Development and Performance of the Wedge-Type Low Specific Speed Compressor Wheel

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INTRODUCTION

The typical low flow, low specific speed compressor impellers are found as last stages of multistage compressors. This occurs with low inlet flows and/or high compression ratio units. The low flow, low specific speed compressor stages have low efficiency levels due to extremely high friction losses, leakages, and disc friction losses. Similar low specific speed large compressor impellers have some advantage in performance level due to higher Reynolds number, while low flow high speed compressor stages, as they come in turbocharger or expander applications, are as a rule moderate or high specific speed stages and maintain relatively good efficiencies.

The demand for low flow compressor stages is rising. Mechanical considerations such as rotor stability, etc., currently limit the operating speeds of such compressors and thus prohibits the usage of very small compressor wheel diameters. The result is that very low specific speed impellers are needed to handle the ever declining volumetric flows in the last stages of the compressors.

In the past, to alleviate some of these problems, the design of the conventional low flow compressor wheels was compromised in such a manner as to keep the width of the impeller passages as large as possible by lowering the blade angles throughout the impeller passage. As a result, a lower head was generated yielding a higher specific speed impeller with better efficiency.

A drastic departure from conventional design of a low specific speed impeller was undertaken in the program to attain acceptable efficiency levels. This objective was accomplished through substitution of the high aspect ratio cross sections of the flow passages in low specific speed impellers with aerodynamically more favorable low aspect ratio
crossections of these passages by means of employing wedge-type impeller blades.

PERFORMANCE LEVELS OF THE LOW SPECIFIC SPEED COMPRESSOR STAGES

It is well known that low specific speed compressor stages have inherently low efficiency. In this paper, the term "specific speed" will be used concurrently with flow coefficient to describe the flow characteristics of the compressor. The performance parameters used are:

\[ N_s = \frac{N}{\sqrt{Q_1 / (g H)^{3/4}}} \]  \hspace{1cm} (1)

\[ \phi = \frac{Q_1}{U_2 \frac{\pi}{4} D_2^2} \]  \hspace{1cm} (2)

\[ \psi = \frac{g H}{U_2^2} \]  \hspace{1cm} (3)

To be able to describe fully the performance of a compressor impeller we need to know the efficiency, the head coefficient, and either the specific speed or the flow coefficient. While the specific speed relates the flow capability and generated head, the flow coefficient describes solely the flow capability. Sometimes, the ratio \( b_2/D_2 \) is used in literature in place of the specific speed or the flow coefficient to show the dependency of efficiency on impeller geometry [1]. These three parameters serve a similar purpose in representing performance and design characteristics of impellers at design conditions.

Figure 1, taken from Reference [2] shows in a solid line typical variation of the compressor stage efficiency with the flow coefficient for radial compressor stages. The broken lines were added to show the approximate range of efficiency variation due to such factors as Reynolds number, diffuser ratio, diffuser type, and quality of design and workmanship. It is understood that the design or type of the compressor impeller changes as the specific speed changes. Impellers on the left end of Figure 1 are very narrow radial type impellers, they change gradually into mixed-flow impellers and finally into axial-type impellers at higher flow coefficients.

In order to determine effective guidelines for the performance improvement of a low flow compressor impeller it is important to identify the various losses in the compressor stage and to establish their order of magnitude. Using Eckert's convention [3] the internal efficiency of a compressor stage can be expressed as a ratio of the useful work to total work input as follows:

\[ \eta_i = \frac{\eta_{ad}}{\eta_{ad} + \Delta \eta} \left( \frac{m}{m + \Delta m} \right) \left( 1 + \frac{\Delta W\phi}{W_i} \right)^{-1} \]  \hspace{1cm} (4)

or

\[ \eta_i = \eta_{ad} \cdot \eta_{vol} \cdot \eta_{DF} \]  \hspace{1cm} (5)

where

\[ \eta_{ad} = \frac{H_{ad}}{H_{ad} + \Delta H} \]  \hspace{1cm} (6a)

\[ \eta_{vol} = \frac{m}{m + \Delta m} \]  \hspace{1cm} (6b)

\[ \eta_{DF} = \frac{1}{1 + \frac{\Delta W\phi}{W_i}} \]  \hspace{1cm} (6c)

It is not the intention of this paper to present a rigorous loss analysis of centrifugal compressor stages and to predict their performance level. The purpose is rather to emphasize that the effect on the efficiency of some losses, like leakage and disc friction, is almost inversely proportional to the mass flow. Another objective of the paper is to look closer at those losses which depend on design and to show where the potential for their reduction is the highest. Still possibilities of reducing the leakage losses and disc friction losses will be discussed.

In the following, individual losses which determine the compressor stage efficiency parameters used in Equation (4) will be discussed in detail. The recommendations for reduction of the internal friction loss in the flow passages of a compressor will be given. A sample calculation showing the effect of losses as a function of the flow size or specific speed on performance of a compressor stage will be presented.
The Friction Losses

The friction losses in the impeller flow passages should be considered first. The classical equation for the friction loss in flow passages

\[ \Delta H_f = \frac{L W^2}{f D_h \eta g} \]  

(7)

where \( D_h = \frac{4A}{p} \)  

(8)

seems to be most appropriate for this study. The expression does not contain the elements pertaining to flow turning, separation, and rotation of the flow passages, but it contains the hydraulic diameter of the passage and other important design and performance parameters. The equation points to several features which, in conventional low specific speed narrow compressor wheels, are the sources of high losses. Each term in the expression may contribute to excessive losses in the impeller channels. The friction coefficient, which is a function of relative roughness, will be high in such wheels since even an exceptionally fine finish will have a higher relative roughness in a narrow passage. The impeller flow passages with low blade angles are relatively long, and the hydraulic diameters of such passages are small.

The areas of flow passages in conventionally designed impellers are reduced as the flow decreases, essentially, by reducing their width. This results in reduced hydraulic diameter of the passages.

One can deduct easily from the above that reduction in friction losses can be attained through increase in the hydraulic diameter of the impeller passages by making them wider but shorter. This can be accomplished by employment of wedge-type impeller blades. It has to be emphasized that the advantages of switching from a conventional blade passage to a passage formed by wedge-type blades becomes more and more attractive the lower the specific speed and the narrower the conventional blade passage. The graph in Fig. 2 shows the variation of the ratio of hydraulic diameter of a flow passage to that of a square having the same area as a function of the aspect ratio where the aspect ratio is defined as the ratio of the blade pitch to blade width. The passages with the same flow areas may be formed as rectangles of different aspect ratios with squares having the largest hydraulic diameters. The coordinate on the right of the chart shows a potential increase in the hydraulic diameter which can be obtained by converting a given aspect ratio rectangular passage into an equivalent area square passage. Since, according to Equation (7), friction losses are inversely proportional to the hydraulic diameter of the passage, considerable gains in performance are possible for the low specific speed impellers with narrow passages while they would be less significant for high specific speed impellers with wide passages. As a matter of fact the wedge-type blades in higher flow coefficient compressor stages most probably may be detrimental to the performance of the stage. Such a case was documented in Ref. [4] where a relatively wide impeller, \( b/D_2 = 0.0555 \), was equipped with wedge-type blades.

This reasoning brought about the idea of departing from the conventional design that impeller flow passages and introducing a shorter but wider passage with less flow diffusion along the flow path. The hydraulic diameter of these flow passages is low but it is substantially higher than that of conventional flow passages. Thus the idea of the wedge-type impeller blade containing an aerodynamically favorable flow passage came into existence.

Disc Friction Loss

Next the disc friction losses were analyzed with respect to their effect in low flow applications. The disc friction loss is usually expressed as in Ref. [3]:

\[ H_{DF} = K \frac{u^2}{2g} \frac{\pi}{4} \frac{D_2^2}{Q} u_2 \]  

(9)

There is not much that can be done to reduce the effect on efficiency of disc friction losses in a stage with impeller of certain flow and certain diameter except producing more pressure rise or head in that stage. Referring to Equation (6c), a higher head would increase the work input \( Wi \) and thus increase the disc friction performance coefficient. It is felt that higher head can be designed into the wedge type wheels without a substantial sacrifice in the stage efficiency. The higher head can be achieved by using a steeper blade angle. The reaction of the impeller should decrease and disc friction loss should remain about the same despite a higher total pressure rise. It was mentioned previously that such a step in a conventionally designed impeller could have been counterproductive in respect to overall efficiency. The experience has shown that it is difficult to maintain reasonable levels of efficiency of the low flow compressor stages the exit blade angles were customarily kept small and the impeller widths relatively large: Steeper blade angles in conventional low flow impellers would lead to narrow passages and high absolute velocity. The combined effect of the latter two conditions in the impeller and diffuser would be a reduced stage efficiency. This is not a case in a wedge-type impeller, where the meridional velocity, passage width, and flow angle are not related to the impeller exit width.
Leakage Loss

The volumetric efficiency enters into general stage efficiency

\[ \eta_{vol} = \frac{m}{m + \Delta m} \]  

expression as another factor, and therefore its significance increases as the mass flow through the compressor stage decreases since the leakage \( \Delta m \) is essentially constant. It becomes very important in low specific speed compressor stages where the flow is small. In this case the improvement in volumetric efficiency is possible through application of improved design seals. Seals are available on the market which have practically a zero clearance. In the case of low specific speed compressor stages it is advantageous to use such seals despite the higher cost and complexity of design.

Diffuser Losses

So far we have seen the greatest potential for improvement of the efficiency of low specific speed compressor impellers in providing impellers with aerodynamically favorable flow passages. Such passages were formed by reducing the pitch between two blades and increasing the blade width of the impeller. Implicit in this change is also a change in the width of vaneless diffuser, crossover, and return vanes since their widths are usually related to impeller exit width. Thus, with performance improvement in the impeller, also a corresponding improvement in stationary passages was anticipated.

Now the question may be asked how the diffuser may react to a nonuniform flow exiting the wedge-type impeller. Would this condition cause additional losses in the diffuser entrance and in the diffuser proper compared with the flow exiting standard impeller?

Before this question should be answered one should realize the fact that flow exiting a standard impeller is not uniform. Dean and Senoo, Ref. [5], present a very informative treatise on jet-wake structure of the flow at the exit of a low specific speed impeller. The wakes may occupy a major portion of the flow passage at the exit, and they may experience a backflow. Lohmann [4] studied the flow structure with hot-wire anemometer of three wheels, one of them (Impeller III) was a wedge-type impeller. The internal flow structure between two blades is more uniform in this impeller than that in a standard impeller with flat exit angle (Impeller I). The efficiency of the compressor stage III is slightly lower than that of Stage I. The author suggests that poor diffuser performance due to large blade wakes was responsible for low performance of the stage. It should be observed that subject stage had a relatively high flow coefficient and that performance was only slightly lower than that of Impeller I. On the other hand, recently, Flynn and Weber, Ref. [6], achieved excellent performance in a low flow, moderate flow coefficient compressor stage, with a mixed-flow type open compressor impeller having the features of the wedge-type blades. According to Dean and Senoo, the total pressure loss in vaneless diffuser varies widely with specific design and operating conditions. They recognize that distortion of the flow decreases quickly at diffuser inlet and that major portion of total pressure loss occurs in that region.

It seems that as development of wedge-type compressor stages is concerned, the flow exiting from the impeller should not be considered as a completely new phenomenon, but rather as an extension and modification of the jet-wake pattern downstream of more conventional compressor impellers. As such, it is not expected to change the diffuser behavior drastically. A closer look may be warranted at the diffuser entrance area with the view of improving its performance.

Sample Calculation

The objective of this exercise is to evaluate the relationship for the internal efficiency of two compressor stages as they are affected by the mass flow and the geometry of the impeller flow passages. The first compressor stage is a medium specific speed or medium flow coefficient stage, and it carries a relatively large flow \( m \). It will be designated with index 1. The second stage, designated with index 2, carries a small flow \( 0.2m \). This stage is consequently a low flow coefficient or low specific speed stage. If the first stage had a flow coefficient of 0.05, the second stage would have a flow coefficient of 0.01. In this simplified comparison model the small flow compressor impeller is derived from the large flow impeller by reducing the meridional width of the blade passages by a factor of five. This is the only major difference in the design of the two compressor stages.

Using Equation [4] the following relationship can be written:

\[ \frac{\eta_1}{\eta_2} = \frac{\eta_{ad,1}}{\eta_{ad,2}} \frac{0.2m}{m} \frac{1}{1 + \frac{WDF}{0.2 W1}} \]

With the following values for the first, medium specific speed compressor stage

\[ \frac{\Delta m}{m} = 0.02, \quad \frac{W_{DF}}{W_1} = 0.03842, \quad \eta_{ad,1} = 0.872 \]

we have

\[ \eta_{1I1} = 0.823, \quad \eta_{vol,1} = 0.98, \quad \text{and} \quad \eta_{DF1} = 0.963 \]

The leakage and disk friction losses are assumed to be the same in both compressors. As a result,

\[ \eta_{vol,2} = 0.909, \quad \eta_{DF2} = 0.839 \]

and

\[ \frac{\eta_{12}}{\eta_{11}} = \frac{1}{1.238} \frac{\eta_{ad,2}}{\eta_{ad,1}} \]

If the adiabatic efficiencies of the two compressor stages were the same, the internal efficiency of the small flow compressor would be

\[ \eta_{12} = 0.823/1.238 = 0.665. \]

It was shown, however, that internal friction losses depend very much upon
the hydraulic diameter of the compressor stage flow passages. As equation (7) indicates, friction losses are inversely proportional to hydraulic diameter of the flow passages. The hydraulic diameter of an impeller flow passage was evaluated for an impeller having 0.050 flow coefficient and also for the same impeller with the blades cut to one fifth of their original width. The reduction in diameter of an impeller flow passage was evaluated also for the same impeller with the blades cut to one fifth of their original width. The sum of the losses $\sum \Delta H$ was evaluated from the value of the adiabatic efficiency

$$\eta_{ad 1} = \frac{H_1 - 0.823}{1 + 1.3868} = 0.872$$

As a result, the ratio of the adiabatic efficiencies in expression (11) may be rearranged as follows:

$$\frac{\eta_{ad 2}}{\eta_{ad 1}} = \frac{H_1 + 0.823}{H_2 + 0.823} = \frac{H_1}{H_2}$$

because the theoretical input head $H_1 + 0.823 = H_2 + 0.823$.

Finally the internal efficiency of the small flow compressor stage will be

$$\eta_{ad 1} = \frac{0.823}{1.3868} = 0.479.$$  

The obtained values of the internal efficiencies are close to those shown in Fig. 1 for the corresponding values of the flow coefficients. Obviously in an actual case the two compressor stages of the comparison sample would differ in more respects than only in the widths of their flow passages. Also the assumptions of constant leakage and disc friction losses were only approximations. The method of determination of losses in the flow passages is more complex than that based on the hydraulic diameter of the passages only. As a result the internal efficiency obtained is only an approximation. Nevertheless the sample illustrates well the trend of the efficiency level with declining flow coefficient.

**Development Work**

The study of the performance problems of the low specific speed compressor stages started in the late 1950's when a new line of the standard compressor stages was routinely tested. At that time the lowest specific speed stage of the standard family, Impeller A-8, Fig. 3, a 0.3556m diameter, high shaft to O.D. ratio, impeller having a specific speed of 0.343, was found to have an adiabatic efficiency of 54.1 percent, see Fig. 4. There were no references in the literature to check performance of these low specific speed small diameter impellers. Lator a comparison with Balje's specific speed - specific diameter chart with efficiency as a parameter [7] indicated that the efficiency of the 0.3556m diameter impeller stage was in agreement with the chart.

Attempts were made to improve the performance of the 0.3556m diameter compressor stage through modifications of the stationary passages: the increased diffuser ratio, vaned diffuser, vaned crossover, and use of a collector in place of vaneless diffuser. Efficiency increase of up to 5 points were realized as a result of various modifications. Some of these modifications were too complex to incorporate in the design in view of moderate improvement in efficiency. The loss analysis favored strongly the design of a wedge-type impeller. The potential for improvement seemed to be greater with that design.

As a first step to verify by simple means the concept of wedge-type compressor wheel, an existing medium-low specific speed wheel, Impeller A-5, was modified in such a manner that 3 out of its 15 passages were blocked by welding in plates at the blade entrance. The exit of those passages was left open. The idea behind this test was to ascertain to what degree the stage performance may deteriorate. The method of restricting the flow area was considered to be very crude since blunt sections between vanes were introduced at the wheel inlet and open spaces were left at the exit of the blades. It was assumed that performance of the stage will be penalized more than if a more careful design was used to design a wedge-type compressor impeller. The impeller was tested in 1961, and the test results of this experiment are shown in Fig. 4, under Impeller A-5-3, where performance of the original A-5 impeller is also shown.

The experiment confirmed the expectations that deterioration in performance was not greater than that which could be expected due to reduction in specific speed in a conventionally designed impeller. The results were also a clear indication that in a really low specific speed range, a well designed wedge-type impeller would show a gain in efficiency compared to standard wheels designed for the same conditions.

As the next step a wedge-type impeller was designed in 1963. The impeller has 13 backward leaning vanes with slight diffusion in the channel. Fig. 5 shows two views of the impeller, referred as A-W Impeller. The impeller was tested in 1965 in a
stage configuration with a vaneless diffuser and standard type return vanes. Fig. 4 shows the test results of the wedge-type impeller along with performance of other reference compressor stages for comparison reasons. The performance of the wedge-type impeller is indicated by a broken line. The specific speed of the impeller was 0.290 and its flow coefficient 0.00617. It was closest in terms of performance to impeller A-8 with specific speed of 0.334 and flow coefficient of 0.00817. The efficiency of the stage with the wedge-type impeller was about 9 percentage points higher than that of the A-8 stage with conventional impeller.

In 1974 a U.S. Patent [8] was obtained for the design of the Low Specific Speed Compressor, envisioning a variety of configurations as follows: wedge-type vanes: backwardly bent, straight, or S-shaped channel forms; with constant or slightly changing cross section areas; rectangular, square, circular, or oval channel cross sections, and a number of ways of streamlining the end portions of wedge-type vanes. As the development work continued, straight flow passage and S-shaped flow passage wedge-type impellers were designed and tested. The change in shape of the channel was introduced in order to steepen the flow angle and to increase the generated head. The design was successful and it was extended down to a flow coefficient of 0.003.

Significant improvement in performance level of the wedge-type impellers was achieved by modifications of the flow channel shape, treatment (rounding off) of the pressure side of the blade trailing edges, and application of tight seals. Interesting tests were conducted on a wedge-type impeller with a straight flow channel; Fig. 6 shows such an impeller without sideplate. Originally both blade walls intersected the impeller circumference as straight lines. Rounding off the pressure side of the trailing edge of a 0.406 m diameter impeller with a radius of 12.7 mm reduced the head only slightly over one percent.

Further rounding off of the pressure side of the trailing edge with a radius of 44.5 mm increased the head by about 3.5 percent from the original level. The efficiency of the stage improved as a result of each modification. A variety of seals was tested with low specific speed impellers, and the pressure balanced teflon seal, similar to that of Ref. [9], was proven to have the best performance. By utilizing the above measures efficiencies of over 70.0 percent were achieved in the low specific speed stages with impellers of 0.347 m diameter having flow coefficients of 0.0078 and above.

Comments on Design and Future Development

It is appropriate to briefly discuss some of the design aspects of wedge-type impellers. It is important in the design of a centrifugal compressor stage to accurately predict the flow and head of the stage. In wedge-type impellers, the flow is determined by
the inlet geometry, as in conventional type impellers. Thus standard design rules should be used. An attempt was made to correlate the head generated in a wedge-type impeller by using one of the standard expressions for the slip factor. Stodola's expression from Ref. [10] was used:

\[
\sigma = \frac{C_u u_2}{u_2} + \frac{C_m}{u_2 \tan \beta} = 1 - \frac{\pi \sin \beta}{Z}
\]  

(12)

where

- \( \sigma \) - slip factor
- \( C_u \) - tangential velocity, m/s
- \( C_m \) - meridional velocity, m/s
- \( \beta \) - blade angle, °
- \( Z \) - number of blades

Following are the values of the slip factor as calculated and as determined from the test:

<table>
<thead>
<tr>
<th>Impeller</th>
<th>Calculation</th>
<th>Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impeller A-W</td>
<td>0.854</td>
<td>0.811</td>
</tr>
<tr>
<td>Impeller A-WN</td>
<td>0.78</td>
<td>0.76</td>
</tr>
</tbody>
</table>

It is felt that the above test values of slip are in good agreement with calculated values despite the fact that such phenomena as disc friction loss, leakage loss, and heat transfer (all of which affect the magnitude of slip, Ref. [10]) are of considerable importance in low specific speed stages.

As previously stated, the two important features which characterize the design of low specific speed compressor impeller are its low deceleration rate in the impeller passages and favorable hydraulic cross sections of these passages. It is realized that this very complex subject requires more fundamental research in both theoretical and experimental areas.

At the present, the development of wedge type impellers continues in two directions: increasing the head and efficiency of the stage, and application to lower specific speed ranges.

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REFERENCES


Fig. 6 Wedge-type impeller without guide plate