18

USE OF MULTIPLE-INLET COMPRESSORS*

In earlier chapters we provided a thorough introduction to compressor performance prediction. In essence, we dealt with *single-inlet* machines. However, applying, analyzing, designing, and testing *multiple-inlet* compressors may differ substantially from working with typical single-inlet centrifugals. Since multi-inlet, or sideload compressors are quite common in the user industry, we will highlight certain aspects of this type of compressor that are mutually important to the user, contractor, and vendor. Further, an understanding of the sideload compressor is essential to provide a matched compressor and process installation.

18.1 CRITICAL SELECTION CRITERIA

Careful consideration of all operating parameters is required to ensure satisfactory compressor and process fit. Unfortunately, these parameters cannot be considered independently; rather, an overall operating analysis is required. This may result in certain operating parameters (or operating levels) being "desired" rather than being "critical" to successful operation of the overall process.

The common design parameter in API 617, the ASME Code, and other codes may require some modification when applied to the sideload compressor. The usual guarantee of flow, discharge pressure, and consumed horsepower probably will not ensure proper compressor and process match.

* Contributed by Kenneth L. Peters (Elliott Company, Jeannette, Pa.), as published in *Hydrocarbon Processing,* May 1981. Adapted by permission of Gulf Publishing Company, Houston, Tex.

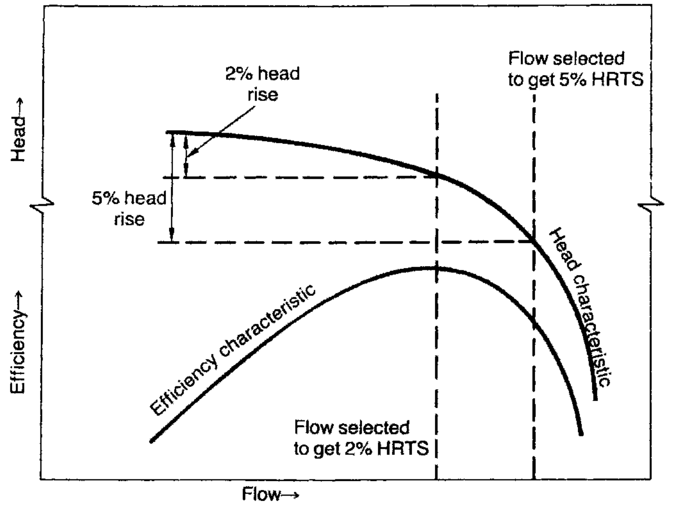

FIGURE 18.1 Effects of 2 and 5% head rise to surge. (*Elliott Company, Jeannette, Pa.*)

Next we point out some parameters that will have a direct effect on compressor selection. This summary is not presented as absolute but rather to demonstrate overall analysis of the specified operating parameters. The discussion includes:

- Head rise to surge, surge margin, overload margin
- Head per compression section
- Compressor parasitic flows (i.e., balance piston leakage)
- Excess margins on other process equipment

18.1.1 Head Rise to Surge, Surge Margin, and Overload Margin

Over the last few years, process engineers have asked for a characteristic curve shape guarantee including *head rise to surge* (HRTS), surge margin, and/or overload margin. Other common terms referring to HRTS include *pressure rise to surge* and *pressure ratio rise to surge*, among others. Depending on other parameters specified, this addition to the guarantee may have no effect or may result in nonoptimum compressor selection. The typical compressor characteristic map shown in Fig. 18.1 illustrates this point. In this example it is evident that a desirable level of 5% head rise to surge will result in nonoptimum efficiency and overload, while a 2% level will give the best efficiency and overload selection.

Another area of interplay on HRTS is with surge margin. The refrigeration process (the widest application of sideload compressors) requires operation at nearly constant discharge pressure (Fig. 18.2). Either driver speed or suction pressure must be reduced as flow moves toward surge. Figure 18.3 shows an actual example where a 15% surge margin was specified with 5% HRTS. Further, the unit had a constant-speed drive with a trip-out if suction pressure dropped below atmospheric pressure. Design suction pressure was 15 psia. With the required 5% HRTS, the unit would trip off stream at 97% design flow. Unfortunately, even 2% would lend the surge margin stipulation immaterial.

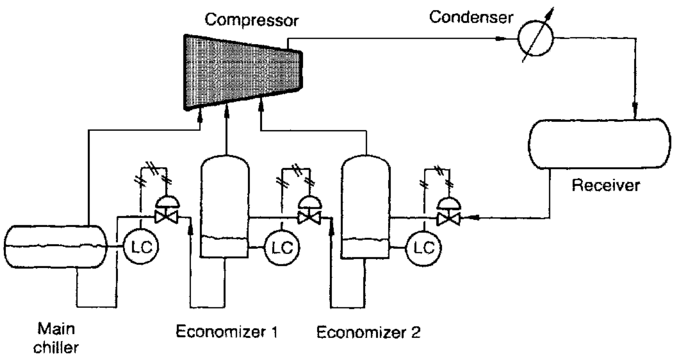

FIGURE 18.2 Simplified multilevel refrigeration process. (*Elliott Company, Jeannette, Pa.*)

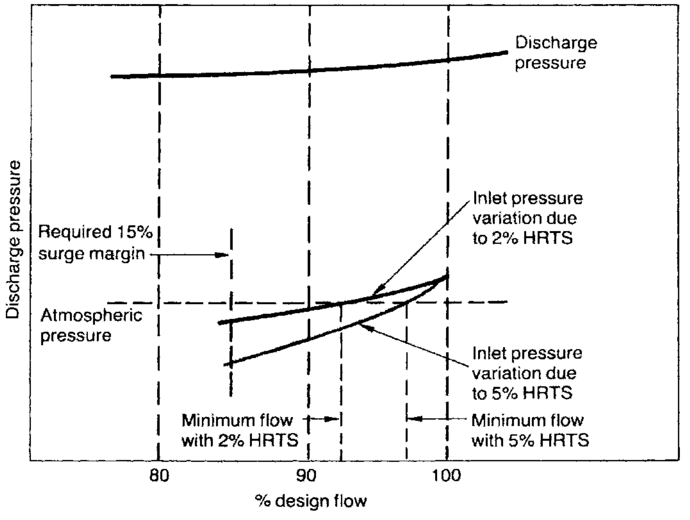

FIGURE 18.3 Effects on pressure for head rise to surge variations (constant-speed drives). (*Elliott Company, Jeannette, Pa.*)

Because of various aerodynamic laws, the surge to stonewall flow range at the conditions of selection for refrigeration service is reduced relative to that seen on various less severe applications. While other applications may have overall flow ranges of 40 to 50%, range on a refrigeration selection is likely to be 20 to 30%. Hence, imposition of an excessive HRTS and/or surge margin criterion may result in only minimal overload capacity, as indicated in Fig. 18.2. Conversely, an excessive overload margin stipulation may result in too low a HRTS or surge margin for safe, reliable, efficient operation.

18.1.2 Head per Section

Another important operating parameter that must be evaluated is the required head per section split on the compressor. Requests for smallest possible bearing spans may create or aggravate various aerodynamic considerations. The fewer impellers per section, the higher the head required per compressor stage. Higher head per stage requires increased rotative speed, resulting in operation at higher Mach numbers. Operation at higher Mach numbers tends to restrict the operating range, flattens the head rise characteristics, and reduces efficiency.

Operation at lower Mach levels is achieved by increasing the compressor stages per section and lowering rotative speed. Rotor dynamics criteria and hardware costs limit this approach. Although there is an optimum sidestream pressure level, as defined by the cycle, the pressure level may require only slight adjustment to allow better compressor selection.

18.1.3 Compressor Parasitic Flows

Reentry of seal equalizing line flow into the main gas stream can also influence compressor selection. If these flows are significant, reentry at a point other than the main inlet may be advantageous so that the sensitivity or tolerance effect on the operating range and efficiency may be reduced. If this option is taken, the manufacturer must take care that its effect on other aspects of the overall mechanical compressor design has been adequately considered. This effect can best be seen by reviewing Fig. 18.4.

One can see that for the same flow entering the main compressor inlet flange, the true operating point on the section characteristic may be significantly different. This problem is readily alleviated by, for instance, putting the parasitic flow into the first sideload. If the parasitic flow is, say, 5% of the first section, it may only be 1 to 2% of the overall flow (also consumed horsepower) entering the first wheel of the next section downstream of the sideload. Hence, variations due to balance piston seal leakage are negligible.

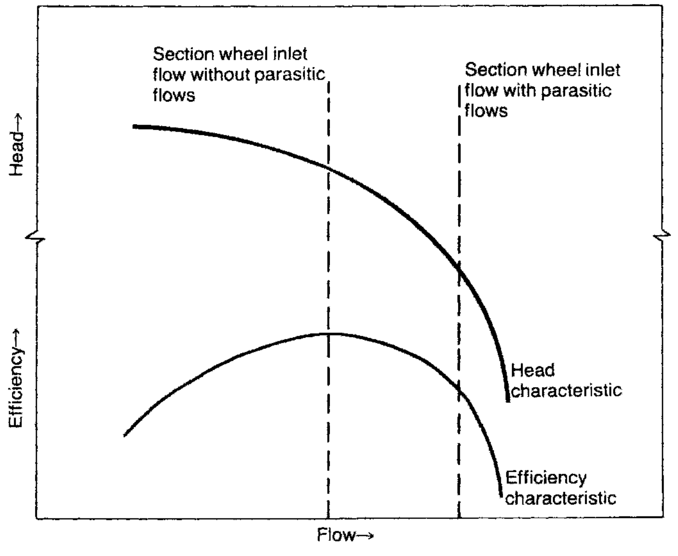

FIGURE 18.4 Possible performance shift due to parasitic flows in a multistage compressor. (*Elliott Company, Jeannette, Pa.*)

18.1.4 Excess Margins on Other Process Equipment

Design of excess margin into the various process components is also a very important consideration. For example, assume that the process designer allows 5% excess in his specification to the contractor, the contractor takes in an additional 5%, and the compressor vendor elects to add 2% in flow. The process design flow now is approximately 88% of the compressor design flow. Operation at the process design flow may greatly reduce the stability margin of the compressor and, in turn, the overall cycle.

Additionally, consider the situation where a constant-speed driver is used. The unit may be operating below atmospheric pressure at the inlet as a result of the margins used in specification of the machine. (This is similar to the situation shown in Fig. 18.3.) A cumulative accounting of all excess margins will aid in realizing successful compressor selection.

18.1.5 Representing Compressor Performance

In the past, typical performance for sideload units has been judged on a set of individual sectional curves based on constant inlet conditions to each section. These curves, however, have no direct relation to the way the unit will operate with a process.

Compressor performance should be presented as individual sectional curves in conjunction with a graph of expected performance based on the type of operation expected in the field. This *constant turndown map* can easily be generated from the individual sectional curves after making assumptions on mode of control, temperature, and other operating conditions.

This map is generated as follows from the sectional curves. Discharge or condenser pressure is assumed constant at design value. The sectional curves then are used to determine exact sideload pressures as a function of mass flow to each section. Normally, a constant turndown in mass flows is assumed. The net result is a characteristic curve for the compressor showing each sideload pressure (and inlet pressure) as a function of mass flow. Speed or inlet pressure (constant-speed units) are varied to meet design discharge pressure requirements. This provides a curve of the actual mode of operation of the unit.

Part load and overload operation data, at least initially, are generated by reducing or increasing all incoming mass flows by the same percentage. Also, initially, design temperatures are assumed constant. A minor modification can account for temperature variations at each inlet due to the flow or pressure change. Figures 18.5 and 18.6 are presented to show constant- and variable-speed applications, respectively.

If generated with input from the customer, a turndown map can be used directly in a simulation. A good simulation analysis provides a much better evaluation of how a unit would operate in the field than sectional curves alone. Review of the turndown map will be a great aid in analyzing interplay of requested levels of operating parameters. Two important items observed directly from the map are pressure levels over the expected operation range of the system and potential stability of the variable-speed driver.

18.1.6 Practical Levels of Critical Operating Parameters

Because no code exists to establish recommended design parameters and limits for sideload units, the following list is recommended as a basis from which pertinent discussions can develop.

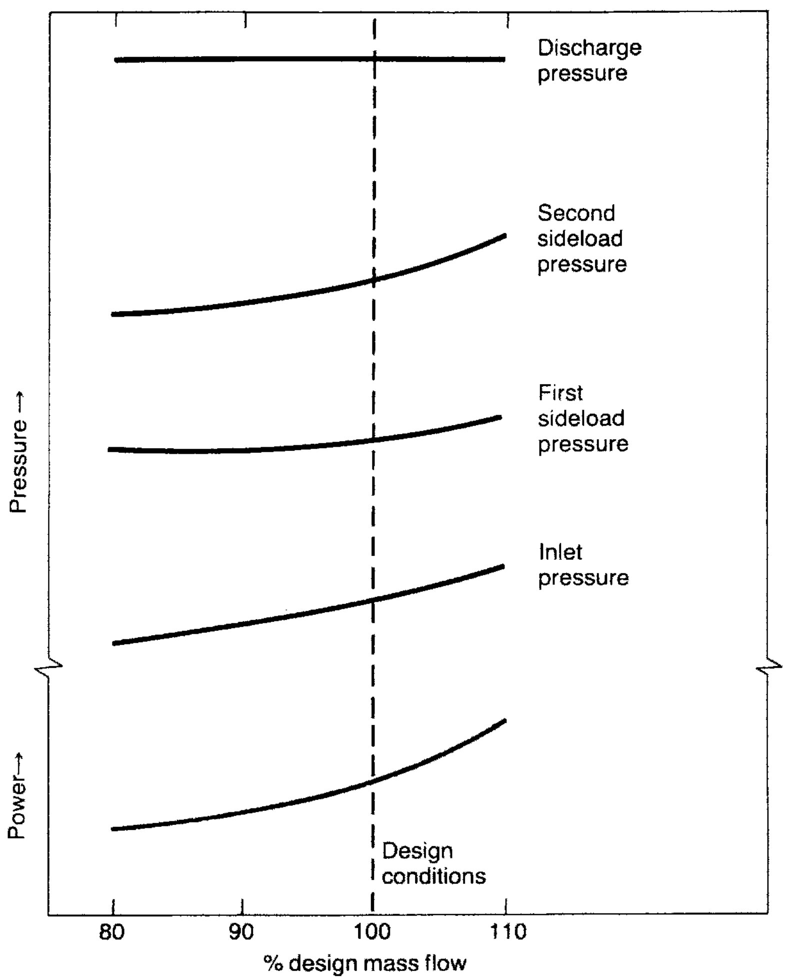

FIGURE 18.5 Typical turndown map with a constant-speed driver. (*Elliott Company, Jeannette, Pa.*)

- *Overall unit horsepower.* This parameter can be made in accordance with the code as $\pm 4\%$, but because of limitations on test instrumentation and calculations, some assumptions may need to be included in the calculations.
- *Sectional head rise.* This particular parameter should be set as low as possible for both constant- and variable-speed applications. Based on experience, a number as low as 2 to 3% overall can be adequate in most applications, assuming that integration of the controls has been done properly.
- *Overload and stability margins.* These quantities are interrelated and collectively should be about 20 to 30% of the design flow value. Minimum margins should be 15% for surge and 5% for overload, or 20% combined range.
- *Sideload pressure level.* This particular parameter is usually the most unstated requirement on the sideload compressor. A reasonable tolerance is $\pm 2\%$ on the section head. Individual process designs may require that these be altered.

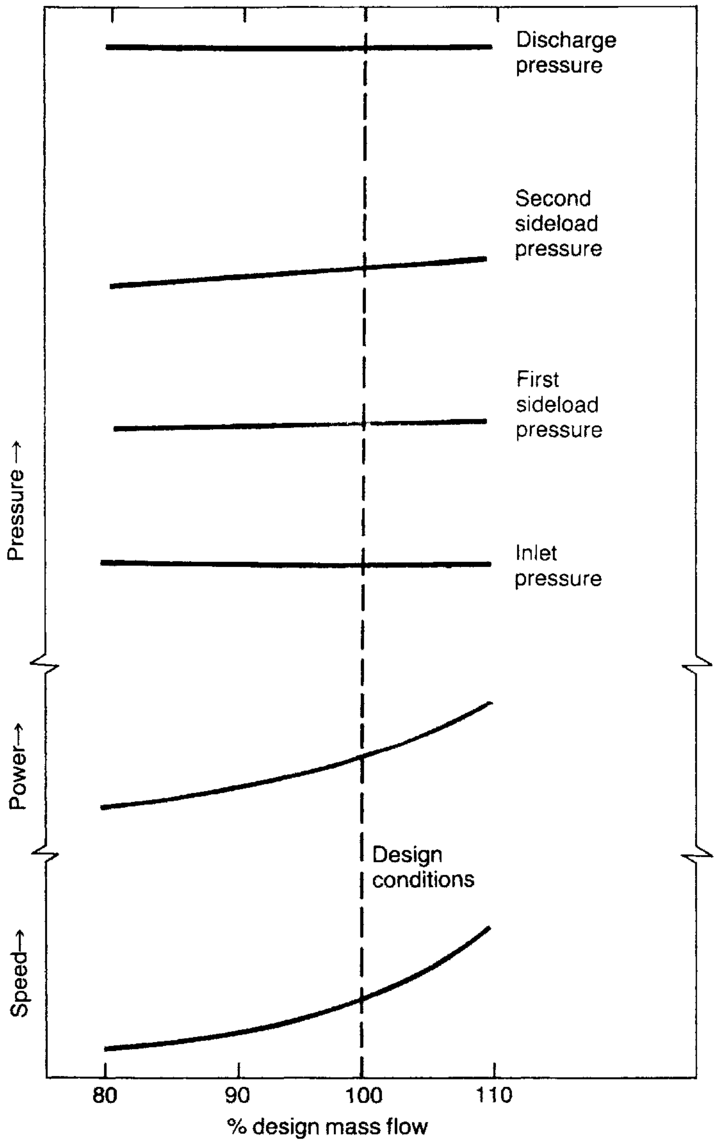

FIGURE 18.6 Typical turndown map with a variable-speed driver. (*Elliott Company, Jeannette, Pa.*)

- Other desired tolerances or requirements need to be reviewed during the application phase of the compressor so that all parties realize their implications.

18.2 DESIGN OF A SIDELOAD COMPRESSOR

Design of each sideload compressor application is dependent on the required parameters discussed previously. However, there are several design areas of a general nature that need to be highlighted.

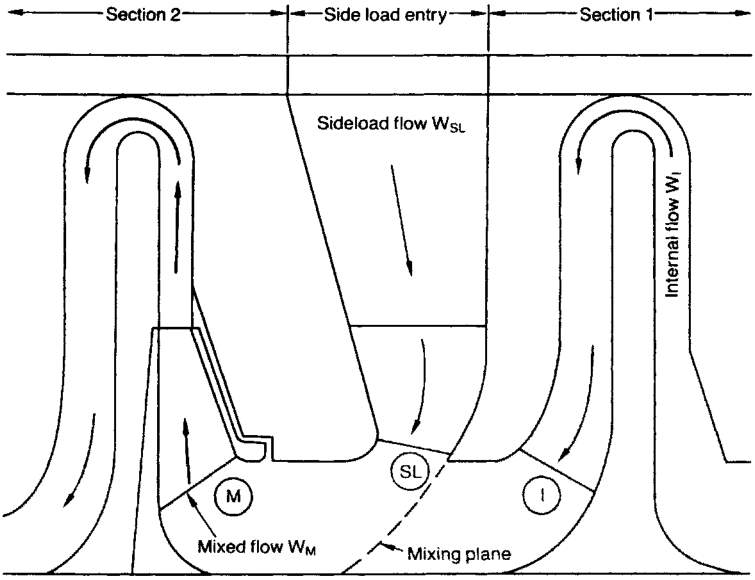

FIGURE 18.7 Cross section of sideload into the main gas stream. (*Elliott Company, Jeannette, Pa.*)

18.2.1 Mixing Area

One of the more complex design areas is in adequate calculation and physical occurrence of the mixing of two streams internally in the compressor. An analysis using conservation of mass, momentum, and energy provides the calculated properties of state following the interaction of the two streams. Any loss due to the mixing process is assumed to be a function of the amount of momentum lost by the main gas stream during the interaction. Also, the mixing process is assumed to occur at a constant static pressure (Fig. 18.7). That is, flowing conditions at station M are obtained by combining conditions of the main and sidestreams at stations I and SL, using conservation of mass, momentum, and energy. This analysis is a function of momentum. Thus, velocities of the two streams at the point of mixing are critical. Close attention to the area ratio of both incoming streams is a must for optimum mixing.

A direct result of improper area control in the mixing chamber is that it may create different operating characteristics, depending on location of the measuring equipment. With the static pressures of both streams equal at mixing, the velocities determine the total pressure of each stream. It is possible that total pressure calculated at station I will be higher than total pressure calculated at the sidestream flange. Hence, if process pressure controls are connected at the compressor flanges, it is possible that the compressor will not be monitored accurately. Figure 18.8 depicts this problem graphically.

In Fig. 18.8, internal performance is measured from the mixing plane to the exit of the section (station I in Fig. 18.7). External flange-to-flange performance is measured at the respective compressor flanges. To match internal performance with external performance, control of the mixing areas is a must.

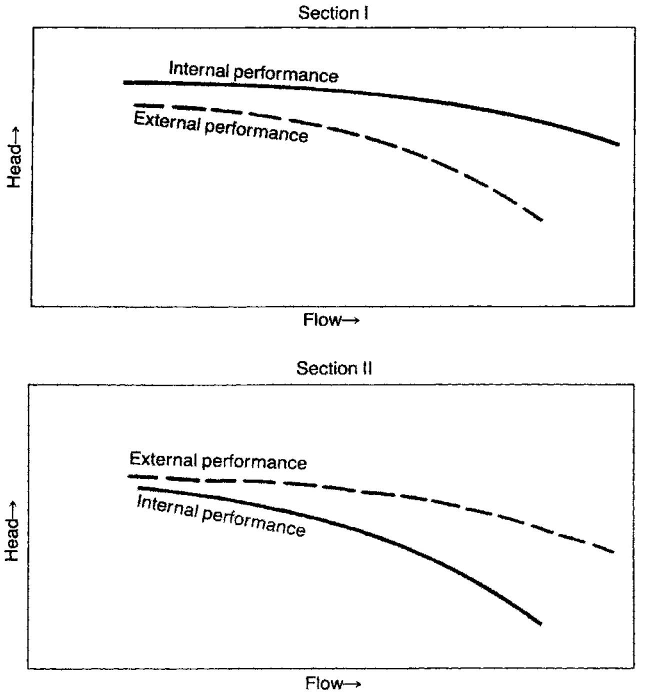

FIGURE 18.8 Possible effects of improper mixing area control as a function of measurement location. (*Elliott Company, Jeannette, Pa.*)

18.2.2 Aerodynamics

Sideload compressor applications are normally used in refrigeration systems that combine high molecular weight and low temperatures. This results in operation at higher Mach numbers. From the equation

$$a_s = \sqrt{kgRT_s} \tag{18.1}$$

where a_s = sonic velocity
k = ratio of specific heats
g = gravitational constant
R = universal gas constant
T_s = static temperature, °R

As molecular weight goes up and T_s decreases, sonic velocity is reduced. Hence, for the same relative velocity of the flowing gas, the relative inlet Mach number is higher than that for, say, air at standard conditions flowing at the same rate.

Notice that relative inlet Mach number is used, not simply Mach number. Care must be taken when reviewing Mach numbers that all values are based on the correct gas velocity. Very simply, the two Mach numbers usually referenced are *inlet Mach number* and *relative inlet Mach number*. Inlet Mach number is typically calculated as the Mach number as flow enters the impeller, measured either at the impeller eye or at the leading edge of the blade passage (Fig. 18.9).

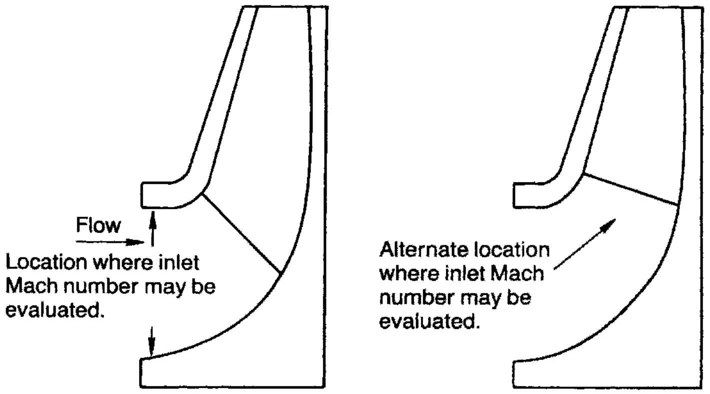

FIGURE 18.9 Locations sometimes used to measure the inlet Mach number. (*Elliott Company, Jeannette, Pa.*)

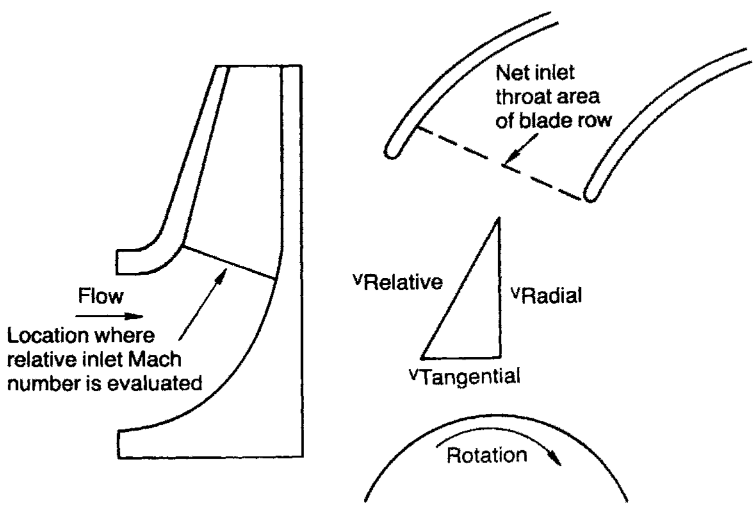

FIGURE 18.10 Impeller geometry for determining the relative inlet Mach number. (*Elliott Company, Jeannette, Pa.*)

In either location the area is unobstructed. Velocity is typically derived by dividing the inlet flow rate by the area, without considering the rotating disk or blade blockage. Conversely, relative inlet Mach number denotes the Mach number of the flow stream perpendicular to the throat section of the impeller (Fig. 18.10). Needless to say, relative inlet Mach numbers are higher than inlet Mach numbers because net throat area is less than the area of the impeller eye or at the leading edge of the blade passage.

Proper application and basic knowledge of proven aerodynamic hardware have permitted successful operation of stages at relative inlet Mach numbers in the area of 0.95 and above. As operating Mach numbers increase, the stable operating flow range decreases and the compressor characteristic curve flattens (Fig. 18.11). *Flattening* is the result of stage component efficiency curves displaying a sharper peak, with the peak moving toward higher flow

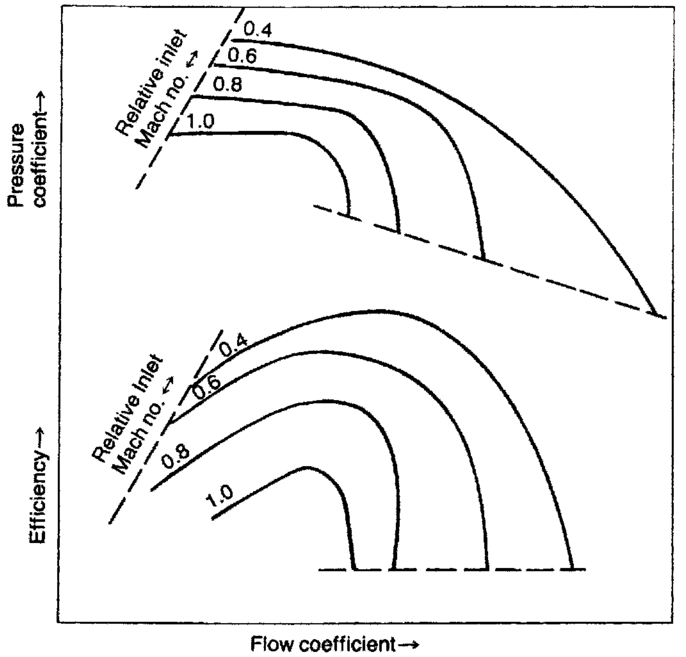

FIGURE 18.11 Effects of the relative inlet Mach number. (*Elliott Company, Jeannette, Pa.*)

coefficients with increased Mach numbers. The result is that head capability at part load is reduced relative to design, and the curve displays a lower slope.

With increasing Mach number, the maximum overload (choke or stonewall) flow coefficient moves steadily toward the design value. Similarly, on the low-flow end, the surge flow coefficient moves toward the design value. Very simply, the higher the Mach number, the smaller the stable operating flow range.

18.2.3 Temperature Stratification

An area of concern is temperature stratification, both radial and circumferential. A good inlet and mixing area design will help minimize circumferential stratification, however, radial stratification will exist to some extent.

18.3 TESTING

The acceptance performance test serves as a confirmation of all of the care taken in the application, design, and manufacture of the sideload unit. The ideal method needed to ensure this confirmation is to perform a controlled ASME-type test in actual service. However, for a number of reasons this is usually impractical. Thus, a modified ASME equivalent performance test may be completed in the compressor vendor's shop prior to shipment.

The ASME equivalent performance test for sideload units per se is not clearly defined. In the absence of concise procedures, each section is generally regarded as an individual

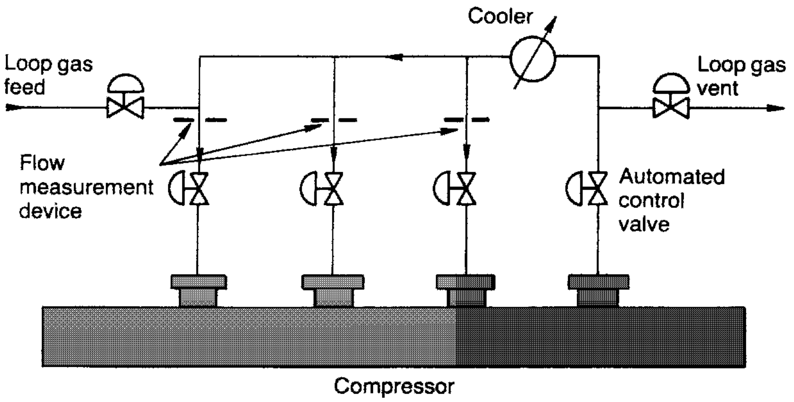

FIGURE 18.12 Typical test loop. (*Elliott Company, Jeannette, Pa.*)

compressor and tested accordingly with the code applied as nearly as possible. However, some general guidelines should be highlighted to help clarify testing procedures.

18.3.1 Test Setup

In actual service each sidestream typically originates at a distinctly different physical location, giving individual properties of state at each entry. To duplicate this condition in the vendor's shop would be extremely costly. In lieu of this, a simplified typical setup for a two-sideload compressor is shown in Fig. 18.12.

Variations to the setup exist, but the basic closed-loop concept is used throughout the industry, with control valves used to control pressures at the various inlets. Under normal conditions a bleed-feed system must be used to maintain loop test gas purity and pressure control. This system, or ones like it, have proven adequate over the past decades.

18.3.2 Instrumentation

Crucial to validity of a shop sideload performance test is the required instrumentation and its proper installation. The following general guidelines on instrumentation have been proven over the past several years and are recommended for general acceptance.

The ASME code provides guidelines for instrumenting the external flanges of the compressor and the flow-measuring sections. This instrumentation will provide adequate readings at the compressor flanges and flow-measuring devices but will not suffice to obtain information on sectional performance or overall unit horsepower consumption. Actual performance depends on the properties of state of gas exiting each section and thus is not directly available from flange readings alone. Internal instrumentation is required, or assumptions must be made on temperature rise across each section. As methods of instrumenting sideload compressors improve, this guide may change.

18.3.3 Testing Procedure

First, as nearly as possible, sidestream and main inlet flows should be held proportionately equal throughout the test and as close as possible to those anticipated in actual service.

Failure to follow this approach may result in conflicting data. Because of varying splits between the sidestream and main gas stream, it is possible to achieve two different levels of performance for the same sectional inlet flow.

Just as important as the testing method is stabilization of individual testing points. Before data are taken time is required at each testing point to permit all system components to become normalized because of heat transfer and other transients. Given constant inlet conditions, stabilization can be assumed when three consecutive readings of discharge pressures and temperatures taken at 3-minute intervals conform within 2%. A good practice to verify stabilization of the first point after each startup of the test is by repeating it after a minimum 5-minute interval. Subsequent readings can be taken when stabilization has been verified.

Because stabilization is a function of mass flow rate, equipment size, total heat capacity of the metal, curve shape, internal leakages, test loop configuration, and heat transfer, it cannot be assumed simply to be based on time. Properties of state of the flowing medium must be satisfied as constants.

18.3.4 Accuracy of Test Results

One question remains on testing: How accurate are the test results? Testing of the entire unit at one speed may not give absolute results because of slight compromises required by code guidelines on volume ratio, Mach number, and Reynolds number. Some believe that testing each section at a given speed or testing partially stacked rotors is a solution. Results of such testing, however, have not shown a sufficiently higher degree of accuracy to warrant the added expense and time.

18.3.5 Evaluation of Results

Individual sectional curves do not depict a reasonable evaluation of the units performance during selection. Similarly, test results cannot be evaluated as individual sections. The sectional curves must be converted to a turndown map before any rational conclusion about overall unit acceptability may be discussed. The turndown map must be used to evaluate results of the sideload test.

Testing a sideload compressor is indeed complicated and involved and requires more expertise than the typical equivalent performance test. Testing requires a total understanding between the witness and the vendor.

The application, design, testing, and analysis of a sideload compressor all differ substantially from a typical centrifugal compressor. Failure to seriously consider the uniqueness and complexity of unit and process interaction—all the way from initial process design through to the actual field installation—may result in an unfavorable installation.

In any event, the topic merits close consideration as is dealt with in Chapter 19.