EFFICIENCY: THE SILENT PARTNER OF MACHINE PERFORMANCE

by

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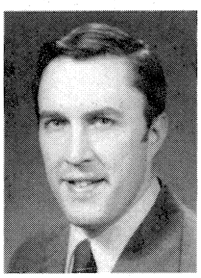

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Dr. Japikse's principal areas of work include the design and development of various centrifugal compressor and radial turbine stages including the development of design tools. His consulting and design work has covered a diverse range of process machinery, turbochargers, gas turbines and cryogenic expanders. He has carried out basic research on both turbomachinery components and systems. He has presented 30 major publications and prepared approximately 100 major reports covering the design, development and troubleshooting of turbomachinery.

ABSTRACT

Careful attention is given to the most common turbomachinery performance problems which result from an inadequate understanding of component and machine efficiency. Examples of several classical problems are presented and the basic definitions and principles involved are carefully reviewed. Common errors in measuring machine efficiency are discussed with examples and suggestions as appropriate. Although machine efficiency frequently plays a secondary role compared with questions of durability, it is pointed out that even small errors in rated efficiency can cause significant stability problems, power/speed mismatch, and even contribute to noise and vibration problems. Three different levels of design analyses are discussed at which new machines can be designed and their efficiency predicted. Typical advantages and disadvantages of each approach are presented. The information presented herein is an experience based summary to guide engineers in their review of machine performance.

INTRODUCTION

Machine efficiency is a popular topic of discussion whenever the cost of fuels and feedstocks is high, but enthusiasm quickly wanes whenever costs drop, even momentarily. As recently as five years ago, concern with the efficiency of industrial turbomachinery was comparatively slight outside of the hydraulic turbine field. In recent years, concern has become comparatively intense due to the high cost of fuel but even this recent concern is comparatively mild when contrasted with the intense concern felt for machine reliability and maintainability which has been foremost in most fields of industrial turbomachinery design and sales. One could realistically ask the rhetorical question: who cares about machine efficiency? Even with the recent upsurge in concern about fuel prices, machine efficiency is a weak partner when contrasted with maintainability and is quickly pushed aside when questions of mechanical performance arise. There are however, many reasons why machine efficiency must always be kept close to the front of an engineer's concern for machine performance. There is a direct relationship between the head rise (or pressure ratio) established by a machine and component efficiencies. Similarly, noise and vibration problems can often be traced to fluid-dynamic processes which reflect poor machine design from a gas dynamic or thermodynamic efficiency viewpoint. Thus the objective of this paper is to provide some of the foundation issues upon which a realistic appreciation for machine efficiency can be based.

PRACTICAL ISSUES OF MACHINE EFFICIENCY

The first efficiency issue is the power required or delivered for a given head or pressure ratio and a given flow rate through a compression or expansion system. If efficiency of one or more important sections of a machine is exceeded or missed in the design or development of a machine, then less or additional power will be required to operate the unit throughout its lifetime. If we consider the definition of efficiency on a component basis, then we recognize the power transferred is given according to the following equations:

$$\eta_{s,\,compressor} = \frac{W_{isentropic}}{W_{actual}} \; ; \; W_{isentropic} = C_p\, T_{oo}\, (pr^{k/(k-1)} - 1) \qquad (1)$$

$$\eta_{s,turbine} = \frac{W_{actual}}{W_{isentropic}} \; ; \; W_{isentropic} = C_p\, T_{oo}\, (1 - {}^{1}/er^{(k/(k-1))}) \qquad (2)$$

To be sure, most industrial machines are comprised of multistage machines for which we must consider the efficiencies of various stages as an aggregate. What we are particularly concerned with is predicting the *inefficiencies* for any given compression or expansion process. We know that we will not achieve the ideal (isentropic) level of work extraction from an expansion process nor will we achieve a desired pressure rise for a compression stage with just the isentropic work. The differences between the isentropic process and the actual process will amount from 10% to 25% of the ideal work level, implying efficiencies anywhere from 75% to 90%, depending on the type of expansion or compression process. We are, therefore, rightfully emphasizing our ability to predict the inefficiency (or loss) of a stage accurately to one or two points out of 10 or 25, which would seem to be a fairly straightforward process. However, for reasons presented in the next section, this frequently can be a very difficult process indeed.

The pressure ratio achieved or required by a given compression or expansion process is directly related to the efficiency of the stage. We can see this in the following equation which is obtained by manipulating equations 1 and 2 above:

$$pr = [1 + \eta\, W_{actual}/(C_p\, T_{oo})]^{k/(k-1)} = [1 + \eta\, (T_{o,exit} - T_{o,o})/T_{oo}]^{k/(k-1)} \tag{3}$$

$$er = [1 - W_{actual}/(\eta\, C_p\, T_{oo})]^{k/(k-1)} = [1 - (T_{o,o} - T_{o,exit})/\eta\, T_{oo})^{-k/(k-1)} \tag{4}$$

$$\Delta pr = k/(k\text{-}1)\, pr^{1/k}\, \frac{(T_{o,exit} - T_{o,o})}{T_{o,o}}\, \Delta\eta \tag{5}$$

$$\Delta er = -\, k/(k\text{-}1)\, er^{\left(\frac{1 + 2k}{k}\right)}\, \frac{(T_{o,o} - T_{o,exit})}{T_{oo}}\, \frac{\Delta\eta}{\eta^2} \tag{6}$$

If we differentiate these expressions as shown by equations 5 and 6, we can find that an increment of one point of stage efficiency will typically mean approximately ½ to 3% on pressure ratio for a typical compression or expansion process for a pr (or er) = 2 per stage, for example, using air as a working medium. Thus, the head rise or pressure ratio of a given stage is influenced by our ability to predict the efficiency, or the inefficiency, of a stage.

The preceding two principles might appear to be obvious applications of the most basic principles from a first course in turbomachinery; but, they gain significant importance when we begin to combine them for a multistage machine. Consider, for example, a compressor characteristic shown in Figures 1a and 1b. The conditions for stage 1 are the parameters which set the inlet conditions for stage 2. A multistage machine is made up of any number (frequently on the order of 6 or 7) stages which must be matched by careful consideration of the characteristics of each individual stage and how they feed from one stage to the next. Consider the conditions of Figure 1a as they feed Figure 1b and imagine that the performance of the stage

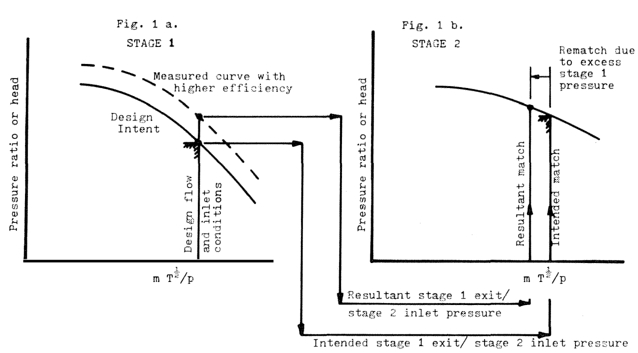


Figure 1. Stage Maps Showing Match Conditions at Design Point and When a Stage Delivers Excess Efficiency and/or Head.

in Figure 1a might actually have been three points higher than rated in the original design intent. This changes the match conditions for Figure 1b to the alternate match point as shown in the figure. In short, one has moved, for example, about 2% toward the surge line. This does not appear to be too great a change but it is magnified through an additional five or six subsequent stages. It results in the overall rematching of all stages to the left, thus implying a *potentially significant reduction in table operating range;* that is the range from the match point to the surge point (for definition of surge and range, see Japikse [1]). Thus, even a small change of a few percent of efficiency when taken in the first stage or two of a multi-stage machine, can lead to significant changes in the overall performance characteristics of the machine as a whole. One simply cannot take efficiency for granted. A little extra efficiency is nice for power savings, but the implications must be carefully considered.

An additional characteristic of machine performance is the trade-off between stage efficiency and stable operating range. An example of performance data for a wide range of centrifugal compressors, including both common process stages and sophisticated high performance gas turbine stages is shown in Figure 2 and Figure 3. Figure 3 illustrates the decay in design point efficiency as machines of greater and greater operating range are required.

In addition to requiring good stable operating range, the slope of a given operating line (on a head vs. flow coefficient

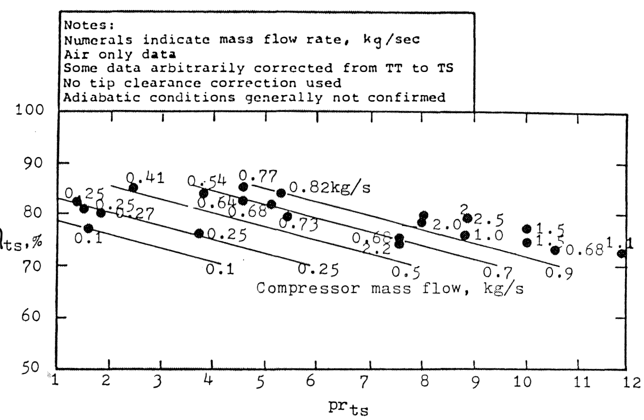


Figure 2. State-of-the-Art Single Stage Centrifugal Compressor Efficiencies on an Isentropic Basis.

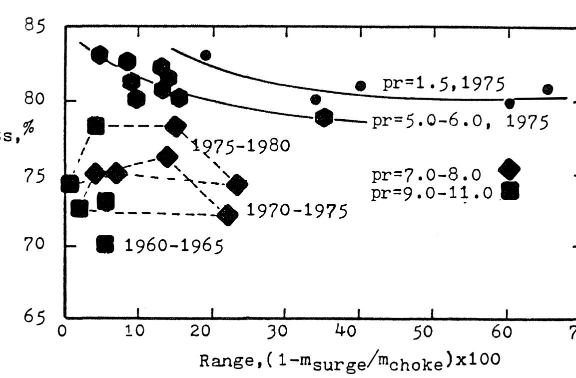


Figure 3. State-of-the-Art Single Stage Centrifugal Compressor Isentropic Efficiency versus Range.

characteristic plot) may frequently be specified to meet a predetermined system requirement. One can appreciate the design requirements for the slope characteristic by considering the following equation, which is deduced from the essential velocity triangles:

$$W_{act} = U_2\, C_{\Theta 2} = U_2\, C_{m2} \tan \beta_{2b} - U_2^2\, \sigma_2 \tag{7}$$

$$pr = [1 + \eta(U_2\, C_{m2} \tan \beta_{2b} - \sigma_2\, U_2^2)/C_p\, T_{oo}]^{k/(k-1)} \tag{8}$$

$$C_{m2} = m/\rho_2\, A_2 \text{ or } C_{m2} = \frac{MRT_2}{P_2 A_2} \tag{9}$$

We see that the slope of the actual performance characteristic is very much dependent upon our design parameters and gas properties.

Finally, gas properties and the correct modelling of gas properties is very important for obtaining realistic predictions of stage efficiency and related machine characteristics. Gas properties have long been a topic of concern for a number of important reasons. A principal reason is that many process machines must work with a fluid whose composition can change over a decade or two of machine operation as the feed stock changes in its basic formulation. In addition, there are seasonal changes which influence the feed stock characteristics. But even beyond this field related problem, significant problems can occur with gas property description. In some instances, thermodynamic data is not known with sufficient accuracy to permit precise design procedures. In the author's experience, one problem occurred where the basic uncertainty in fundamental thermodynamic data was equivalent to one stage in the overall machine design! In another experience, the author was forced to work with basic h-s data where the total entropy change through the compressor was equivalent to less than ten units of the least significant figure given in the property data. Problems of this type require very careful consideration by the designer so that the resulting machine is not conceived foolishly. In addition, the performance of various components of a machine will vary widely depending on the actual fluid used. For example, the slope of the head rise versus flow coefficient shown previously in equations 8 and 9 is very much dependent on the fluid dynamic properties. Further examples will be given in subsequent sections.

PREDICTING STAGE EFFICIENCY

It is instructive to consider three different levels of design analyses when the question of predicting machine performance is considered. The first level is a direct scaling of an existing stage to new applications; a second level of design analysis employs a correlated data base of various impellers and various additional machine elements. By contrast, a third level uses a detailed fundamental flow modelling in order to predict the level of component efficiency. Before considering these three levels in some detail, it is appropriate to point out that no one of these three levels is more valid or more correct than any other. The level to be used is dependent on many particular application considerations and the level selected should be the one which will yield the most realiable end results.

The first level of design analysis uses the fundamental principles of similitude which are based on Buckingham's Pi theory as presented in all basic turbomachinery textbooks (for example, the references by Shepherd [2] and Horlock [3]). From these we derive the following nondimensional parameters:

$$\pi_1 = \frac{Q}{ND^3} = \frac{m}{ND^3\rho} = \frac{m}{UD^2\rho} \tag{10}$$

$$\pi_2 = \frac{H}{N^2D^2} \tag{11}$$

$$\pi_3 = \frac{P}{\rho N^3 D} \tag{12}$$

$$\pi_4 = \frac{\mu}{\rho N D^2} \tag{13}$$

$$\pi_5 = \frac{a^2}{N^2D^2} = \frac{a^2}{U^2} \tag{14}$$

$$\pi/\sqrt{\pi_5} = \frac{m}{U\, D^2\rho}\frac{u}{a} = \frac{m}{a\, D^2\rho} = \frac{m}{D^2 p}\sqrt{\frac{RT}{kg}} \tag{15}$$

The basic dimensionless Pi parameters are shown as groups 1-5 above. From these all essential turbomachinery dimensionless parameters can be derived. The first grouping is a flow coefficient whereas the second grouping is a head coefficient. The third grouping provides a power coefficient whereas the fourth grouping is an inverse Reynolds number based on wheel speed and the final group is proportional to the inverse of Mach number squared. It is clear that one or more dimensions tie these groupings together and, if a proper relationship is chosen for a scale parameter, then one can carry a given design from one application to another, within the restraint of physical properties. One can accommodate some changes in properties, for example if one is changing from one perfect gas to another, but if significant changes in properties are encountered, then they can interact strongly with these coefficients, thus compounding the scaling process. For the simple case of a perfect gas, the traditional scaling route becomes immediately apparent. The last grouping which is formed by the ratio of Pi_1 divided by $(Pi_5)^{.5}$ gives us the familiar m $T^{0.5}/P$ relationship used extensively in gas turbine design and many other areas of industrial machinery. Clearly the molecular weight of the gas and the gas constant will influence the scaling choice if one moves from one type of gas to another. However, it is clear that one chooses the scale factor (ratio of wheel diameter from one stage to the wheel diameter of a new stage) according to the square root of the mass flow. Then, by considering the Pi_5 group it is clear that the speed will be determined in a linear relationship with the wheel diameter. Thus, two of the dimensionless parameters are frozen and then all other parameters follow, except for the Reynolds number. The Reynolds number cannot be scaled precisely from one case to another unless one can control the inlet density independently. For most industrial applications, this is not possible.

As long as these fundamental similarity parameters are preserved from application to application, and precise geometric scaling is used according to the resultant scale factor, then one will obtain an extremely high degree of precision in the resultant machine characteristics. The only corrections which must be made are for a second order effect which is not included in the basic groups given above, and that is due to the Reynolds number effect. Various Reynolds number corrections are recommended in the ASME PTC-10 [4] and the API [5] codes. An example of this type of data is shown in Figure 4 for a variety of centrifugal compressors.

In cases where we are scaling from very large to very small sizes, however, it may not be possible to maintain a precise geometric scaling. Then we may very well introduce some important deviations (from correct scaling) which could cause errors in predicting the machine performance. For example, blade thicknesses might not scale, fillet radii might not scale, angles may be modified slightly, and operating clearances may change noticeably. These effects could cause significant deviations which should be watched closely. However, the biggest problem would probably exist if the machine was scaled from one operating fluid to another. While this could be done with relative impunity for ideal gases, it could also be the source of considerable difficulty for real gases.

At the second level of design analysis, a manufacturer can consider the use of a wide variety of different components in a mix and match mode. This approach gives a manufacturer maximum flexibility in meeting design requirements while maintaining a very high degree of confidence in the resultant stage performance. The central element of such a stage is the rotor (see Figure 5). An example of correlations of rotor performance are shown in Figures 6 and 7 for the rotors

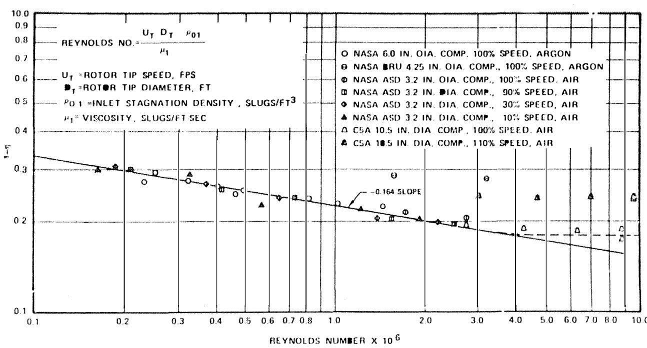


Figure 4. Compilation of Centrifugal Compressor Efficiency Decrement as a Function of Reynolds Number, from Pampreen (11).

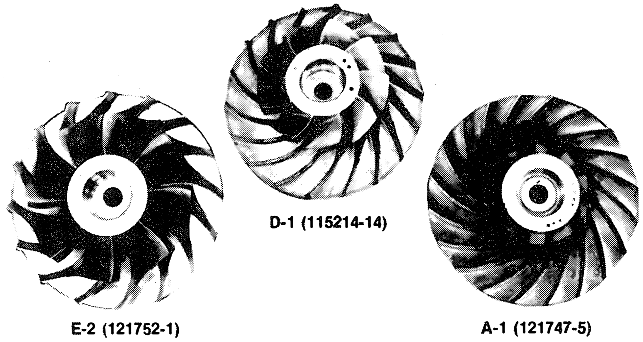


Figure 5. Solar Centrifugal Compressor Impellers Which Form a Family for Various Design Requirements. Rodgers (6)

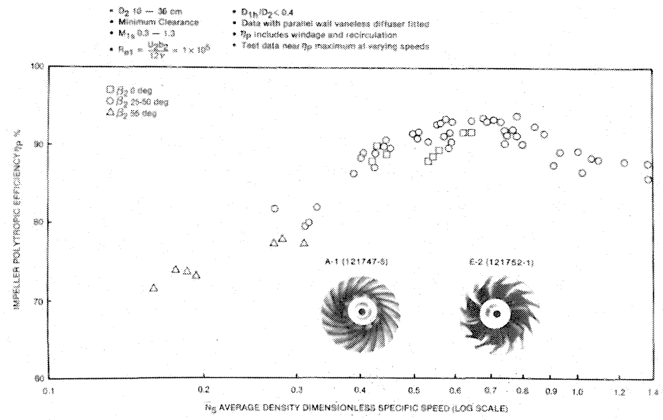
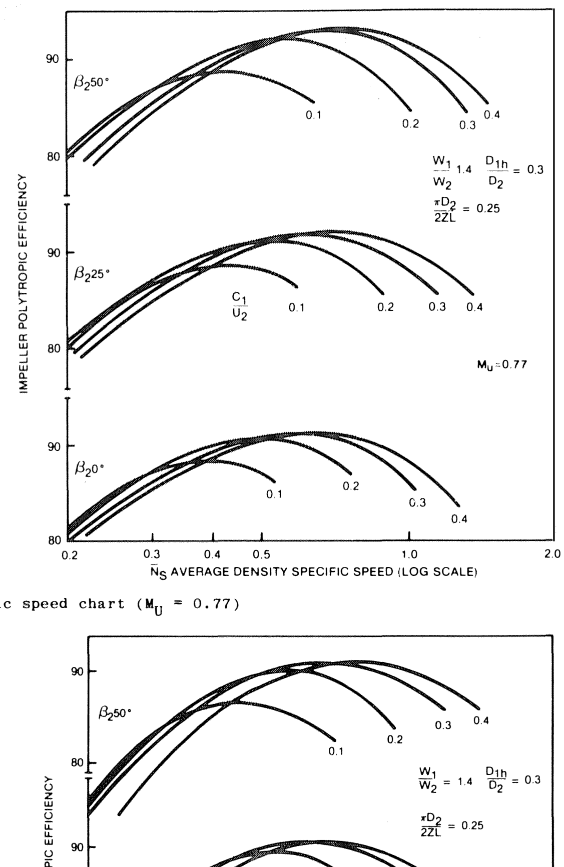
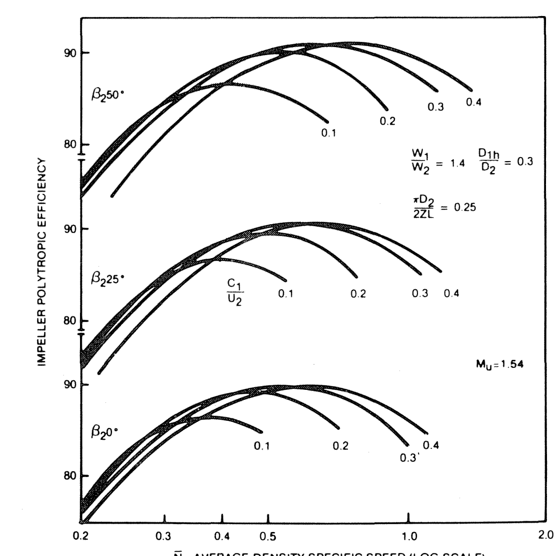


Figure 6. Solar Centrifugal Compressor Impeller Efficiencies. Rodgers (6)



Specific speed chart (M_U = 0.77)



Specific speed chart (M_U = 1.54)

Figure 7. Solar Centrifugal compressor Impeller Efficiencies as a Function of Important Design Variables. Rodgers (6)

displayed in Figure 5. This particular example is from Colin Rodgers [6] and displays a series of Solar compressor rotors. Around this rotor many different inlet configurations and diffuser configurations can be employed. Additional downstream elements, such as return channel cascades or volutes, can also be used. Each of these additional elements must similarly be mapped in terms of its basic performance characteristics or computed from first principles (level 3 approach). By combining these various elements, an overall stage characteristic can be prepared.

However, at this level the problem becomes quite a bit more complicated. The actual performance of the rotor will depend on the type of inlet flow which it receives. If the flow is clean (as in the case of Figures 5 - 7) then comparatively high performance will result. However, if the inlet flow is distorted, then the performance of the rotor must be downgraded significantly. Similarly, the performance of any element after the rotor, such as a diffuser and subsequent elements like a volute or return channel, will depend on the detailed structure of the flowfield entering those elements. For example, the type of velocity gradient entering a vaneless diffuser has been shown to have significant influence on the expected performance of the diffuser itself. An example is shown in Figure 8, which is taken from fundamental research by Professor Senoo in Japan; it shows that the minimum angle permitted into the vaneless diffuser depends significantly on the level of inlet velocity profile distortion. Angles lower than this level are susceptible to rotating stall. Many other examples of diffuser performance subject to inlet velocity profile variations can be presented. Most process machines are prepared by using a variety of different stages and stage elements which are either scaled by the Level I approach or remixed/matched according to a Level II approach. No manufacturer has yet reported sufficiently detailed information upon which this mix and match process can be carried out with complete precision. To do so would require comprehensive and sophisticated traversing of the inlet and outlet conditions of each element used in the stage matching process and a comprehensive data base of correlated data from which to choose for new designs. This has not been done and would be prohibitively expensive. Instead, such problems must be dealt with from a more general and overall

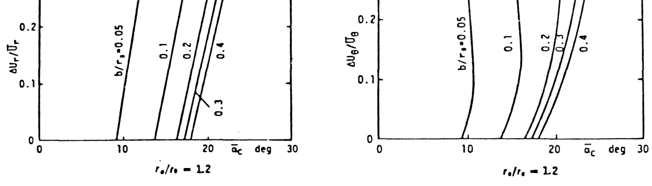


Figure 8. Example of Velocity Profile Effects on a Critical Diffuser Performance Parameter. $\Delta U_r/U_r$ and $\Delta U_\Theta/U_\Theta$ are Profile Distortions in Meridional and Tangential Velocity Entering a Vaneless Diffuser. α_c is the Critical Diffuser Inlet Flow Angle Below Which Rotating Stall and Possibly Surge Could Be Initiated. From Senoo (7).

experience base from the manufacturer's prior design and development history. The simple fact that the vast majority of process turbomachinery manufacturers achieve the desired specifications in most cases, is a strong credit to the skill and experience which they bring to bear on the design and development problems. To be sure, noteworthy exceptions exist where the desired performance has not been achieved. When this occurs, it is most often due to the unusual coupling of different elements which could not be sufficiently perceived before the machine was constructed.

At the third level of design analysis, one moves away from empirical data bases and attempts to predict the performance of a proposed stage by using basic, conceptual flow models of the different phenomena involved in the stage according to fundamental thermodynamic and fluid dynamic principles at hand. A certain amount of empirical information is needed for this process but it is empirical information which describes the essential flow processes and not correlations of component performance per se. In many cases it is possible to attack a design problem with approximately the same level of overall uncertainty with either a Level II or a Level III technique for a typical process stage. However, the Level II has traditionally been preferred since one can at least take comfort in using information which comes from components very similar to those which will be used in the final design. However, if a completely new operating fluid is to be used or if significant departures from previous geometric combinations are to be considered, then the comprehensive Level III modeling may be preferred. Details of Level III modeling are presented by Japikse [8].

OVERALL PERFORMANCE LEVELS

Much of the historical confusion surrounding a proper understanding of machine efficiency can be traced to the techniques for defining and computing efficiency, for testing machines to determine efficiency, and the methods used to report both computed and measured efficiency levels. Each of these issues will receive attention in this section.

The basic isentropic efficiency was defined in equations 1 and 2, which simply relate the actual work input or work extraction for a process to the ideal work transfer which could be obtained in an isentropic process. This efficiency is a very clear definition which can be used in many ambiguous ways. In order to have a meaningful comparison between the idealized isentropic process and any actual process of work exchange, it is necessary that we maintain an apples-to-apples comparison. The most common error which is introduced in turbomachinery testing is to measure the actual work input with a temperature change, without being careful as to whether or not heat may have been lost or added to the system through environmental heat transfer. Thus, in order for any efficiency values to be valid, it is necessary to thoroughly insulate the test facility when measuring actual work transfer according to temperature change. This includes all paths of possible heat transfer, not only through casings, but through adjacent oil baths, and so forth. Of course, if the power is being measured directly by means of shaft torque measurements adjacent to the turbomachinery rotors, then these problems can be avoided. However, such measurements are comparatively rare due to their complexity and expense. The adiabatic isentropic efficiency is perhaps the most commonly employed definition of efficiency for the turbomachinery field as a whole. However, for process compressors, it may be preferable in many cases to use a polytropic efficiency.

Figure 9a. shows the essential characteristic of the isentropic efficiency calculation for a compression process, whereas

Figure 9b. shows the equivalent parameters for an expansion process. We observe that if any given compression or expansion process is broken into a series of smaller processes, that the method of bookkeeping will become quite important for determining actual levels of the equivalent isentropic work transfer. Taking the compression process as an example, we can observe that the summation of several smaller, subsequent, compression processes has an isentropic work which is greater than the reference isentropic work taken from the initial entropy level. We can define an efficiency with either definition and observe that the overall isentropic efficiency is less than the stage efficiency which would be obtained by summing up the various segments of the compression process. The difference boils down to one important fact: as a compression or expansion process proceeds, the temperature of the gas changes. For a compressor this means that the inlet temperature to a subsequent compression process is higher, thus making the subsequent compression effort more difficult. By breaking the process into a sequence of individual steps, this change in temperature is correctly recognized and each subsequent step is rated realistically according to the conditions at the beginning of its process. This effect has often been called a preheat effect for compression or a reheat effect for an expansion process.

If an effort is made to consider a compression process as a sequence of infinitesimal isentropic processes, then we are led to the definition of polytropic efficiency. This result is obtained by integrating the following equation:

$$\eta_p = \frac{W_s}{W_{actual}} = \frac{pr^{k/(k-1)} - 1}{tr - 1} \qquad (16)$$

$$= \frac{\left(\frac{dp}{p} + 1\right)^{\frac{k-1}{k}} - 1}{\left(\frac{dT}{T} + 1\right) - 1} = \frac{T}{dT}\left[\left(\frac{dp}{p} + 1\right)^{\frac{k-1}{k}} - 1\right] \qquad (17)$$

so as to obtain the following relationship for polytropic efficiency (after expanding the power term with a series expression):

$$\int \frac{dT}{T} = \frac{1}{\eta_p}\,\frac{k-1}{k}\int \frac{dp}{p} \qquad (18)$$

$$\eta_p = \frac{k/(k-1)\ln(p_2/p_1)}{\ln(T_2^1/T_2)} \qquad (19)$$

To evaluate this expression, it is necessary to know the temperature T_2' which can be obtained from the conventional definition of isentropic efficiency (equation 1) thus giving:

$$T_2^1 - T_1 = \frac{T_1}{\eta_s}\left[\left(\frac{p_2}{p_1}\right)^{k/(k-1)} - 1\right] \qquad (20)$$

Thus we have seen that there are several ways of rating efficiency which will give, in fact, different numerical values depending on the choice which has been made. The polytropic efficiency is perhaps the most commonly used in the process field and it is clear that one must always understand when an isentropic or polytropic efficiency definition is being employed. This distinction can be emphasized by replotting the data shown previously in Figure 2 (now as Figure 10) in terms of polytropic efficiency. It may be observed that some (about half) of the efficiency decay with increasing pressure ratio (for a given flow) has been eliminated, which is a consequence of the basic idea of the polytropic efficiency definition.

A second important issue in understanding machine efficiency is the requirement of proper testing. An aspect of this problem was indicated previously: if repeatable tests are to be performed on any turbomachinery where the work transfer is determined by a temperature change, then the rig must be thoroughly insulated. This single problem is perhaps the most prevalent source of error in turbomachinery test data and, hence, in reported efficiencies. In some cases the amount of heat rejected into the atmosphere is comparatively negligible; in other cases, it can influence the reported efficiency by 5, 10, or 15 points of stage efficiency. The problem is circumvented by thoroughly insulating the test rig when access can be obtained. However, it is very common to find situations where access is not possible. In this case, it is necessary to repeat tests with various environmental factors in order to permit a correction to be made to the reported data. This approach has worked well in a number of instances, but does require increased test commitments. However, this is only one of a number of serious problems which can occur in turbomachinery testing.

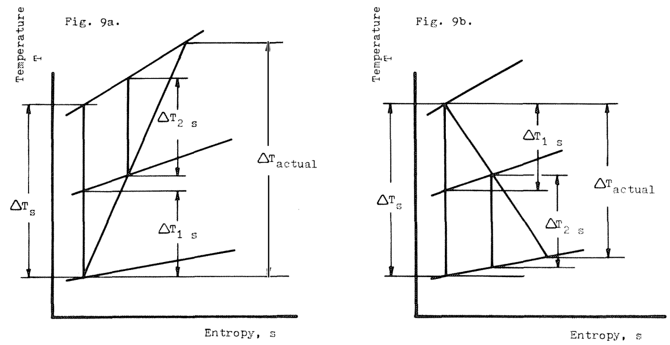


Figure 9. Examples of Overall Isentropic Efficiency and a Cycle or Process Efficiency for a Compressor (9a) and a Turbine (9b). If the Process is Broken into a Series of Infinitesimal Steps, a Polytropic Process is Described.

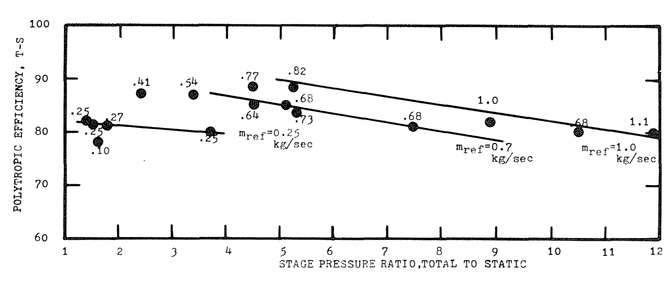


Figure 10. Recomputation of the Figure 1 Isentropic Data in Terms of Polytropic Efficiency. Note that the Slope of the m=Const. Curves has Decreased by a Factor of Two. The Remaining Slope Is Due to Scale Effects.

Additional problems include the following:

1. Instrumentation calibration (including torque-meters, thermocouples and pressure transducers);
2. Correction for recovery factor effects, stem conduction effects, and radiation effects on thermocouples;
3. The use of correct thermodynamic property data.

Examples of several of these effects are given with more complete information available in detailed lecture notes (Japikse, [9]).

Recovery factor is one of the most often omitted corrections from thermocouple measurements. Even shielded thermocouples require a recovery factor correction for flows with Mach numbers over approximately 0.3. For example, if a temperature of 600°R is measured (140°F) in a stream of Mach 0.3, then a typical recovery factor correction of 0.6° is required. This correction may or may not be significant, but in many tests it is important. If there is a temperature change of 100° through the stage, then we are dealing with 0.6 points of stage efficiency as an uncertainty. If the Mach number was 0.6, then the error would be about 1.2 points. Unshielded thermocouples have much larger errors. Indeed, many compressor and turbine tests require measurement of inlet and outlet temperatures in this Mach number range. In addition to this often forgotten correction, it is necessary to allow for stem conduction and radiation in certain installations. Preferably, one chooses the thermocouple installation so as to minimize or eliminate such correction factors.

Basic calibration of equipment is essential, but not always carried out accurately. Pressure transducers must be properly calibrated so as to have a traceable reference with an acceptable degree of precision. In actual usage, instruments can drift and lose calibration and test procedures must allow for these variations. Usually pressure errors are less significant in determining stage efficiency than temperature measurement errors, but they still can be important. Finding a realistic inlet and outlet pressure can be a source of trouble quite apart from basic calibration errors. If an inappropriate measurement location is selected, it is possible to influence reported efficiencies by one or two points of stage efficiency.

Finally, the issue of appropriate thermodynamic property data is important. The best way to compute an isentropic efficiency is to use gas table data directly and work with enthalpies and entropy. This gives a precise calculation which is traceable to the basic thermodynamic data. However, if a semiperfect gas law relationship is employed, it is possible to obtain different levels of efficiency for exactly the same process. Errors of one or two points of stage efficiency can occur. Thus if a semiperfect gas relationship is employed, then it is important to state this preference and to record the level of gas constant employed. Similarly, when a polytropic process is used, the computational procedure and the coefficients employed must be recorded.

Finally, the issue of reporting efficiency data is important. The preceding sections should make it clear that experimentally determined efficiency information requires an accurate statement of the computational method employed, the instrumentation used for measuring each of the essential parameters, specifications for each instrument employed in obtaining the measured data, uncertainty statements for the different parameters, source of thermodynamic information employed, and finally, an uncertainty calculation. The uncertainty calculation is frequenty based on a root mean square statistical approach as set forth by Kline and McClintock [10]. In this approach, one must itemize the uncertainty of each measured parameter and introduce this into the overall calculation. An example is shown in Table 1; the specific uncertainties are introduced into the basic equation as follows:

$$\omega_R = \sqrt{\sum \left(\frac{\partial R}{\partial v_i}\omega_i\right)^2} \tag{21}$$

ω_R = uncertainty of result, R

ω_i = uncertainty due to each variable, v_i

We see that this calculation then implies an uncertainty of approximately 0.45 points of efficiency for a 1 sigma uncertainty or 0.7 points of efficiency for 2 sigma uncertainty band for the case shown in Table 1. Although most laboratory scientific work is carried out with a 2 sigma uncertainty band, (20:1 odds), it is the author's experience that most industrial work is not carried out at this level. Instead, it is the *opinion* of this author that most work tends to be carried out with approximately a 1 sigma uncertainty band.

TABLE 1. SAMPLE VALUES FOR A RADIAL TURBINE UNCERTAINTY CALCULATION.

Parameter Name	Value	ω (Parameter) 20:1 odds	ω (Parameter) 7:1 odds
P_{oo} (psia)	36.81	0.08	0.05
T_{oo} (°R)	766.20	0.50	0.30
P_{out} (psia)	17.83	0.08	0.05
T_{out} (°R)	546.00	0.70	0.50
Patm (psia)	14.36	0.01	0.005
Results:			
ETA		0.007	0.0045

CLOSURE

Attention has been given to both design calculation methods for stage efficiency and experimental procedures as employed widely in the turbomachinery industry. Three different levels of design analysis are possible and are used in various applications depending on the problem at hand. No one level of design analysis is more appropriate than the other in the general sense but the level of design analysis employed should be chosen based on the requirements of the given design problem. Frequently, the scaling of an existing stage to a new application is the most reliable approach possible. Regardless of the level of design analysis selected, it is possible to introduce significant errors in the prediction of a machine efficiency. These errors can be minimized by careful attention to the information employed and the thermodynamic relationships used for the efficiency calculations. Similar care must be taken when an experimental measurement is made of stage efficiency. Efficiency measurements have numerous opportunities for significant error. The most frequent sources of error are heat transfer to or from the test facility and the overlooking of significant temperature correction factors. However, other errors can be made as detailed in this survey. An investigator can focus these experimental problems quickly if a comprehensive uncertainty analysis is made of the reported performance data.

NOMENCLATURE

A area

a speed of sound
b passage width
C absolute velocity (relative to a Newtonian frame, e.g. compressor casing)
c_p specific heat at constant pressure
D diameter
er turbine expansion ratio, p_{oo}/p_{exit}
h static enthalpy/unit mass; also, annulus height
h_o stagnation enthalpy/unit mass
k ratio of specific heats
M Mach number
m mass flow rate
N shaft speed
N_s specific speed: $N_s = NQ_o^{1/2}/(\Delta h_o)^{3/4}$,

$N_{ss} = NQ_o^{1/2}/(\Delta h_{os})^{3/4}$

where N = rotational speed in rpm

Q_o = inlet flow = m/ρ_{oo} in ft^3/sec, (Note: for turbines use exit Q)

Δh_o = stage enthalpy change in $ft\text{-}lb_f/lb_m$

Δh_{os} = stage isentropic enthalpy change in $ft\text{-}lb_f/lb_m$

P power
p static pressure
p_o stagnation pressure
pr pressure ratio: $pr = p/p_{oo}$
Q volumetric flow rate
R gas constant
r radius
^oR degrees Rankine
s entropy/unit mass
T static temperature
T_o stagnation temperature
U impeller (metal) velocity
V_s slip velocity: $V_s = C_{m2} \tan \beta_{b2} + W_{\theta 2}$
W_x total shaft work per unit mass of fluid
α absolute flow angle
η efficiency $\eta_c = \dfrac{h_{os} - h_{oo}}{W_x}$, $\eta_t = \dfrac{W_x}{h_{os} - h_{oo}}$

(measuring stations must be specifically defined)

μ dynamic viscosity
π dimensionless groupings
ρ density
ρ_o stagnation density
σ slip factor: $\sigma = 1 - V_s/u_2$
ϕ flow coefficient:
ω uncertainty in a specified parameter

Subscripts

b blade property
c compressor
m meridional
o stagnation, also inlet station
p polytropic
ref reference state or station (must be specifically defined)
s indicates that process follows an isentropic path
t turbine
ts total-to-static
tt total-to-total
θ tangential
2 impeller tip

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