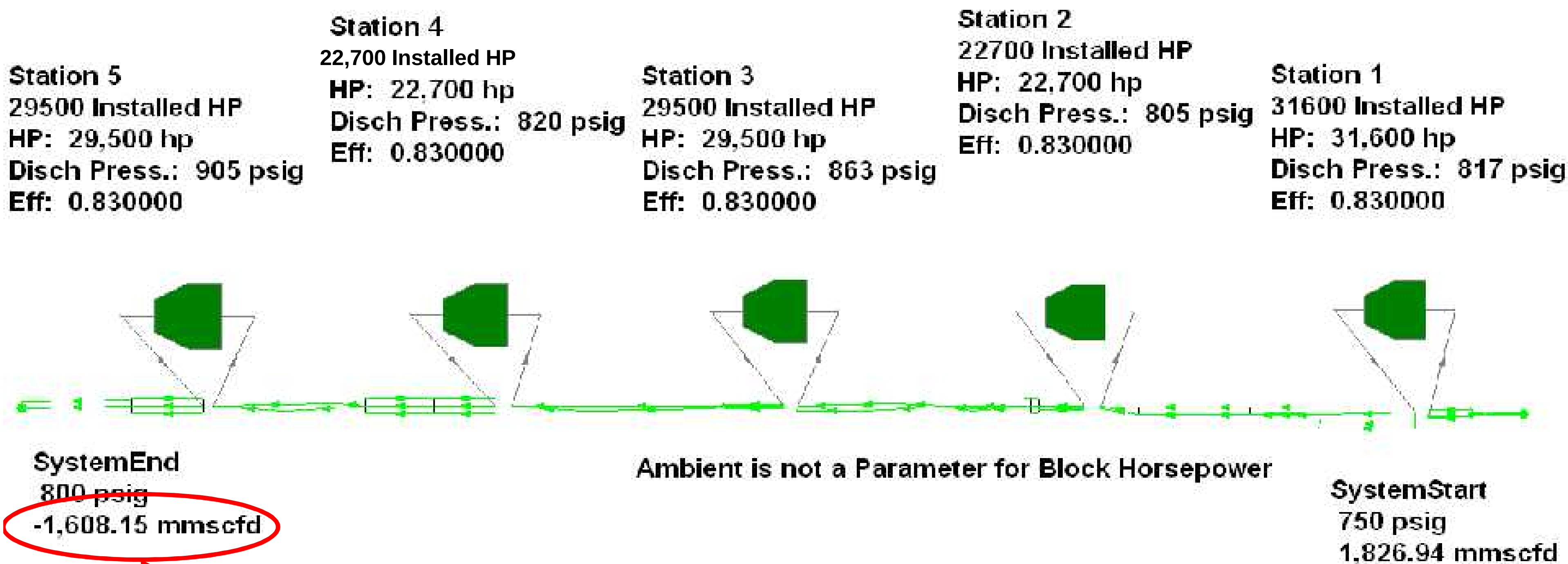
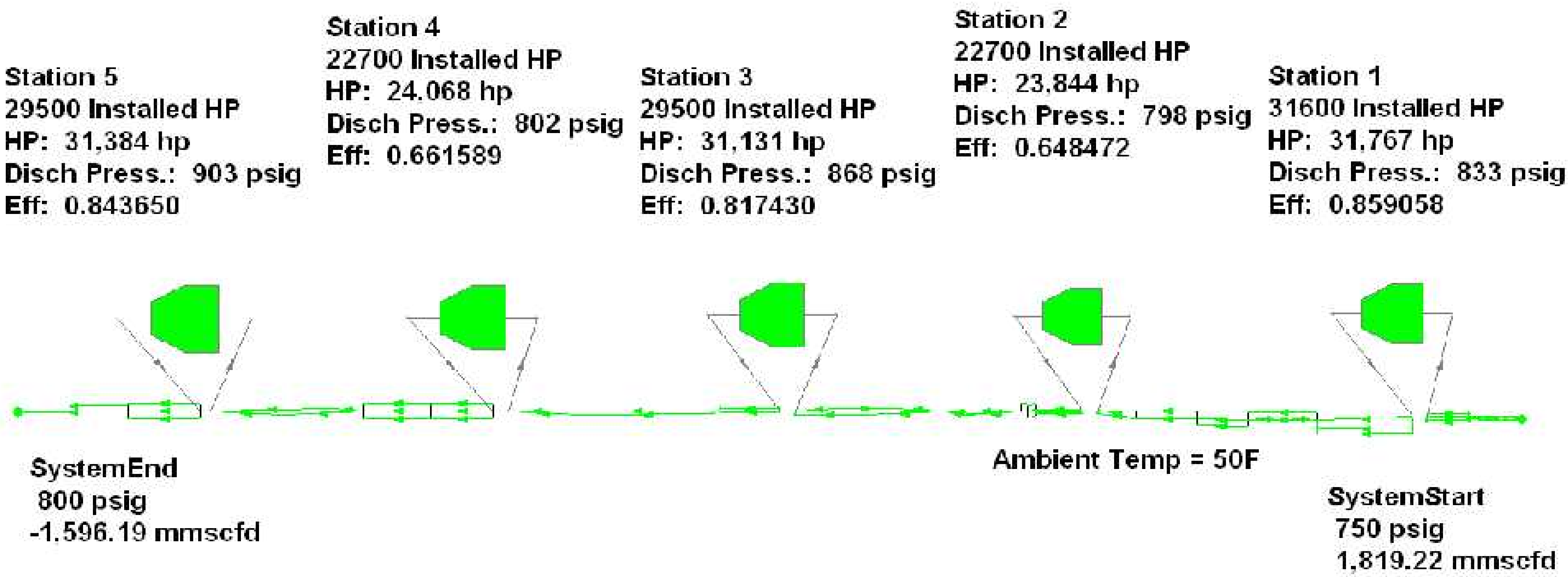
Example 1: Capacity with Installed HP

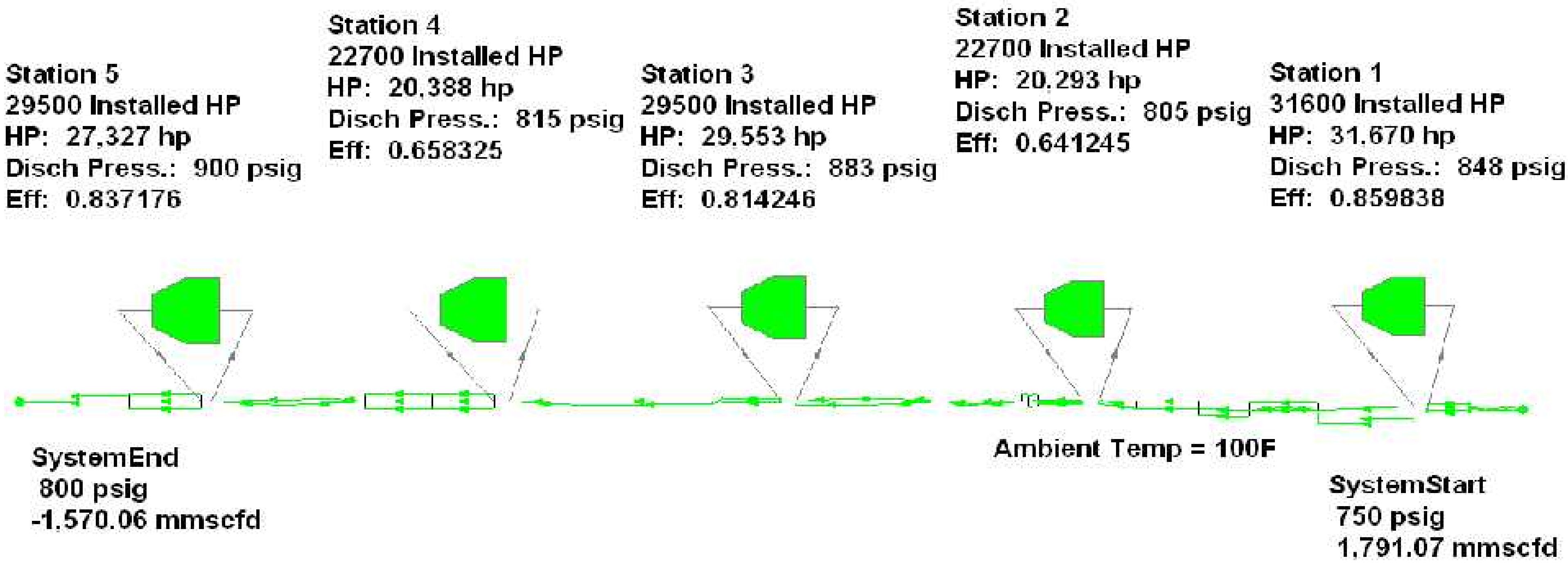


Assume that throughput represents sold capacity

Example 1: Stations with DSC at 50F



Example 1: Stations with DSC at 100F



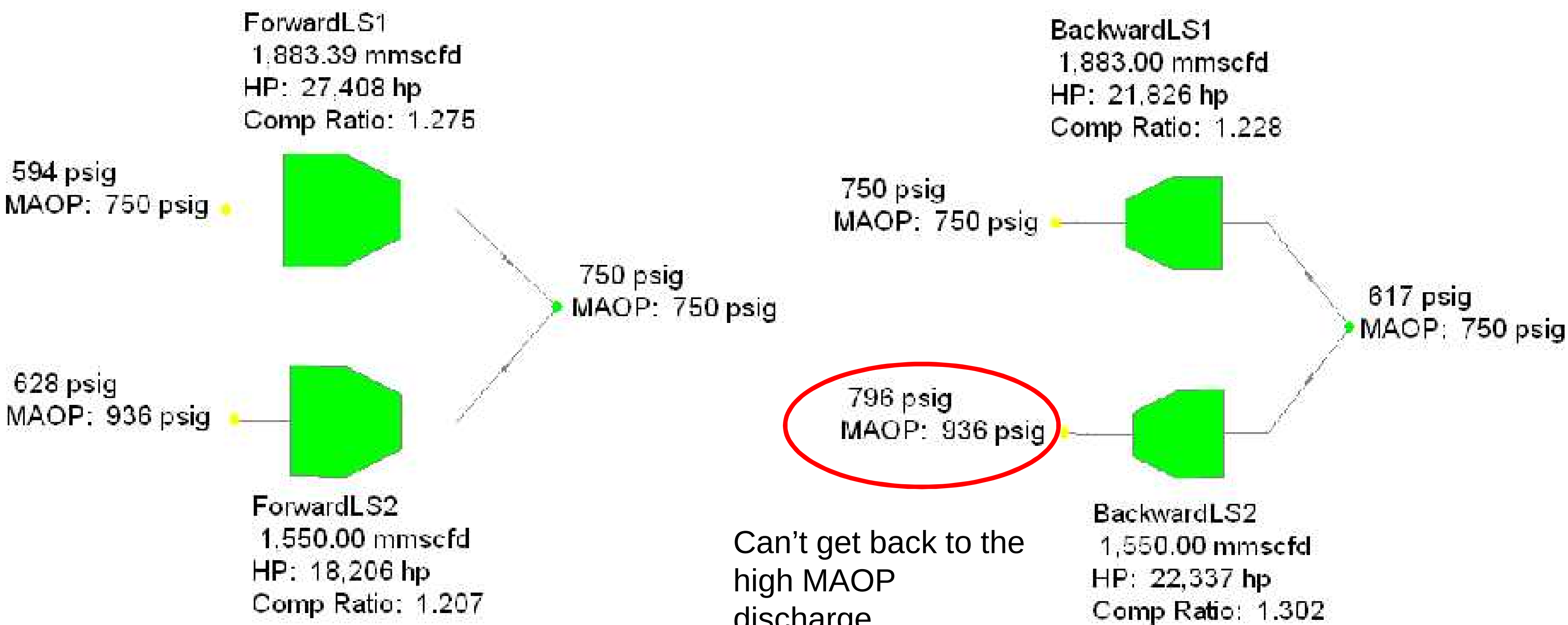
Block Versus DSC: Example 1

Mode	Ambient Temp	Volume	Difference from Block	
Block Station Horsepower	NA	1,608		
Detailed Station Calcs	50	1,596	-12	Oversold
Detailed Station Calcs	100	1,570	-38	Oversold

Turbine/centrifugal stations 2 and 4 have low efficiency and the ambient temperature reduces capacity.

Example 2: Repurpose Existing Station

Total Installed Horsepower 48100
30 Small Recips 1946 - 1966 Vintage
Station was designed for forward flow from high MAOP



Can't get back to the high MAOP discharge

Example 2: Repurpose Existing Station (cont'd)

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STATION IN FORWARD HAUL MODE

45,613 Horsepower of 48,100 installed

Line Service 1

Unit	Torque	HP
1	101%	1,312
2	101%	1,312
4	85%	1,104
5	85%	1,104
6	85%	1,104
7	85%	1,104
8	85%	1,104
9	85%	1,104
15	81%	1,048
16	81%	1,048
17	81%	1,048
18	86%	1,037
19	94%	1,125
20	102%	1,575
21	102%	1,574
22	102%	1,574
23	102%	1,574
24	102%	1,574
26	101%	1,659
28	101%	1,665
29	101%	1,659
		27,408

Line Service 2

Unit	Torque	HP
3	92%	1,194
10	92%	1,194
11	92%	1,194
12	97%	1,942
13	98%	1,952
14	99%	1,983
25	96%	1,578
27	101%	1,668
30	100%	5,500
		18,206

STATION IN BACKHAUL MODE

44,163 Horsepower of 48,100 installed

Line Service 1

Unit	Torque	HP
1	97%	1,255
2	97%	1,255
12	97%	1,945
13	98%	1,955
14	98%	1,959
25	97%	1,605
26	97%	1,605
27	97%	1,608
28	98%	1,613
29	98%	1,623
30	98%	5,402
		21,826

18, 19 are off

Line Service 2

Unit	Torque	HP
3	100%	1,304
4	93%	1,208
5	93%	1,208
6	93%	1,208
7	93%	1,208
8	93%	1,208
9	93%	1,208
10	100%	1,304
11	100%	1,304
15	88%	1,145
16	88%	1,145
17	88%	1,145
20	100%	1,546
21	100%	1,550
22	100%	1,550
23	100%	1,550
24	100%	1,550
		22,337

Different station conditions in the backhaul mode prevent LS1 units from fully loading and 18, 19 can't come on.

KINDER MORGAN

Example 2: Repurpose Existing Station (cont'd)

Example 2 take-aways:

1. Using block horsepower in the forward haul direction would grossly overestimate the station capability (45,613 max usable versus 48,100 installed)

2. Units 18 and 19 cannot run in the backhaul condition. This is most likely because there are units which need additional unloading steps to allow all units to unload enough to let 18 and 19 come on. Different spreads prevent units in Line Service 1 from fully loading.