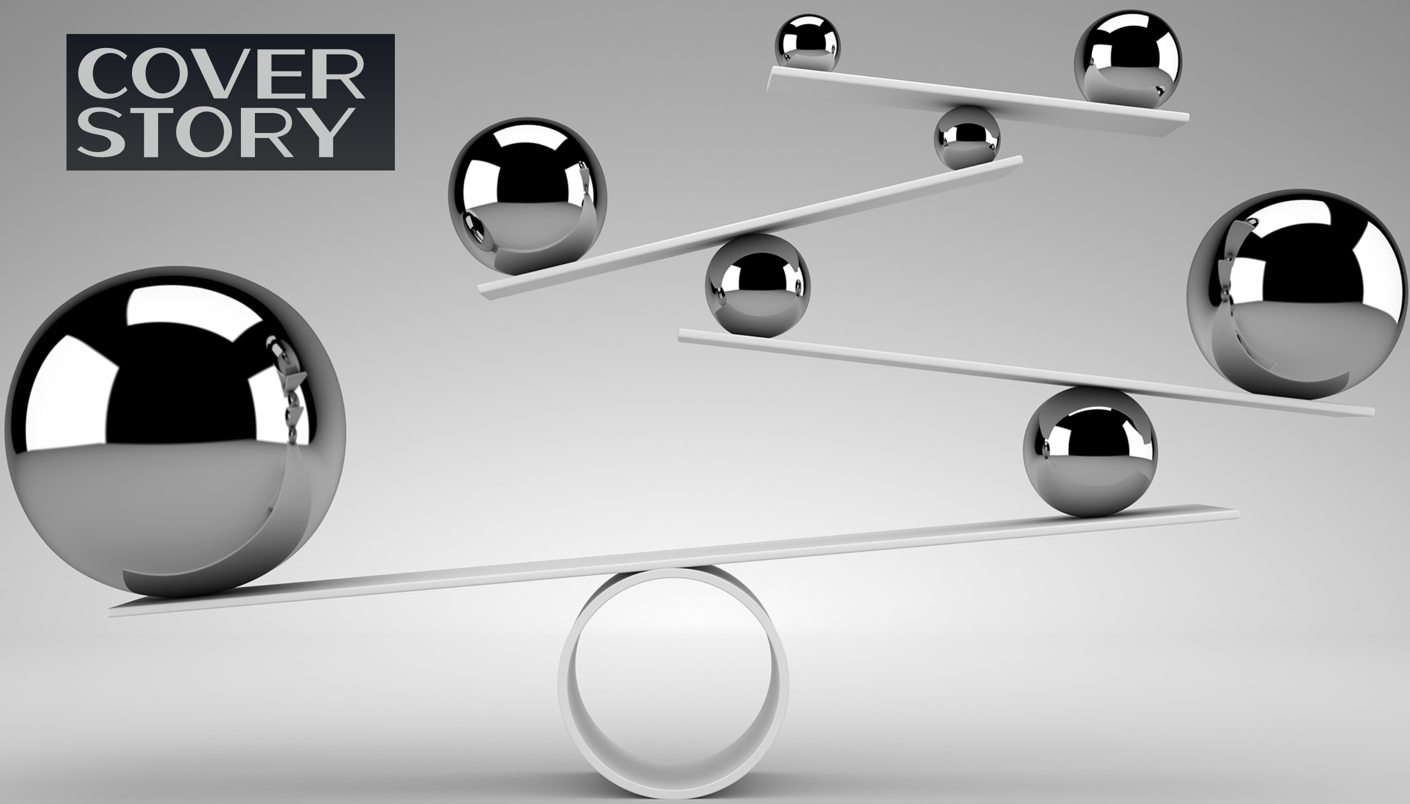
COVER STORY



BALANCING THE ECONOMIES OF SCALE

Shane Harvey, Elliott Group, USA, discusses how ethylene producers can meet future plant capacity needs and maintain economies of scale without increasing cracked gas compressor size.

Profitable ethylene production involves high-volume, low-cost operation with uninterrupted production runs. Cracked gas compressor (CGC) performance is critical to achieving these goals. As global demand for plastic products increases, ethylene plants continue to grow in size as economies of scale are realised at larger capacities (Figure 1). A 1.5 million tpy ethylene plant is common now, and 2 million tpy equipment is already operating out in the field. Customer requests for 2.5 million tpy and even 3 million tpy equipment are expected.

Beyond plant capacity, there has been a general trend towards decreasing suction pressure at the CGC inlet. Suction pressure improves selectivity, so in some cases compressor volume flow is outpacing plant capacity increases. As volume plays a direct role in compressor frame size, continuing to scale the same compressors ever larger to match capacity means compressor frames will outpace plant capacity.

The scaling problem

The problem is if the industry continues to simply increase plant capacities and CGC inlet volume flows, expecting to increase compressor size to match, economies of scale will disappear. Manufacturers cannot meet ever-increasing plant capacities and volume flows by simply continuing to build larger and larger machines (Figure 2). As compressors get larger, not only do they get more expensive and challenging

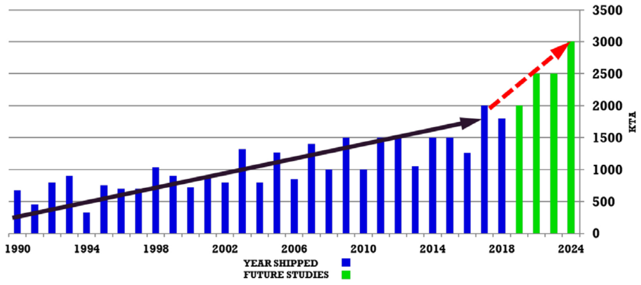


Figure 1. Growth in plant capacity.

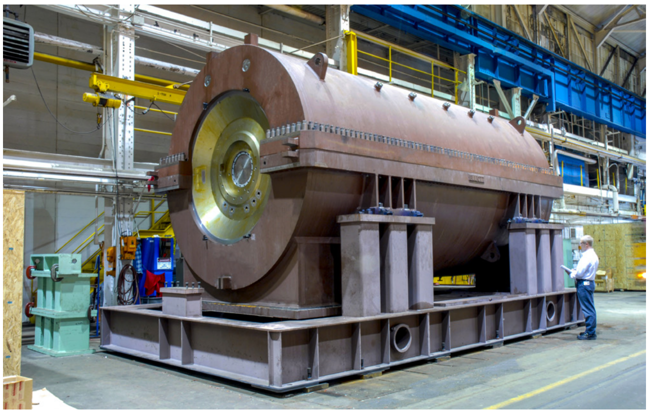


Figure 2. An Elliott 110M compressor during manufacturing.

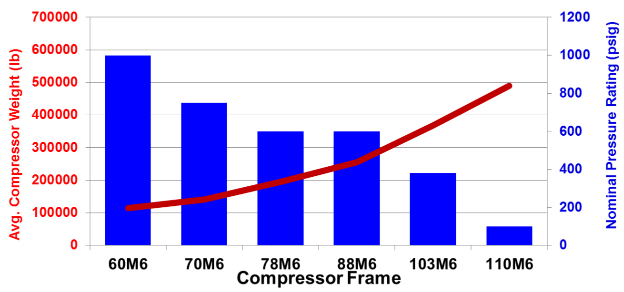


Figure 3. Growth in compressor frame size.

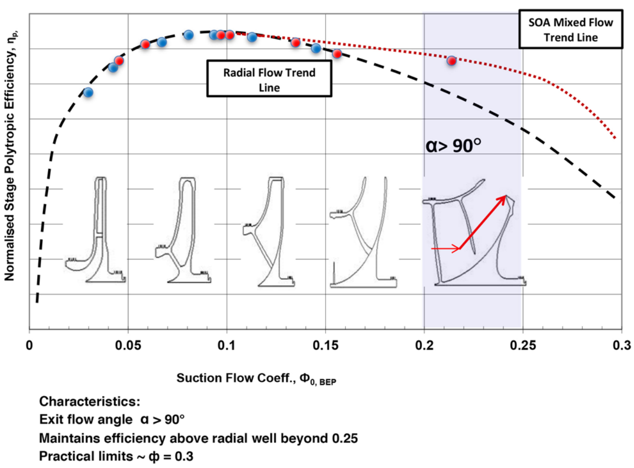


Figure 4. Radial to mixed-flow transition.

to build and maintain, but so do their large steam turbine drivers, as power must be increased with plant capacity.

Challenges of larger frames

Elliott's 88M frame olefins compressor represents the divide between a 'large' and an 'extremely large' compressor. Above the 88M frame, each increase in frame size corresponds to a 35% increase in unit weight (Figure 3). The cost per ft^3/min. of flow no longer improves at these large frame sizes. Diaphragms are larger, and rotors are heavier and longer, increasing installation and maintenance costs. The physical size of machinery cannot be infinitely scaled. Along with meeting process demands, plant management should consider the costs of safely and reliably installing and maintaining these larger machines for the decades to come, including the costs associated with increased maintenance times, larger crane capacity requirements, additional logistical concerns during a turnaround, and reduced availability of repair facilities capable of handling large components.

Three-part solution

Through multiple research and development projects and product reviews, Elliott developed a three-part solution for increasing compressor flow capacity without increasing the frame size. Adopting higher flow coefficient stages is the first part. The second part is increasing the size of the compressor frame nozzles to accommodate higher flows on smaller frames, while reducing the bearing spans. Third is conducting a complete review of the stage line-up using modern computational fluid dynamics (CFD) tools to reduce the existing stage spacing.

Compressor flow coefficient

Flow coefficient is a dimensionless characteristic used in compressor design that represents the flow capacity of an impeller.

There is a constant interplay between impeller diameter and compressor speed. Traditionally, as flow increases, the diameter increases, the speed decreases, and the flow coefficient remains within the same range. Some use Equation 1 to define flow, however, Equation 2 is an alternative used by some industry suppliers, so care must be taken when comparing data among vendors.

$$\Phi = \frac{700Q}{Nd^3} \tag{1}$$

Φ = Flow coefficient
Q = Flow [ft^3/min.]
N = RPM [1/min.]
d = Impeller dia. [ft]

$$\Phi = \frac{Q}{DU} \tag{2}$$

Φ = Flow coefficient
Q = Inlet flow [m^3/s]
D = Impeller dia. [m]
U = Tip speed [m/s]

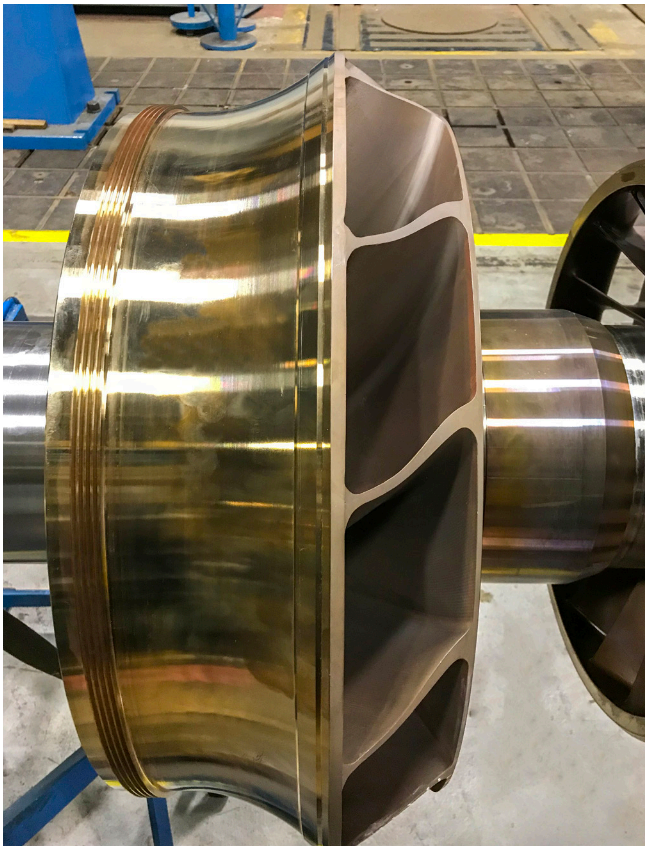


Figure 5. Second-generation mixed-flow impeller installed on shaft.

In the radial stages traditionally associated with centrifugal compressors, flow enters the impeller parallel to the shaft and exits the impeller perpendicular to the shaft. Gas accelerates radially, and the increased velocity is recovered as pressure in the diffuser section of the diaphragm. Often, compressor design incorporates first-stage flow coefficients of 0.12 to 0.15 to keep overall string efficiency high.

Well-designed mixed-flow stages, which represent the transition from radial to axial compression, offer a 30% increase in flow capacity within the same frame size at similar string efficiency. Figure 4 illustrates the trend of efficiency for radial staging as the flow coefficient increases, and the ability of mixed flow to maintain efficiency levels. Efficiency levels are maintained above traditional radial stages, well beyond 0.25.

Using modern CFD tools, Elliott designed, built, and validated through full-scale, single-stage testing, a new series of state-of-the-art, second-generation mixed-flow stages (Figure 5). These new, high-flow stages offer excellent flow path sizing and aerodynamic throat control using splitters. They are available in multiple frame sizes for olefins and refinery applications.

Increased nozzle sizes

In parallel to the impeller development project, additional nozzles were added to those already available for existing frame sizes. This led to both an expansion of frame size capability and reduced bearing spans for the existing nozzles as shown in Table 1.

Table 1. Increased nozzle sizes for existing frames

Frame	Previous maximum inlet	Revised maximum inlet size	Previous maximum discharge	Revised maximum discharge size
	in.	in.	in.	in.
56	36	42	24	30
60	42	48	36	36
70	48	60	30	36
78	54	66	36	42
88	66	78	42	48
103	72	90	54	60
110	84	102	54	66

Table 2. Comparison of existing plant with high-flow coefficient plant

Casing		First stage flow co Φ	Bearing span (m)	Rotor weight (kg)	Unit weight (kg)
LP	88MD3-3	0.146	5.4	12 000	188 000
	78MD3-3	0.214	4.8	9000	145 000
	Delta	46.50%	-12%	-25%	-43 000
					-23%
MP and HP	88M4I	0.0915 and 0.0431	4.4	11 500	168 000
	78M4I	0.1351 and 0.0593	4.2	8600	125 000
	Delta	48% and 35%	-5%	-25%	-43 000
					-26%

Review of existing stages

In addition to enabling larger flows with larger compressor nozzles and higher flow coefficients, Elliott also reviewed existing, well-proven staging and flow path elements. By applying a new generation of state-of-the-art CFD tools and design methodologies, the spacing of numerous stages was reduced while maintaining the high performance level industry demands.

Case study

A case study was carried out which compared an existing 1.5 million tpy CGC plant using typical 3D radial staging against a selection using the high-flow coefficient strategies discussed earlier. Table 2 illustrates the benefits.

The reference plant consists of a two-body CGC string driven by an Elliott steam turbine. The low-pressure (LP) section is a double-flow 88MD3-3 with three impellers on each side. The medium- and high-pressure (MP and HP) sections are handled by an 88M4I, with two impellers per section. The compressor rated point is approximately 58 MW at 3600 RPM.

The hypothetical plant with a high-flow coefficient selection has a double-flow 78MD3-3 compressor in section 1, and sections 2 and 3 are in a 78M41 casing. The compressor rated point is 58 MW at 4300 RPM. High-flow coefficient staging is applied in the double-flow machine only.

As is typical for large volume compressors, nozzle size plays a major role in rotor length. As shown in Table 2, application of the external volute discharge on the double-flow compressor reduces the bearing span. Reducing the bearing span shortens the string and significantly reduces casing weight. While this case study only looks at the impact on the compressors, the shorter and faster string would be driven by a shorter, more efficient turbine as well. While the inlet stages may be less efficient, overall the entire string does not lose efficiency. In plants with a fourth and even a fifth section of CGC compression in a third body, overall string power can be improved. The power improvement comes from the third body being much better speed matched to the now faster LP/MP casings. Increased speed also tends to increase performance from the steam turbine drivers; as such, steam rates may improve beyond just the compressor power improvement, which may allow for more flexibility on steam balance or opportunities for overall energy savings.

Elliott is already building compressors using high-flow coefficient staging, and this staging is not limited to 2 million tpy plus plants. These designs are being applied to plants of all sizes to reduce plant cost and equipment weight and footprint.

Conclusion

Equipment manufacturers need to work together with stakeholders (owners, operators, process licensors, and engineering, procurement, and construction contractors) to meet ethylene plant challenges. All stakeholders need to apply sound engineering to the design of their critical rotating equipment. Far too often, outdated rules of thumb and engineering guidelines get in the way of doing real engineering to meet ethylene producers' ever-increasing expectations of improvement in performance, capacity, cost, and safety. To meet customer demands for higher flow compressors for large-scale ethylene plants, a number of impellers and design improvements have been developed that significantly increase the flow capabilities of the existing compressor product range.

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