压缩机约束在管道建模和优化中的作用

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本白皮书最初由作者提交至“管道模拟兴趣小组（PSIG）”年会。在此特别感谢中石化石油工程设计有限公司李欣泽工程师和中国石油天然气管道工程有限公司裴娜工程师对中文翻译的校正和建议。

摘要

本文旨在讨论压缩机站约束对管道建模和优化结果的影响。管道模型中压缩机的定义包括了限制其运行区的约束。这些约束可以包括最大和最小速度、压比、气缸余隙和功率。离心式压缩机的运行也受到喘振和音速流的限制。这些约束用于反映实际的压缩机限制，或者将运行区限制在更有效的范围。在现场的启动和停机程序中，压缩机需要暂时打破其中一些约束。在优化算法中，强制执行最小约束有时会带来次优答案，因为它们会在可行空间中产生“孔”，导致局部最小值。天然气管道在运行中必须要满足各种约束，本文将探寻一种方法，将这些约束所产生的影响，进行可视化。本文还将探讨管道模拟和优化软件中包含最小约束的稳态压缩机模型，阐述其使用时的相关问题。

研究压缩机约束对模拟结果的影响，要检查它们如何影响最大和最小压力包络。这种方法的价值在于，它使得不同管道约束对于管道运行可能范围的影响方式，更容易可视化。本文还探讨管道模拟和优化过程中对压缩机站使用最小约束的相关问题，提醒建模人员注意。

引言

压缩机站布置在管道沿线，能够提供能头，允许气体长距离输送。管道分析和优化软件用数学模型来表征压缩机站。这些模型估算压缩机运行成本，预测给定水力条件下压缩机运行的可行性。管道分析结果的优劣，取决于压缩机模型对于压缩机站能力和燃料消耗的预测水平。

许多压缩机模型近似于现场运行的压缩机。这些模型中既有相当简单的理论压缩机模型，描述了具有最大功率和效率的压缩机的性能，也有复杂的动态压缩机模型。鉴于管道建模和优化软件中需要大量的压缩机计算，所选的压缩机模型应在计算费用和准确性之间取得平衡。因此，管道优化和建模软件经常使用稳态压缩机模型。详细的稳态压缩机模型将会准确预测运行成本，以及它所表征的压缩机的可行运行区。

有些压缩机约束界定了无法达到的运行条件，而有些压缩机约束则可选择用于将压缩机运行限制在更有效的运行区。了解约束是如何影响模拟和优化结果，有助于选择要实施的压缩机约束。

本文一开始将讨论往复式压缩机和离心式压缩机的一些约束。接下来，本文将描述一种管道约束的可视化方法。本文还将讨论在稳态分析和优化中实施这些约束的效果。在本文最后，将提出一些对于压缩机建模的建议。

往复式压缩机约束

往复式压缩机利用活塞压缩气缸内的气体，类似于自行车打气筒的工作方式。往复式压缩机可由许多不同的驱动机提供动力，包括天然气发动机、电动机、柴油发动机等。这些驱动机可以是变速的，也可以是定速的。

随着管道条件的变化，控制器必须调整压缩机的运行，以满足所需的压力条件。控制往复式压缩机运行的三种主要方式是：变换压缩机速度、改变气缸余隙量和改变机器的扫气容积。变速驱动机可通过加快或减慢活塞速度，控制通过气缸的流量。此外，许多往复式压缩机的气缸有固定或可变的余隙腔。当腔体打开时，气缸的容积增加，而扫气容积量不增加。这种额外的气缸容积减少了气缸的压缩量，从而减小了流量和所需的驱动机功率。最后，通过停用气缸端，减少压缩机的总扫气容积，进而减小流量和压缩机运行所需功率。

定速定余隙往复式压缩机

有些往复式压缩机的燃气驱动机运行速度范围非常小。当这些压缩机的速度降至最低运行速度以下时，它们在负载下就无法有效运行。一些压缩机驱动机的运行速度范围很窄，以至于基本属于固定速度的机器。

如上所述，添加可变或固定余隙腔可增加气缸余隙。配备固定余隙腔的定速压缩机，具有与每个余隙腔对应的离散运行点表面。“RC_fixed_s_fixed_cl”就可证明这一点，它是一台单级 3000hp（2237kW）往复式压缩机，有 5 个独立的扫气容积，每个容积包含 8 个级别的余隙，这样余隙和扫气容积共产生了 40 种组合。如需了解更多关于 RC_fixed_s_fixed_cl 的信息，请参见附录 A 中的表 2。

表 1 显示了扫气容积为 13.5 平方英寸（87.1cm^2）、吸入压力为 800psig（5516kPa）、排出压力为 950psig（6550kPa）的余隙级别中，往复式压缩机可达到的流量。压缩机不能在表中所示值之间输送流量。因此，在这些压力条件下，压缩机流量可达 299.7 mmcfd（8.487 Mm^3/d）和 290.6 mmcfd（8.228 Mm^3/d），但不能达到 295 mmcfd（8.353 Mm^3/d）。

余隙（%）	流量（mmcfd）（Mm^3/d）	功率（hp）（kW）
90	304.3 8.617	2709 2020
100	299.7 8.487	2668 1990
120	290.6 8.228	2587 1929
140	281.4 7.969	2505 1868
160	272.3 7.710	2424 1807
180	263.1 7.451	2342 1747
200	254.0 7.192	2261 1686
220	244.8 6.933	2179 1625

表 1 - 吸入压力为 800psig（5516kPa）、排出压力为 950psig（6550kPa）时，压缩机 RC_fixed_s_fixed_cl 在余隙范围的运行。

图 1 显示了压缩机 RC_fixed_s_fixed_cl 在 500psig（3447kPa）压力范围下的排气量。要注意，排气量的界定是基于所有吸入压力小于排出压力。在较低压力比下，压缩机的排气量受到最小余隙和最大扫气容积的限制。在较高压力比下，以最小余隙运行压缩机所需的功率超过了驱动机的最大功率。在排气量图中级别所示的水力条件下，必须要增加余隙量，以保持所需功率低于最大功率。

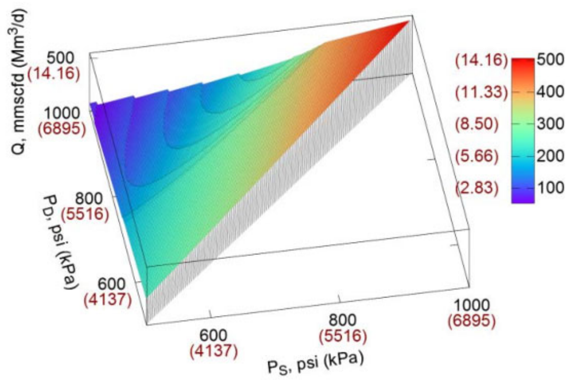

图 1 – 随吸入压力（P_S）和排出压力（P_D）变化的压缩机 RC_fixed_s_fixed_cl 排气量（Q）

该压缩机在给定流量下运行时，结果是不连续的。图 2 是压力离散为 2.5psig（17kPa）、流量为 300mmcfd（8.50 Mm³/d）时，随吸入压力（P_S）和排出压力（P_D）变化的压缩机利用率百分比。

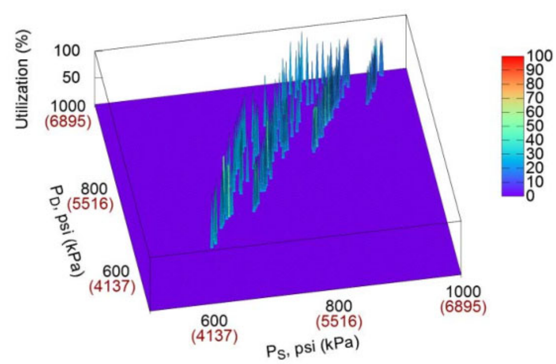

图 2 - 流量为 300 mmcfd（8.50 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的利用率百分比

分散的可行点表示在可用余隙级别对应的压力和流量组合下进行的计算。要注意，该区域大部分都是不可行的。

但是，如果允许连续的余隙而不是固定的余隙，则该区域的更大段将填充进来。下图（图 3）绘制的是流量为 300mmcfd（8.50 Mm3/d）、吸入和排出压力变化的情况下，RC_fixed_s_var_cl 的利用率。该压缩机与前一个压缩机相同，除了允许余隙在给定扫气容积的最大值和最小值之间有连续变化。

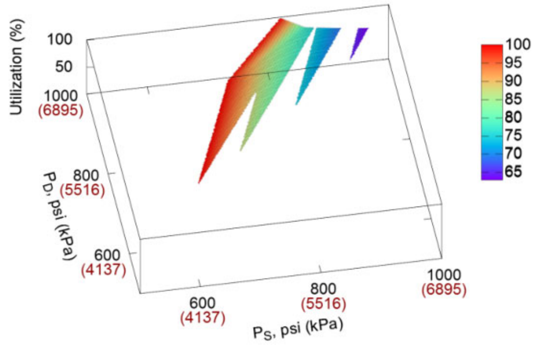

图 3 - 流量为 300 mmcfd（8.50 Mm³/d）、允许连续余隙的情况下，随吸入压力（PS）和排出压力（PD）变化的利用率百分比

在现场，管道是在瞬变条件下运行。虽然模型会解决两个余隙/腔组合之间的运行，但在现场，管道运行员将通过交替运行两个相邻的余隙级别，来获得类似的水力结果。作者建议，在管道建模和优化软件中对具有固定余隙的往复式压缩机建模时，要始终使用连续余隙选项。

可行区域的剩余间隙是扫气容积的变化造成的。由于扫气容积是以步进方式变化的，因此在该流量时，存在着从一个扫气容积过渡到下一个扫气容积所对应的运行间隙。这些间隙可以通过允许压缩机在最低速度以下运行来填补。

图 4 是流量为 300 mmcfd（8.50 Mm³/d）、压力范围相同时的 RC_var_s_var_cl 图。这台压缩机和前一台压缩机的唯一区别是，它可允许速度在 250 到 300 rpm 之间连续变化。当允许一个速度范围时，可行运行区的不连续性就消失了。作者建议，用管道软件模拟往复式压缩机时，要使用现场可达到的最大速度范围。

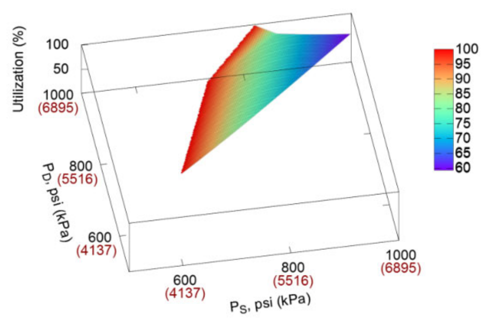

图 4 - 流量为 300 mmcfd（8.50 Mm³/d）、允许连续余隙、允许速度在 250 到 300 rpm 之间变化时，随吸入压力（PS）和排出压力（PD）变化的利用率百分比

最小功率

压缩机运行的成本之一是使用的燃料量。压缩机的燃效可以用进入压缩机用作燃料的气体百分比来描述。压缩机在低功率运行时所用燃料的百分比，比较高功率运行时增加 130%以上[1]。

压缩机建模人员有时会指定一个较低限的功率，试图以最可能有效的方式运行压缩机。如果最小功率被当作一个约束，则功率低于最小限制时，压缩机会关闭。如果最小功率被当作警告，在最小功率以下运行的压缩机将发出警报。作者已经看到，为了以最可能有效的方式运行机器，有些往复式压缩机模型把最小功率设置到高达最大功率的 80%。

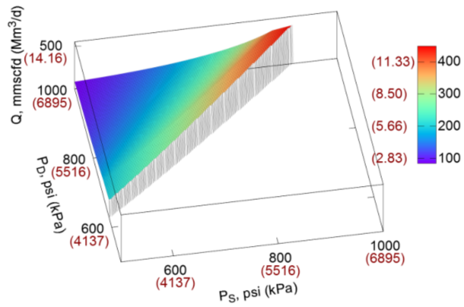

图 5 - 随吸入压力（PS）和排出压力（PD）变化的压缩机 RC_fixed_s_var_cl_min_power 排气量

图 5 显示了压缩机 RC_fixed_s_var_cl_min_power 在可行区域的排气量，这台压缩机与 RC_fixed_s_var_cl 相同，但其强制最小功率约束为 2400hp（1790kW）。要注意，在排气量条件下运行的压缩机对应的可行区域相当大，但是它开始时压比明显大于 1.0。然而，任何给定流量的可行区域都非常窄，并且随着流量的变化，压力会转换。图 6 至图 12 显示了在 350mmcfd（9.91Mm³/d）和 50mmcfd（1.42Mm³/d）之间的一系列流量下，压缩机 RC_fixed_s_var_cl_min_power 在可行区域运行的利用率百分比。要注意，对于每个单独的流量，可行区域都非常窄，并且随着流量的减少，可行区域会从较低的压比移向较高的压比。

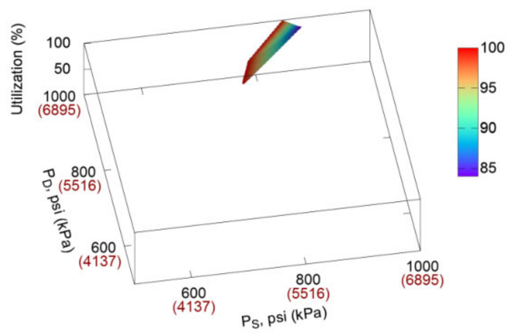

图 6 - 流量为 350 mmcfd（9.91 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的压缩机 RC_fixed_s_var_cl_min_power 利用率百分比

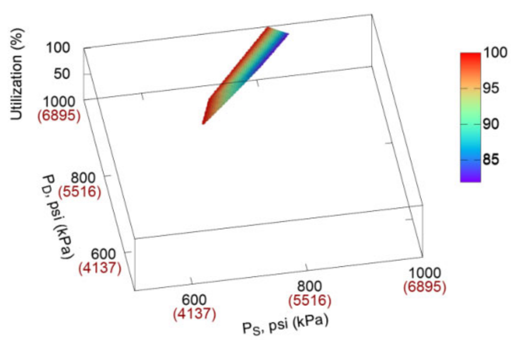

图 7 - 流量为 300 mmcfd（8.50 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的压缩机 RC_fixed_s_var_cl_min_power 利用率百分比

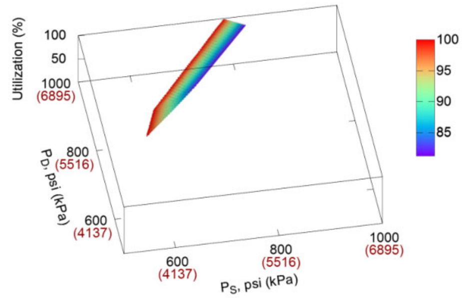

图 8 - 流量为 250 mmcfd（7.08 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的压缩机 RC_fixed_s_var_cl_min_power 利用率百分比

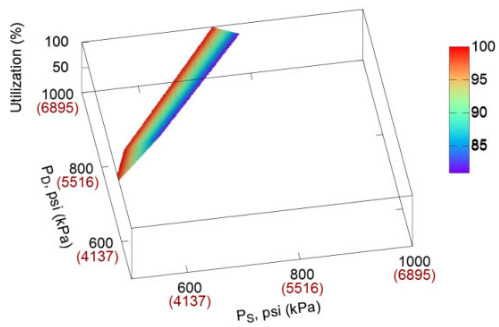

图 9 - 流量为 200 mmcfd（5.66 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的压缩机 RC_fixed_s_var_cl_min_power 利用率百分比

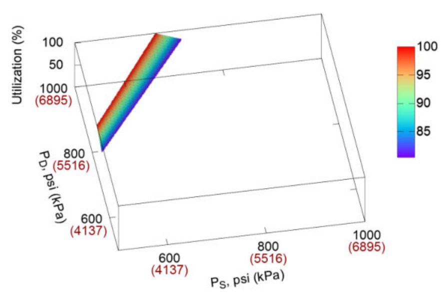

图 10 - 流量为 150 mmcfd（4.25 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的压缩机 RC_fixed_s_var_cl_min_power 利用率百分比；计算是在 500psig（3447kPa）至 1000psig（6895kPa）的压力范围进行的

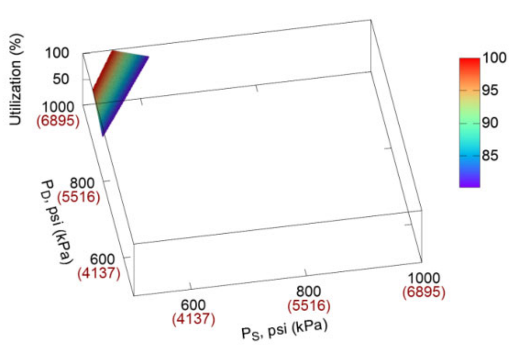

图 11 - 流量为 100 mmcfd（2.83 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的压缩机 RC_fixed_s_var_cl_min_power 利用率百分比

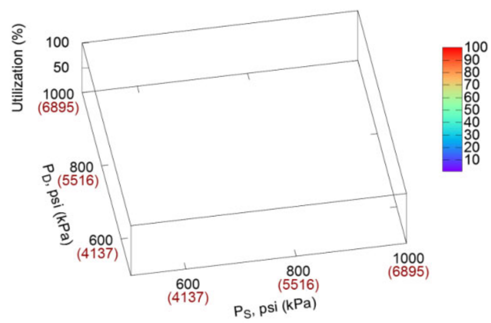

图 12 - 流量为 50 mmcfd（1.42 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的压缩机 RC_fixed_s_var_cl_min_power 利用率百分比

如果去除了最小功率约束，各个流量的可行区域会显著增加，如图 3 和图 7 比较所示。

离心式压缩机限制

离心式压缩机的运行可以用扬程流量图来表示。这些图描述了基于绝热缸头和压缩机流量条件的运行效率和速度，并给出了可行的压缩机运行边界。

管道使用的离心式压缩机通常由天然气燃料的涡轮机驱动。像往复式压缩机一样，这些驱动机也可能具有受限的运行速度范围。作者已经看到过有的最小速度设置为最大速度的 40%-75%。

下面的讨论将集中在一台离心式压缩机模型“CC”，其最大功率为 6000hp（4474kW），扬程流量图如图 13 所示：

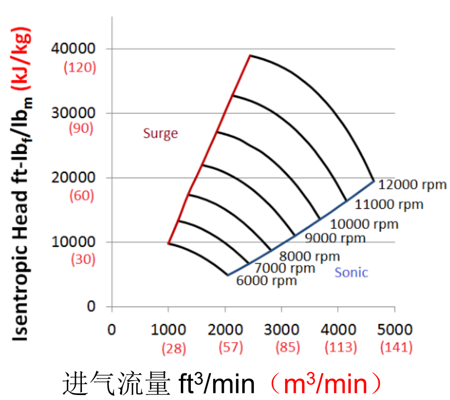

图 13 - 压缩机 CC 的扬程流量图

这台离心式压缩机图右侧的蓝线表示音速限制。压缩机可以在这条线的右侧运行，但排出温度会升高，效率未知。顶部曲线表示最大速度时的运行，底部曲线表示最小速度。图左侧的曲线表示喘振区域，压缩机不能在该区域运行。本文演示的离心式压缩机设置为循环气流，以防止在喘振线左侧区域运行。

图 14 显示了压缩机 CC 的离心式压缩机排气量。要注意，由于最低速度要求，压缩机直到压比明显高于 1.0 时才开始传输流量。还要注意，排气量一开始会随着压比的增加而增加。这相当于压缩机沿着音速线运行。

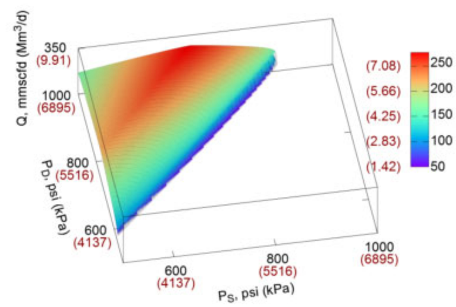

图 14 - 随吸入压力（PS）和排出压力（PD）变化的离心式压缩机排气量

最小功率

像往复式压缩机一样，离心式压缩机经常有最小功率要求。这些最小功率要求通常源于效率和排放问题。图 15 显示了上述压缩机的排气量，其最小功率设置为最大功率的 60%。

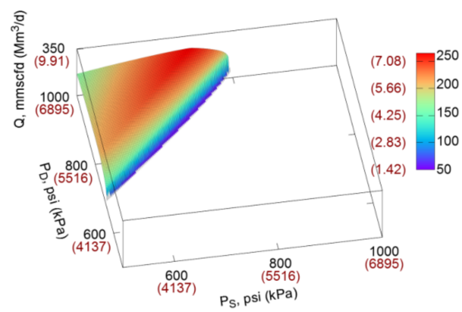

图 15 - 随吸入压力（PS）和排出压力（PD）变化的离心式压缩机排气量；最小功率设置为最大功率的 60%

图 14 和图 15 之间的比较表明，与没有设置最小功率限制的压缩机相比，具有最小功率限制的压缩机在开始运行之前必须具有明显更高的压力比。

图 16 至图 18 显示了压缩机“CC”在不同流量条件下的利用率图。要注意，由于最小功率限制，可行区域会随着流量的减少而缩小。相比之下，图 19 显示了在流量为 100 mmcfd（2.83 Mm^3/d）时，没有最小功率约束的同一压缩机的利用率百分比。要注意，当最小功率约束解除时，更大的压力区域是可行的。

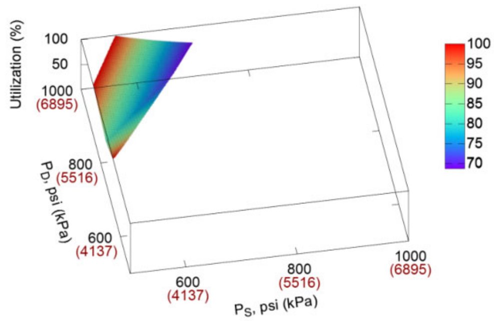

图 16 – 在流量为 200 mmcfd（5.66 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的离心式压缩机利用率百分比；最小功率设置为最大功率的 60%

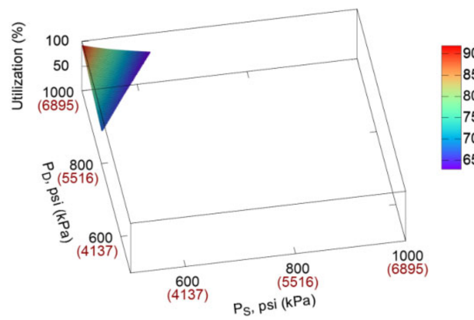

图 17 - 在流量为 150 mmcfd（4.25 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的离心式压缩机利用率百分比；最小功率设置为最大功率的 60%

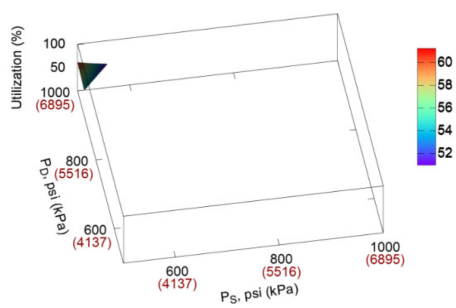

图 18 - 在流量为 100 mmcfd（2.83 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的离心式压缩机利用率百分比

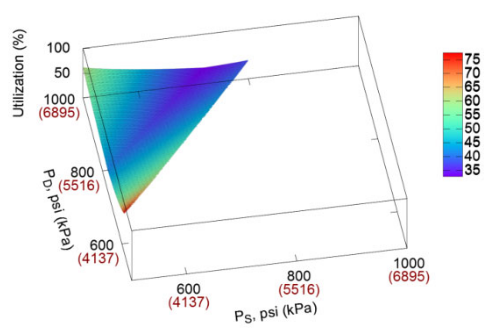

图 2 - 在流量为 100 mmcfd（2.83 Mm³/d）时，随吸入压力（PS）和排出压力（PD）变化的离心式压缩机利用率百分比，未设置最小功率

如果允许音速运行，排气量图将根据图 20 所示变化。

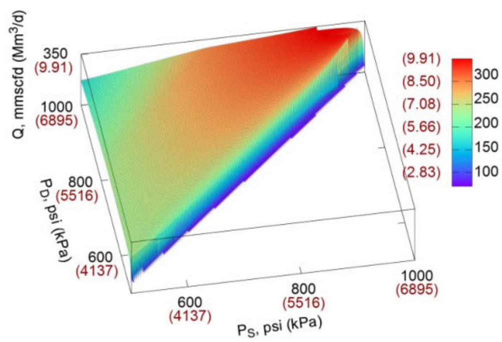

图 3 – 未设置最小功率、不允许音速运行的离心式压缩机排气量（Q）

排气量现在随着压比的增加而减少。在讨论压缩机约束对稳态分析的影响时，这一点将会很重要。

寻找可行的管道运行

除了压缩约束之外，天然气管道还有多种运行约束，比如管道最大允许工作压力（MOP）、管道最小允许工作压力（MINOP）以及供气和送气约束。当压缩机约束与整个管道拓扑和约束相结合时，有时甚至很难确定管道是否具有运行可行性。这个问题对于稳态分析显然很重要，对于管道优化也很重要，因为大多数优化算法需要一个可行的运行作为起点。

将约束可行性进行可视化的一种方法，是把所有约束转化为整个管道的总体最大和最小压力包络。如果最小压力包络低于最大压力包络，则可能存在运行可行性。否则就没有可行的管道运行。我们可以通过一次考虑几个约束条件来逐步构建这些压力包络，这样就可以确定使压力包络不可行的具体约束条件。

我们分三步讨论如何建立压力包络。第一步，我们将考虑管道 MOP 和 MINOP 约束。由此生成的压力包络称为 1 级压力包络。第二步，我们将对压缩机站引入最小吸入压力和最大排出压力，生成 2 级压力包络。第三步，我们将引入压缩机站最大极限约束，生成 3 级压力包络。

管道 MOP 和 MINOP

考虑一个如图 21 所示的带有规定 MOP 和 MINOP 的管道。为了简化图纸又不失普遍性，我们假设 MOP 和 MINOP 沿管道是分段常数，并且仅取两个离散值。当流体沿着这条管流动时，流体的压力随着距离的变化而变化。压力-距离曲线的形状取决于流速、流体、管道特性以及高度的变化。

对于给定的流速值和流体以及管道特性，*有效的* MOP（EMOP）曲线，是从 MOP 曲线下方垂直平移压力-距离曲线，直到其刚好接触到 MOP 曲线而获得的。如果 MOP 沿管道是常数，两条曲线的接触点将落在压力最高的管道上。对于具有可变 MOP 值的管道，两条曲线的接触点可能落在压力小于最高压力的管道上。有效的 MOP 曲线告诉我们，在给定条件下，管道中给定点允许的最大压力通常小于该点的 MOP 值。

同样，*有效的* MINOP（EMINOP）曲线，是从 MINOP 曲线上方垂直平移压力-距离曲线，直到其刚好接触 MINOP 曲线来获得。如果 MINOP 沿管道是常数，两条曲线的接触点将落在压力最低的管道上。对于具有可变 MINOP 值的管道，两条曲线的接触点可能落在压力大于最低压力的管道上。有效的 MINOP 曲线告诉我们，在给定条件下，管道中给定点允许的最小压力通常大于该点的 MINOP 值。

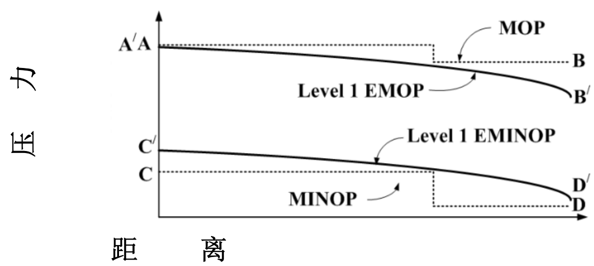

图 **21** - 1 级压力包络

如果沿管道所有点上的有效 MOP 曲线都高于有效 MINOP 曲线，则存在一个满足有效 MOP 和有效 MINOP 要求的运行策略。

图 22 显示了在较高的流速下有效 MOP 和有效 MINOP 曲线的样子，以及随之而来的较高的摩擦压降速度。在该图中，有效 MOP 曲线完全低于有效 MINOP 曲线，并且没有可行的管道运行。

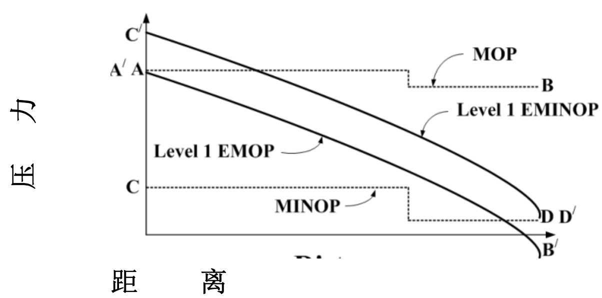

图 **22** - 不可行的 1 级压力包络

压缩机站的最小吸入压力和最大排出压力

在这一步，我们将考虑管道上游和下游端的压缩机站，并确定上游压缩机站的最大排出压力和下游压缩机站的最小吸入压力对有效 MOP 和 MINOP 曲线的影响。

在图 23 中，A/ - B/是 1 级有效 MOP，C/ - D/是 1 级有效 MINOP。如果位于该管道上游端的压缩机站的最大排出压力高于 A/，则对有效 MOP 曲线没有影响。然而，如果压缩机站的最大排出压力低于 A/，则会降低有效 MOP 曲线，如图 23 所示，生成 A// - B/，即 2 级有效 MOP 曲线。

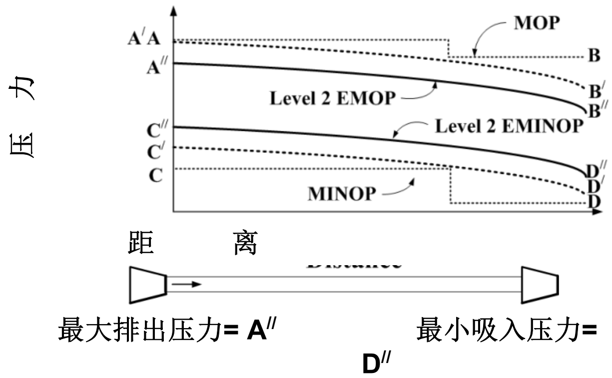

图 **4** - 2 级压力包络

如果管道下游端的最小吸入压力低于 D'，则对有效 MINOP 曲线没有影响。但是，如果最小吸入压力高于 D'，那么最小压力包络就必须升高，生成线 C'' - D''，即 2 级有效 MINOP 曲线。

压缩机站的最大极限约束

各种压缩机站最大极限约束（最大功率、最大速度、最大流量、最大压比等），限制了给定流速下，压缩机站允许输出的最大压升。最大允许压升会影响最大和最小压力包络。到目前为止，我们已经考虑了两个连续压缩机站之间的管段。确定压缩机站允许输出的最大压升，需考虑各站之间压力包络的相互作用。

以下图 24 所示的场景为例，其中有三个压缩机站和两个管段，我们已经对其计算了 2 级 EMOP（A'' - B''）和 2 级 EMINOP（C'' - D''）曲线。把第二个压缩机站能够达到的最大压升设为△Pmax。我们将考虑△Pmax 是如何影响两个管段中的 2 级 EMOP 和 EMINOP 曲线。对于压缩机站的给定吸入压力 Ps、排出压力 Pd 和压升△Pmax，我们可以认为：

Pd = Ps + △P

上述等式表示，我们可以通过最大化 Ps 和△P（因为二者都是正的）来最大化 Pd。这样，第二个压缩机站吸入侧的最大压力以及最大压升，决定了最大排出压力。如果该最大排出压力大于第二个管段的 A''，那么其 EMOP 不会受到影响。但是，如果最大排出压力低于第二个管段的 A''，则其 EMOP 需要降低，生成第二个管段的 3 级 EMOP（A'''-B'''）。

压缩机站的吸入、排出压力与其压升之间的关系可改写如下：

Ps = Pd - △P

该等式表示，我们可以通过最小化 Pd 和最大化△P 来最小化 Ps。这样，第二个压缩机站排出侧的最小压力以及第二个站的最大压升，就决定了该站的最小吸入压力。如果该最小吸入压力小于第一个管段的 D''，那么其 EMOP 不会受到影响。但是，如果最小吸入压力大于 D''，则其 EMOP 需要升高，生成第一个管段的 3 级 EMINOP（C'''-D'''）。

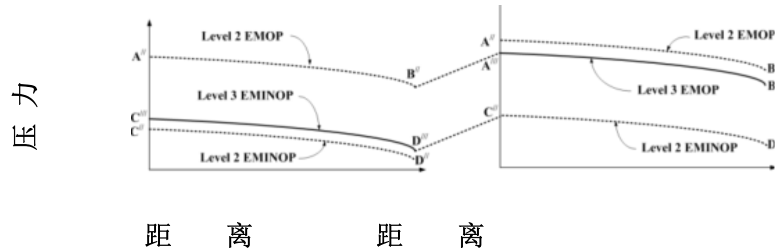

图 **24** - 3 级压力包络

如果压缩机仅包含最大极限约束，则 3 级最大和最小包络本身是运行管道的可行方式。在 3 级最大和最小包络之间的区域内，使用类似于生成最大包络的方法，可以容易地发现其他可行的管道运行。每个压缩机站的运行都可选择所需的设定压力（每个站的最大和最小包络压力之间的任何值）。然后，可以使用像最大包络计算的压缩机约束，对这些设定压力进行调整。

但是，如果压缩机包含最小约束，包络就不一定表示可行的运行场景。以下章节描述了最小约束对稳态分析、稳态优化和瞬态前视模拟的影响。

最小约束对模拟和优化的影响

以上关于管道包络的讨论，提出了一种用于确定潜在最大可行区域的方法。在由最大和最小压力包络限定的区域之外，任何设定压力的选择都是不可行的。然而，与之相反的情况也不一定是真的。在最大和最小包络界定的范围内选择设定压力，既可能可行，也可能不可行。为了了解给定管道流量下的最大和最小包络之间的可行区域，可以离散最大和最小包络之间的压力，并求解每种压力组合下压缩机的可行性。可以通过分析数据结果，选择通过管道的可行路径。下面描述了一种方法，可对给定流量下整个管道的管道运行可行性进行可视化。这种可视化技术用于指出最小压缩机限制对于稳态分析和优化的影响。

为了说明这个过程，请考虑图 25 和图 26 所示的虚拟管道。该管道将天然气从管道最西边的单一供应源运输到东部的单一输送点。

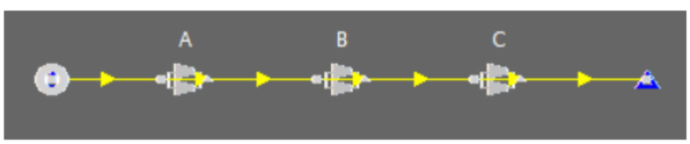

图 25 - 包含 3 个压缩机站的虚拟管道，每个站由 75 英里（121km）、28 英寸（711mm）管道分隔开；所有管道都是 75 英里（121km）长

每个压缩机站包含单台往复式压缩机“RC_fixed_s_var_cl_min_power”，如本文“往复式压缩机限制”一节所述。回想一下，该压缩机的最小功率等于最大功率的 80%，这使得它的可行运行区相当窄。

对于给定流量为 275 mmcfd（7.79 Mm^3/d）的管道，其最小和最大压力包络如图 27 所示。

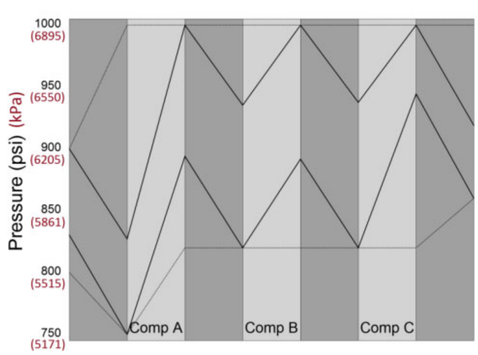

图 5 – 流量为 275 mmcfd（7.79 Mm³/d）时，最大和最小压力包络

所有三台压缩机的源节点压力和排出压力均离散为 10 个压力，最大和最小离散压力对应于模型中该点的最大和最小压力包络值。其余 8 个压力在最大和最小压力之间均匀隔开。给定这些离散压力和 275 mmcfd（7.79 Mm³/d）的流量，管道稳态方程可以求解下游压力，得到所有压缩机的离散吸入压力。图 28 第一列显示了压缩机 B 的离散吸入压力，第一行显示了该压缩机的离散排出压力。内部单元格表示在给定吸入和排出压力下压缩机计算的结果。行表示离散吸入压力（单位为 psig），而列表示离散排出压力（单位为 psig）。表格的红色部分对应着低于最小功率的压缩机计算，而表格的左上绿色角表示旁路的情况，右下绿色角表示可行的压缩区域，其中压缩机燃料以 mmcfd 为单位在表格内报告。

0.00	891.75	902.56	913.40	924.28	935.15	951.33	959.45	967.57	983.77	999.98
935.29	0.00	0.00	0.00	0.00	0.00	-2.00	-2.00	-2.00	-2.00	-2.00
921.00	0.00	0.00	0.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
906.68	0.00	0.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
892.31	0.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
877.94	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
863.48	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
856.22	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
848.97	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
834.49	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	0.48
820.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	0.48	0.52

图 6 - 流量为 275 mmcfd（7.79 Mm³/d）时，压缩机 B 的燃料和可行性表

该表在图 29 中以图形表示。

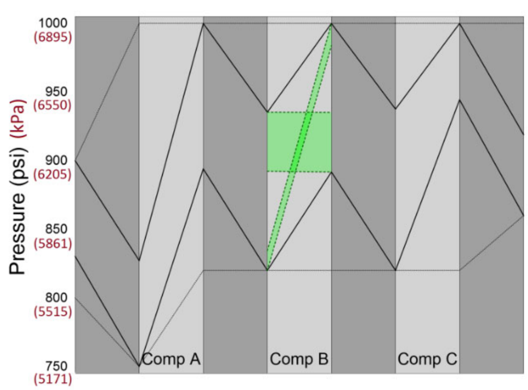

图 7 - 流量为 275 mmcfd（7.79 Mm³/d）时，压缩机 B 可行性表的图示；水平绿色框表示可行的压缩机旁路状态，而倾斜的绿色部分表示可行的压缩状态

压缩机 B 区域左侧处的深灰色和浅灰色边界表示压缩机 B 的吸入压力，同样，压缩机 B 部分的右侧浅灰色和深灰色边界表示压缩机 B 的排出压力。横跨浅灰色区域的绿线连接着与可行的压缩机运行相对应的吸入和排出压力。水平绿线对应着可行的旁路运行，而倾斜绿线跨越可行压缩对应的吸入和排出压力。注意，由于最小功率要求，只有在高压力比下压缩才有可能。还需注意，旁路运行必须对应压缩机的压降。所有低于压缩机排出侧最小压力包络的吸入压力都会导致旁路状态不可行。

图 30 将绿色部分延伸穿过压缩机 B 下游的管道

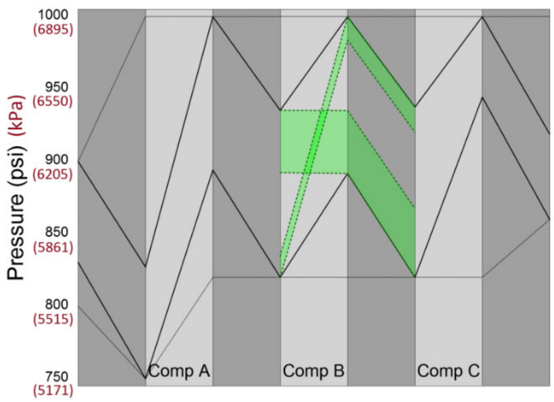

图 30 - 流量为 275 mmcfd（7.79 Mm³/d）时，压缩机 B 可行性表的图示；压缩机 B 排放口对应的可行区域，通过下游管道延伸至压缩机 C 的吸入口

压缩机 B 和压缩机 C 之间穿过深灰色区域的绿线，表示压缩机 C 吸入压力的可能选择，它对应于压缩机 B 的可行排出压力。

图 31 显示了所有压缩机的可行区域。

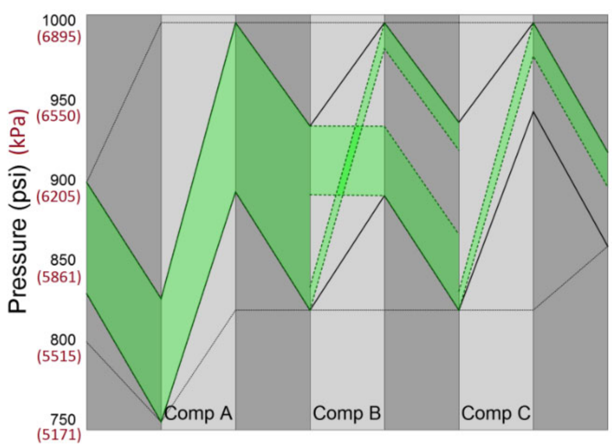

图 31 - 275 mmcfd（7.79 Mm³/d）流量下，所有压缩机可行性表的图示

压缩机 C 有一个狭窄的可行区域。由于压缩机 C 的最小功率限制，只有最大压比是可行的。压缩机 C 没有可行的旁路状态，因为压缩机排出节点处的最小压力包络大于压缩机吸入节点处的最大压力包络。穿过压缩机 B 和 C 的可行路径，只有在压缩机 C 吸入区域左侧的绿色区域与压缩机 C 吸入边界右侧的绿色区域的相交之处，才能形成。从压缩机 C 的吸入侧看，这相当于绕过压缩机 B 并通过压缩机 C 压缩气体。为了找到贯穿多个压缩机的可行运行压力，必须找到一条连接每个压缩机站可行区域的路径。图 32 蓝线之间的区域描绘了管道的潜在可行路径。

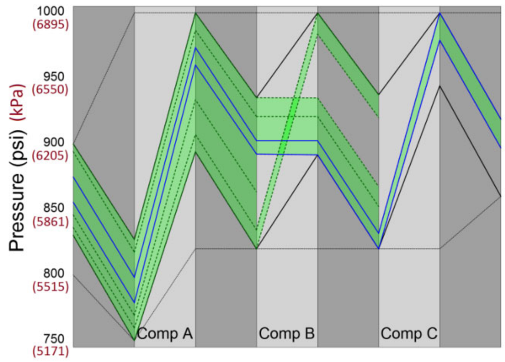

图 32 - 275 mmcfd（7.79 Mm³/d）流量下，所有压缩机可行性表的图示；蓝线之间的区域表示可行的管道运行区。

使用动态编程来寻找管道最佳控制压力时，要选择一条穿过蓝线之间绿色部分的路径，以最小化该流量下的目标函数（燃料）。有关动态编程的更多信息，请参见以下参考资料，这些资料描述了动态编程在稳态管路运行优化中的应用[2, 3]。。

为了说明随流量变化的管道可行性，管道流量从 5 mmcfd（0.14 Mm³/d）到 400 MMC F3（11.3 Mm³/d）之间变化，每次递增为 5 mmcfd（0.14 Mm³/d）。对于每个流量值都进行动态编程燃料最少化。不可行区域以红色表示。图 33 以黑色绘制所需燃料，以红色绘制不可行区域。

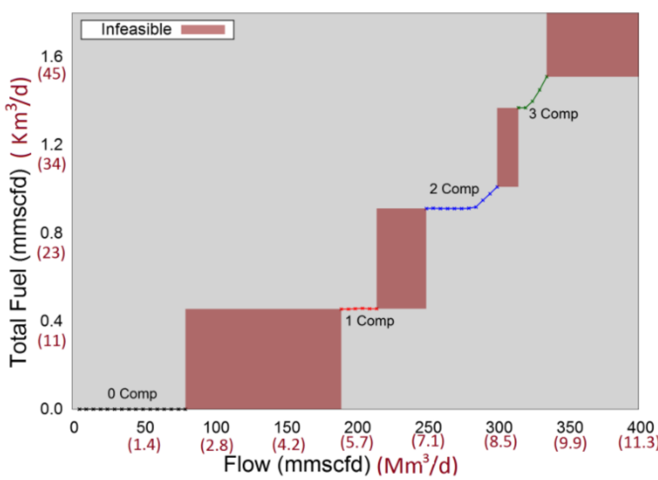

图 33 - 流量从 0 变化到 400 时的管道燃料用量。红色区域表示没有潜在管道运行可行性的流量

在 338 mmcfd（9.57 Mm³/d）流量下，最大和最小管道压力包络相交。这相当于管道的最大排气量。可行解决方案中的间隙是打破最小约束的结果，可通过检查生成的可行性图予以解释。图 34 至图 40 显示了来自每个可行区域和不可行区域的流量样本的可行性图。

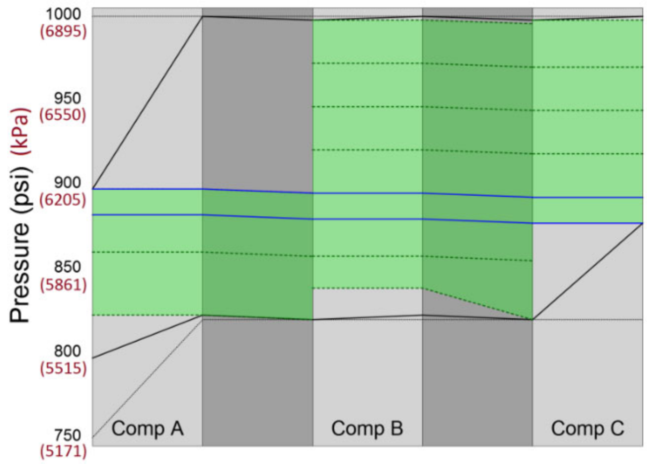

图 34 - 管道流量为 50 mmcfd（1.42 Mm³/d）时的可行性图。唯一可行的运行模式是完全旁路所有压缩机。

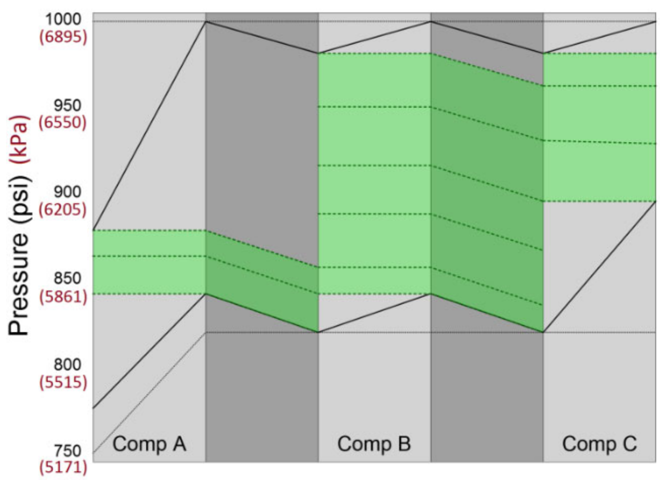

图 8 – 150 mmcfd（4.25Mm³/d）管道流量时的可行性图。管道无法运行，因为整个管道的可行区域之间没有连接。管道的摩擦压力损失太大，无法满足最小输送压力，并且由于最小功率要求，压缩是不可能的。

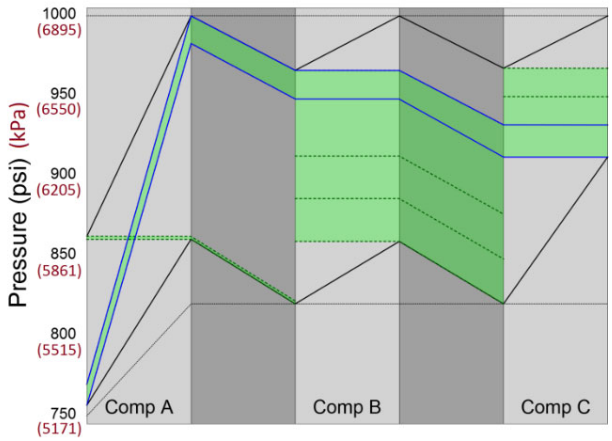

图 9 - 200 mmcfd（5.66 Mm³/d）管道流量时的可行性图。压缩机 A 正在压缩，而压缩机 B 和 C 被旁路

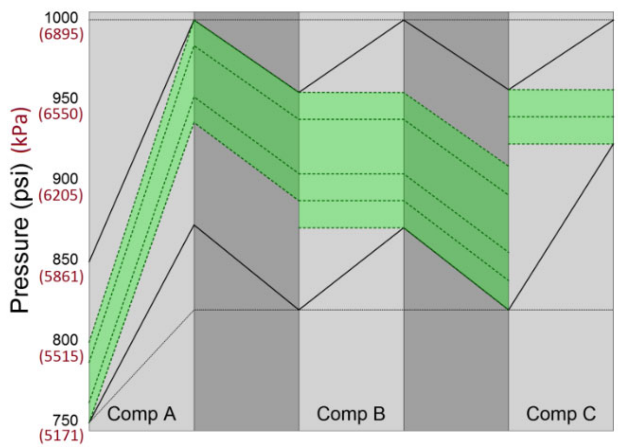

图 10 - 230 mmcfd（6.51 Mm³/d）管道流量时的可行性图。管道运行是不可行的，因为压缩机 B 的旁路状态与压缩机 C 的旁路状态不相交

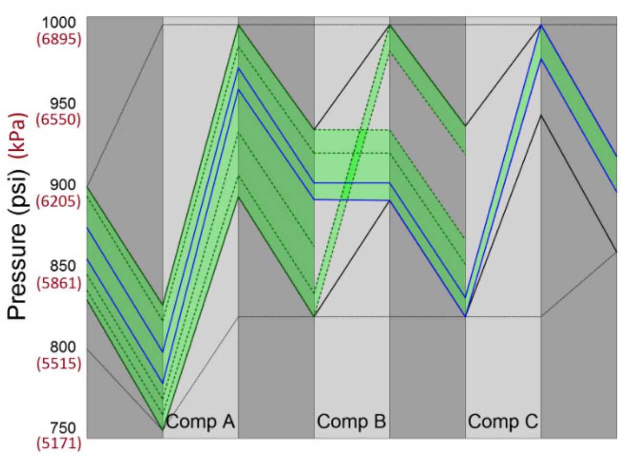

图 11 - 275 mmcfd（7.79 Mm³/d）管道流量时的可行性图。可行运行包括旁路压缩机 B 并通过压缩机 A 和 C 进行压缩

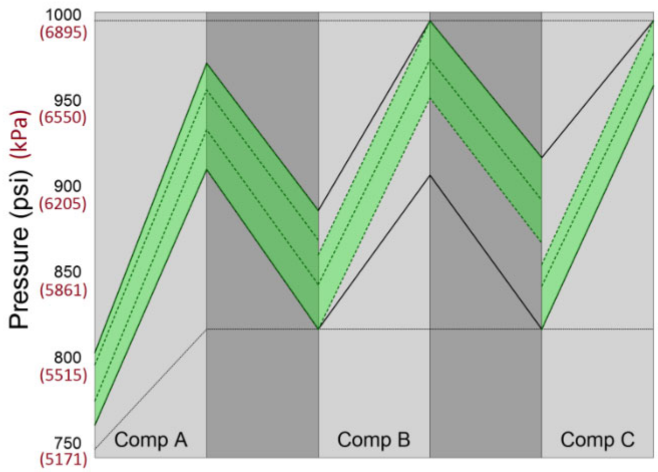

图 12 - 310 mmcfd（8.79 Mm³/d）管道流量时的可行性图。管道运行是不可行的，因为气体以过高的压力进入压缩机 C

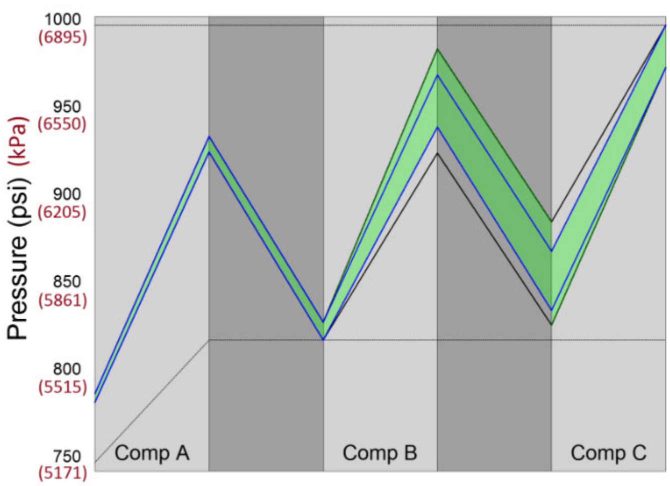

图 13 - 335 mmcfd（9.49 Mm³/d）管道流量时的可行性图。所有三个压缩机都在压缩

可行区域中的这些间隙是压缩机最小约束的直接结果，导致解决方案不可行。第一个可行区域包括通过三个旁路站在整个管道自由流动。随着流量的增加，摩擦压力损失增加到了打破最小输送压力 860 psi（5930kPa）的程度。这时需要压缩来提高气体压力。但是，一台压缩机在管道 MAOP 时排放所需的功率小于压缩机的最小功率，这使得运行没有可行性。

流量为 185 mmcfd（5.24 Mm³/d）时，管道流量和压降就足以使压缩机 A 在其最小功率之上运行。然而，当流量超过 215 mmcfd（6.09 Mm³/d）时，就有必要启动另一台压缩机，以避免打破最小输送压力。在这种流量下运行两台压缩机，无法不打破最小功率或最大压力要求。当流量增加到 245 mmcfd（6.94 Mm³/d）时，可以在不打破最小功率限制的情况下，启动第二台压缩机。类似的推理可用于解释 300（8.50）和 320 mmcfd（9.06 Mm³/d）之间的可行性间隙，这种情况必须运行三台压缩机以满足最低压力要求。

稳态优化

以上讨论说明的是，压缩机站的最小限制，是如何随着管道流量的变化，给可行的管道运行带来间隙的。这些可行性中的间隙，有可能导致稳态优化给出次优答案。稳态优化通常使用两步优化过程。给定一组环流，使用动态编程求解压缩机运行压力，使目标最小化。使用外部流量优化算法，改变模型的环流，获得最佳管网流量分布。通过交替进行流量更新步骤和动态编程步骤，推进优化。

流量更新步骤通常使用直接搜索法或下降法来改变流量，搜寻最佳目标。这些方法从最初的猜测开始，逐步走向最佳。如果函数是不连续的，它们会陷入局部极小值。在上述示例中，如果目标是找到系统中的最大流量，则流量优化可能无法"跨越"流量为 85（2.4）、220（6.2）或 305 mmcfd（8.6 Mm^3/d）时的不可行的间隙。由于本文引入的示例问题是一维的，因此可以使用更稳健的技术，通过更完整地搜索解决方案空间，找到最优解。然而，随着系统中能够优化的流量数量的增加，在不可行空间中寻找样点的计算代价更高，因此解决方案更有可能最终成为局部极小值。

这一示例，重点在于往复式压缩机最小功率约束所产生的可行性间隙。其他的最小约束也可能导致可行性间隙：如本文前面所述，离心式压缩机可能具有最小速度和最小功率约束。这些约束也会减少可行区域，影响管道的可行性。一个更极端的例子是固定速度、固定式往复式压缩机。固定速度、固定余隙的压缩机，几乎总能导致动态编程无法找到可行的运行条件（除了不重要的旁路状态）。如上所述，动态编程要求，对于给定的流量，压缩机运行是在一系列离散的吸入和排出压力下计算的。如果离散的流量、吸入和排出压力与图 2 所示的余隙腔组合不完全一致，这个点就不可行。

我们建议，在稳态优化过程中，如果会带来不可行的点，则最小压缩机约束应予以忽略。只要最大和最小压力包络不交叉，就允许所有管道流量可行。然而，动态编程这个步骤应该设置成在不打破最小限制的情况下，优先选择运行条件。

稳态分析

下一个问题是：最小压缩机约束如何影响稳态分析？

稳态分析中的压缩机通常设置为保持吸入或排出压力，而不是设置为以排气能力运行。如果压缩机没有足够的排气量来输送所需的设定压力，则压缩机须切换到排气量控制，并放弃设定压力。

我们考虑一下"往复式压缩机限制"一节中的压缩机"RC_var_s_var_cl"。由于没有最小功率限制，排气量随着吸入压力的增加和排出压力的降低而增加。我们假设排出压力设定为 900 psig（6210 kPa），管道水力学要求流量为 200 mmcfd（5.66 Mm^3/d），吸入压力为 575 psig（3960 kPa）。在这些条件下，压缩机"RC_var_s_var_cl"的排气量为 129 mmcfd（3.65 Mm^3/d），因此稳态会将压缩机切换到排气量控制，压比会降低，直到压缩机的排气量足以满足流量。

在这种情况下，如果吸入压力或多或少保持不变，排出压力将降至 775psig（21.9kPa）以下。

然而，如果强制执行最小功率限制，则只有非常窄的可行运行区间。对于上述示例，在流量为 200 mmcfd（5.66 Mm^3/d）、吸入压力为 575 psig（3960kPa）时，可行区域在排出压力 725 psig（5000kPa）和 750 psig（5170kPa）之间。如果选择的设定压力在该狭窄可行区域之上，排气量控制就可以允许压缩机找到可行的管道解决方案。相反，如果选择的排出设定压力在这个可行区域之下，获得可行性的唯一选择是增加设定点或关闭该站。

当压缩机的排气量随着压比的增加而增加时，也会出现类似的问题。离心式压缩机的排气量由音速线的运行来确定时（见图 14），这种情况就会发生。同样，如果压缩机运行时的水力条件是流量达到了音速线，那么将压缩机切换到排气量控制会导致压比增加而不是降低。因此，找到运行可行性的唯一方法是关闭压缩机或增加压比，从而打破设定压力。如前所述，如果压缩机运行在音速区域内得到许可，这个问题就会消失（见图 20）。

由于最小压缩机限制导致可行区域可能狭窄，作者建议，对于稳态分析，低于最小值的运行应通过警告而不是约束来处理。另一种可能性是，如果压缩机运行低于最小限制，则允许用算法来增加压缩机的设定压力。

瞬态分析

最小约束影响瞬态分析的一个主要方式是在压缩机站启动过程中。如果压缩机受到最小功率或最小速度的限制，在启动过程中，压缩机将需要打破这些最小限制，以便从 1.0 的压比前进到期望的运行比。

压缩机启动需要将压缩机从静止状态带到可行的运行点。这是一个内在的动态过程，以受控的方式进行，期间可能会涉及到压缩机循环气体，直到压缩机达到速度。通常，压缩机在达到运行条件之前不会处于负载状态。

解决这个问题有多种方法。压缩机的启动过程可以详细建模。另一种选择是将最小限制视为警告而非约束，类似于我们对稳态的建议。在压缩机站启动顺序中，这些警告可以忽略。第三种选择是完全忽略不可行的压缩机运行，直到找到可行区域。

结论

我们建议，在管道模拟或优化软件中使用压缩机模型之前，要把压缩机模型描述的可行区域可视化。如果在运行区内有任何“孔”，或者如果有非凸或非连续区域，那么在管道建模和优化过程中，应该予以注意。在使用优化软件中的管道建模时，对压缩机最小限制带来的“孔”进行填充，始终是个好主意。

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致谢

作者要感谢 Nicholas Russ 先生，感谢他提供的宝贵图表和他对本文的深切关心。作者还要感谢 DNV GL 管理层对这项工作的大力支持。

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附录 1

属性	RC_fixed_s_fixed_cl
最大功率	3000 hp（2237 kW）
最小功率	0 hp（0 kW）
扫气容积的数量	5
余隙/扫气容积的数量	8
扫气容积	15.5, 13.5, 11.5, 10, 7.5 ft^3 0.439, 0.382, 0.326, 0.283, 0.212 m^3
余隙（%）	90,100,120,140,160,180,200,220
最高速度	300 rpm
最低速度	300 rpm

表 2 – 往复式压缩机数据

WHITEPAPER

Synergi^{TM} Gas

The effects of compressor constraints in pipeline modeling and optimization

AUTHOR: Andrew Daniels, DNV GL, Sanjay Yadav, DNV GL, Richard Carter, DNV GL

DATE: May 2014

This whitepaper was originally presented by the authors at the annual Pipeline Simulation Interest Group (PSIG) conference.

ABSTRACT

The purpose of this paper is to discuss the effects of compressor station constraints on pipeline modeling and optimization results. Compressor definitions within pipeline models contain constraints which limit their region of operation. These constraints can include maximum and minimum speeds, compression ratios, cylinder clearances and powers. Operation of centrifugal compressors is also limited by their surge and sonic flows. These constraints are used to reflect either actual compressor limitations or to limit the regions of operation to more efficient areas. In the field, compressors temporarily need to violate some of these constraints during startup and shutdown sequences. In optimization algorithms, enforcing minimum constraints can sometimes cause sub-optimal answers because they produce “holes” in the feasible space which can lead to local minima. This paper will investigate a methodology for visualizing the effects of various constraints that a gas pipeline must satisfy for feasible operation. It will also lay out the problems associated with using steady state compressor models containing minimum constraints within pipeline simulation and optimization software.

The effect of compressor constraints on simulation results will be studied by examining how they impact the maximum and minimum pressure envelopes. The value of this approach is that it makes it easier to visualize how different pipeline constraints impact the available range of pipeline operation. This paper will also alert modelers to the issues associated with using minimum constraints on compressor stations during pipeline simulation and optimization.

INTRODUCTION

Compressor stations are placed throughout transmission pipelines to boost pressure allowing gas to be transported over long distances. Mathematical models represent compressor stations within pipeline analysis and optimization software. These models estimate compressor operating cost and predict the feasibility of compressor operation under the given hydraulic conditions. Pipeline analysis results are only as good as the compressor model’s ability to predict the station’s capability and fuel usage.

A number of compressor models exist which approximate the operation of compressors in the field. These models range from fairly simplistic theoretical compressors which describe the capacity of the compressor with a maximum power and efficiency, to sophisticated dynamic compressor models. Given the large numbers of compressor calculations required within pipeline modeling and optimization software, the compressor models chosen should strike a balance between computational expense and accuracy. For this reason, pipeline optimization and modeling software frequently use steady state compressor models. A detailed steady state compressor model will accurately predict operating cost as well as the feasible operating region of the compressor it is representing.

Some compressor constraints bound unachievable operating conditions, while other compressor constraints are optionally used to constrain compressor operation to more efficient operating regions. An understanding of how constraints affect simulation and optimization results is useful in the process of selecting which compressor constraints to enforce.

This paper will begin by discussing some of the constraints that reciprocating and centrifugal compressors have. Next, it will describe a method for visualizing these constraints within the pipeline. The paper will discuss the effects of enforcing these constraints within steady state analysis and optimization and will conclude with some compressor modeling recommendations.

RECIPROCATING COMPRESSOR CONSTRAINTS

Reciprocating compressors use pistons to compress gas within cylinders similar to the way a bicycle pump works. A number of different drivers are used to power reciprocating compressors including natural gas engines, electric motors, diesel engines, etc. These drivers can be either variable speed drivers or fixed speed drivers.

As pipeline conditions change, controllers must adjust the compressor's operation to meet desired pressure conditions. Three of the main ways for controlling reciprocating compressor operation are through varying compressor speed, changing the amount of cylinder clearance, and changing the swept volume of the machine. Variable speed drivers speed or slow the pistons, thus controlling the flow through the cylinders. Also, many reciprocating compressors have either fixed or variable clearance pockets on the cylinder. When these pockets are opened, the volume of the cylinder is increased without increasing the amount of swept volume. This excess cylinder volume reduces the amount of compression in the cylinder thus reducing the flow and required driver power. Lastly, by deactivating cylinder ends, the total swept volume of the compressor is reduced thus reducing the flow and power required to operate the compressor.

Fixed-speed fixed clearance reciprocating compressors

Some gas-fired reciprocating compressor drivers have a very small range in operating speeds. These compressors cannot operate efficiently under loads when their speed drops below the minimum operating speed. Some compressor drivers have such a narrow operating speed range that they are essentially fixed speed machines.

As mentioned above, cylinder clearance can be increased by adding either variable or fixed clearance pockets. Fixed speed compressors with fixed clearance pockets have discrete operating point surfaces corresponding to each clearance pocket. This is illustrated using "RC_fixed_s_fixed_cl", a single stage 3000 hp (2237 kW) reciprocating compressor having 5 unique swept volumes each containing 8 clearance steps. This gives a total of 40 clearance and swept volume combinations. For more information about RC_fixed_s_fixed_cl, please see Table 2 in Appendix A.

Table 1 shows the achievable reciprocating compressor flows for the clearance steps with a swept volume of 13.5 in2 (87.1 cm2), a suction pressure of 800 psig (5516 kPa) and a discharge pressure of 950 psig (6550 kPa). The compressor cannot deliver flows between the values represented in the table. Thus, the compressor can flow at 299.7 mmcfd (8.487 Mm3/d) and at 290.6 (8.228 Mm3/d) mmcfd but not 295 mmcfd (8.353 Mm3/d) under these pressure conditions.

Clearance	Flow		Power	
(%)	(mmcfd)	(Mm^3/d)	(hp)	(kW)
90	304.3	8.617	2709	2020
100	299.7	8.487	2668	1990
120	290.6	8.228	2587	1929
140	281.4	7.969	2505	1868
160	272.3	7.710	2424	1807
180	263.1	7.451	2342	1747
200	254.0	7.192	2261	1686
220	244.8	6.933	2179	1625

Table 1 - Compressor RC_fixed_s_fixed_cl operation at a suction pressure of 800 psig (5516 kPa) and a discharge pressure of 950 psig (6550 kPa) over a range of clearances.

Figure 1 shows the capacity of compressor RC_fixed_s_fixed_cl over a pressure range of 500 psig (3447 kPa). Notice that the capacity is defined for all suction pressures less than the discharge pressure. The capacity of the compressor at lower pressure ratios is limited by the minimum clearance and maximum swept volume. At higher ratios, the power required to operate the compressor at its minimum clearance exceeds the driver's maximum power. The steps in the capacity plot indicate hydraulic conditions under which clearance amounts must be increased to keep the required power lower than the maximum power.

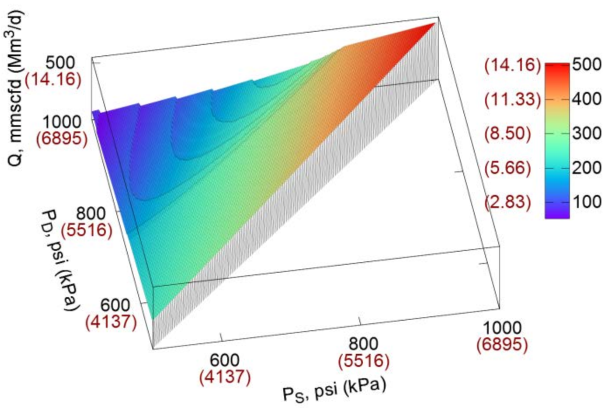

Figure 1 - Capacity (Q) of compressor RC_fixed_s_fixed_cl as a function of suction (P_S) and discharge (P_D) pressures

When this compressor is operated at a given flow, the results are discontinuous. Figure 2 is a plot of the compressor's percent utilization as a function of suction pressure (PS) and discharge pressures (PD) for a flow of 300 mmcfd (8.50 Mm3/d) with a pressure discretization of 2.5 psi (17 kPa).

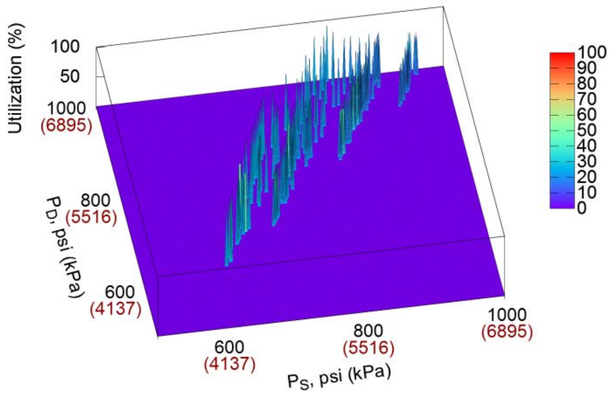

Figure 2 - Percent utilization as a function of suction (PS) and discharge (PD) pressures for a flow of 300 mmcfd (8.50 Mm3/d)

The scattered feasible points indicate calculations which happen to be performed at pressure and flow combinations that corespond to available clearance steps. Note that most of the region is infeasible.

However, if one allows a continuum of clearances instead of fixed clearances, a larger section of the region fills in. The following chart (Figure 3) is a plot of utilizations for a flow of 300 mmcfd (8.50 Mm3/d) at varying suction and discharge pressures for RC_fixed_s_var_cl. This compressor is identical to the previous one except that the clearance is allowed to vary continuously between the maximum and minimum values for a given swept volume.

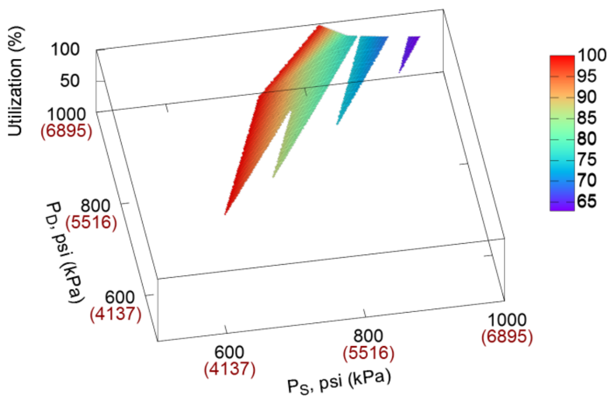

Figure 3 - Percent utilization as a function of suction (PS) and discharge (PD) pressures for a flow of 300 mmcfd (8.50 Mm3/d) allowing continuous clearance

In the field pipelines operate under transient conditions. While a model will solve for operation between two clearance/pocket combinations, in the field the operator of the pipeline will achieve similar hydraulic results by alternating operation between the two adjacent clearance steps. The authors recommend always using a continuous clearance option when modeling reciprocating compressors with fixed clearance pockets in pipeline modeling and optimization software.

The remaining gaps in the feasible region are due to swept volume changes. Since swept volumes are changed in a stepwise fashion, at this flow there are gaps in operation corresponding to the transition from one swept volume to the next. These gaps may be filled in by allowing the compressor to operate below minimum speed.

Figure 4 is a plot of RC_var_s_var_cl under the same range of pressures at a flow of 300 mmcfd (8.50 Mm3/d). The only difference between this compressor and the previous one is that the speed is allowed to vary continuously between 250 and 300 rpm. The discontinuities in the feasible operating region vanish when a range of speeds is allowed. The authors recommend using the maximum speed range achievable in the field when modeling reciprocating compressors in pipeline software.

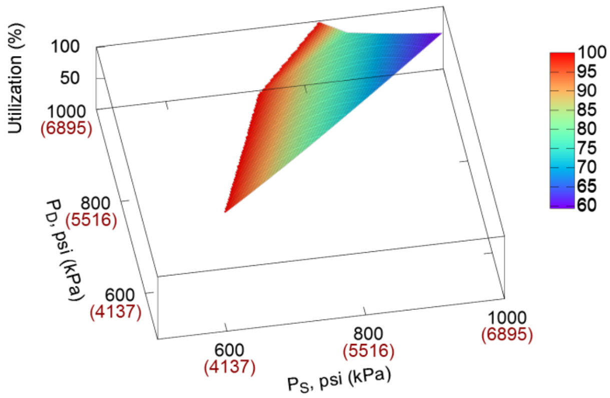

Figure 4 - Percent utilization as a function of suction (PS) and discharge (PD) pressures for a flow of 300 mmcfd (8.50 Mm3/d) allowing continuous clearance and allowing the speed to vary between 250 and 300 rpm

Minimum power

One of the costs of operating a compressor is the amount of fuel used. The compressor's fuel efficiency can be described by the percentage of the gas entering the compressor that is used as fuel. When compressors are operated under low power conditions, the percentage of fuel used can increase by more than 130% over the usage at higher power conditions [1].

Compressor modelers sometimes specify a lower limit to the power in an attempt to operate the compressor in the most efficient manner possible. If minimum power is treated as a constraint, the compressor will shut down if the power is less than the minimum limit. If minimum power is treated as a warning, compressors operating under minimum power will generate an alarm. The authors have seen models of reciprocating compressors with the minimum power set as high as 80% of the maximum power in an attempt to run the machines in the most efficient manner possible.

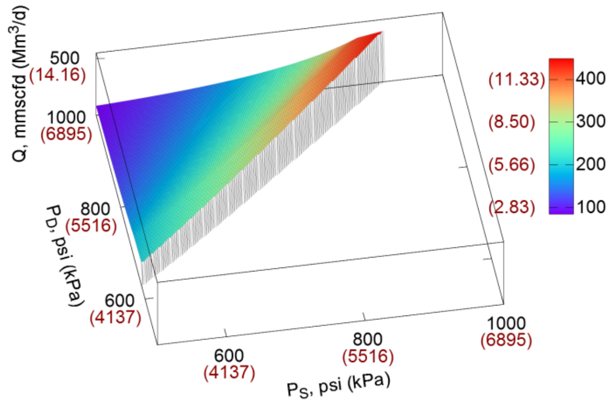

Figure 5 - Capacity as a function of suction (PS) and discharge (PD) pressures for compressor RC_fixed_s_var_cl_min_power

Figure 5 shows the capacity of the feasible region of compressor RC_fixed_s_var_cl_min_power, a compressor identical to RC_fixed_s_var_cl except for an enforced minimum power constraint of 2400 hp (1790 kW). Notice that the feasible region corresponding to the compressor operating under capacity conditions is fairly large, but it begins at a compression ratio significantly larger than 1.0. However, the feasible region for any given flow is quite narrow and shifts in pressure as the flow varies. Figure 6 through Figure 12 show the percent utilization for the feasible operating region of compressor RC_fixed_s_var_cl_min_power operating at a series of flows between 350 mmcfd (9.91 Mm3/d) and 50 mmcfd (1.42 Mm3/d). Notice that the feasible region is quite narrow for each individual flow, and it moves from lower compression ratios to higher compression ratios as the flow decreases.

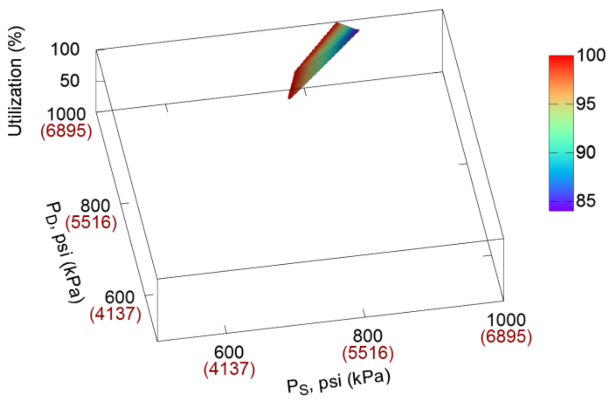

Figure 6 - Percent utilization as a function of suction (PS) and discharge (PD) pressures at a flow of 350 mmcfd (9.91 Mm3/d) for compressor RC_fixed_s_var_cl_min_power

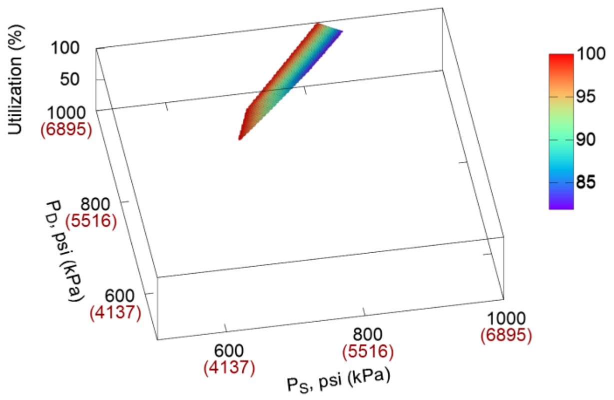

Figure 7 - Percent utilization as a function of suction (PS) and discharge (PD) pressures at a flow of 300 mmcfd (8.50 Mm3/d) for compressor RC_fixed_s_var_cl_min_power

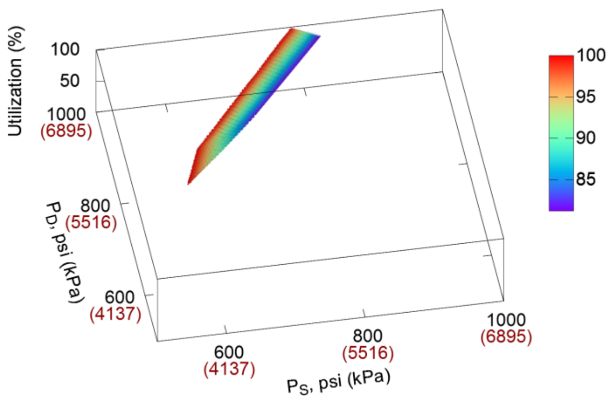

Figure 8 - Percent utilization as a function of suction (PS) and discharge (PD) pressures at a flow of 250 mmcfd (7.08 Mm3/d) for compressor RC_fixed_s_var_cl_min_power

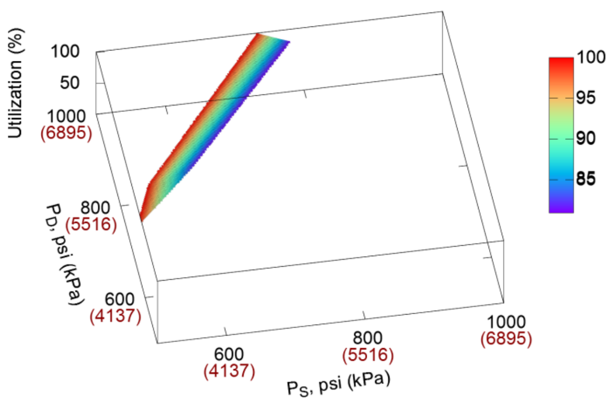

Figure 9 - Percent utilization as a function of suction (PS) and discharge (PD) pressures at a flow of 200 mmcfd (5.66 Mm3/d) for compressor RC_fixed_s_var_cl_min_power

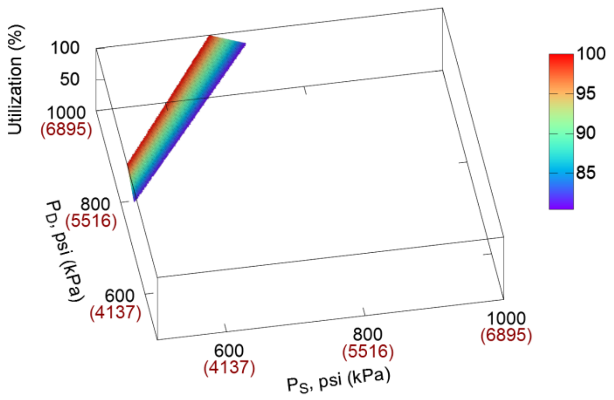

Figure 10 - Percent utilization as a function of suction (PS) and discharge (PD) pressures at a flow of 150 mmcfd (4.25 Mm3/d) for compressor RC_fixed_s_var_cl_min_power; calculations were performed over pressures ranging from 500 psig (3447 kPa) to 1000 psig (6895 kPa)

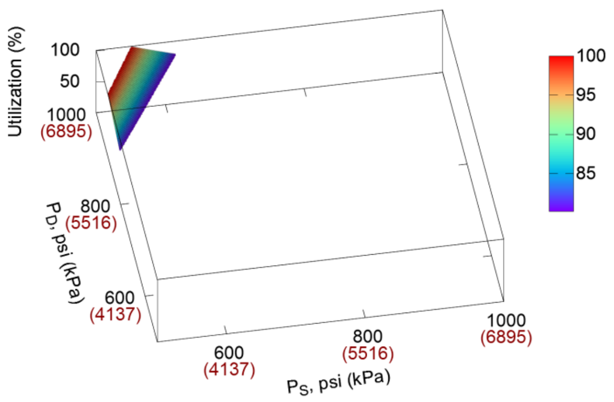

Figure 11 - Percent utilization as a function of suction (PS) and discharge (PD) pressures at a flow of 100 mmcfd (2.83 Mm3/d) for compressor RC_fixed_s_var_cl_min_power

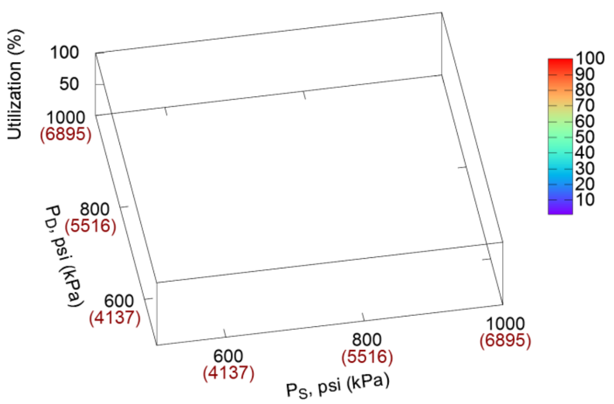

Figure 12 - Percent utilization as a function of suction (PS) and discharge (PD) pressures at a flow of 50 mmcfd (1.42 Mm3/d) for compressor RC_fixed_s_var_cl_min_power

If the minimum power constraint is removed, the feasible regions of the individual flows increase significantly as can be seen by comparing Figure 3 with Figure 7.

CENTRIFUGAL COMPRESSOR LIMITS

Centrifugal compressor operation can be represented by head flow maps. These maps describe the operating efficiency and speed as a function of the adiabatic head and flow conditions through the compressor, and give the boundaries of feasible compressor operation.

Centrifugal compressors used in pipeline applications are typically driven by natural gas fired turbines. Like reciprocating compressors, these drivers also may have a limited range of operating speeds. The authors have seen the minimum speed set to 40%-75% the maximum speed.

The following discussion will center on a centrifugal compressor "CC" modeled with a maximum power of 6000 hp (4474 kW) and a head flow map as described in Figure 13:

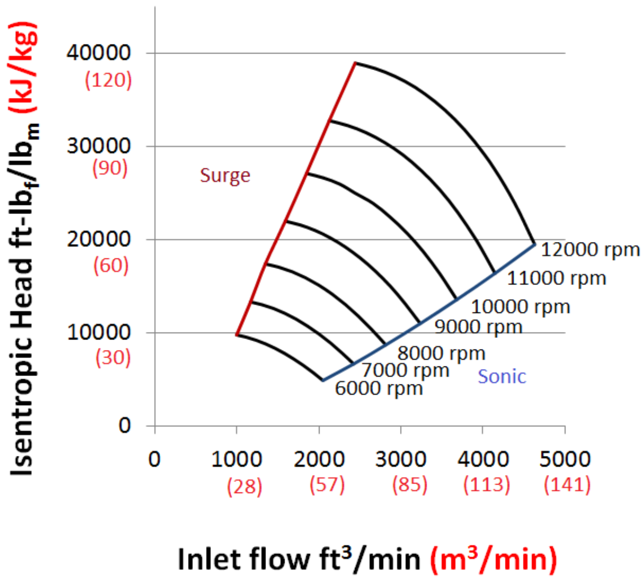

Figure 13 - Head flow map for compressor CC

The blue line at the right hand side of the centrifugal compressor map represents the sonic limit. The compressor can operate to the right of this line but discharge temperatures increase and the efficiency is unknown. The top curve represents the operation at maximum speed and the bottom curve represents the minimum speed. The curve at the left of the chart represents the surge region, where the compressor cannot operate. The centrifugal compressor demonstrated in this paper is set up to recycle gas flow to prevent operation in areas left of the surge line.

Figure 14 shows a centrifugal compressor capacity map for compressor CC. Notice that the compressor does not begin to flow until the compression ratio is significantly above 1.0 due to minimum speed requirements. Also notice that the capacity initially increases with increasing compression ratio. This corresponds to the compressor operating along the sonic line.

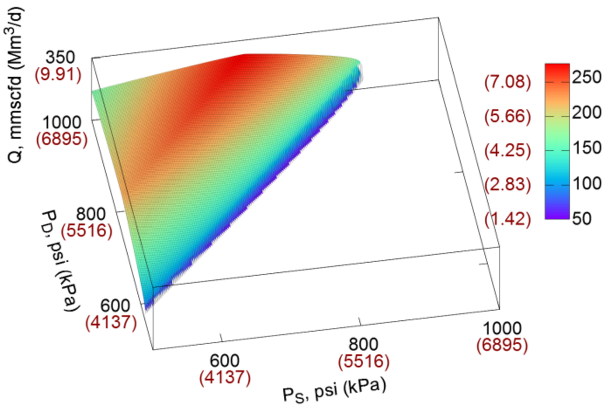

Figure 14 - Centrifugal compressor capacity as a function of suction (PS) and discharge (PD) pressures

Minimum Power

Like reciprocating compressors, centrifugal compressors often have minimum power requirements. These minimum power requirements typically stem from efficiency and emissions concerns. Figure 15 illustrates the capacity of the above compressor with a minimum power set to 60% of the maximum power.

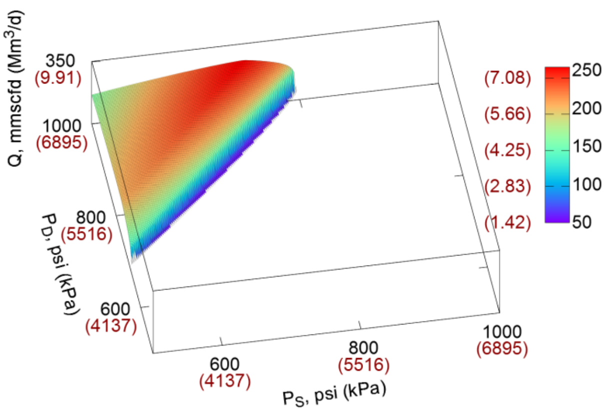

Figure 15 - Centrifugal compressor capacity as a function of suction (PS) and discharge (PD) pressures; minimum power is set to 60% of maximum power

A comparison between Figure 14 and Figure 15 suggests that compressors with minimum power limits must have significantly higher pressure ratios before they begin operating than compressors without minimum power limits defined.

Figures 16 through 18 show utilization plots of compressor "CC" under various flow conditions. Notice that the feasible region shrinks with decreasing flow due to the minimum power limits. By contrast, Figure 19 shows the percent utilization of the same compressor without the minimum power constraint at a flow of 100 mmcfd (2.83 Mm3/d). Notice that a much larger pressure region is feasible when the minimum power constraint is removed.

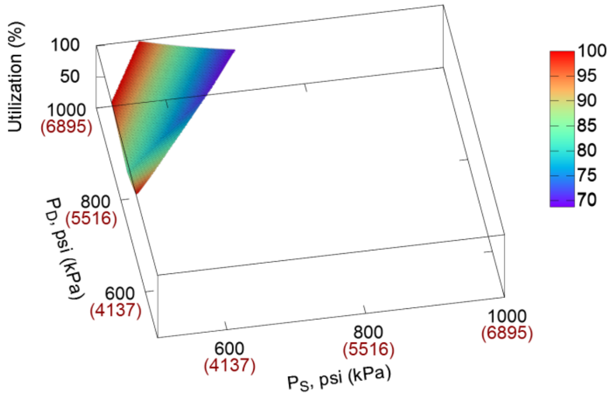

Figure 16 - Centrifugal compressor percent utilization at a flow of 200 mmcfd (5.66 Mm3/d) as a function of suction (PS) and discharge (PD) pressures; minimum power is set to 60% of maximum power

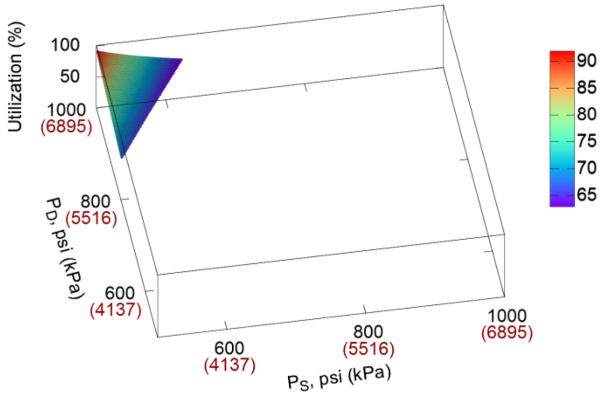

Figure 17 - Centrifugal compressor percent utilization at a flow of 150 mmcfd (4.25 Mm3/d) as a function of suction (PS) and discharge (PD) pressures; minimum power is set to 60% of maximum power

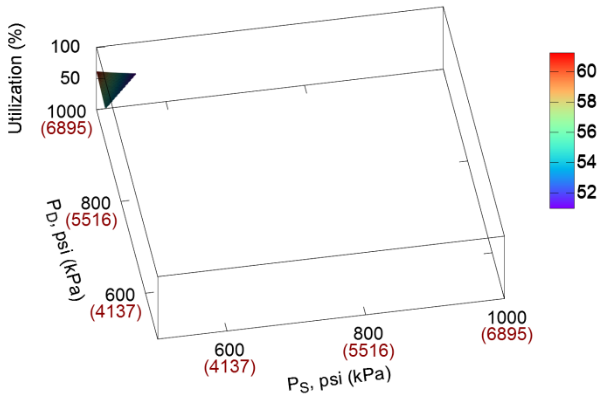

Figure 18 - Centrifugal compressor percent utilization at a flow of 100 mmcfd (2.83 Mm3/d) as a function of suction (PS) and discharge (PD) pressures

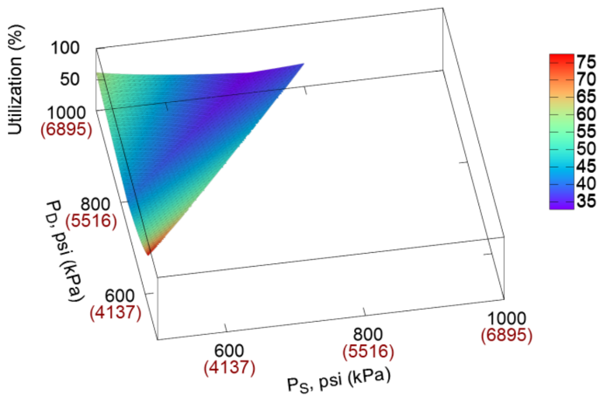

Figure 2 - Centrifugal compressor percent utilization at a flow of 100 mmcfd (2.83 Mm^3/d) as a function of suction (P_S) and discharge (P_D) pressures with no minimum power applied

If sonic operation is allowed, the capacity map changes to the plot shown in Figure 20.

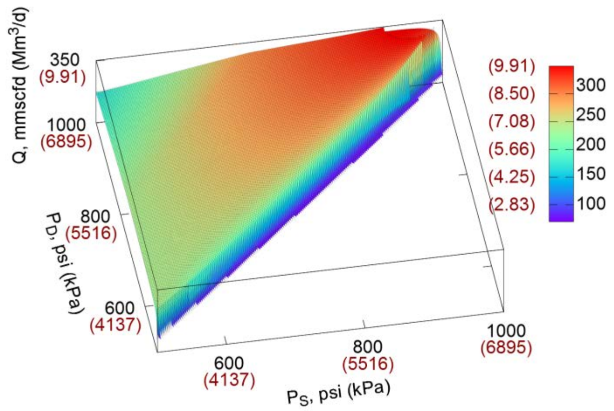

Figure 3 - Centrifugal compressor capacity (Q) with no minimum power and sonic operation allowed

The capacity now decreases with increasing compression ratio. This will become important in the discussions on the effects of compressor constraints on steady state analyses

FINDING A FEASIBLE PIPELINE OPERATION

In addition to compression constraints, gas pipelines have multiple operational constraints such as maximum allowed pressure in pipes (MOP), minimum allowed pressure in pipes (MINOP) as well as supply and delivery constraints. When compressor constraints are combined with overall pipeline topology and constraints, it is sometimes difficult to determine if the pipeline even has a feasible operation. This issue is obviously important for steady state analyses, and is also important for pipeline optimization since most optimization algorithms require a feasible operation as a starting point.

One approach to visualize the feasibility of constraints is to translate all constraints into overall maximum and minimum pressure envelopes for the entire pipeline. If the minimum pressure envelope is everywhere below the maximum pressure envelope, then there may be a feasible operation available. Otherwise there is no feasible pipeline operation. We can build these pressure envelopes incrementally by considering a few constraints at a time, which allows us to determine the specific constraint that makes the pressure envelopes infeasible.

Let us discuss how to build the pressure envelopes in 3 stages. In stage 1, we will consider pipe MOP and MINOP constraints. The pressure envelopes resulting from these will be referred to as Level 1 pressure envelopes. In stage 2, we will introduce minimum suction and maximum discharge pressures for compressor stations resulting in Level 2 pressure envelopes. In the third stage, we will introduce compressor station maximum limit constraints resulting in Level 3 pressure envelopes.

Pipe MOP and MINOP

Consider a pipe with specified MOP and MINOP as shown in Figure 21. To simplify the drawings without loss of generality we will assume that MOP and MINOP are piecewise constant along the pipe and take on only two discrete values. As fluid flows along this pipe, the pressure in the fluid changes with distance. The shape of the pressure versus distance curve depends on flow rate, fluid and pipe characteristics and elevation changes.

For a given value of flow rate and fluid and pipe characteristics, the *effective* MOP (EMOP) curve is obtained by translating the pressure versus distance curve vertically from below the MOP curve until it just touches the MOP curve. If the MOP was constant along the pipe, the contact between the two curves would occur at a point along the pipe where the pressure was highest. For a pipe with variable MOP values, the contact between the two curves can occur at a point along the pipe where the pressure is less than the highest pressure. The effective MOP curve tells us that under the given conditions, the maximum pressure allowed at a given point in the pipe is usually less than the MOP value at that point.

Similarly, the *effective* MINOP (EMINOP) curve is obtained by translating the pressure versus distance curve vertically from above the MINOP curve until it just touches the MINOP curve. If the MINOP was constant along the pipe, the contact between the two curves would occur at a point along the pipe where the pressure was lowest. For a pipe with variable MINOP values, the contact between the two curves can occur at a point along the pipe where the pressure is more than the lowest pressure. The effective MINOP curve tells us that under the given conditions, the minimum pressure allowed at a given point in the pipe is usually greater than the MINOP value at that point.

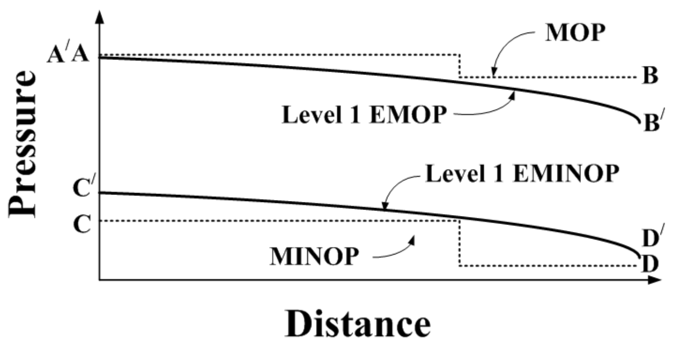

Figure 21 - Level 1 pressure envelopes

If the effective MOP curve is above the effective MINOP curve at all points along the pipe, then there exists an operational strategy that will satisfy the effective MOP and effective MINOP requirements. Figure 22 shows what the effective MOP and MINOP curves might look like at a higher flow rate with consequent higher rate of frictional pressure drops. In this figure, effective MOP curve is completely below the effective MINOP curve and there is no feasible pipeline operation available.

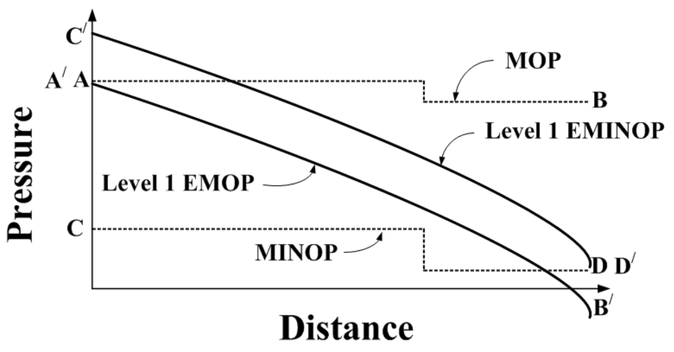

Figure 22 - Infeasible level 1 pressure envelopes

Minimum suction and maximum discharge pressures for compressor stations

In this stage we will consider the compressor stations at the upstream and downstream ends of the pipe and determine the effect of the maximum discharge pressure of the upstream compressor station and the minimum suction pressure of the downstream compressor station on the effective MOP and MINOP curves.

In Figure 23, $A^{/}$ - $B^{/}$ is the Level 1 effective MOP and $C^{/}$ - $D^{/}$ is the Level 1 effective MINOP. If the compressor station at the upstream end of this pipe has a maximum discharge pressure higher than $A^{/}$, then there is no effect on the effective MOP curve. However, if the station maximum discharge pressure is lower than $A^{/}$, then this would lower the effective MOP curve as shown in Figure 23 leading to $A^{//}$ - $B^{//}$, which would be the Level 2 effective MOP curve.

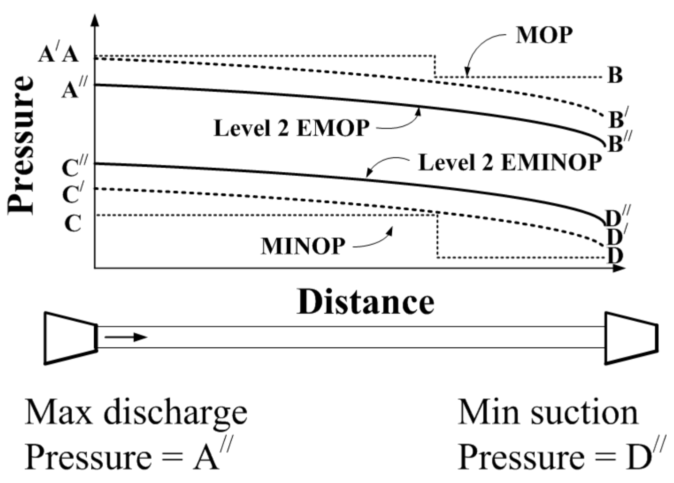

Figure 4 - Level 2 pressure envelopes

If the minimum suction pressure at the downstream end of the pipe is lower than D/, then there is no impact on the effective MINOP curve. However, if the minimum suction pressure is above D/, then the minimum pressure envelope has to be raised leading to line C// - D//, which would be the Level 2 effective MINOP curve.

Maximum limit constraints for compressor stations

Various compressor station maximum limit constraints (maximum power, maximum speed, maximum flow, maximum ratio, etc.), limit the maximum amount of pressure rise a station can deliver at a given flow rate. The maximum allowable pressure rise can impact the maximum and minimum pressure envelopes. Until now we have considered pipe segments between two consecutive compressor stations. Consideration of the maximum amount of pressure rise a station can deliver causes interaction among the pressure envelopes across stations.

Consider the scenario shown in Figure 24 where there are three compressor stations and two pipeline segments for which we have calculated Level 2 EMOP (A// - B//) and Level 2 EMINOP (C// - D//) curves. Let the maximum pressure rise that can be achieved by the second compressor station be ΔPmax. We will consider how ΔPmax impacts the Level 2 EMOP and EMINOP curves in the two pipeline segments. For a given suction pressure Ps, discharge pressure Pd and pressure rise ΔP across the station, we can say the following.

Pd = Ps + ΔP

The above equation dictates that we can maximize Pd by maximizing Ps and ΔP (since both are positive). Thus the maximum pressure on the suction side of the second compressor station along with the maximum pressure rise dictates a maximum discharge pressure. If this maximum discharge pressure is greater than A// for the second pipeline segment, then its EMOP is not impacted. However, if the maximum discharge pressure is lower than A// for the second pipeline segment, then its EMOP will need to be shifted down resulting in Level 3 EMOP (A/// - B///) for the second pipeline segment.

The relationship between station suction and discharge pressures and station pressure rise can be rewritten as follows.

Ps = Pd - ΔP

This equation dictates that we can minimize Ps by minimizing Pd and maximizing ΔP. Thus the minimum pressure on the discharge side of the second compressor station along with the maximum pressure rise for the second station dictates a minimum suction pressure for this station. If this minimum suction pressure is less than D// for the first pipeline segment, then its EMINOP is not impacted. However, if the minimum suction pressure is greater than D//, then its EMINOP will need to be shifted upward resulting in Level 3 EMINOP (C/// - D///) for the first pipeline segment.

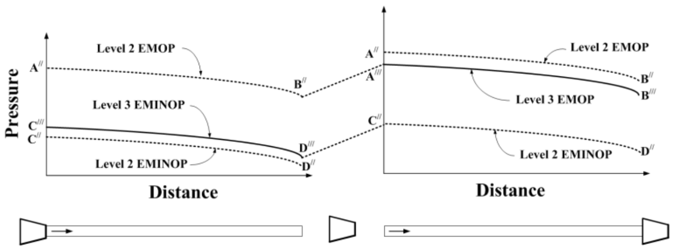

Figure 24 - Level 3 pressure envelopes

If compressors contain only maximum limit constraints, the level 3 maximum and minimum envelopes themselves are feasible ways to operate the pipeline. Other feasible pipeline operations can be readily found within the region between the level 3 maximum and minimum envelopes using a method similar to the way the maximum envelope is developed. Desired set pressures (any value between the maximum and minimum envelope pressures at each station) can be selected for each compressor station's operation. These set pressures can then be trimmed by applying the compressor constraints as in the maximum envelope calculations.

However, if compressors contain minimum constraints, the envelopes do not necessarily represent feasible operating scenarios. The following sections describe the effect of minimum constraints on steady state analyses, steady state optimization, and transient look forward simulations.

EFFECTS OF MINIMUM CONSTRAINTS ON SIMULATION AND OPTIMIZATION

The above discussion on pipeline envelopes proposes a method for determining the maximum range of the potentially feasible region. Any selection of set pressures outside of the region bounded by the maximum and minimum pressure envelopes is infeasible. However, the converse is not necessarily true. A selection of set pressures within the bounds of the maximum and minimum envelopes may or may not be feasible. To gain an understanding of the feasible region between the maximum and minimum envelopes for a given pipeline flow, one can discretize the pressures between the maximum and minimum envelopes and solve for compressor feasibility at each pressure combination. A feasible path through the pipeline can be selected by analyzing the resulting data. A method for visualizing feasible pipeline operation through the entire pipeline for a given flow is described below. This visualization technique is used to point out the effects minimum compressor limits have on steady state analysis and optimization.

To illustrate this process consider the fictitious pipeline illustrated by Figure 25 and Figure 26. The pipeline transports gas from a single source in the far western side of the pipeline to a single delivery in the east.

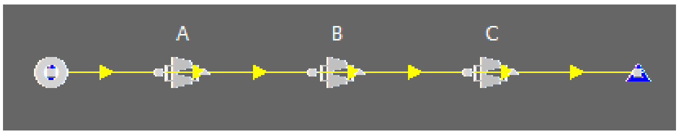

Figure 25 - Fictitious pipeline containing 3 compressor stations each separated by 75 miles (121 km) of 28 inch (711 mm) pipeline; all pipes are 75 miles (121 km) long

Each compressor station contains a single reciprocating compressor "RC_fixed_s_var_cl_min_power" described in the section of this paper entitled "Reciprocating compressor limits". Recall that this compressor has a minimum power equal to 80% of the maximum power which gives it a rather narrow feasible operating region.

For a given flow of 275 mmcfd (7.79 Mm^3/d) through the pipeline, the minimum and maximum pressure envelopes for this pipeline are shown in Figure 27.

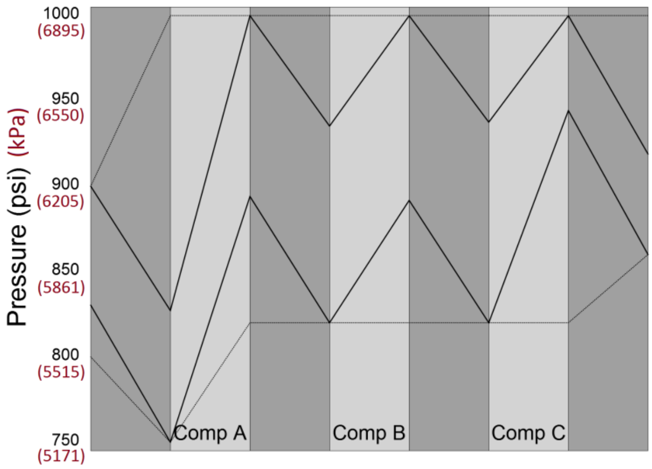

Figure 5 - Maximum and minimum pressure envelopes for a flow of 275 mmcfd (7.79 Mm^3/d)

The source node pressure and the discharge pressures from all three compressors are each discretized into 10 pressures, with the maximum and minimum discretized pressures corresponding to the value of the maximum and minimum pressure envelopes at that point in the model. The other 8 pressures are spaced evenly between the maximum and minimum pressures. Given these discretized pressures and a flow of 275 mmcfd (7.79 Mm^3/d), the steady state pipe equations can be solved for the downstream pressures yielding the discretized suction pressures for all of the compressors. Figure 28 shows the discretized suction pressures for compressor B in the first column and the discretized discharge pressures for that compressor in the first row. The internal cells represent the results of compressor calculations at the given suction and discharge pressures. The rows represent discretized suction pressures (units of psig) while the columns represent discretized discharge pressures (units of psig). The red portion of the table correspond to compressor calculations below minimum power, while the upper left hand green corner of the table represents bypassed scenarios and the lower right hand green corner represents feasible compression regions with compressor fuel in units of mmcfd reported in the cells.

0.00	891.75	902.56	913.40	924.28	935.15	951.33	959.45	967.57	983.77	999.98
935.29	0.00	0.00	0.00	0.00	0.00	-2.00	-2.00	-2.00	-2.00	-2.00
921.00	0.00	0.00	0.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
906.68	0.00	0.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
892.31	0.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
877.94	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
863.48	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
856.22	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
848.97	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00
834.49	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	0.48
820.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	-2.00	0.48	0.52

Figure 6 - Fuel and feasibility table for compressor B at a flow of 275 mmcfd (7.79 Mm^3/d)

This table is represented pictorially in Figure 29.

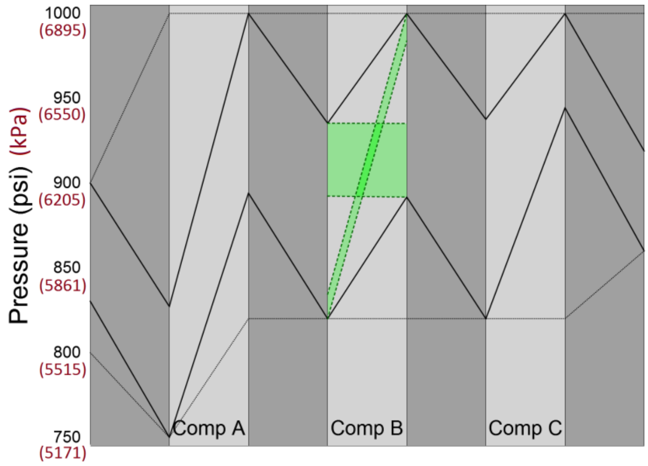

Figure 7 - Pictorial representation of the feasibility table for compressor B at a flow of 275 mmcfd (7.79 Mm^3/d); the horizontal green box represents feasible compressor bypassed states, while the sloped green section represents feasible compression states

The boundary of the dark gray and light gray areas at the left side of compressor B's region represents the suction pressures of compressor B. Similarly, the boundary of the light and dark gray area on the right side of Compressor B's section represents the discharge pressures of compressor B. The green lines across the light gray area connect suction and discharge pressures corresponding to feasible compressor operation. The horizontal green lines correspond to feasible bypass operation while the sloped green lines span suction and discharge pressures corresponding to feasible compression. Note that compression is only possible at high pressure ratios due to minimum power requirements.
Also note that bypass operation must correspond to a pressure drop across the compressor. All suction pressures lower than the minimum pressure envelope at the discharge side of the compressor result in infeasible bypass states.

Figure 30 extends the green section through the pipeline downstream of compressor B.

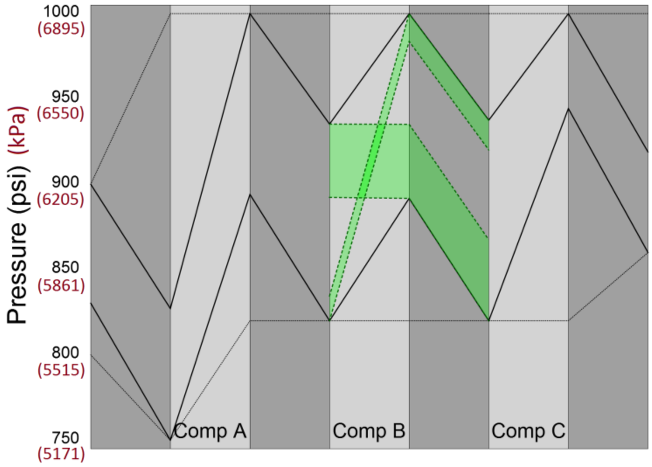

Figure 30 - Pictorial representation of the feasibility table for compressor B at a flow of 275 mmcfd (7.79 Mm3/d); the feasible region corresponding to the discharge of compressor B is extended through the downstream pipes to the suction of compressor C

The green lines between compressor B and compressor C through the dark gray region indicate possible choices for the suction pressure of compressor C corresponding to the feasible discharge pressures from compressor B.

Figure 31 shows the feasible regions for all of the compressors.

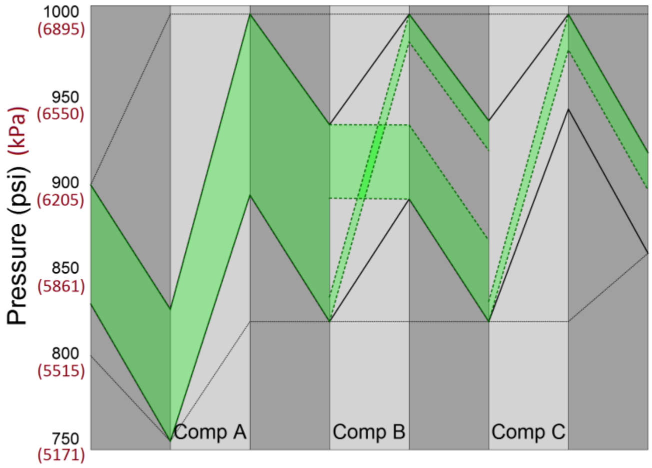

Figure 31 - Pictorial representation of the feasibility table for all compressors at a flow of 275 mmcfd (7.79 Mm3/d)

Compressor C has a narrow feasible region. Only the largest compression ratio is feasible due to the minimum power limit on compressor C. Compressor C has no feasible bypass states because the minimum pressure envelope at the compressor's discharge node is greater than the maximum pressure envelope at the compressor's suction node. A feasible path through compressors B and C can only occur where the green areas to the left of compressor C's suction region intersect with green areas to the right of compressor C's suction border. Looking at the suction side of compressor C, this corresponds to bypassing compressor B and compressing gas through C. To find feasible operating pressures through multiple compressors one must find a path connecting feasible regions through each compressor station. The area between the blue lines in Figure 32 depicts the possible feasible paths through the pipeline.

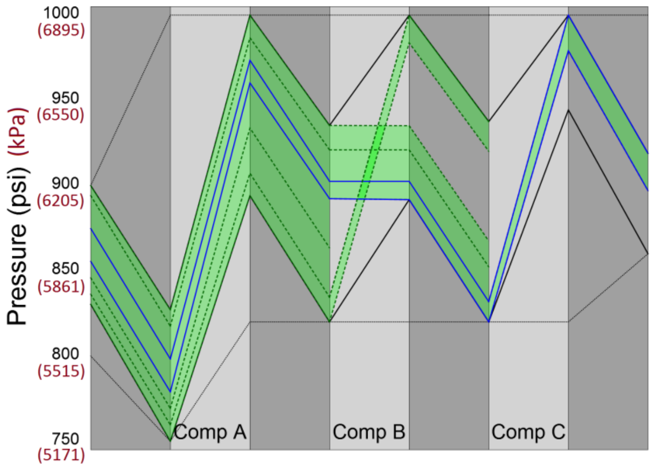

Figure 32 - Pictorial representation of the feasibility table for all compressors at a flow of 275 mmcfd (7.79 Mm3/d); the feasible pipeline operating region is indicated by the area between the blue lines

When dynamic programming is used to find the optimum control pressures through the pipeline it selects a path through the green section between the blue lines that minimizes the objective function (fuel) for this flow. For more information on dynamic programming please see the following references describing the application of dynamic programming to optimize steady state pipeline operation. [2, 3].

To illustrate pipeline feasibility as a function of flow, the pipeline flow is varied from 5 mmcfd (0.14 Mm^3/d) to 400 mmcfd (11.3 Mm^3/d) in increments of 5 mmcfd (0.14 Mm^3/d). Dynamic programming fuel minimizations were performed at each flow value. Figure 33 plots the required fuel in black with infeasible regions in red.

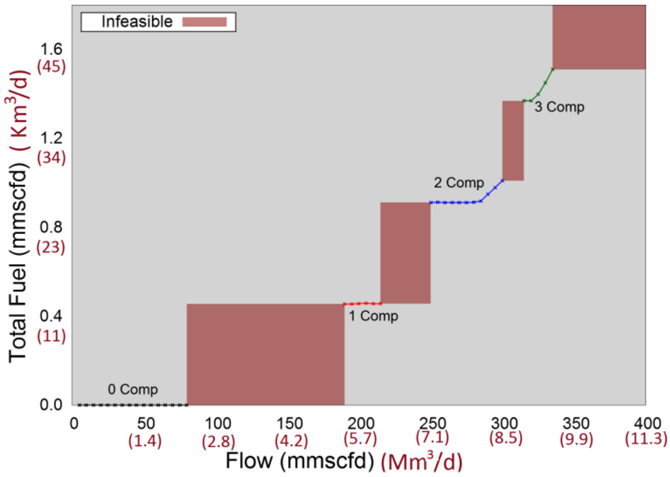

Figure 33 - Pipeline fuel usage as flows are varied from 0 to 400. The red regions represent flows with no possible feasible pipeline operation

The maximum and minimum pipeline pressure envelopes cross at a flow of 338 mmcfd (9.57 Mm3/d). This corresponds to the maximum capacity of the pipeline. The gaps in the feasible solutions are a result of minimum constraint violations and can be explained by examining the feasibility plots generated. Figures 34 through 40 show feasibility plots for a sample flow from each of the feasible and infeasible regions.

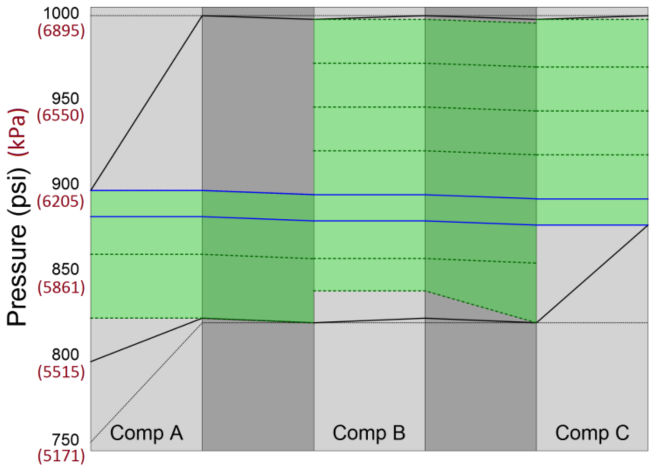

Figure 34 - Feasibility chart for a pipeline flow of 50 mmcfd (1.42 Mm3/d). The only feasible operating mode is to fully bypass all compressors.

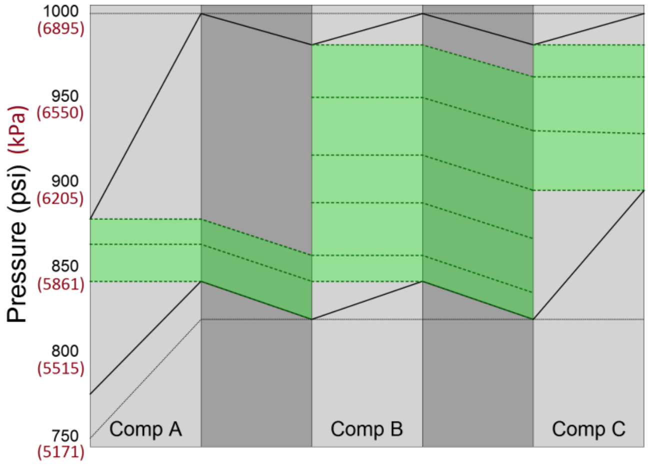

Figure 8 - Feasibility chart for a pipeline flow of 150 mmcfd (4.25 Mm^3/d). The pipeline cannot be operated since there is no connection between feasible regions through the entire pipeline. The frictional pressure loss through the pipes is too large to meet the minimum delivery pressure and compression is not possible due to minimum power requirements.

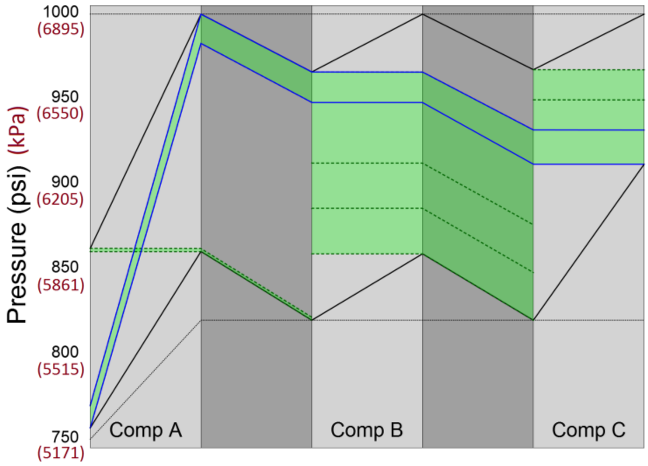

Figure 9 - Feasibility chart for a pipeline flow of 200 mmcfd (5.66 Mm^3/d). Compressor A is compressing while compressors B and C are bypassed

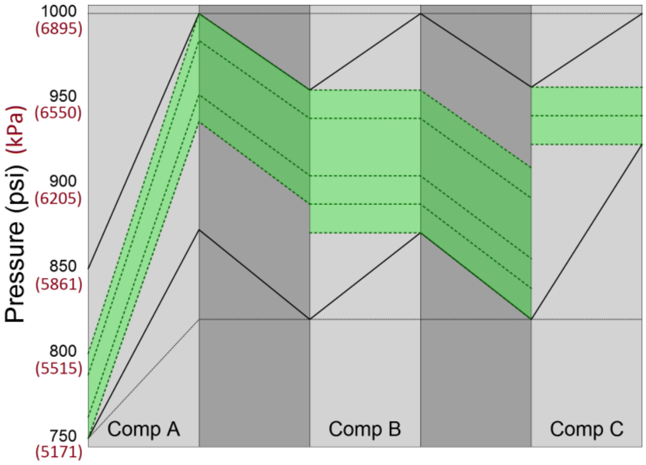

Figure 10 - Feasibility chart for a pipeline flow of 230 mmcfd (6.51 Mm^3/d). Feasible pipeline operation is not possible since the bypass states from compressor B do not intersect with bypass states on compressor C

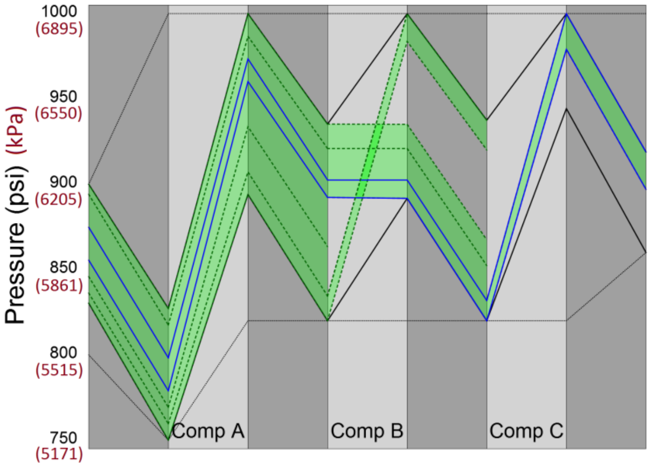

Figure 11 - Feasibility chart for a pipeline flow of 275 mmcfd (7.79 Mm^3/d). Feasible operation consists of bypassing compressor B and compressing through compressors A and C

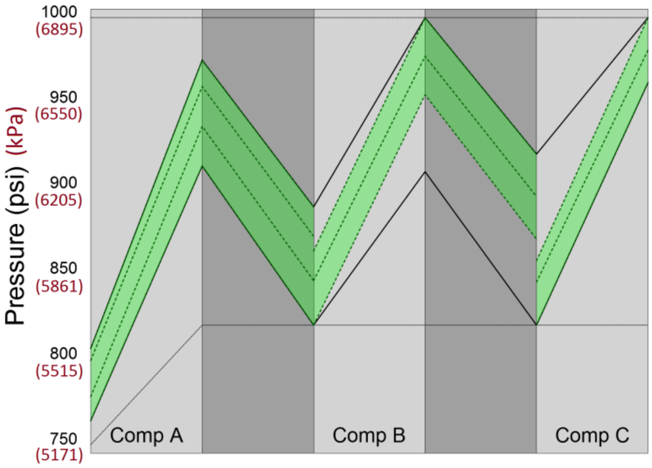

Figure 12 - Feasibility chart for a pipeline flow of 310 mmcfd (8.79 Mm^3/d). Feasible pipeline operation is not possible since gas enters compressor C at too high a pressure

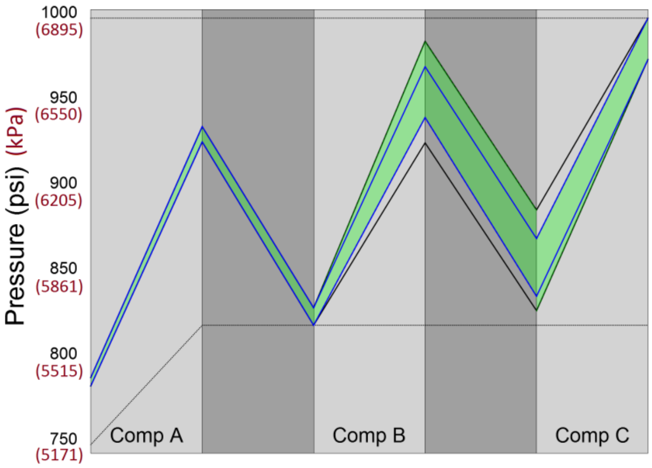

Figure 13 - Feasibility chart for a pipeline flow of 335 mmcfd (9.49 Mm^3/d). All three compressors are compressing

These gaps in the feasible region are direct results of the minimum constraints on compressors causing infeasible solutions. The first feasible region consists of free-flowing the entire pipeline through three bypassed stations. As the flow increases, the frictional pressure losses increase to the point where the minimum delivery pressure of 860 psi (5930 kPa) is violated. Compression is needed to boost the pressure of the gas. However, the power required for one compressor to discharge at the MAOP of the pipeline is less than the minimum power of the compressor, making operation infeasible.

At a flow of 185 mmcfd (5.24 Mm3/d), the flow and the pressure drop through the pipeline become sufficient to operate compressor “A” above its minimum power. However, when the flow exceeds 215 mmcfd (6.09 Mm3/d), it becomes necessary to start another compressor to avoid minimum delivery pressure violations. Two compressors cannot be run at this flow without violating minimum power or maximum pressure requirements. When the flow is increased to 245 mmcfd (6.94 Mm3/d), a second compressor can be started without violating minimum power limits. Similar reasoning can be used to explain the feasibility gap between 300 (8.50) and 320 mmcfd (9.06 Mm3/d), where three compressors must be run to meet the minimum pressure requirements.

Steady state optimization

The above discussion illustrates how minimum limits in compressor stations may cause gaps in feasible pipeline operation as a function of pipeline flow. These gaps in feasibility can cause steady state optimizations to give sub-optimal answers. Steady state optimization typically uses a two-step optimization process. Dynamic programming is used to solve for the set of compressor operating pressures that minimize the objective given a set of loop flows. An outer flow optimization algorithm is used to vary the model's loop flows in such a way as to obtain the optimal network flow distribution. The optimization proceeds by alternating between flow-update steps and dynamic programming steps.

The flow-update step typically uses a direct search method or a descent method to vary the flows in a search for the optimum objective. These methods start with an initial guess and take steps toward the optimum. They can get stuck in local minima if the function is not continuous. In the above example, if the objective is to find the maximum flow in the system, the flow optimization could potentially fail to "step over" the infeasible gaps at flows of 85 (2.4), 220 (6.2), or 305 mmcfd (8.6 Mm3/d). Since the sample problem introduced in this paper is one dimensional, more robust techniques can be used to find the optimum by a more complete search of the solution space. However, as the number of optimizable flows in the system increases, it is more computationally expensive to sample points in the infeasible space, so the solution is more likely to end up in local minima.

This illustration focused on feasibility gaps generated by minimum power constraints in reciprocating compressors. Other minimum constraints can cause feasibility gaps as well: as outlined earlier in the paper, centrifugal compressors may have minimum speed and minimum power constraints. These constraints also produce a reduced feasible region, which can affect pipeline feasibility. A more extreme example is the fixed speed, fixed reciprocating compressor. Fixed speed, fixed clearance compressors will almost always cause dynamic programming to fail to find a feasible operation condition (other than the trivial bypassed state). As illustrated above, dynamic programming requires that for a given flow, compressor operation be calculated over a discretized series of suction and discharge pressures. If the discretized flow, suction and discharge pressures do not exactly line up with a clearance pocket combination as illustrated in Figure 2, the point will be infeasible.

We recommend that minimum compressor constraints should be ignored during steady state optimization when it results in infeasible points. This will allow all pipeline flows to be feasible as long as the maximum and minimum pressure envelopes do not cross. However, the dynamic programming step should be configured in such a way as to preferentially select operating conditions with no minimum limit violations.

Steady state analysis

The next question is how do minimum compressor constraints affect steady state analyses? Compressors in steady state analyses are normally set up to hold either the suction or discharge pressure as opposed to being set up to operate at their capacity. If the compressors do not have sufficient capacity to deliver the required set pressures, the compressors switch to capacity control and back down from the set pressure.

Let's consider compressor "RC_var_s_var_cl" in the section entitled "Reciprocating compressor limits". Since there is no minimum power limit, the capacity increases with increasing suction pressure and decreasing discharge pressure. Let's assume that the discharge pressure is set to 900 psig (6210 kPa) and the pipeline hydraulics dictate that the flow is 200 mmcfd (5.66 Mm3/d) and the suction pressure is 575 psig (3960 kPa). Compressor "RC_var_s_var_cl" has a capacity of 129 mmcfd (3.65 Mm3/d) under those conditions, so the steady state would switch the compressor to capacity control, and the compression ratio would decrease until the capacity of the compressor is sufficient to meet the flow.

In this case, if the suction pressure is more or less constant the discharge pressure would decrease to below 775 psig (21.9 kPa).

However, if the minimum power limit is enforced, there is only a very narrow band of feasible operation. For the above example, at a flow of 200 mmcfd (5.66 Mm3/d) and a suction pressure of 575 psig (3960 kPa), the feasible region is between discharge pressures of 725 psig (5000 kPa) and 750 psig (5170 kPa). If the set pressure is selected above this narrow feasible region, capacity control can allow the compressor to find a feasible solution for the pipeline. Conversely, if a discharge set pressure is selected below this feasible region, the only options to obtain feasibility are to increase the set point or to shut down the station.

A similar problem arises when a compressor's capacity increases with increasing compression ratio. This can occur in centrifugal compressors when their capacity is defined by operation along the sonic line (see Figure 14). Again, if the compressor is operating under hydraulic conditions such that the flow reaches the sonic line, switching the compressor to capacity control will result in increasing the compression ratio instead of decreasing the compression ratio. Thus, the only ways to find a feasible operation is to either shut down the compressor or increase the compression ratio so that the set pressure is violated. As previously stated, this problem will vanish if compressor operation is allowed within the sonic region (see Figure 20).

Because of the potentially narrow feasible region due to minimum compressor limits, the authors recommend that for steady state analyses, operation below minimum values should be treated via a warning as opposed to a constraint. Another possibility is to allow the algorithm to increase compressor set pressures if compressor operation falls below minimum limits.

Transient analysis

One main way that minimum constraints affect transient analyses is during the compressor station startup process. If a compressor is limited by minimum power or minimum speed, during the startup process the compressor will need to violate these minimum limits to proceed from a compression ratio of 1.0 to the desired operating ratio.

Compressor start up requires taking compressors from rest to a feasible operating point. This is an inherently dynamic process that proceeds in a controlled manner which may involve the compressor recycling gas until the compressor is brought up to speed. Typically, compressors are not placed under load until they reach operating conditions.

There are multiple ways to work around this solution. The compressor start-up process could be modeled in detail. Another option is to treat minimum limits as warnings instead of constraints, similarly to our steady-state recommendations. These warnings can be ignored during station startup sequences. A third option would be to simply ignore infeasible compressor operation until the feasible region is found.

TAKEAWAY THOUGHTS

We recommend visualizing the feasible region depicted by compressor models before using them within pipeline simulation or optimization software. If there are any holes within the operating region or if there are non-convex or non-continuous regions, one should be aware of the effects of these during pipeline modeling and optimization. When using pipeline modeling in optimization software, it is always a good idea to fill in the "holes" relating to minimum compressor limits.

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ACKNOWLEDGEMENTS

The authors would like to thank Nicholas Russ for his valuable support in supplying figures and input for this paper. The authors would like to thank the management of DNV GL for supporting this work.

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APPENDIX 1

Attribute	RC_fixed_s_fixed_cl
Maximum Power	3000 hp (2237 kW)
Minimum Power	0 hp (0 kW)
Number of swept volumes	5
Number of clearances / swept volume	8
Swept volumes	15.5, 13.5, 11.5, 10, 7.5 ft^3 0.439, 0.382, 0.326, 0.283, 0.212 m^3
Clearances (%)	90,100,120,140,160,180,200,220
Maximum speed	300 rpm
Minimum speed	300 rpm

Table 2 – Reciprocating compressor data

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