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Revision Memo

- 04/04
- Page 4 Increased sensitivity of axial versus centrifugal compressor to liquid carryover and solids.

Page 8-9 Modified axial versus centrifugal comparison using lessons learned from the GTL project.

Page 21 Corrected Figures 4&4B for units and to more accurately reflect operating conditions of axial compressors.

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SCOPE

This section presents comparative information on the types of compressors commercially available and provides the bases for selection of compressor type. Additional comparative data relating to serviceability and sparing can be found in [Design Practice XI Section C](#). Details on construction styles within the basic type categories and on specific mechanical design features are presented in the subsections on specific compressor types.⁽¹⁾

REFERENCES

DESIGN PRACTICES

Other subsections of [Section XI](#).

OTHER REFERENCES

Cost Estimating Manual

BACKGROUND

Compressor type selection requires comparison of the compression service requirements with the capabilities and limitations of the available machine types to determine the optimum economic choice. Type selection is sometimes so obvious it does not require methodical consideration. However, where significant competition exists between rival types, a comparative study and selection should be made early in the process design phase, to allow basic design of the utility systems and other installation facilities, and to allow valid project cost estimates.

BASIC DESIGN AND SELECTION CONSIDERATIONS

COMPRESSOR TYPE CLASSIFICATION

All compressors fall into one of three categories, either dynamic, positive displacement or ejector. See [Figure 1](#). The dynamic category can be split into three groups, i.e., centrifugal compressors, axial flow compressors and fans. Positive displacement compressors exist in two basic groups which are characterized by the motion of the machines; either reciprocating or rotary. The reciprocating compressor group can be further broken down into three further sub groups covering piston, plunger (single acting high pressure), and diaphragm compressors. Rotary compressors include sliding vane, liquid ring, lobe and screw machines.

The term blower is often used to refer to services that have high volumetric flow rates with small pressure increases and is applied regardless of compressor type. The use of this term is more commonly applied in air services such as supply of cat plant regeneration air, blast furnace air and sewage aeration.

FACTORS AFFECTING SELECTION OF COMPRESSOR TYPE

Local Experience - The experience that a particular refinery or geographical area has acquired with a particular compressor type may influence type selection, particularly in cases where one or more compressor types could reasonably be applied. Favorable or unfavorable prior experience with a particular type may be related to any of the following:

1. The appropriate compressor type selection for the previous application.
2. The particular model previously applied.
3. The proximity of manufacturers service facilities and personnel.
4. The size and skill resources of the plant maintenance staff.
5. The availability of proper maintenance tools and facilities.



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Commercial Availability of Compressor Models - A fairly fundamental step in selecting the compressor type for a given service is to ensure that a suitable compressor is commercially available. [Figures 2 through 12](#) are application range charts for each of the major compressor types, using actual inlet volume flow rate and discharge pressure level as the parameters.

There are two ranges defined for each compressor type. They are the feasible range, where commercial models are offered by at least one major manufacturer and the economic range, where models are offered by at least two manufacturers. For the application which lies within the feasible range, but outside the economic range, operating experience is often very limited, supply sources are limited, and some degree of special (rather than standardized) design is required by most manufacturers.

- (1) **Note:** API compressor data sheets specify pressure in psi, bar, and kPa. In this subsection, bar is the metric unit used for pressure.

For services where the volume flow rate and discharge pressure fall within the economic range of only one compressor type, the type selection process is straightforward. For services where the volume flow rate and discharge pressure fall in the economic range of more than one compressor type, a compressor type selection study must be made.

See [Appendix A, Comparative Advantages of Various Compressor Types](#).

Power Level - In addition to establishing the feasibility of applying a particular compressor type to a particular combination of inlet volume flow and discharge pressure, the power transmission capability of the compressor type must also be evaluated. [Table 1](#) provides a summary of typical and maximum possible power ratings for each compressor type.

Serviceability of Compressor Types - For run length requirement between maintenance opportunities and reliability potential. See [Design Practice XI Section C, Compression Machinery Serviceability and Sparing](#).

Service Conditions Affecting Compressor Type Selection - The following service conditions have the strongest influence on selection of the optimum compressor type, and also impact construction style and design features:

- **Head requirement** - Positive displacement compressors tend to be more economical than dynamic types for high head requirements. See [Table 2](#).
- **Pressure ratio** - Multistage reciprocating compressors tend to be most economical for high pressure ratios. See [Table 2](#).
- **Head Capacity Performance Characteristics** - A general comparison of the shapes of the head (or pressure) versus capacity characteristic curves for the various compressor types is shown in [Figure 12](#). Further details on centrifugal and axial compressor curve shapes are included in [Design Practice XI Section E and F](#).
- **Lube Oil Injection into Process Streams** - Compressors which require internal lubrication (lubricated reciprocating, oil injected screw and sliding vane) are unsatisfactory for oxygen service and certain reactor feed services where oil deposits seriously contaminate catalysts. Internal compressor lubrication is undesirable (although frequently used) for instrument air and refrigeration services.
- **Liquid Carryover in Process Gas** - The compressor types with the least mechanical sensitivity are ejectors, liquid ring compressors, and high pressure helical screw compressors, in that order. The most adversely affected types are lubricated and unlubricated reciprocating, sliding vane and axial compressors. See [Table 2](#).
- **Solids in Process Gas** - Liquid ring compressors are least sensitive, followed by high pressure helical screw compressors. The most sensitive types are sliding vane, non-lubricated and lubricated reciprocating, and axial. See [Table 2](#).
- **Molecular Weight Swings** - When compared with dynamic compressors, the performance of positive displacement compressors is less sensitive to swings in gas composition. Dynamic compressors are not suited to large swings in molecular weight. **Caution: for all machine types, the full range of gas compositions must be specified as it may effect staging arrangements and mechanical design.**
- **Discharge Temperature Sensitivity of Gas** - All types can be designed with multiple staging to limit temperature rise. The rotary screw and straight lobe types can be designed for liquid injection cooling. Liquid ring compressors hold discharge temperature close to the compressant liquid inlet temperature.
- **High Inlet Temperature** - Centrifugal compressors and fans can be specially designed for inlet temperatures in the 350 to 1000°F (175 to 540°C) range. High pressure helical screw compressors can be designed for up to 450°F (230°C), through the use of oil cooled rotors.
- **Fouling Tendency of the Gas** - Axial and high speed, single stage centrifugal are not suitable for fouling service. Flushing allows helical screw and conventional centrifugal compressors to be used in a fouling service. See [Table 2](#).

For a summary of how performance parameters affect compressor design See [Table 3](#).

Factors Directly Influencing Comparative Economics

- **Machine Prices** - See Cost Estimating Manual. Consult machinery specialists for supplemental estimates.
- **Installation cost level** - This is dependent upon physical size, mechanical complexity and multiplicity. Dynamic and high pressure helical screw compressors have significantly lower costs than reciprocating compressors.



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- **Efficiency** - This influences operating cost. See [Design Practice XI Section D, Compression Calculations](#) for typical efficiencies for each compressor type.
- **Maintenance requirements** - Typical maintenance costs and operating costs for all utilities should be considered. See [Design Practice XI Section C, Compression Machinery Serviceability and Sparing](#).

COMPRESSION SERVICE MACHINE TYPE SELECTION PROCEDURE

Selection of the compressor type for each application should be made by working through the following procedure:

- Compare the power requirement level with the range of power ratings defined in [Table 1](#) to eliminate some types from consideration and judge whether application is routine or unusual.
- Note which compressor types have the inlet volume flow and discharge pressure point within the economic field, and which types have the point within the feasible field, using the compressor application range charts. See [Figures 2 through 12](#).
- Eliminate types which are technically unsuitable due to particular service requirements.
- Consider prior local experience for similar services.
- Obtain machinery specialist assistance to decide whether specific investigation of any of the compressor types which fall in the feasible category is justified.
- For the remaining compressor types, decide how many units and of what capacity would be installed, and what capital spares would be required.
- Determine what compressor types have been applied to this service in recent projects, noting what the flow rates on the other projects have been.
- Review advantages of each compressor type being considered. See [Appendix A, Comparative Advantages of Various Compressor Types](#).
- Make a brief economic comparison of competitive cases, using performance data from the appropriate subsections in this DP section, cost data from the **Cost Estimating Manual** and include operating and maintenance costs typically over a 3-5 year period.
- If the brief economic study shows strong competition between two or more types, obtain machinery specialist assistance to get current cost and performance data from at least two manufacturers of each compressor type.
- Perform a detailed economic study using the manufacturer's current data.
- If the results of the economic study indicate strong competition between compressor types, write the Design Specification on the basis of the more likely choice and specify that complete (technical and commercial) proposals should also be solicited for the alternate type. The proposals should be evaluated during the detailed engineering phase.



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FACTORS AFFECTING SELECTION OF UTILITY AIR COMPRESSOR TYPE

- ♦ **Type Selection** - Because of wide choice, rapidly changing technology, and high competition between compressor types, it is usually more advantageous to make the final compressor type selection for plant air service during detailed engineering, based on competitive commercial proposals, rather than to make it during the plant design phase. See the application range charts to determine the types most likely to be competitive.

The compressor type selection procedure previously used for process compressor service can also be used to select compressor type for utility air service.

An oil-free compressor type is preferred over the lubricated reciprocating type, where any part of the compressed air will be used for instrument air. This is because the cleanliness of the plant instrument air system will not then depend on the maintenance and performance of oil removal equipment.

Applicable Types - The following six compressor types actively compete for plant utility and instrument air applications, i.e., 110 to 130 psig (7.6 to 9.0 barg) discharge pressure, across the capacity range commonly needed for refineries and chemical plants:

- **Lubricated Reciprocating** - This was by far the most frequently used type until the mid 1960's. Models are well developed and highly standardized; efficiency is high and reliability good for reciprocating machines. The disadvantages are:
 1. Cylinder lube oil removal facilities must be provided for the portion of the discharge supplying the instrument air distribution system; this equipment requires frequent maintenance and is not always completely effective.
 2. Maintenance shutdowns are more frequent than for rotary and centrifugal types.
 3. Costs are high for high flow rates.
- **Non-Lubricated Reciprocating, PTFE Piston Rings** - Materials and wearing parts designs have improved to the point where the maintenance requirement of fully developed commercial models is only moderately higher than for conventionally lubricated reciprocating models.
- **Labyrinth Piston Reciprocating** - Oil-free; higher in cost and marginally lower in efficiency than the PTFE ring non-lube style, but distinctly lower in maintenance requirements. High initial cost is difficult to justify for plant air service, despite low maintenance requirements.
- **High Pressure Helical Screw** - This type has been widely used for plant and process air services in Europe (including Exxon refineries) since the late 1950's. Costs are low for high capacities; operation is oil-free for unlubricated screw designs. Oil flooded screw designs suffer the same disadvantages with oil removal facilities as the lubricated reciprocating compressors.
- **Packaged Integrally Geared Centrifugal** - Typically multistage, high speed compressors. Most models use a single impeller for each process compression stage, an integral gearbox with impellers driven by multiple high speed pinions. The compressor is packaged with a pre-piped inter-cooler system. Most of the major compressor manufacturers now have commercial models with large numbers of installations for as low as 700 ICFM (1220 m³/h) and as high as 18,000 ICFM (30,600 m³/h). Higher flow models have been developed and applied in the air separation industry with inlet volume flows in excess of 70,500 ICFM (120,000 m³/h). The units are intercooled for high efficiency; they are low in cost for high volumes; operation is oil-free. Reliability records are good but do not yet match those of conventional process centrifugal compressors.
- **Intercooled High Volume Centrifugal** - Horizontally split single casing, low speed. Developed in the 1950's for low power compression of high volumes of air for process services (liquefaction, ammonia plants, etc.), but also applicable and economical for plant air service in large refineries.

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FACTORS AFFECTING SELECTION OF MACHINE TYPE FOR VACUUM SERVICE

Applicable Types - The term "Vacuum pump" refers to any compressor type with an inlet pressure well below atmospheric. To obtain very low absolute inlet pressure ("high vacuums") - for instance, below 100 Torr - two or more vacuum pumps, often of different types, are placed in series. Multi-staging permits optimum application of each type. [Figure 14](#) shows the pressure ranges in which the various types are applied. Evacuation service for ejectors is covered in [Design Practice XI Section J, Ejectors](#).

Special Characteristics - Vacuum services have several characteristics significantly different from other compression services, such as the following:

- Compressors are physically large for a given mass flow rate, because of the low density of the gas at vacuum inlet conditions.
- Inlet silencers, filters, and piping must be generously sized for very low pressure drop because of the significant effect on pressure ratio and head requirement.
- Compressor shaft seals must prevent air leakage into the machine, as well as prevent gas leakage to atmosphere.
- The mass and inertia of the moving parts tend to be high in relation to required driver rating because the low gas density causes the normal power requirement to be low. Over sizing of motor drivers is often required to provide sufficient torque for acceleration of the unit to full speed.
- For reciprocating compressors, cylinder jacket cooling tends to be unusually effective at removing compression heat, due to low gas mass flow with respect to cooling water flow rate.
- Driver sizing must allow for the maximum load during the inlet system evacuation period, immediately after startup. This operation of startup evacuation is also known as "pump-down" or "hogging." As the inlet pressure decreases during evacuation, the horsepower required reaches a peak between the atmospheric inlet condition and the operating inlet condition. This power versus inlet pressure characteristic must be evaluated by the vacuum pump vendor to determine the minimum allowable driver size.

MACHINE TYPE SELECTION FOR VACUUM SERVICE

Compressor type selection for vacuum services is somewhat more difficult than for other services because a wide choice exists between ejectors, reciprocating, centrifugal, and various types of rotary compressors. Standard packaged equipment sets are commercially available for various capacities and vacuum levels. Final type selection may not be practical before complete vendor proposals are solicited. The following procedure is recommended. If choice is not obvious after completing this review, then make a detailed economic comparison of investment and operating costs of the applicable types.

- Use [Figure 14](#) to determine applicable types, based on inlet pressure. For higher than atmospheric discharge pressure, select applicable types based on comparable pressure ratios.
- Eliminate technically unsuitable types, for reasons such as the following (See [Design Practice XI Section E through J for the characteristics of each type](#)):
 - Reciprocating and sliding vane compressors are not suitable where liquid carryover is likely.
 - Steam, electric power, or cooling water may be unavailable at the installation site.
 - Capacity requirement may be out of the range feasible for some types (see application range charts).
 - Large variation in molecular weight is not suitable for dynamic compressors.
 - Oil-free characteristics may be required.
 - Non-condensing multistage ejectors are inefficient for continuous service.
- Decide between ejectors and compressors based on:
 - Reliability and sparing needs.
 - Rough investment costs.
 - Utility consumption and costs.
 - Ejectors are very reliable with low first cost, but other types are 3 to 10 times more efficient.



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APPENDIX A

COMPARATIVE ADVANTAGES OF VARIOUS COMPRESSOR TYPES

- ◆ The following paragraphs review the advantages and disadvantages of a particular machine type as compared to its major competitor(s). The listed comments, advantages and disadvantages are intended as guidance for individual case studies for machine type selection.

AXIAL COMPRESSORS VERSUS CENTRIFUGAL COMPRESSORS

Axial compressors compete directly with centrifugal compressors in the inlet flow range of 30,000 to 300,000 ft³/min. (51,000 to 510,000 m³/h). Specific economic comparisons of the two types are usually necessary throughout this range. Below inlet volumes of 70,000 ft³/min. (119,000 m³/h), the centrifugal usually proves to be more attractive; above inlet volumes of 130,000 ft³/min. (220,000 m³/h), recent experience has shown that multiple centrifugal compressors can compete economically with a single axial compressor. (For example, ManTurbo's isotherm compressor was measured against an axial compressor for the GTL project.) Reliability of multiple centrifugal bodies against a single axial should be weighed for the specific project. Results tend to depend on the specific circumstances of the case, rather than on generalized comparisons of the two machine types. The following list of general advantages and disadvantages of axial compressors is intended to help guide case studies:

• Advantages

1. Very high inlet volume flow capacity per machine up to 700,000 ft³/min. (1,200,000 m³/h). More designs are available for inlet volume flows above 200,000 ft³/min. (340,000 m³/h) than for centrifugal compressors.
2. Efficiency can be as much as 10% higher than that of centrifugal compressors, resulting in lower power consumption, lower driver rating, and smaller utilities supply system. However, individually cooled stages in a centrifugal compressor have significantly narrowed this gap.
3. Smaller physical size and lighter weight than for centrifugal compressors, allowing lower installation costs, i.e., due to shelter size, crane size, plot area, foundations, installation handling effort, etc.
4. With a steam or gas turbine driver, the higher speed usually allows direct drive (no gear unit) and efficient turbine designs.
5. Rotor/casing design can provide flexibility to make minor performance modifications somewhat more conveniently (by adding, removing and shifting stages and by adjusting stator vane angles) than with centrifugal compressor construction.
6. Higher pressure ratio per casing (limited by discharge temperature) due to higher efficiency.
7. Easier than centrifugal compressors to operate in parallel with other compressors of all types, due to the steep head capacity curve.
- ◆ 8. Higher flows can be achieved in an axial versus a centrifugal body. For most applications, a single body is the most cost effective solution considering initial, operating and outage costs.

• Disadvantages

1. Narrower range of flow for stable operation, especially with constant speed drive, unless costly variable stator vane design is used.
2. Flow control systems and surge protection controls are more complex and costly than for centrifugal compressors. Surge control must be very reliable, since surge can quickly damage an axial compressor.
3. Performance deterioration due to gas path fouling and erosion is more severe than in centrifugal compressors. This requires better inlet filtration and makes axial compressors unsuitable for long continuous runs in fouling services.
4. Damage by ingested objects tends to be more extensive than in centrifugal compressors.
5. At present, developed models for process applications generally have lower pressure limitations than do centrifugal compressors. Upper pressure limit of most manufacturers is 170 psig (12 barg). (However, axial compressors have the potential for development to pressure levels at least as high as those of centrifugal compressors.)
6. Very limited experience to date in other than air service, making justification for application to new gas services difficult.
7. The cost of capital spares (including stator blades) for the axial tends to be higher than for the centrifugal.
8. Higher noise level requires more extensive acoustical treatment than for a centrifugal.
- ◆ 9. Reliability of multiple stages of variable stator vanes can be a concern. Movement failure in a single variable stator may lead to uncontrolled surging and failure of numerous blade rows.
- ◆ 10. Overall, an axial compressor tends to be less rugged and less forgiving of mistakes due to operation, maintenance and foreign object ingestion. Failures modes tend to be more severe with longer downtime and higher repair costs.

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• **Standoff Factors**

1. Prices for axial compressors and centrifugal compressors tend to be highly competitive in the flow range to which they are both applicable, especially in the inlet volume flow range of 70,000 to 130,000 ft³/min., when stator vane controls is provided on the axial.
2. Reliability factor records for two types are equally high.
3. Sensitivity to over temperature and rotor vibration are about the same for both types of machines.
4. Risks of blade/impeller vibration failures, assuming equally careful design and thorough component testing are roughly equivalent.
5. Piping and valving costs are about the same for axial compressors and centrifugal compressors.

CENTRIFUGAL COMPRESSORS VERSUS RECIPROCATING COMPRESSORS

The principal advantages and disadvantages of centrifugal compressors, relative to reciprocating compressors, may be summarized as follows:

• **Advantages**

1. Long continuous run lengths (typically 5 years) are possible with high reliability, eliminating the need for multiple compressors and installed standby capacity.
2. For the same operating conditions, machine prices are lower for high volume flow rates.
3. Machine is small and light weight with respect to its flow rate capacity, and requires less plot area for installation.
4. Installation costs are lower due to smaller size, absence of reciprocating forces, and because only one unit is normally installed.
5. Total maintenance costs are lower than for reciprocating compressors.
6. When a turbine is selected as a driver, the centrifugal compressor's speed level allows direct drive (no gear unit), thereby minimizing equipment cost, reducing power requirements, and increasing unit reliability.
7. Flow control is simple, continuous, and efficient over a relatively wide flow range.
8. No lube (or seal) oil contamination of process gas, as occurs with lubricated reciprocating compressors.
9. Smooth flow characteristics, without pressure pulsation above surge point.

• **Disadvantages**

1. Lower efficiency (by 7 to 13%) than most positive displacement types for the same flow rate and pressure ratio, especially for pressure ratios over 2.
2. Operation is not efficient below the surge point, since recycle is necessary.
3. The differential pressure capability is sensitive to changes in gas properties, especially molecular weight, because the machine has a definite head limitation. This makes service design very critical for variable molecular weight streams. This becomes less of a concern for variable speed applications.
4. For low molecular weight gases, the pressure ratio capability per stage is low, tending to require a large number of machine stages, hence mechanical complexity.
5. As a guide, assume that conventional centrifugal models are generally not available for actual volume flow rates at the discharge of the last compressor stage below the following:

<u>Gas MW</u>	<u>Minimum Discharge Flow - ACFM (m³/hr)</u>
2	270 (459)
16	240 (408)
30	210 (357)
44	180 (306)
58	150 (255)

For lower flow rate applications, consult a machinery specialist.



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RECIPROCATING COMPRESSORS VERSUS OTHER COMPRESSOR TYPES

Reciprocating compressors compete with all types of compressors except centrifugal compressors and axial compressors at very high flow rates. Their principal advantages and disadvantages follow:

- Advantages**

1. Available for capacities below the economic flow range of centrifugal compressors.
2. Economic for the very high heads typical of low molecular weight gas services.
3. Available for extremely high pressures; often used for discharge pressures above 3,600 psig (250 barg).
4. Much less sensitive to gas composition and average property changes than dynamic compressors.
5. Suitable for 0 to 100% stepwise flow variation, via clearance pockets and valve unloaders, with minimum power wastage at low rates.
6. Higher overall efficiency than centrifugal compressors for pressure ratios over 2.
7. Flow rate insensitive to discharge pressure level changes.
8. Lower discharge temperature than centrifugal compressors, due to higher efficiency and jacket cooling.
9. Much less sensitive to coupling misalignment and piping forces than centrifugal, axial and rotary compressors, which operate at higher rotational speeds.

- Disadvantages**

1. Large foundation masses are required to avoid high vibrations due to the reciprocating forces.
2. In continuous services, multiple units are required to avoid plant outages for compressor maintenance.
3. Maintenance costs are 2 to 3 times as high as those of centrifugal compressors.
4. Continuous run length potential is much shorter than for centrifugal compressors; outage frequency is much higher, primarily due to valve failure.
5. Reciprocating compressors are sensitive to solids carryover, due to use of rubbing parts.
6. Lubricated types machines are sensitive to liquid carryover, due to destruction of the lubricant film.
7. Large plot area is required, compared to that for rotary and centrifugal types.
8. Lubricated type machine injects lube oil into gas stream; non-lube type requires wearing parts which must be renewed frequently.
9. More frequent operator inspection is required than for other types, due to susceptibility to valve failure and force feed lubrication failure.

ROTARY COMPRESSORS VERSUS RECIPROCATING COMPRESSOR

Rotary compressors, as a class, compete with reciprocating compressors in services below 500 psig (35 barg). Most types of compressors within the rotary class have the following advantages and disadvantages when compared to reciprocating compressors:

- Advantages**

1. Lower cost for most moderate pressure services in the 0 to 2,000 hp (1500 kW) range.
2. Longer continuous run length potential.
3. Somewhat lower maintenance costs.
4. Smaller foundations due to absence of reciprocating forces.
5. Marginally higher mechanical efficiency than reciprocating compressors.

- Disadvantages**

1. Lower compression efficiency, due to slip, resulting in lower overall efficiency.
2. Machine is more closely matched to design conditions and less versatile for off design pressure ratios.
3. Much lower discharge pressure limitations.
4. Flow control normally limited to speed or recycle.