ABSTRACT

This paper describes the development of a subscale single stage centrifugal compressor with a dimensionless specific speed (Ns) of 1.8, originally designed for full size applications as a high volume flow, low pressure ratio, gas booster compressor. The specific stage is noteworthy in that it provides a benchmark representing the performance potential of very high specific speed compressors of which limited information is found in open literature. Stage & component test performance characteristics are presented together with traverse results at the impeller exit. Traverse test results were compared with recent CFD computational predictions, for an exploratory analytical calibration of a very high specific speed impeller geometry. The tested subscale (0.583) compressor essentially satisfied design performance expectations with an overall stage efficiency of 74% including, excessive exit casing losses. It was estimated that stage efficiency could be increased to 81% with exit casing losses halved.

NOMENCLATURE

\[ \text{b} \quad \text{Impeller Blade Height} \]
\[ \text{C} \quad \text{Absolute Velocity} \]
\[ \text{Cp} \quad \text{Diffuser Static Pressure Recovery} \]
\[ \text{D} \quad \text{Diameter} \]
\[ \text{g} \quad \text{Gravitational Acceleration} \]
\[ \text{H} \quad \text{Head} \]
\[ \text{IGV} \quad \text{Inlet Guide Vane} \]
\[ \text{k} \quad \text{Specific Heat Ratio} \]
\[ \text{P} \quad \text{Total Pressure} \]
\[ \text{Mu} \quad \text{Mach Number} = \frac{U}{\sqrt{g \cdot R}} \]
\[ \text{Ns} \quad \text{Dimensionless Specific Speed} \]
\[ \text{q} \quad \text{Work Factor} = \frac{H}{U^2} \]
\[ \text{R} \quad \text{Gas Constant} \]
\[ \text{T} \quad \text{Total Temperature} \]
\[ \text{U} \quad \text{Tangential Velocity} \]
\[ \text{W} \quad \text{Massflow} \]
\[ \alpha_2 \quad \text{Impeller Tip abs Air Angle} \]
\[ \beta_2 \quad \text{Backsweep Angle} \]
\[ \Delta \quad \text{Difference} \]
\[ \phi \quad \text{Inlet Flow Coefficient} = \frac{C_1}{U_2} \]
\[ \eta \quad \text{Efficiency} \]
\[ \nu \quad \text{Poisson's Ratio} \]
\[ \sigma \quad \text{Stress} \]
\[ \rho \quad \text{Density} \]
\[ \omega \quad \text{Angular Velocity} \]

Subscripts

1 \quad \text{Impeller Inlet} \\
2 \quad \text{Impeller Tip} \\
3 \quad \text{Diffuser Exit} \\
ad \quad \text{Adiabatic} \\
c \quad \text{Compressor} \\
e \quad \text{Exit} \\

Note all angles are with respect to the meridional plane.

1. INTRODUCTION

The wide diversity of applications for centrifugal compressors & pumps cover impeller designs ranging from dimensional inlet specific speeds as low as 0.3, to as high as 2.0 where:

\[ \text{Specific Speed} \quad N_s = \frac{\omega \cdot (W^2 / \rho)}{(g \cdot H)} \]

The classic specific speed charts of Balje(1) for single stage centrifugal compressors show peak efficiencies close to \( N_s = 0.8 \), with efficiency diminishing at lower & higher specific speeds. Since the majority of centrifugal compressor designs cluster the optimum efficiency region, limited open literature exists relating to the performance characteristics of the lower efficiency, very low & very high specific speed compressor designs. In a recourse to quantify compressor performance levels of very low specific speeds the author described the results of experiments with a partial emission centrifugal compressor in (2). The intent of this complementary paper is to quantify the performance characteristics of centrifugal compressors at the opposite end of the \( N_s \) spectrum, by relating the results of performance calibrations on a single stage compressor with a design \( N_s \) of 1.8.

The particular \( "F" \) stage to be discussed was an extension of the work described in (3) & (4) relating to the component characteristics of a family of compressor stages encompassing an \( N_s \) range from 0.3 to 1.3, & was conducted using the same compressor test facility. The \( F \) stage is noteworthy in that it offers to serve as a benchmark representing the performance potential of high specific speed centrifugal compressors, of which limited information is found in literature, with some exceptions notably (5), besides serving as an intriguing test case for special calibration of the Dawes CFD code. As will be discussed later the adiabatic efficiency attained for the \( F \) stage was quite low at 74% yet only 1% point below the design test goal, in a \( N_s \) range which is best suited by mixed flow compressor designs. The development...
intent was, however, to extend the flow capability of an existing family of stages using the maximum amount of existing hardware. Recourse to a mixed flow stage would have required almost complete redesign. The "Achilles Heel" (or knee-so to speak) of very high Ns centrifugal impellers is the high shroud curvature in the knee region, caused by the characteristic geometry with a large inducer eye & large impeller exit tip height, being susceptible to flow separation aft of the inducer throat.

The use of high blade backsweep can assist in delaying the separation susceptibility, in that the stream surface curvatures are relaxed. As a consequence the F impeller was designed with 50 deg backsweep (relative to the radial direction). A second problem in the design of high Ns stages is the relatively high radial velocity component at the diffuser exit which can result in a relatively high discharge loss if not recovered in the scroll or discharge collector. As will be discussed later the exit loss of the F stage amounted to several percentage points loss in stage efficiency.

During the initial design of the F stage it was realized that impeller structural constraints could present a limitation in eventual application as a consequence of increased blade & disc stresses, together with high hoop stress for a shrouded impeller option. Preliminary evaluation of the stresses anticipated during developmental air test calibrations with an open shrouded impeller revealed acceptable stress levels commensurate with short time low speed operation. The relationship between specific speed & impeller stresses is addressed later in this paper.

2. COMPRESSOR DESIGN FEATURES.

The anticipated application of the F stage was as a low pressure ratio, high volume flow, gas booster pipeline compressor, in a size 1.71 larger than the air test model stage. Following design of the full size compressor, hardware compatible for installation & ambient suction testing in an existing subscale compressor rig was designed & procured. The subscale design point air test performance parameters are listed in Table 1.

Table 1 Performance Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Ratio</td>
<td>1.21</td>
</tr>
<tr>
<td>Corrected Airflow pps</td>
<td>8.3</td>
</tr>
<tr>
<td>Adiabatic Efficiency</td>
<td>75</td>
</tr>
<tr>
<td>Rotational Speed</td>
<td>14</td>
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<tr>
<td>Specific Speed</td>
<td>1.81</td>
</tr>
<tr>
<td>Inducer Hub Diameter</td>
<td>3.5</td>
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<tr>
<td>Inducer Tip Diameter</td>
<td>8.75</td>
</tr>
<tr>
<td>Impeller Tip Dia (D2) (mean)</td>
<td>11.9</td>
</tr>
<tr>
<td>Blade Exit Height</td>
<td>1.63</td>
</tr>
<tr>
<td>Impeller Backsweep β1</td>
<td>50</td>
</tr>
<tr>
<td>Number of Blades</td>
<td>14</td>
</tr>
<tr>
<td>Inlet Flow Coefficient</td>
<td>0.44</td>
</tr>
<tr>
<td>Work Factor</td>
<td>0.44</td>
</tr>
</tbody>
</table>

(Note efficiency flange to flange total-to-total, less downstream exit diffuser).

The open face impeller is shown installed in the test rig on Fig 1, together with the parallel wall diffuser for both vaneless & vaned diffuser testing. Note that the inducer blading was heavily leaned in order to relax the passage curvatures.

3 TEST RIG DESCRIPTION.

The compressor test rig depicted schematically on Fig 2 was used to conduct F stage performance calibrations at ambient suction conditions. The rig was driven by a 500hp direct current electric variable speed motor driving through a speed increaser gearbox. The inlet airflow was measured with a bellmouth venturi, & subsequently ducted to the axial inlet guide vanes (IGV's) of the compressor. Airflow regulation was achieved with a butterfly valve at the compressor discharge.

The pressure & temperature at the inlet & discharge were measured with Kiel probes & resistance temperature devices. Table 2 lists the test rig instrumentation. Automatic traverse capability with a Cobra probe at 1.29 the impeller tip diameter was provided, together with shroud pressure transducers & thrust transducers to measure axial thrust.

Table 2. List of Instrumentation

<table>
<thead>
<tr>
<th>Station</th>
<th>Temperatures</th>
<th>Pstatic</th>
<th>Ptot</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Bellmouth</td>
<td>3</td>
<td>3</td>
<td>1</td>
</tr>
<tr>
<td>Inlet Flange</td>
<td>3</td>
<td></td>
<td>3</td>
</tr>
<tr>
<td>Impeller Shroud</td>
<td>11</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Impeller Tip</td>
<td>5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Diffuser Exit</td>
<td>3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Discharge Flange</td>
<td>4</td>
<td>3</td>
<td>3</td>
</tr>
</tbody>
</table>

Overall compressor efficiency could be determined from both temperature rise & input power measurements.
Analysis of potential measurement errors at the low speed test conditions revealed that the efficiency uncertainty based on temperature rise was +/- 1.2% points compared to +/-1.7% points based on torque. Efficiency levels quoted are those based on temperature measurements.

The mixed impeller exit vector conditions were computed using insulated casing discharge total temperature, impeller tip static pressures, and the continuity equation assuming an impeller tip blockage and recirculation plus windage correction according to (3). Impeller tip traversing with a small cobra probe mounted on an electronically modulated traverse actuator, was conducted at several points along a selected constant speed line to determine impeller exit flow profile conditions. The axial inlet (zero camber) IGV’s shown on Fig 3, were manually adjusted to permit calibrations at vane angles of zero, -10, & -25 degrees to the axial direction. The impeller was tested at a DeLaval number of Mu = 0.65, & was allowed to thermally stabilize in the insulated casing between all test points. The nominal cold shroud clearance between the impeller & abradable stationary shroud was 0.070 in (1.8mm). Very slight shroud rubbing during testing affirmed near zero running clearance.

Prior to performance calibrations of the subscale F stage tests were conducted on the impeller with the tip diameter originally machined at 14.0 inch (356mm), both with a vane & vaneless diffuser configuration, as shown on Fig 4. These initial tests indicated essentially the same overall efficiency with either diffuser type as a consequence of high discharge collector losses amounting to complete loss of the diffuser exit dynamic head.

The impeller tip diameter was subsequently machined down to a mean diameter of 11.9 inch, & performance calibrations initiated on the design subscale geometric configuration, commencing with tests at zero IGV prewhirl, & Mu = 0.65. Table 3 lists the test sequence followed with IGV settings, & modifications to the vaneless space width & 0.25 inch increased inducer leading edge undercut.

Test performances for the various configurations of the subscale F stage are shown on Fig 5 in terms of overall stage efficiency & work factor versus inlet flow coefficient. Test peak efficiencies & corresponding work factors are compared to design goals on Table 4.
Adjustment of the IGV's to -10 & 25 deg on Build 10A, both decreased the stage efficiency as shown on Fig 7. Examination of the impeller passage geometry near the inducer throat, at the hub, revealed the possibility of high blockage. It was therefore decided to test the effect of increasing the leading edge sweep angle (negative sweep) from 10 to 35 deg. Subsequent testing on Build 10C showed 1% point efficiency loss with the leading edge modification.

Cobra probe traverse results at flow coefficients $\phi = 0.29, 0.41, & 0.52$ are shown on Fig 8. Near design flow coefficient the flow generally fills the central passage, flipping towards the shroud near stall, while flipping towards the hub as choke is approached. These general patterns of flow migration were recently matched by CFD modelling of the F impeller using the Dawes Code. Satisfactory agreement of the test flow profiles was obtained with the CFD results, baring exception of the near stall flow coefficient $\phi = 0.29$. 

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**Fig 5. F Stage Test Performances**

Decreasing the vaneless space width on Build 10B increased the stage efficiency from 72 to 74.0% basically due to improved diffuser performance. Impeller & diffuser performances for Build 10A & 10B are shown on Fig 6, & indicate peak diffuser recovery $C_p 2$ increased above 0.6 with the smaller width, while impeller efficiency remained essentially unchanged. Note however that the Slip Factor also increased slightly as reflected by the change in work factor in Table 4.

The casing exit loss coefficient $\lambda$ defined by total pressure loss from the vaneless diffuser exit to exit flange, divided by the diffuser exit dynamic head is also shown on Fig 6 & exceeds unity at flow coefficients higher than design.

**Fig 6. Build 10A & 10B Performances**

**Fig 7. Performance with IGV -10 & 25 deg**

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**Fig 8. Traverse Results**
The results of performance calibrations on the F Stage confirmed the design expectations & indicated that further performance improvements would be attainable with enlargement plus additional diffusion in the exit collector.

Fig 2 shows the rig collector was of the "folded over" type for ease of fabrication & installation. Such collectors are known to lose a high fraction of the exit dynamic head as discussed in (7). It was estimated that if the exit loss coefficient was reduced 50%, the stage efficiency would increase by 7% points, illustrating the importance of the exit casing design on high specific speed centrifugal & mixed flow compressors performance.

5 SPECIFIC SPEED & IMPELLER STRESSES.

As mentioned previously the two major design constraints that limit the application of high specific speed centrifugal compressors are excessive impeller stresses & high casing pressure losses.

It can be shown as (6) extensively describes, that the specific speed concept can be extended beyond impeller aerodynamic characterization, to also impeller stresses, inertial, & dynamic properties. For instance the blade bending stress at the impeller tip can be approximated by the following expression;

\[
\sigma = \frac{1}{2g} b_r \left( \frac{h}{r} \right)^2 \left( 1 + \frac{2 \times \text{Taper ratio}}{13} \right) \frac{h}{r}^2 \quad (2)
\]

where \( h \) = blade root thickness

This approximate expression ignores the important effect of blade rake, root fillet radius & load sharing from adjacent blading, but nevertheless provides a useful hand check in preliminary design iterations, & fingerpoints blade features controlling the bending stresses. For example high Ns impellers with intrinsically larger blade height to tip diameter ratios \( (b_2/D_2) \), will propagate stresses increasing in proportion to the blade height squared.

The direct relationship between Ns & \( b_2/D_2 \) is illustrated by the trends shown on Fig 9 with stage pressure ratio as a parameter. Note that although increasing pressure ratio requires higher tip speed, the impeller tip height diminishes due to the higher density ratio.

The stress relationship can be further simplified by assuming a typical blade taper ratio of 4:1, as;

\[
\sigma = \frac{1}{2g} U_2^2 \sin \beta_2 \left( \frac{b_2}{D_2} \right) \left( \frac{b_2}{b_2} \right) / g \quad (3)
\]

Likewise the impeller hub stress is a function of the tip speed, impeller shape (specific speed) & size of the bore.

An approximate expression for the hub stress is given by;

**Hub Stress**

\[
\tau = \text{Impeller Shape Factor} \frac{U_2^2}{b_2^2} (3 + \nu) / g \quad (4)
\]

where \( \nu \) is Poisson's ratio & the impeller shape factor is dependent upon specific speed & shroud configuration, as typified by the data shown on Fig 10. Since increasing specific speed enlarges the size of the inducer eye the impeller axial (forward) thrust increases, thus specific speed is also related to the axial thrust characteristics of the impeller.

These specific speed relationships are generally only useful in the preliminary design of centrifugal compressors, as their range of application is limited compared to a myriad of impeller geometries that may be possible for any given design case. Nevertheless the Ns connection can provide a rapid sanity check to prepackaged automated computational design programs now popular.
CONCLUDING REMARKS

The development of a single stage high specific speed centrifugal compressor has been described to provide a benchmark representing the performance capabilities of high Ns stages, & to indicate potential pitfalls in their application. The information also serves to complement an earlier paper by the primary author relating experiments on a very low specific speed partial emission centrifugal compressor, thereby completing the "end game". In this respect both low & high Ns impeller performance cases are superimposed against the polytropic impeller efficiency correlation of Ref 8 on Fig 11, indicating reasonable agreement with the correlation trends.

The adoption of a high Ns design is not always an independent design choice, & in fact almost invariably arises as an extension to an existing compressor, turbocharger, or gas turbine product line, as was indeed this particular compressor.

One example of this is the practice of impeller tip diameter cropping to match lower head requirements, which intrinsically increases Ns.

A second example is the continuing demand for increased power in small intermittent operation gas turbines, or expendable turbojets, through higher airflows within the same engine envelope, thereby also increasing compressor specific speeds.

REFERENCES


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