Restage of Centrifugal Gas Compressors for Changing Pipeline Landscape

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Abstract

Compressor installations in the gas transmission and production industry are subject to changing operating conditions. Many pipelines typically experience seasonal variations in conditions, as well as different daytime and night time operations. A well designed, wide range compressor can provide operational flexibility to handle changing conditions.

Recently, due to shale gas exploration and increased demand from high gas price regions like Europe and Asia, pipeline compressors have been challenged to meet new operating conditions and increased gas capacity. Some existing pipelines are actually reversing gas transmission directions from importing duty to exporting. Even though centrifugal gas compressors driven by gas turbines provide tremendous operational flexibility, the economics of restaging make it a great option to optimize for new conditions.

Restaging considerations are not only driven by gains in compressor efficiency, flow capability or improvements in fuel efficiency. Important considerations also include the ease of restaging, downtime and cost.

This paper discusses the design principles and solutions for centrifugal gas compressors in a changing landscape. The criteria for restage and case studies are also presented.

Introduction

Most turbomachinery OEMs use similar design tools such as CFD, FEA and CAD, manufacturing technologies as well as development testing in their design process, gas compressor products are distinctly different due to different design philosophies. For example, some OEMs design compressors with high efficiency within a narrow range by using low solidity airfoils (LSA) vaned diffusers, while other OEMs design compressors that can be operated across a wide flow range with decent efficiency.

It is important to offer high efficiency gas compressor solution to meet the initial conditions: pressure, temperature, gas composition, flow etc. However, changes in operating conditions such as gas field depletion and natural gas demand increase, are the nature of either production or pipeline compressors applications. It is equally important that the gas compressors can be restaged easily to reduce life-cycle costs and minimize downtime costs.

Gas Compressor Design Philosophies

Aerodynamics, rotordynamics, and mechanical design are the three main technical areas for gas compressor design. Serviceability must also be part of the design criteria, as gas compressors can be in service for many years or even decades. The operating conditions are different for every compression project. Depending on the specific requirements, compressors have to handle different operating parameters such as flows, suction and discharge pressures, suction temperatures, gas compositions, power levels, train configurations, as well as specific customer requirements. To cover all different applications in the oil and gas industry, manufacturers either employ pre-engineered standardized compressors, or more or less customized designs. The difference between these two approaches is not in the sophistication of the design, nor in the capability to show better performance at project specific operating conditions. The main difference is that for customized designs the engineering is performed for a specific project, while for pre-engineered designs, most if not all engineering is performed long before an order is placed. This also includes the testing required to verify the designs.

The standardized design method uses pre-engineered components and pretested designs. For a specific application, aero components are selected from a family of impellers and stators, which have already been tested individually or tested inside of a compressor. This pre-engineered design approach allows for shorter response time as well as proven performance and durability. Compressor aero performance characteristics are predicted and continuously refined through test data. Since a particular family of aero components can cover a large flow range as shown in Figure 1, different staging combinations can replace the existing staging to optimize the operation around new conditions. This is called compressor restage. The rotordynamic stability envelope also has to be verified by testing as shown in Figure 2. The standard design method reduces the risk for both users and OEMs to a minimal level.

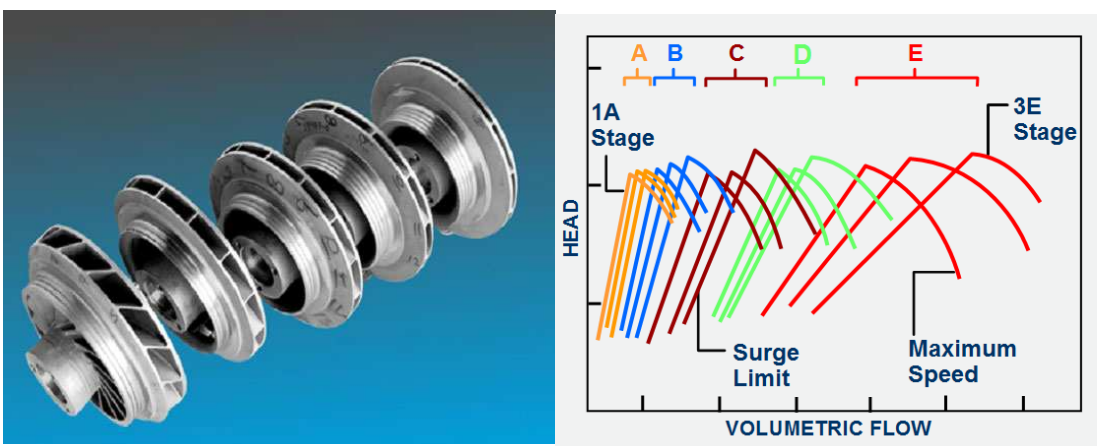

Figure 1 A Typical Impeller Family and Performance Curves

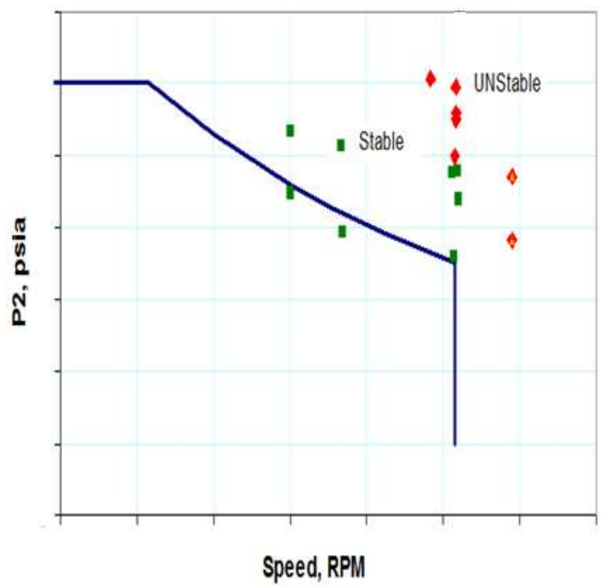

Figure 2 A Typical Stability Envelope

After new equipment is installed, the maintenance requirements for centrifugal compressors are minimal. Two major events have to be considered during the operation of the machines:

- Damage due to foreign object debris (FOD)
- Large change in operating conditions to the extent that the economics favor a restage of the compressor (or, in extreme cases, the addition or removal of a compressor body).

Although rare, damage due to FOD creates the need for an immediate response to restore the capability to operate the station, especially if there is limited or non-existent stand-by capacity. If only stationary components (like inlet vanes) are damaged, the operation may continue. If impellers are damaged, they will have to be replaced. In situations like this, often only the first stage is damaged. Designs that allow quick replacements offer an advantage.

The latter event is usually a planned event. Frequently, operating conditions change gradually, and the point where a compressor restage makes economic sense is predictable.
In pipeline applications, the economics of restage are most likely dictated by the potential improvement in throughput capacity, or the chance to reduce fuel consumption. Avoidance of recycle, as well as avoiding operation in choke, while neither damaging nor disruptive can fall in either category. For oil production applications, the ability to lower suction pressure drives compressor restages. We will address these questions with case studies later in this paper.

Economics also evolve around questions like cost and downtime. While many OEMs recommend the replacement of the entire rotor and stator components if a restage is required, there are opportunities to make use of existing hardware if the compressor design allows.

Modular Design

The modular rotor design was recognized by API (7th edition of API 617) [1]. As shown in Figure 3, stub shafts, impellers, and spacers (if needed) are bolted together to form a modular shaft. The tie-bolt is stretched to a level that the torque can be transmitted through the interface between components.

One of the concepts of standard design method is that all the aerodynamic components from one compressor family must be mechanically interchangeable. Modular design is a way to take full advantage of interchangeable aerodynamic components. With modular rotor design and interchangeability of aero components, the compressor can have thousands of combinations within a common mechanical design.

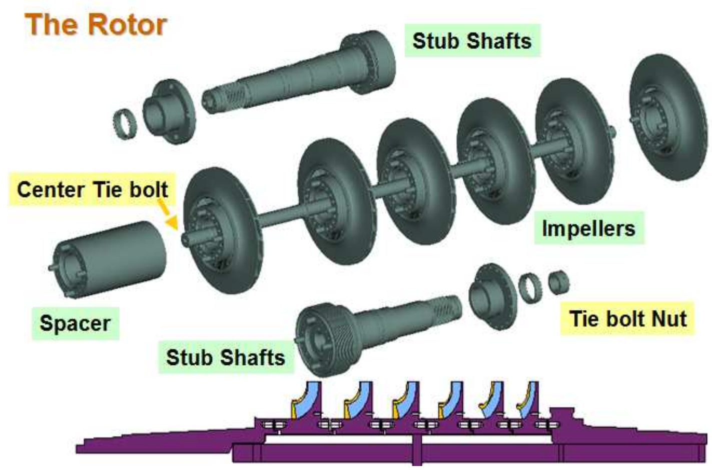

Figure 3 A Typical Modular Rotor

Traditionally, the industry has used solid shaft rotor construction methods. The impellers are shrink-fitted onto a solid shaft. The rotor is centered by two halves of stationary components, which is called a split case design. When the compressors run as designed under clean gas and at design conditions, the difference between solid shaft and modular shaft designs is negligible. But when the compressor requires overhaul or restage, there is a significant difference in terms of cost, lead time, and sustainability between both designs. Modular rotor design is easier to disassemble since it does not require the expensive and difficult shrink-fit process. Thus, it is more restage and overhaul friendly. The impellers that can be reused are easily salvaged to reduce the cost and minimize the down time for customers. The impellers that are displaced by the compressor restage can be put into storage for use in future restages.

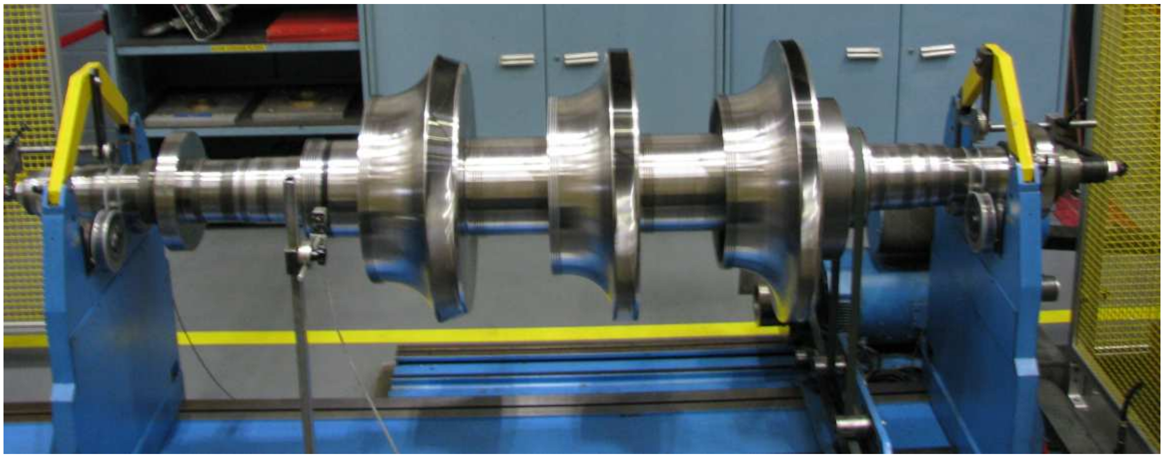

Figure 4 A typical Modular Rotor on the Balance Machine

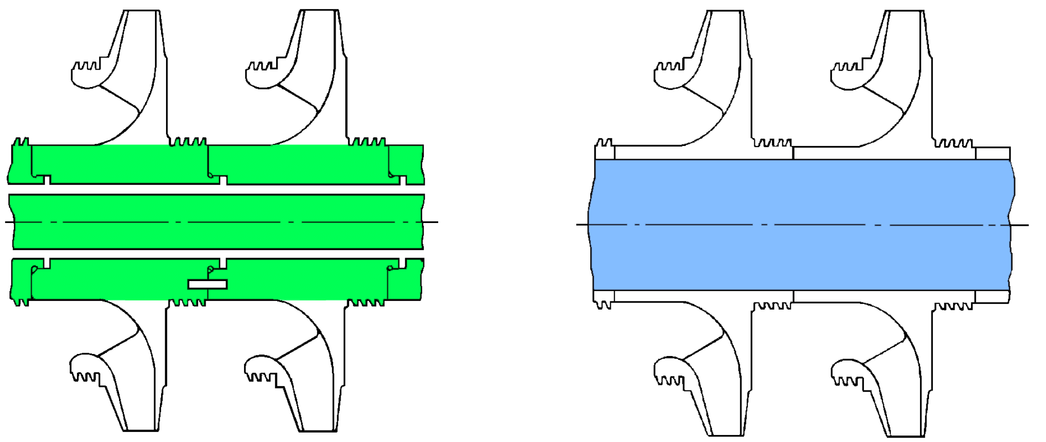

Figure 5 Comparison of Modular rotor (Left) to Solid Shaft (right) rotor

The main concern with the modular rotor is rotor stiffness. In 2009, J. Moore and A. Lerche [2] evaluated an industrial tie-bolt rotor against an equivalent solid rotor and concluded that modular rotor design met the required API separation margin criteria. Not only that, the solid rotor has higher amplification factor and unbalance response. The solid rotor has about 10% lower log decrement value and lower stability threshold than the modular rotor.

Engine matching

The centrifugal compressor and its driver have to be matched regarding speed and power consumption. When using electric drives, the match between compressor and driver is done via a gearbox, which also adds torsional damping to the system. When matched with a two shaft gas turbine, it is good practice to match the compressor speed at the design point or the rated point with the power turbine operating in the vicinity of its optimum speed. In general, the more powerful the gas turbine, the slower its power turbine wants to run. For example, while a 6000 hp class gas turbine has a maximum power turbine speed of 16500 rpm, a 20000 hp class gas turbine may have a maximum power turbine speed roughly half that number.

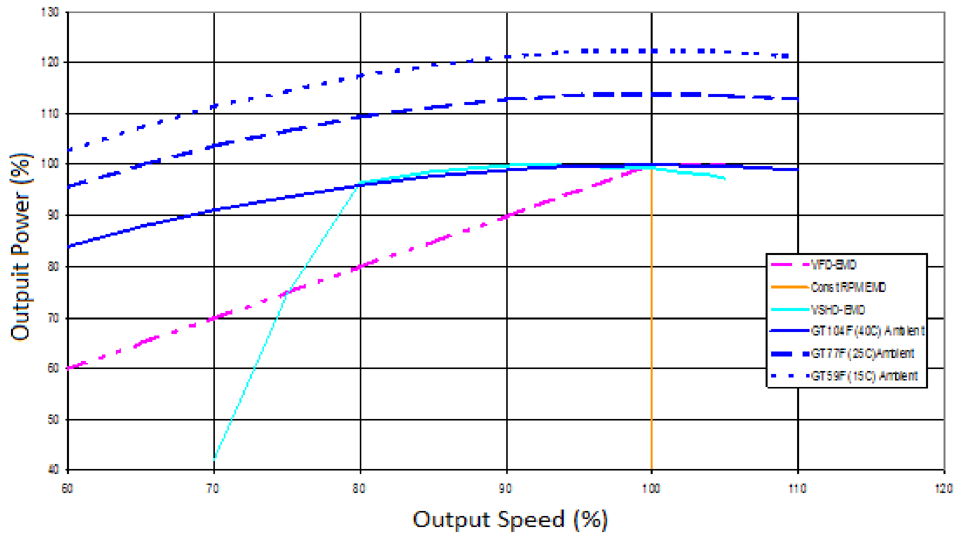

Figure 6: Speed –Power Characteristics of Compressor Drivers

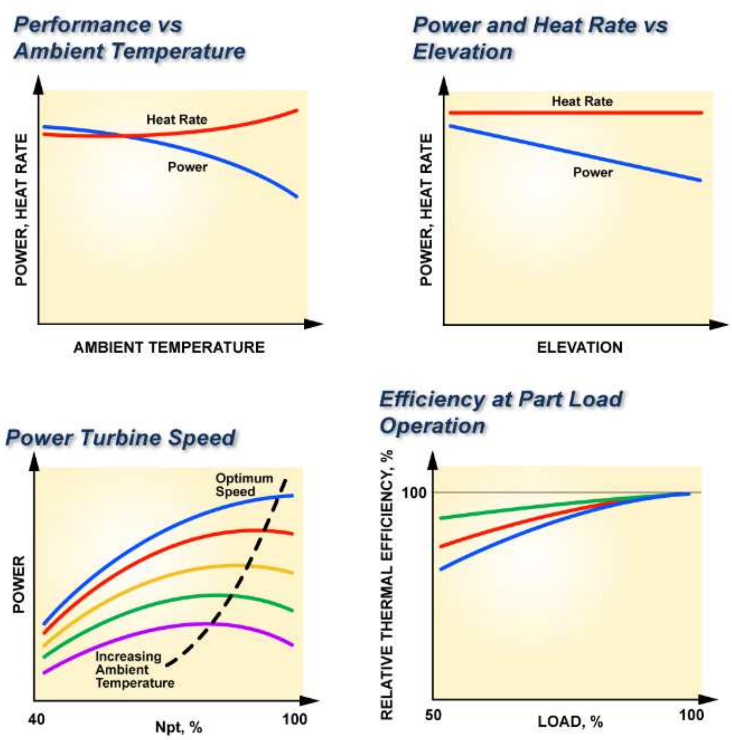

Figure 7: Gas Turbine Performance Characteristics.

It is also important to understand the speed-power characteristic of different drivers: While a power turbine actually produces more torque at low speeds than at high speeds, a VFD driven electric motor produces constant torque at best (Figure 6).
If the driver is a gas turbine, a number of issues need to be considered: Gas turbine uprates at engine overhauls may provide more power than originally installed. One manufacturer over the years brought an engine that was originally introduced at 10,000 hp in several steps to a power

level of 16,000 hp today. The gas turbine provides far more power on cold days. It is also advantageous to operate the gas turbine fully loaded.

Principles of Gas Compressor Restage

Gas Compressor Performance

In reality, Gas conditions always change in either pipeline or production compressors. If conditions oscillate around the design point, for a typical wide range compressor, no restaging is needed. But when conditions change in one direction away from the design point, compressor restaging should be considered.

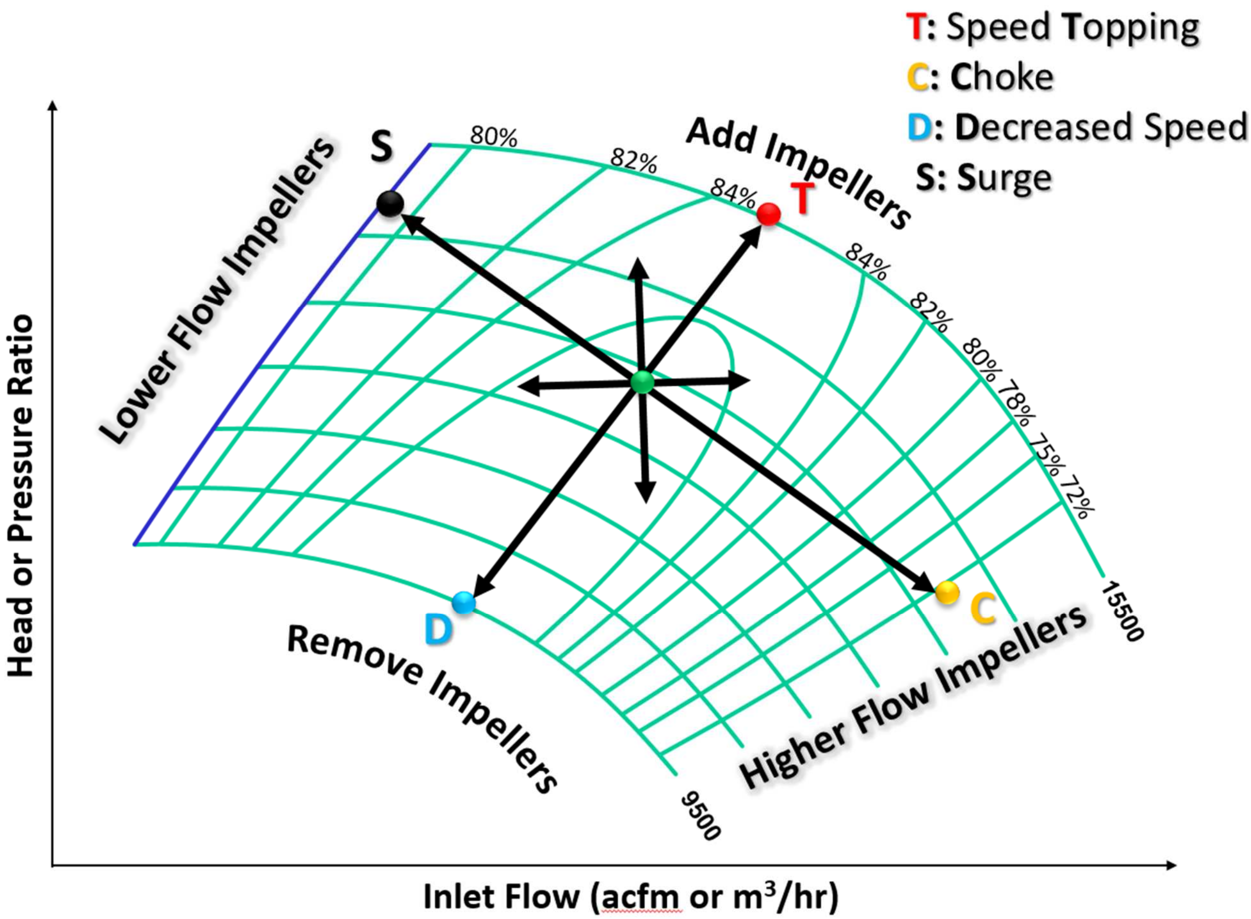

Figure 8 A Typical Multistage Compressor Flow-Head Map

Essentially, 6 key parameters define the gas compressor performance: Inlet/discharge temperature/pressure, flow, and speed for a given gas composition. Gas properties such as specific gravity, specific heat ratio, specific heat and compressibility also affect the compressor performance. Changes of above mentioned parameters may require speed and power change.

$$H_{isen} = \frac{29.27}{SG} \times \frac{k}{k-1} \times Z \times T1 \times \left[\left(\frac{P2}{P1} \right)^{\frac{k-1}{k}} - 1 \right] \qquad (1)$$

$$H_{actual} = Cp \times (T2 - T1) \qquad (2)$$

$$\eta_{isen} = \frac{H_{isen}}{H_{actual}} \qquad (3)$$

Where,

- -Cp is specific heat ratio at constant pressure
- -H_{isen} is isentropic head
- -H_{actual} is actual head
- -k is specific heat ratio
- -P1/P2 is inlet/discharge pressure
- -T1/T2 is inlet/discharge temperature
- -SG is specific gravity
- -Z is compressibility factor
- –η_{isen} is isentropic efficiency

The temperature, pressure, and gas properties are combined into two terms: isentropic head and isentropic efficiency as shown in equations 1 to 3. The two combined parameters plus flow and speed are the 4 key parameters to evaluate the compressor performance, which is typically shown in a Head-Flow map (Figure 8).

The effect of temperature, pressure, and gas composition mainly move the operating point in the T (speed Topping) or D (speed Decreasing) direction as these parameter mainly affect head as shown in Equation 2.

When suction temperature is increased from original design point, more head will be created for the same pressure ratio and higher speed will be required to move the new flow point to T direction. The temperature also changes the map slightly. Higher temperature tends to tilt the map in counter-clockwise direction.

Suction pressure moves the point in the T or D direction also. For a typical declining gas field, the suction pressure reduces over time. To reach the same discharge pressure, more pressure ratio requires higher speed and more flow will pass through the compressor as gas density reduces. The point moves in the T direction. There are also cases where the suction pressure is increased. In this case, the point moves to the D direction, as the required head reduces. Same principle applies to discharge pressure: when it increases, the pressure ratio increases with the same suction pressure. More head is needed and the point moves in the T direction. If pressure ratio decreases, the point moves in the D direction.

Gas composition will change over time, especially for production applications. Heavier gas (larger Specific Gravity) requires less power to reach the same pressure ratio, thus decreases the speed requirements. Since the flow does not change much, the flow point moves vertically down. Heavier gas also tilts the map in clockwise direction. Therefore, the flow point moves in the D direction. Lighter gas behaves opposite and the flow point moves in the T direction.

The flow change effect is easier to explain. If more flow is needed, the flow point will move in the C direction to the Choke side of the map. If more flow is needed at constant power consumption, the flow point moves downward to the choke side in the C and D direction. If more flow is required at constant head, the flow point moves horizontally to the choke side.

The compressor efficiency is mainly a function of flow. When more flow is needed, the flow point moves in the C direction, and efficiency drops fast from the best efficiency point. At lower isentropic efficiency, the discharge temperature increases quickly at the same level of head. More power is lost due to a less efficient compressor. If less flow is needed, the flow point moves in the S direction to the Surge side of the map, where decay in efficiency is less rapid. As typical in production applications, insufficient flow may put the operating point to the left of the surge line, requiring the anti-surge valve to open to protect the compressor from surge. In this situation, power is wasted by recycling the gas through the compressor.

Gas Compressor Restage Principles and Value Proposition

The energy balance of the whole power train from engine (or other drivers) to the compressor can be expressed in Equation 4. The isentropic head and efficiency as well as the flow of the gas compressor were discussed in above session. The standard flow is a function of actual flow under standard conditions.

The power needed to produce the head is also affected by the engine efficiency and mechanical efficiency. Mechanical efficiency is relatively constant and engine efficiency is mainly a function of the speed.

$$Power = C \times \frac{SQ}{\eta_{isen}\eta_{mech}} H_{isen} = FuelEnergy\,\eta_{engine} \qquad (4)$$

Where,

Power is driver (engine) output power
C is a constant
SQ is standard flow
η_{mech} is mechanical efficiency
η_{engine} is engine efficiency

At the design point, the efficiency terms are optimized so that the compressor can produce the required flow and head with minimum power. When the flow point stays away from design point for an extended time, the compressor or engine are running less efficient, which requires more power. The purpose of a gas compressor restage is to re-optimize the compressor staging in order to maximize efficiency at the new conditions to minimize the power consumption or maximize flow, head, or both.

Increasing discharge pressure for gas injection and gas gathering for a declining field (lower suction pressure) are two typical scenarios in which the flow point moves in the T direction. The compressor has to be rotated faster to keep the same pressure ratio until eventually power turbine or compressor itself reaches maximum speed. This is a typical speed toping case. By adding additional stages to the compressor, the required speeds can be reduced to generate the required pressure ratio, or the speed can remain the same in order to generate higher pressure ratio. For gas gathering in a declining field, that means extended field life. For gas injection, higher pressure means more oil production. These are two cases where the investment for restaging can be recovered quickly. For example, if there is 8% extra power left due to speed topping, by restaging the compressor, the site can produce extra 8% flow or 8 % head increase. For a typical 12000 hp engine driven pipeline compressor, 8% flow increase is approximately 30 mmscfd of natural gas. The incremental revenue due to restaging correlates to about $90,000/day based on the gas price of $3/mmbtu.

When the operating point moves in the D direction, the compressor is running at much slower speed. Normally, the compressor speed is designed to require the power turbine to run over 90% of max speed, in order to reach highest efficiency levels. The engine efficiency drops as speed reduces. When the operating point consistently requires engine speeds lower than optimum levels, removing one or two stages will increase the required compressor speed and improve engine efficiency. This type of restaging reduces engine fuel consumption. By restaging the compressor, up to 10% fuel can be saved. For a 3.5MW turbine, 10% fuel saving is about $80,000 a year by assuming gas price of $3/mmbtu.

During seasons of high flow demand, it is normal to require maximum flow from a compressor. In this scenario the running point moves in the C direction, where efficiency drops quickly. Although the compressor may not be physically choked, the available power can limit the capacity throughput, and in some instances, a package may not be able to deliver the required flow. For this case, typically, smaller flow stages are replaced by larger flow stages. Figure 9 shows how restaging to larger flow stages changes the performance map (Green for restaged compressor performance map) to better match the conditions. Both compressor efficiency and flow capacity are improved. For example, if the restaging can improve the efficiency from 75% to 81% (6% improvement), the flow can be increased by 8% according to Equation 4. This restaging is just like the speed topping case above that can increase the revenue by $90,000 a day for a 12000 hp engine driven compressor.

Opposite to a choke situation, when there is not enough gas, the point moves in the S direction. When the compressor cannot get enough flow, the anti-surge valve opens to avoid surge and the compressor runs in recycle mode. A portion of compressed gas will be cooled to feed back to the compressor. This is the only way to keep the compressor out of surge. Surge can cause violent vibration and catastrophic damage to the compressor. The energy going consumed by recycling gas is wasted and also extra energy is needed at site to pump cooling water or drive fans for gas cooling. This is equivalent to drop the compressor efficiency. Restaging can solve this problem by replacing higher flow stages with smaller stages to accommodate the lower volumetric flows. For a 3.5MW industrial gas turbine driving a compressor with 20% recycle flow, given gas price of $3.00/mmbtu gas price and 300 days of operation, the savings can be potentially up to $130,000 a year. If this application is for oil production, the saved 20% power can be used to increase head about 20%. The increased oil production could pay back the restage investment within months or even weeks.

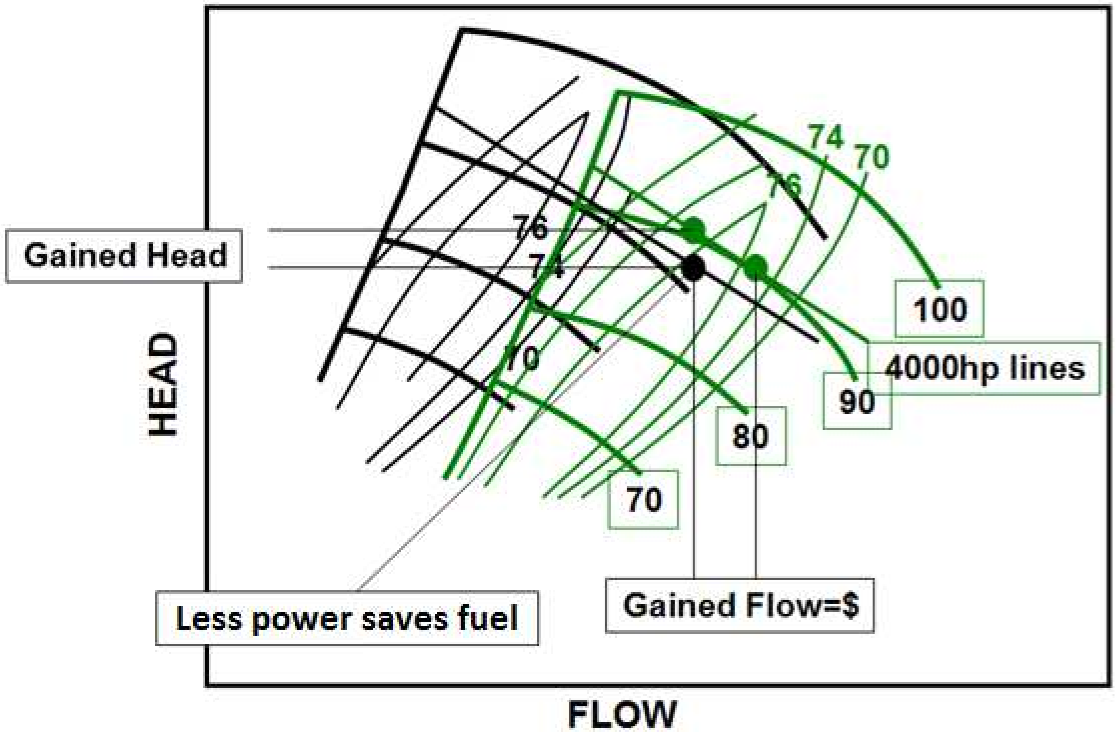

Figure 9 Gas Compressor Restage Principles

Besides economic reasons, running in recycle mode could cause high discharge temperatures if insufficient cooling is supplied in deep recycle mode. Dry gas seals, balance piston Babbitt and Anti Surge Valves can be damaged in periods of extended recycling.

In summary, the main benefits for restaging are: more oil/gas production, less fuel consumption, and better equipment health.

<u>Restage Criteria</u>

A restage is generally recommended at the next time of overhaul if the investment can be recovered within 5 years. If the restage investment can be paid off less than 1 year, restage should be considered immediately.

The economic study of payback period requires interaction between the user and the OEMs. A study based on 379 recently sold compressor restages by Solar Turbines is described below. Four parameters stood out as good indicators for restaging: inlet flow coefficient (Φ), isentropic head coefficient (Ψ), inlet pressure (P1), and the required power. The changes between the conditions before restaging and the original design were calculated. The detailed criteria for each parameter are shown in Table 1 below. The <25%, 25%-50% and >50% ranges identify the percentages of the 379 compressors restaged. For example, less than 25 % of compressors were restaged when Suction Pressure changed by less than 5%, but more than 50% of the compressors were restaged when suction pressure changed by 15% or more. These variation change regions thus established the “trigger points” for restage recommendations. Roughly speaking, for power, suction pressure, and head coefficient, the trigger point for restage consideration (Yellow warning) for next overhaul is when the parameter drifted 5% to 15%. If they drifted more than 15%, that is the trigger (Red warning) for immediate restage consideration. The flow coefficient trigger points are 25% for next overhaul and 50% for immediate consideration. If any of the Red Warning is triggered, the compressor should be restaged. If all four Yellow Warning are triggered, the compressor should also be restaged.

The other general rule is that a compressor restage is recommended when the efficiency is less than 6% of peak efficiency and power is a limiting factor. Regaining this 6% efficiency with a restage typically results in 8% or more flow gain.

Percent Change	<25%	25%-50%	>50%
Φ_1	<15%	15%-31%	>31%
Ψ_1	<5%	5%-19%	>19%
P_1	<5%	5%-15%	>15%
HP	<3%	3%-13%	>13%

Table 1 Trigger Points for Restage Parameters

$\Phi_1 = \dfrac{Q_1}{(D_2)^3 N}$ is the inlet flow coefficient, for compressors, using the first compressor inlet flow coefficient

$\Psi_{isen} = \dfrac{H_{isen}}{(D_2 N)^2}$ is the isentropic head coefficient for single body compressor,

$\Psi_{isen} = Cp \dfrac{T1}{(D_2 N)^2}\left[\left(\dfrac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right]$ for compressors using the total pressure ratio and the first compressor speed and impeller tip diameter

Case Study 1: Extra Capacity for Pipeline Application

This package was original sold in 1998 for a pipeline application in the US. The original design points are listed in the first column of Table 2 below and are marked as Point 1 on the compressor maps in Figure 10.

The customer wanted to relocate this package from the existing site to another due to increased demand at the new site. The new site conditions had lower suction and discharge pressure requirements (2.7 and 17.12% respectively) but the flow demand at new site was 67.85% higher than current design conditions. Column 2 shows the max flow capability of the current staging for new site conditions at max power. The current staging could only provide 959.16 MMSCFD of flow at max power, 31.47% less than the requirement. All four key parameters were in red zone indicating a restage would be required to perform the new duty point. The last column shows the delta between the Original Design Point and New Design Requirement for Original Staging. Besides all four key parameters being in the red and current staging not being able to perform the required flow duty, the efficiency with Max power and original staging would have been 46.91% lower than original.

	Original Staging		Restaged	Delta Max Power To Design for Original Staging
	Design Point	Max Power		
Point on Map	1	2	3	
Phi	0.045	0.081	0.114	44.88%
Psi	2.043	0.747	0.644	-173.53%
P1 (PSIA)	885.00	861.77	861.77	-2.70%
HP Total (HP)	6225.00	11025.00	11080.00	43.54%
Efficiency (%)	85.50	58.20	84.50	-46.91%
P2 (PSIA)	1185.00	1011.77	1011.77	-17.12%
SQ (MMSCFD)	450.03	959.16	1399.57	53.08%
Flow (ACFM)	4490.41	10298.01	15026.39	56.40%
P2/P1	1.34	1.17	1.17	-14.05%

Table 2: Change in Operation

In 2013 the compressor was restaged to higher flow staging and the number of stages was reduced from 2 to 1 to increase speed, efficiency and flow capability. The new staging is 26.3% more efficient than the original staging at max power and also provided a flow increase from 959.16 MMSCFD to 1399.57 MMSCFD at max power conditions. Figure 10 shows the new performance map in green and the old performance map in red along with all three points from Table 2. As it can be seen, the restage allowed the compressor to have increased flow throughput while maintaining operation in peak efficiency zones.

This is a typical pipeline application restage where increased flow demand along with maximum power consumption and efficiency gain, a compressor restage can be paid back in weeks if not days if the customer owns even a percentage of the gas.

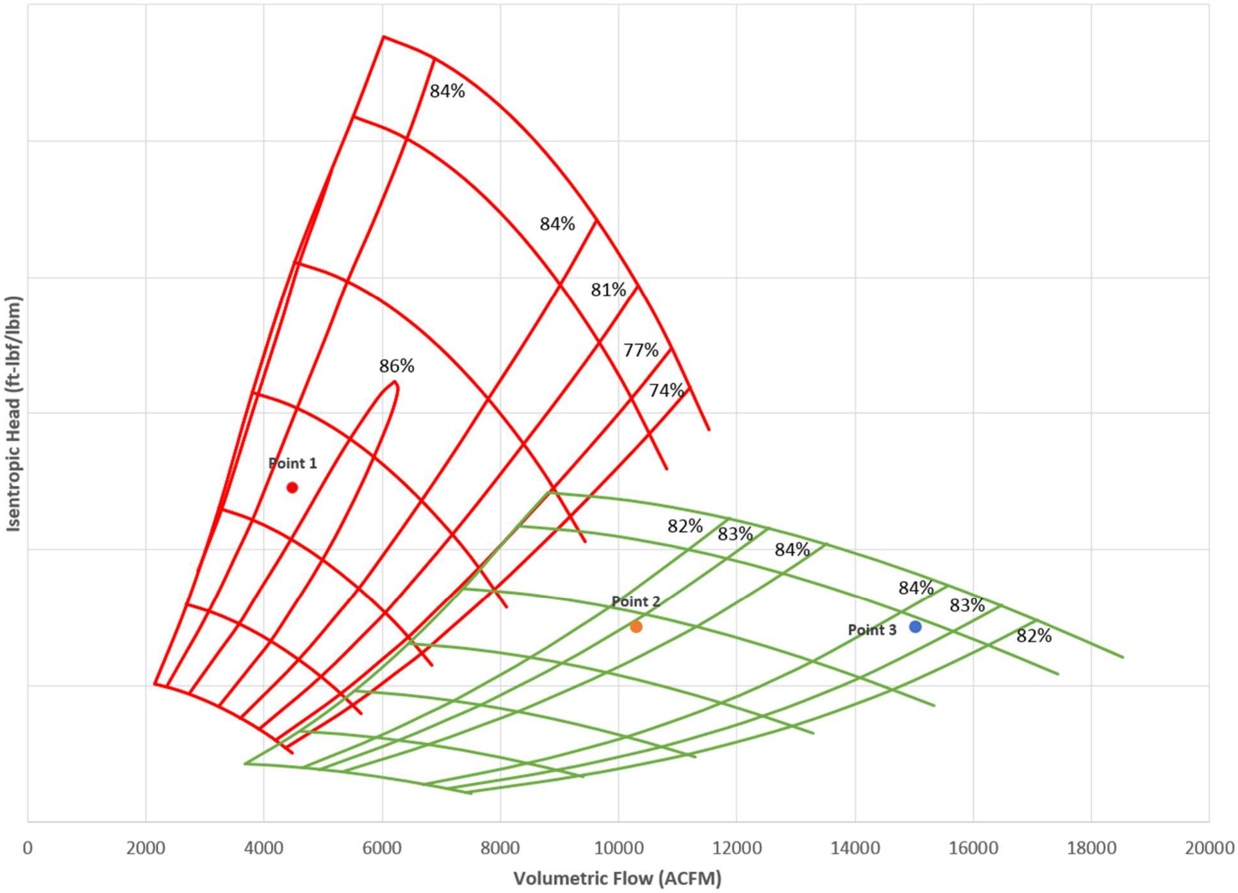

Figure 10: Existing and Restage Staging Performance Map for Case 1

Case 2: Package Relocation to Increase Station Capacity and Discharge Pressure

In today's evolving marketplace, the need for additional gas compression has increased. Although the purchase of new turbomachinery equipment is preferred in some instances, the reallocation of unused or standby turbomachinery packages provides opportunities to reduce capital investment, delivery schedules and sourcing activity. From an inventory management perspective, package relocation increases unit asset value, drives down operational costs and helps maintain optimum inventory levels. It also allows for performance improvements and increased operational flexibility for either current or future conditions. In both gas gathering and transmission applications, reallocation of existing turbomachinery packages can be a very convenient and economical method to meet new site conditions; particularly increasing station discharge pressure and gas throughput. Similar to the sourcing of new equipment, close collaboration with the OEM is pivotal in ensuring that the relocated package is properly sized, upgraded accordingly, and most importantly, makes sound financial sense. This case study provides an example of the various benefits of package relocation.

In gas gathering applications, there are generally multiple sources of gas volumes that change over time. The addition of new wells or gas streams may sometimes be prohibited if the available power at site is not sufficient. This particular station was designed with 3 stages of compression, with two primary sources of gas. Original design conditions can be seen in Table 3 below. New process conditions at site required an additional 19 MMSCFD of side stream gas flow, and an increase in station discharge pressure to 730 PSIA. Although the existing gas compression equipment had a very wide range of operating efficiency, the combination of increased gas flow and discharge pressure was not achievable primarily due to the available power at site. The maximum achievable flows and pressure with the existing packages can also be seen in Table 3.

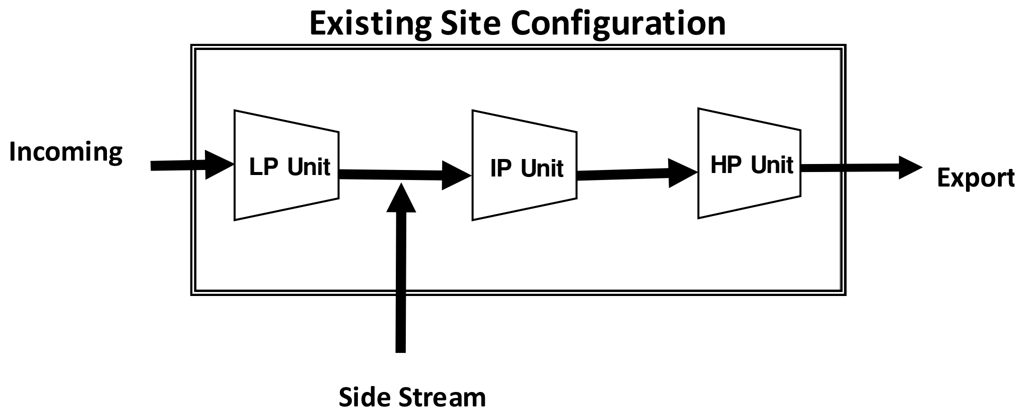

Figure 11: Existing Site Configuration

			Existing Equipment	
	Design	Site Goal	Max P2	Max Flow
P1	36.6 PSIA	45 PSIA	45 PSIA	45 PSIA
P2	606.6 PSIA	730 PSIA	730 PSIA	490 PSIA
SQ	61 MMSCFD	80 MMSCFD	50 MMSCFD	67 MMSCFD

Table 3: Original Design Conditions and Existing Station Maximum Capacity

Close coordination with the aftermarket applications team identified an existing package that could be reallocated as the 4th stage of compression to increase both discharge pressure and gas flow at site, as shown in Figure 12. The addition of a 4th stage of compression decreased the head requirements across the existing units, which allowed for increased gas throughput with the same available horsepower. The 4th stage would serve as a booster to meet the required station discharge pressure.

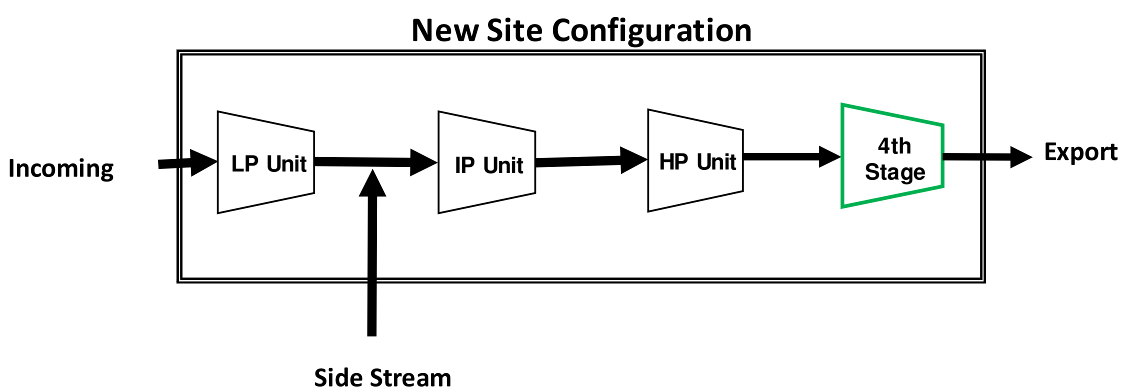

Figure 12: New Site Configuration

	Goal	With 4th Stage
P1	45 PSIA	45 PSIA
P2	730 PSIA	730 PSIA
SQ	80 MMSCFD	71 MMSCFD

Table 4: New Station Maximum Capacity

The gas compressor on the 4th stage package was originally designed for very different process conditions as shown in Table 5. The new site conditions required a much higher suction pressure, which decreased the inlet flow coefficient by 58.4%. Analysis of the existing staging showed that the 4th stage package would need to recycle 81% of the flow to maintain positive surge margin. The new design point plotted on the original performance curve can be seen far to the left of the surge line. The amount of recycling needed would increase fuel consumption and site emissions. At higher ambient temperatures, the required amount of recycling through the 4th stage package would not be possible due to limited power.

Parameter	Original	New
P1 (PSIA)	267	400
P2 (PSIA)	432	731.6
Power (HP)	3348	2454
SQ (MMSCFD)	67.86	71
Inlet Flow (ACFM)	4212	1680
Head (FT-LBF/LBM)	20204	22838

Parameter	% Change
P_1	49.8%
Power	41.7%
Φ_1	58.4%
Ψ_1	23.2%

Table 5: 4th Stage Package Conditions

A gas compressor restage of the 4th stage package optimized around the new site conditions can be seen in the green performance curve below in Figure 13. The selected staging increased package performance at the new conditions, and also provided enough turndown and speed margin to increase gas volumes beyond 71 MMSCFD. Additional upgrades to the IP and HP compressors will allow the customer to reach the 80 MMSCFD target. Keeping in mind the large increase in gas flow and discharge pressure at site, the limited number of package upgrades needed to meet the new conditions were very minimal. The reallocation significantly reduced the capital investment needed to meet the new conditions, and improved project timeline significantly.

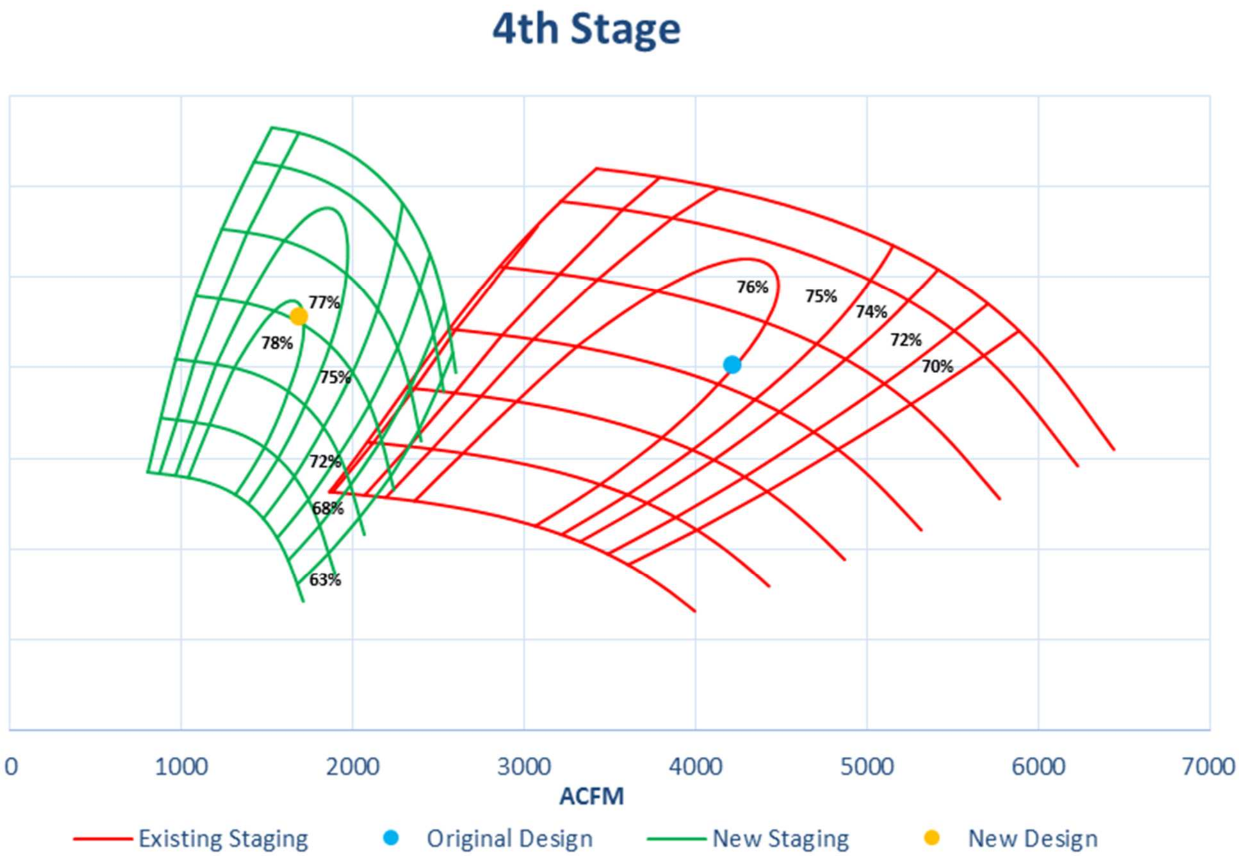

Figure 13: 4th Stage Package Original vs New Staging

Conclusions

This paper presents the gas compressor design principles and restage fundamentals. A simple set of criteria for the economic evaluation of restages are suggested and evaluated through real case studies.

Two case studies represent 2 typical scenarios for restage: Increased capacity for pipeline applications and package relocation to increase station capacity with minimal capital investment. For all these cases, the investment to restage can be paid back fast.

Reference

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