离心压缩机的调级原理和案例研究

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摘　要:作为供气和产油的主要设备之一,压缩机被广泛应用于石油和天然气工业领域。由于油田和天然气管道中气量压力和温度等不停变化,压缩机面临工况改变的挑战。虽然离心压缩机,特别是燃气涡轮机驱动的压缩机,可以在不同工况下有很大的灵活性,但通常为了适应长期非设计工况,压缩机调级是更经济的选择。压缩机调级时不仅需要考虑压缩机效率、输气能力或者整机燃料效率的改进,还应考虑调级成本和调级导致的停机对输气的影响。讨论了具有低生命周期成本的离心压缩机的设计原理和解决方案,提出了压缩机调级和案例研究的标准。

关键词:压缩机;优化;调级

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Gas Compressor Restage Principles and Case Studies

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Abstract: Compressor installations in the oil and gas industry are subject to continuously changing operating conditions. While centrifugal compressors, especially when driven by gas turbines, provide a tremendous flexibility, it is often economic to restage the equipment to optimize for new process 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 compressors with a low life-cycle cost. The criteria for restage and case studies are also presented.

Keywords: Compressor; Optimization; Restage

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0　Introduction
前言

Although most turbomachinery OEMs are using similar design tools like 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 airfoil (LSA) vaned diffusers, while other OEMs design the compressors that can be operated in a wider flow range with decent efficiency.

虽然大多数涡轮机设备厂家在其设计过程中均使用相似的设计工具,如计算流体力学(CFD)、有限元分析(FEA)和计算机辅助设计(CAD)、制造技术和测试方法,但由于设计理念不同,压缩机产品有着明显的不同。例如,一些设备厂家通过使用有叶式扩压器(LSA)来提高设计工况点的效率,这一设计会减小压缩机变工况的工作范围;而其他设备厂家采用无叶式扩压器,这样的设计可以扩大压缩机变工况的工作范围。

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

在压缩机选型的时候,一般第一要考虑的是选择合适的压缩机以满足初始的工况条件:压力,温度,气体组成,流量等。然而,在油气行业,工况条件的改变是必然会发生的,例如气田消耗和天然气需求增加。所以,压缩机使用期间的调级也是要考虑的一个重要因素。压缩机调级造价和容易程度对于生命周期成本有着至关重要的影响。

1　Principles of Gas Compressor Restage
压缩机调级原理

1.1　Gas Compressor Performance
压缩机的性能特性

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.

实际上,不管是产油还是供气,压缩机的运行工况都是不断变化的。如果工况在设计点周围振荡,对于典型的压缩机而言则不需要调级,但是当工况沿远离设计点的一个方向变化时,就应考虑压缩机的调级。

Essentially, six 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.

基本上,气体压缩机的性能可由 6 个关键参数来定义:压缩机入口、出口的压力和温度,流量和转子转速。气体性质,如比重、比热比、比热和压缩性,也会影响压缩机的性能。上述参数的改变可能要求转速和功率做出相应改变。

$$H_{isen} = \frac{29.27}{SG} \times \frac{k}{k-1} \times Z \times T_1 \times \left[\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] \quad (1)$$

$$H_{actual} = Cp \times (T_2 - T_1) \quad (2)$$

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

Where, Cp is specific heat ratio at constant pressure; H_{isen} is isentropic head; H_{actual} is actual head; k is specific heat ratio; p_1/p_2 is inlet /discharge pressure; T_1/T_2 is inlet / discharge temperature; SG is specific gravity; Z is compressibility factor; η_{isen} is isentropic efficiency.

式中:Cp 为定压比热;H_{isen} 为等熵压头;H_{actual} 为实际压头;k 为比热比;p_1 /p_2 为入口 /出口压力;T_1 /T_2 为入口 /出口温度;SG 为比重;Z 为压缩因子;η_{isen} 为等熵效率。

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 four key parameters to evaluate the compressor performance as shown in a Head-Flow map(Figure 1).

如式(1)~(3)所示,温度、压力和气体性质可以合并成 2 个参数:等熵压头和等熵效率。2 个新组成的参数加上流量和转速是用于评估压缩机性能的 4 个关键参数,可以形成压缩机性能曲线(图 1)。

The effect of temperature, pressure, and gas composition mainly move the operating point inthe T(speed Topping) or D (speed Decreasing) direction as these parameter mainly affect head as shown in Equation 1.

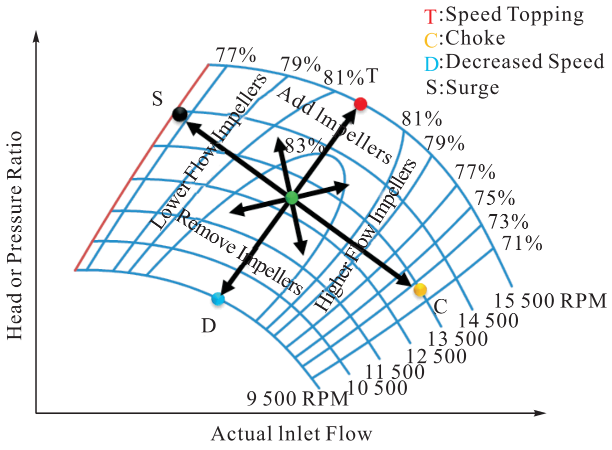

Figure 1 A Typical Multistage Compressor Flow-Head Map

图 1 典型多级离心压缩机性能曲线

温度、压力和气体组成的影响主要在 T(最大转速线)或 D(低速区)方向上移动工况点,因为如式(1)所示,这些参数主要影响压头。

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.

当压缩机入口温度从原始设计点增加时,相同的压力比将会导致压头和压缩机速度的增加,这样新的工况点就会移向 T 方向。较高的入口温度会使得性能曲线沿逆时针方向稍微倾斜。

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. More flow will pass through the compressor as gas density reduces. Thus, the operating point moves in the T direction. There are also cases where the suction pressure is increased. In this case, the design 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.

压缩机入口压力有时候也沿着 T 或 D 方向移动。典型的气田在寿命中后期时压力会降低,为了达到相同的出口压力,需提高压缩机压比,这样就会要求压缩机转速提高。气田压力降低也会导致入口气体密度降低,对应的容积流量就会增加,因此工况点就会沿着 T 方向移动。当然,有时新的气田来气会增加目前气站的入口压力,在这种情况下,随着所需等熵压头的减小,工况点会向 D 方向移动。相同的原理也适用于压缩机出口压力:当其增加时,压力比也增加,就需要更高的等熵压头,工况点在 T 方向上移动。如果压力比降低,则在 D 方向上移动。

Sometimes, gas composition changes 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.

气体的组成成分也会随时间变化。较重的气体(较大的比重)需要较小的功率以达到相同的压比,从而降低对压缩机转速的要求。由于流量变化不大,工况点就会垂直向下移动。较重的气体还沿顺时针方向倾斜压缩机的性能曲线。因此工况点沿 D 方向移动。较轻的气体情况相反,会导致工况点在 T 方向上移动。

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.

压缩机流量变化效应容易解释,如果需要更多流量,工况点将沿 C 方向移动到压缩机的堵塞工况区。如果在定功率下提高流量,工况点将沿 C、D 方向移动到堵塞工况区。如果在定压头提高流量,工况点将水平移动到堵塞工况区。

Compressor efficiency is mainly a function of flow. When more flow is needed, the flow point moves to the choke side. The efficiency drops fast from the best efficiency point. At lower efficiency, the discharge temperature increases quickly. More power is lost due to a less efficient compressor. If less flow is needed, the flow point moves to the Surge side of the map. The efficiency reduction is less rapid in the S direction. But 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.

压缩机效率主要随流量变化。当需要更多流量时,

工况点移动到堵塞工况区，效率从最佳效率点快速下降。效率的降低导致更多功损变为热量，提高了出口温度，如果流量降低，工况点移动到 S 方向，即喘振区方向。效率并没有大幅度降低，但是流量不足可能将工况点置于喘振线的左侧，需要打开防喘振阀以保护压缩机免于喘振。在这种情况下，防喘振阀通过将压缩机出口气体回流到压缩机入口来使得叶片有足够的流量通过，避免喘振，这样的气体再循环将浪费能量。

1.2 Gas Compressor Restage Principles and Value Proposition 压缩机调级原理和经济性分析

The energy balance of the whole power train from engine (or other drivers) to the compressor can be expressed in Equation 4. 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.

从燃气轮机（或其他驱动器）到压缩机的整个动力系的能量平衡可用式（4）表示。标准条件下实际流量可以换算为标准流量。燃气轮机的输出功由燃气轮机效率和机械效率决定。一般而言，动力传输的机械效率是相对不变的，燃气轮机的效率主要由速度决定。

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

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.
式中：Power 为驱动器的输出功；C 为常数；SQ 为标准流量；η_{mech} 为机械效率；η_{engine} 为驱动器效率。

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. The purpose of a gas compressor restage is to reoptimize the compressor staging in order to maximize efficiency at the new conditions to minimize the power consumption or maximize flow, head, or both.

整机效率一般在设计工况附近被最优化，以确保整机以最小功率保证所需的流量和压头。当实际运行工况长时间不在最优化的设计工况附近时，整机效率就会大打折扣。压缩机调级的目的就是要重新优化压缩机分级，以便在新的工况条件下实现整机优化。

CASE 1：Compressor /Engine Running too Fast
情况 1：压缩机转速太快

The two typical scenarios in which the flow point moves in the T direction are increasing discharge pressure for gas injection and decreasing suction pressure for gas gathering. The compressor has to be rotated faster to keep up with the increasing 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.

增加压缩机出口压力或减小入口压力是工况点在 T 方向上移动的典型情况。这种情况导致压缩机必须增加转速来增加压比，直到最终动力透平或压缩机本身达到最大速度，这就是典型的转速封顶。对于这种情况，增加压缩机的级数，就可以把速度降下来且同时满足所需的压比；从另外一个角度讲，可以达到同转速提高压比的效果。对于处于晚期的气田，气田的压力降低，要求收集站机组的压比增加，调级意味着延长气田的寿命。对于用于油田增压的压缩机设备，调级增加的压力意味着石油产量的增加。这两种情况下，可快速收回压缩机调级的投资。

CASE 2：Engine Running too Slow
情况 2：燃气轮机转速过低

When theoperating 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.

当工况点沿 D 方向移动时，压缩机低速运行。通常，压缩机速度在设计工况下要达到或超过燃气轮机最大速度的 90 %，以使得燃气轮机在高效区运行。发动机效率随着速度的降低而降低。当工况点始终要求发动机转速低于最佳水平时，压缩机移除一个或多个级将增加所需的压缩机速度并提高发动机效率。这种类型的重新分级减少了发动机燃料消耗。

CASE 3：Choked Compressor
情况 3：堵塞运行

During seasons of high flow demand, it is normal torequire 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. 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.

在高流量需求的季节,压缩机通常满载运行。在这种情况下,工况点沿 C 方向移动,压缩机效率迅速下降。有些压缩机就会因堵塞而不能满足要求的大流量,即便压缩机实际上可能没有物理阻塞,但其低效率使得可用功率降低,从而限制了压缩机吞吐量。在一些情况下,整机可能无法达到吞吐量的要求。对于这种情况,通常可以用小流量级替代大流量级来给压缩机调级。

CASE 4: Compressor Running in Recycle Mode
情况 4:喘振保护再循环情况运行

Opposite toa 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 to keep the compressor out of surge. Surge can cause violent vibration and catastrophic damage to the compressor. The energy 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 a drop in the compressor efficiency. Restaging can solve this problem by replacing higher flow stages with smaller stages to accommodate the lower volumetric flows.

与堵塞情况相反,当压缩机流量降低时,工况点沿 S 方向移动。一旦压缩机的流量小于喘振保护线,防喘振阀就会打开以避免喘振,且压缩机以再循环模式运行,一部分压缩气体将被冷却以返回到压缩机入口,以保证压缩机没有喘振。喘振会导致压缩机剧烈振动和灾难性损坏。这个过程中,再循环气体消耗的能量被浪费了,且额外的能量也会用来驱动泵送冷却水或驱动风扇用于气体冷却。这等效于压缩机效率的下降。通过用较小流量的级代替较高流量的级,压缩机调级可以解决这个问题。

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.

除了经济原因,如果在深度再循环模式中气体没有得到足够的冷却,再循环模式运行会导致较高的排出温度。干气密封件、平衡活塞巴氏合金和防喘振阀可能由于长期高温而被损坏。总之,压缩机调级的主要好处是:提高石油和天然气产量,降低燃料消耗,使设备运行更健康。

1.3 Restage Criteria
压缩机调级标准

A restage is generally recommended at the 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.

如果可以在 5 年内收回成本,一般建议在压缩机大修时进行调级。如果不到 1 年内能收回成本,则应立即考虑压缩机调级。

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 (P_1), and the required power. The changes between the conditions just 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. Generally, 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.

用户和生产商共同合作才能更好地计算出压缩机调级的经济性。最近索拉透平公司根据以往 379 台压缩机调级的经验总结出了一套科学的调级指数来指导调级。4 个参数可以作为判断是否调级的指标:入口流量系数(Φ)、等熵压头系数(Ψ)、入口压力(p_1)和所需功

率。这 4 个参数调级前的工况和原始设计条件的差异可以作为调级的晴雨表。每个参数的详细标准见表 1。通常,对于功率、入口压力和压头系数,如果偏移了原始设计工况的 5 %~15 % 时,下次大修时就要考虑调级(黄色警告)。如果这些参数偏移了 15 %,那就应该立即考虑调级(红色警告)。流量系数如果偏移了 25 %,下次大修时考虑调级,如果超过 50 %,应立即考虑调级。如果触发任何一个参数的红色警告,则应考虑调级,如果触发了所有 4 个参数黄色警告,也应该考虑立即调级。

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.

另一个比较简单的规则是目前压缩机效率和压缩机最高效率之差,如果差值高于 6 % 且燃气轮机达到最大功率的话,推荐压缩机调级来恢复这 6 % 的效率,或者是大致 8 % 吞吐量增益。

Table 1 Trigger Points for Restage Parameters

表 1 调级参数标准

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 %

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

是入口流量系数,对于多台串联的压缩机组,采用首台低压压缩机入口流量系数。

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

是压缩机等熵压头系数。

$\Psi_{isen} = Cp \frac{T_1}{(D_2 N)^2}\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right]$ for compressors using the total pressure ratio and the first compressor speed and impeller tip diameter.

用于采用总压比和首台低压压缩机转速、叶轮缘直径的压缩机。

2 Case Studies
案例分析

Case 1: Extra Capacity for Pipeline Application
案例一:天然气管道吞吐量增加

Two identical packages were commissioned in 1994 for a pipeline application in US. The maximum power from the engine with the given conditions onsite was 11 910 hp. The original design point parameters are listed in the first column of Table 2 and marked as point 1 on the performance map in Figure 2.

1994 年两台相同压缩机机组开始在美国的一条天然气运输管道上投入运行。燃气轮机的最大功率为 11 910 马力。表 2 第 1 列列出了当时的设计工况参数,图 2 中的性能曲线上这个设计工况标记为点 1。

Over the years, both downstream flow demand and the suction pressure were increased. In the year 2000, the compressor could only deliver 426.1 mmscfd using the full engine power (the 2nd column in the table). The operating point drifted from point 1 to the choke side to point 2 in Figure 2. The efficiency dropped to 76.3 %. The 3 of the 4 key parameter changes were in the red zone indicating a restage opportunity.

自从这两台压缩机投入运行以来,这两台机组下游流量需求和入口压力均有所增加。到了 2000 年,压缩机只能提供 426 mmscfd(1 mmscfd = 1 116.3 Nm^3/h)的吞吐量(表 2 第 2 列)。按图 2 所示,工况从点 1 漂移到阻塞区的点 2,效率由 81.2 % 下降到了 76.3 %。4 个关键参数更改中的 3 个位于红色警报区域,表示压缩机应该尽快调级。

Table 2 Change in Operation

表 2 案例一参数变化

	Original Staging		Restaged	Changes Max Power to Design
	Design Point	Max Power		
Point Perf. Map	1	2	3	
Φ	0.062 3	0.076 6	0.081 3	23.0 %
Ψ	4.06	3.198 0	2.992 0	−21.3 %
p_1(psia)	725.0	752.6		3.8 %
Power total(hp)	9 086.0	11 910.0		31.1 %
Efficiency(%)	81.2	76.3	83.7	5 %
p_2(psia)	1 216	1 207.6		−0.7 %
SQ(mmscfd)	312.0	426.1	467.2	36.6 %
Flow(acfm)	4 136.7	5 400.1	5 921.0	30.5 %
p_2/p_1	1.68	1.60		−4.3 %

In 2000, the compressors were restaged to higher flow staging. Meanwhile the stage number is reduced from 4 stages to 3 stages to increase the speed, efficiency and flow capability. To reduce the restage cost, one stage was reused. The restaged compressors were 7.4 % more efficient than the off design efficiency and the maximum flow was increased by 9.6 % from 426.1 mmscfd to 467.2 mmscfd. The same 3 points are shown on the restaged compressor performance maps (Figure 3). As can be seen, the maximum flow point was positioned in the area of peak isentropic efficiency.

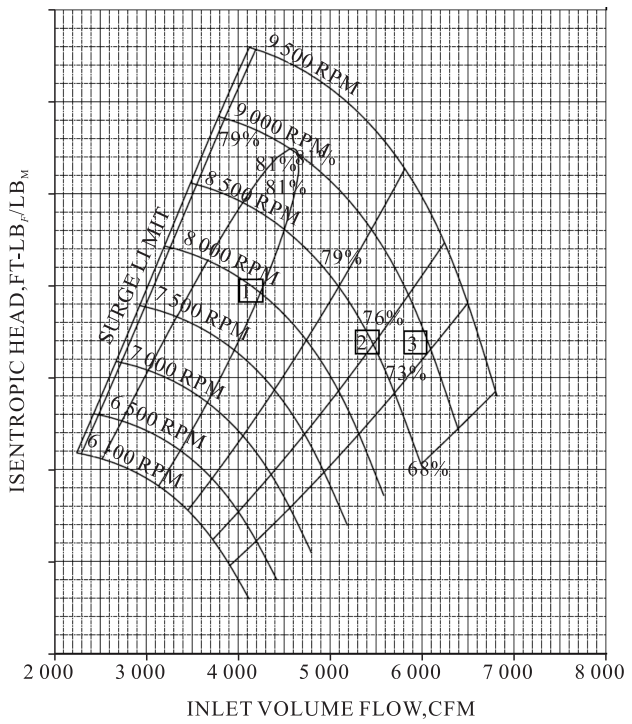

Figure 2 Existing Staging Performance Map for Case 1

图 2 案例一调级前压缩机的性能曲线

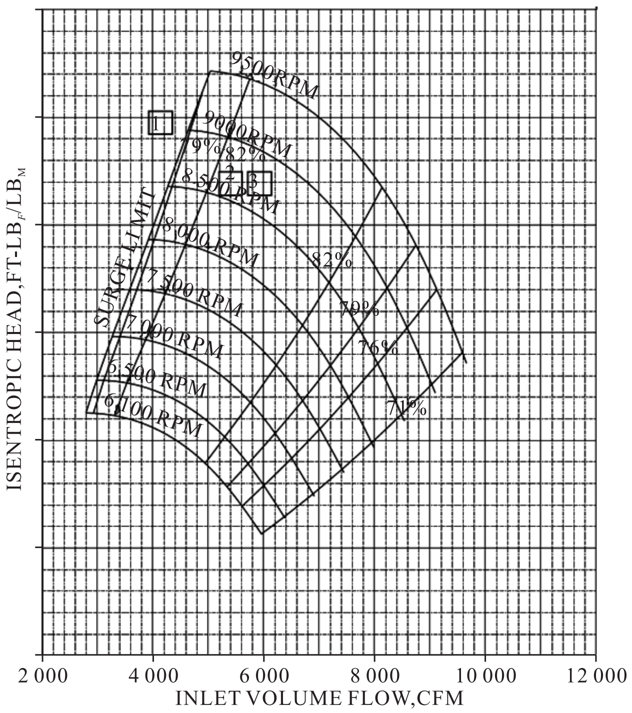

Figure 3 Restaged Performance Map for Case 1

图 3 案例一调级后压缩机性能曲线

按照建议，压缩机用更高流量的级替代了低流量级，同时压缩机由以前的 4 级压缩改成了 3 级压缩，以提高压缩机转速、效率和吞吐能力。为了减少调级成本，调级前压缩机的其中一级被清理后重新使用在调级后的压缩机里。调级后的压缩机的效率提高了 7.4 %，最大流量从 426.1 mmscfd 增加了 9.6 %到 467.2 mmscfd。调级后的工况点如图 3 的点 3，可以看出，调级后的工况点在压缩机的最佳效率区。

Based on the well head gas price, the increased revenue due to restage is about $150,000 /day. This is a typical pipeline application restage. A compressor restage can be paid back in weeks if not days if the customer owns the gas.

基于井口天然气的价格，调级后客户的收入增加了约 15 万美元 /d。这是一个典型的天然气管网中调级的例子。压缩机调级的投资几周内甚至几天内即可收回。

Case 2: Oil Production Increase

案例二：油产量增加

A turbine drivinga two body compressor package was commissioned in 1997 on a platform in the Gulf of Mexico. This is a typical application for offshore oil production. The package includes a gas turbine (ISO power: 7 600 hp) driving six stage Low Pressure (LP) Compressor, which through a Gear Box (speed increaser) drives an eight stage High Pressure (HP) compressor.

1997 年，1 台燃气轮机驱动 2 两台并联压缩机在美国墨西哥湾的一个钻井平台上投入生产。这是石油生产中压缩机调级的典型例子。燃气轮机的 ISO 功率为 7 600 hp(1 hp = 0.75 kW)，直接驱动低压压缩机(LP)，然后通过齿轮变速驱动高压压缩机(HP)，低压压缩机是 6 级压缩，高压压缩机是 8 级压缩。

The condition changes are shown in Figure 4. Initially, the field pressure was stable after the compressor was commissioned in 1997. Then gas production was increased to move the original design point (red dot) to the choke side (C direction) and speed topping side (T direction). The required running point eventually moved out of the range the original compressor staging can handle.

图 4 显示了投入生产后压缩机经历的工况改变。最初，在 1997 年压缩机调试后，油田的天然气压力稳定，但是随着气量增加，工况(红点)移动到堵塞区域(C 方向)，并且不断提速(T 方向)，压缩机无法满足增加的气量要求，需要调级。

In 2000, both the LP and HP compressors were restaged to larger flow staging combinations. The LP

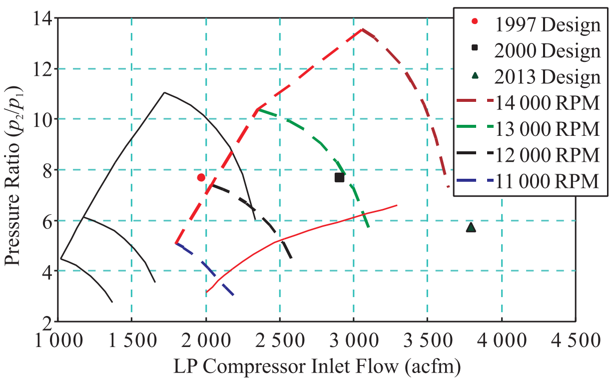

Figure 4　Restage Maps for Case 2

图 4　案例二中调级前后压缩机的性能曲线

compressor was restaged to 7 stages with 4 stages being reused in the new configuration. The HP compressor was kept at 8 stages, 3 of which were from the original compressor. The performance map is shown in the middle of Figure 12 with the new design point (black square) in the middle of the staging operating range. Due to the restage, the production was increased from 30 mmscfd to 44 mmscfd. The increased production correlated to net incremental revenue of ＄51 520 /day based on ＄3.68 /kscf (averaged well head price of 2000 from EIA).

2 台压缩机都在 2000 年完成了调级来满足更大的流量需求。低压压缩机由 6 级压缩改成了 7 级压缩，且重新使用了调级前的 4 个级的部件。高压压缩机还是 8 级压缩，保留了调级前的 3 级的部件。这样重复利用原来压缩机的部件大大降低了成本。图 4 中标出了调级后的压缩机性能曲线，用虚线表示，新的工况点用黑色方块表示。调级后的压缩机将整机吞吐量从 30 mmscfd 增加到 44 mmscfd。这次调级的经济效果非常显著，每天增加的收入达到了 5.1 万多美元。

After several years of operating with these restaged compressors, the field pressure started declining. As pressure dropped to 89.7 psia in 2012, the gas density was reduced by 45.6 %. For the same mass flow or standard flow, the actual inlet flow was almost doubled. The compressors were chocked at the maximum speed of 14 300 rpm and flow of 3 750 acfm (29.7mmscfd).

这台压缩机调级后的机组投入使用几年后，油气的压力开始下降，到了 2012 年，压力降至 89.7 psia (1 psia = 6.895 kPa)，油气密度降低了 45.6 %。压缩机仍然要吞吐相同的质量流量或标准流量来满足生产需要，由于压力和密度降低，入口容积流量几乎增加了一倍。压缩机在 14 300 r /m 的转速和 3 750 acfm (29.7mmscfd) 的流速出现了堵塞。

To accommodate the new conditions, the two compressors underwent restage analysis. Although the compressors could meet the flow requirement, they could not meet the required discharge pressure due to the low suction pressure associated with the depletion of the well. In order to meet the higher head requirements, another tandem compressor package was relocated to add additional compression ability, while the two existing compressors were designed as a booster package. The restage solution with both packages in series not only met the current requirements, but also left room to allow the field pressure to drop another 10 psi and still deliver the required standard flow and discharge pressure.

为了适应新的条件，满足生产需要，2 个压缩机再次进行了调级（图 4 的右侧实线和黑色三角点）。调级后的压缩机可满足流量需求，解决了堵塞问题。但是由于油井进入晚期，压力过低，调级后的机组不能满足所需的出口压力。为了满足更高的压比要求，1 台闲置的相同机组的压缩机也被调级后重新启用，串联在这台调级后压缩机的下游，进一步增压。这样的平台压缩总体规划不仅满足了当前产油的需求，还为油气压力进一步降低留有余地，能够保证未来油气压力降到 10 psi 的产油。

The outgoing gas in this case was used for oil lift application. The restage solution increased oil production by about 300 barrels a day. The entire project including restaging both packages (4 compressors) and expensive piping changes, was paid back within a year's operation. It also added flexibility in operation and margins for future field depletion.

这一解决方案使石油产量每天增加约 300 桶。整个项目包括 2 台整机的 4 台压缩机（包括闲置机组的调级）和昂贵的管道更换，客户不到一年就收回了成本。这个方案还增加了操作的灵活性，有效延长了油井寿命。

Table 3　Condition Changes for Case 2

表 3　案例二工况变化 (1997 ~2013 年)

	1997	2000	2013	2000	2013
Φ	0.054 7	0.074 6	0.089 7	36.4 %	20.3 %
Ψ total	11.16	11.158 3	9.457 5	0.0 %	−15.2 %
p_1 (psia)	164.7	164.7	89.7	0.0 %	−45.5 %
HP total (hp)	4 588.0	6 591	4 040	43.7 %	−38.7 %
p_2 (psia)	1 264.7	1 264.7	514.6	0.0 %	−59.3 %
SQ (mmscfd)	30.0	44.1	30.0	47.1 %	−32.0 %
Flow (acfm)	1 973.7	2 904.9	3 794.8	47.2 %	30.6 %
p_2/p_1	7.68	7.68	5.74	0.0 %	−25.3 %

Detailed conditions are shown in Table 3. Two key indicators were in the red zones according the criteria for restages in 2000 and 2013. The payout period also fell into the 1 year financial criteria.

详细的工况变化见表 3。根据 2000 年和 2013 年的调级标准,其中两项关键指数已进入红色警报区,回收期也已满足 1 年财政标准。

Case 3: Life Cycle Management for Declining Field
案例三:油田设备生命周期管理

This is another typical field depletion case. The compressor was commissioned in 1996 off shore of Indonesia. After initial commissioning, the field pressure started to decrease as shown in Table 4. The compressor was restaged twice to adapt to the changing conditions in 2002 and 2006. Each time, the restage criteria justified the restage.

这是另一个典型的油田晚期的例子。这台机组于 1996 年在印尼海上钻井平台投入生产。如表 4 所示,机组运行后不久油田的气压开始降低。压缩机分别在 2002 和 2006 年进行了调级以适应工况变化。每次在调级之前,都有两项指数的变化处于红色警报区。

After several successful years of operation after the 2006 restage, the pressure and standard flow decreased rapidly and the compressor could not perform with the low pressure of the well. As a result, the Anti-Surge Valve (ASV) opened to increase suction pressure. To improve the conditions for the compressor, some high pressure side stream gas was injected to increase the suction pressure to 124.7 psi. By the end of 2012, the compressor was running with the ASV approximately 80 % open, resulting in 17.9 mmscfd out of 25.9 mmscfd throughput being recycled. The net though flow was only 8 mmscfd, or 31 % of the total flow.

2006 年调级后的机组经过几年运行,油气的压力和标准流量都迅速下降,压缩机已经不能满足这样低压的流量需要了,防喘振阀(ASV)不得不打开来进行回流。为了改善这一情况,客户不得不从一些高压气源里引入一些油气来把压缩机入口压力增加到 124.7 psi。到了 2012 年底,压缩机运行时放喘振阀要 80 % 左右打开,这样导致了 17.9 mmscfd 的油气充分循环,占了 25.9 mmscfd 压缩机吞吐量的 69 %,净通吞吐量只有总流量的 31 %。

Table 4 Well Condition Variation for Case 3

表 4 案例三油井工况条件变化(1996 ~2012 年)

Staging	1996	2002	2006	2012	2012	2002	2006	2012
Before or after restage	Design	after	after	before	after			
Φ	0.031 9	0.048 8	0.052 1	0.053 2	0.021 1	53.0 %	6.8 %	−60.3 %
Ψ	6.93	5.806 4	6.045 0	4.729	7.768	−16.2 %	4.1 %	64.3 %
p_1(psia)	350.0	285.0	84.7	124.7	124.7	−18.6 %	−70.3 %	0.0 %
HP total(hp)	3 471.5	3 651.0	1 836.8	2 079.2	616.2	5.2 %	−49.7 %	−70.4 %
Efficiency(%)	70.7	75.5	67.3	60.1	64.1			
p_2(psia)	1 115	800	304.7	323.7	323.7	−28.3 %	−61.9 %	0.0 %
SQ(mmscfd)	40.0	49.9	17.0	25.9	8.0	24.8 %	−65.9 %	−69.1 %
Flow(acfm)	1 218.3	1 912.8	2 252.4	2 144.8	663.7	57.0 %	17.8 %	−69.1 %
p_2/p_1	3.19	2.81	3.60	2.60	2.60	−11.9 %	28.1 %	0.0 %

The compressor was restaged in 2013 with the performance map at the new condition shown in Figure 5. The restage saved about 12.9 mmbtu/hr fuels by eliminating recycling. With the price of natural gas significantly higher in this region of the world, the payback period with fuel cost savings resulted in a 9 month payback period.

这台压缩机在 2013 年进行了调级,其性能曲线见图 5。这一调级节约了大约 13 mmbtu/hr(1 mmbtu/hr = 3.73×10^8 J/h)燃料。燃料成本节省使得客户在短短的 9 个月收回调级成本。

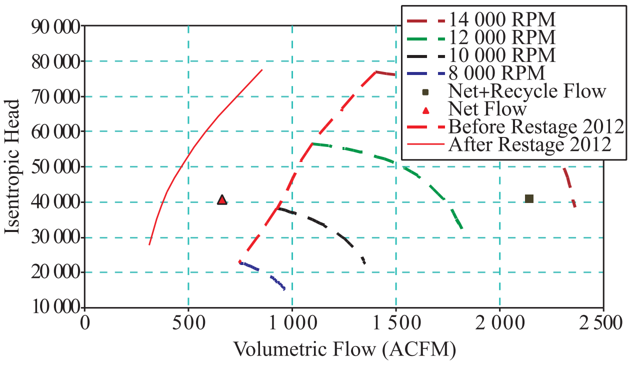

Figure 5 Restage Map for Case 3

图 5 案例三的调级性能曲线

Case 4: Complex Project, Life Cycle and Inventory Management

案例四:生命周期和库存备机管理

The operational challenges associated with well depletion are considerably more difficult for customers with offshore installations. Space limitations, downtime criticality, geographic location and harsh environments play a major role in how compressor restages are executed. Additionally, declining revenue recognition associated with depleting wells require gas compressor restage projects to be cost effective for the customer; making existing asset reallocationand inventory consumption vital to project success.

油井衰竭导致的工况变化对于海上平台设备来说是相当困难的。空间限制、停机停产的损失以及地理位置和恶劣环境为压缩机调级增加了很大难度和复杂程度。此外,停机的收入下降要求压缩机调级要严格控制成本;重新利用已有和库存的压缩机组对于调级项目的成功与否至关重要。

Two successful complex projects involving depleting wells have been recently executed in the North Sea, an area which provides a significant portion of Europe's natural gas supply. The first project involved a two body tandem compressor configuration operating in series. (Turbine ISO 6100 HP-Speed Increasing GBX-LP Compressor-HP Compressor) As is typical with well depletion, over three years of operation, falling suction pressure drove the as designed compressor staging to speed topping as shown in Figures 6 − 7. Although standard gas volumes decreased by approximately by 42 %, the decrease in suction pressures caused a net increase in inlet gas flow, which drove the operating points of both the LP and HP compressors to the choke side of the compressor map. Projections of future gas conditions confirmed that suction pressures would continue to fall along with standard gas volumes over the course of the following years. A comparison of original design data to operations 3 years after can be seen in Tables 5 and 6. The P1 and Power parameters exceeded the 50 % trigger points, confirming the need for compressor restage to meet the new operating duties.

最近在欧洲北海就有两个成功的复杂调级项目的例子,这些地区是欧洲的主要天然气产区。第一个项目 1 台燃气轮机驱动的 2 台串联压缩机机组,由 1 台 ISO 6100 马力通过变速箱驱动 1 条低压压缩机和 1 台高压压缩机。见图 6 ~ 7,衰竭期的油井在 3 年的时间里,油气压力的降低使得压缩机接近最大转速。尽管标准气体流量减少了约 42 %,但是由于压力降低,入口气体体积流量却增加了,使得 2 台压缩机在阻塞区附近运行。根据客户对未来油气参数变化的预测,在接下来的几年中,油气压力和标准流量都会继续下降。表 5 ~ 6 中数据与 3 年后的操作相比,入口压力和功率参数都超过 50 %变化的触发点,确认了压缩机调级的必要性。

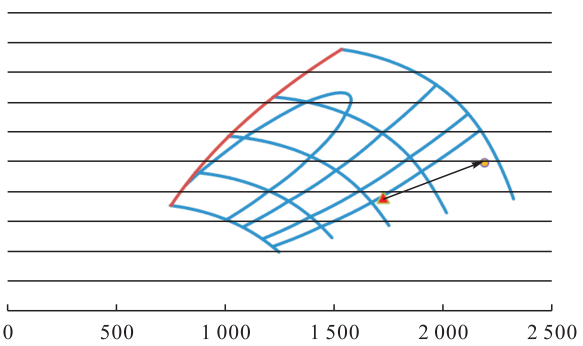

Figure 6　Existing LP Compressor Peformance Map

图 6　调级前低压压缩机性能曲线

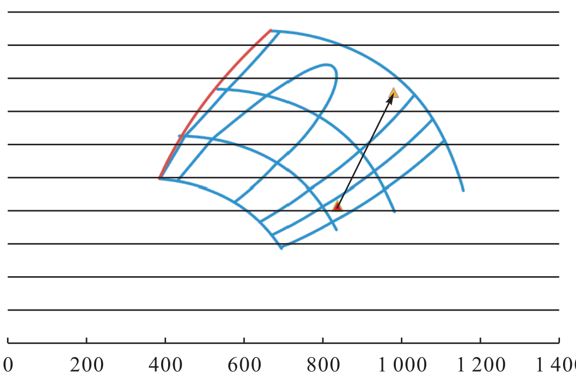

Figure 7　Existing HP Compressor Peformance Map

图 7　调级前高压压缩机性能曲线

Table 5　Changing Process Conditions for Case 4

表 5　案例四的工况

Parameter	Design Winter 2008		Operation Winter 2011	
	LP	HP	LP	HP
p_1 (Bara)	35	65. 5	16	35
p_2 (Bara)	66. 5	137	36	114
Isentropic Head (m-kgf /kgm)	7 625	8 352	9 934	15 153
SQ(nm^3 /day3)	1 410	1 410	811. 2	811. 2
Inlet Flow (m^3 /min)	1 726	838	2 190	979
T_1(C)	30	27	21	26
Speed(RPM)	18 524	18 524	21 879	21 879
HP(kW)	2 583	2 984	1 505	2 028

Table 6 Parameter Change for Case 4

表 6 案例四的工况变化幅度

Parameter	Delta	
	LP	HP
p_1(Bara)	54 %	47 %
HP(kW)	42 %	32 %
PHI	7 %	1 %
PSI	7 %	30 %

In addition, suction pressures were expected to decline from 16 Barg to levels as low as 4 Barg, while Standard Gas flows were expected to drop from 800 Nm^3 /day-e3 down to 500 Nm^3 /day-e3. As the existing compressors were fully staged and near speed topping, the ability to drop P1 lower while maintaining discharge pressure constant would require the addition of a 3rd stage of compression. In addition, increasing inlet flow volumes would require that the 3rd stage of compression be a larger, higher volume compressor. This created a particularly challenging situation because the customer was on an extremely tight production schedule, and also had budget and space constraints.

此外,油气压力预期从 16 Barg(1 Barg = 0. 1 MPa)下降到低至 4 Barg 的水平,同时,标准流量预期从每天 800×10^3 m^3 的流量下降到每天 500×10^3 m^3。现有转速已经接近压缩机最大允许转速,而且压缩机已经没有再增加级数的空间了。进一步降低油气压力将需要增加 1 台压缩机。此外,这 1 台增加的压缩机要能够吞吐由于油气低压而增加的体积流量。这使得实际项目的方案和操作都非常具有挑战,因为客户的生产进度非常紧张,并且还有预算和空间限制。

The technical solution identified was a 3 - body tandem skid operated in series. In order to keep project costs down, the 3 - body tandem skid was designed to keep the two existing compressors as is, for IP and HP duty. No restage of the IP and HP compressors were needed. The added LP compressor was designed to compress gas from 4 Barg to 12 Barg, with enough turndown percentage to handle the 800 - 500 Nm^3 /day-e3 flow decay. The driven equipment skid was designed specifically to fit within the customer's tight space constraints, and has successfully been in operation for the last year achieving the customer's production goals.

最后所确定的解决方案是加入了 1 个串联的低压压缩机,整个机组变成了 3 体串联压缩机,保留已有的 2 个压缩机作为中压和高压压缩机。为了进一步降低成本,中压和高压压缩机位置不变,不需要重新安装。添加的低压压缩机可以将油气从 4Barg 压缩 12Barg,而且机组具有足够宽的工况范围可以应付每天 800×10^3 到 500×10^3 m^3 的流量范围。整台机组占地面积很小,满足了客户空间小的难题。机组成功在 2013 年开始运行,实现了客户的生产目标。

3 Conclusions 结论

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.

本文介绍了压缩机设计和调级的基本原理,并且通过实际案例研究给出了一套压缩机调级的简单标准。

Four case studies represent 4 typical scenarios for restage: increasing gas production and oil production, life cycle management and inventory management for depletion field. For all these cases, the investment to restage can be paid back fast.

4 个案例研究代表性地示范了 4 种压缩机调级的典型应用:增加天然气运输吞吐量、增加石油产量、生命周期管理和库存管理。对于类似的情况,压缩机调级的成本收回得很快。

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