USE OF GRADE 80 STEEL IN FABRICATED CENTRIFUGAL IMPELLER

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SUMMARY

Typical backward curved centrifugal fan impeller construction consists of a solid backplate, a spun or fabricated mid-span intermediate shroud, a spun or fabricated front plate and rolled blades welded in between the shrouds and backplate. An alternative method to an intermediate shroud is proposed, wherein a high strength low alloy steel is used (ASTM A656 Grade 80). The Grade 80 steel's higher yield point provides sufficient strength to eliminate the need for an intermediate shroud, resulting in an efficiency gain of 3% while reducing manufacturing cost by 42%.

INTRODUCTION

Backward curved centrifugal impellers are used in many industrial blower applications. Centrifugal fans have many embodiments which depend on factors including the required duty point, air density, the presence of particulate or other conveyed material, fan power and efficiency concerns (Daly [1]). Some of the advantages of backward curved fans include high efficiency, lack of overloading horsepower characteristics, stable performance throughout a large range of the characteristic pressure curve and low noise levels near the fan's peak efficiency.

The example that is examined in the reported research required a backward curved centrifugal fan as a consequence of the required minimum total efficiency of 72.7% at the selected duty point of 4.60 m^3/sec at 3363 Pa. Additionally, the available space envelope for the fan allowed a maximum impeller diameter of 508 mm.

There is little data available in the published literature on the effect of a mid-span intermediate shroud on fan performance. The shroud is mentioned by Eck [2] as a design improvement, though there is no discussion of the aerodynamic effects. An impeller design, suggested by Eck [3], incorporating an angled gusset projecting from the backplate to approximately the middle of the blade allowed for a forty percent increase in tip speed. However, efficiency effects, as compared to an unreinforced impeller were not discussed, as the main focus was increasing tip speed. Historical unpublished data suggested that other backward inclined centrifugal impellers with similar intermediate shroud can be expected to show an increase in efficiency up to 6% when the intermediate shroud is removed.

AERODYNAMIC EVALUATION

The inclusion of one or more intermediate shrouds in a fan impeller, from an aerodynamic perspective, is detrimental to the air performance and efficiency of a fan. Test data from a model test demonstrate this, Figure 1. These previously unpublished results were identified as Test Numbers P-0472-T2 and P-0472-T4 and were undertaken at Garden City Fan in November, 1973. The fan tested was a Garden City Fan 222BF fan having flat backward inclined blades. Two impellers were tested, one with an intermediate shroud consisting of gusset-type blade reinforcements, the other without. The two impellers were tested in the same test setup. Figure 2 shows the type of impeller tested with the intermediate shroud in place.

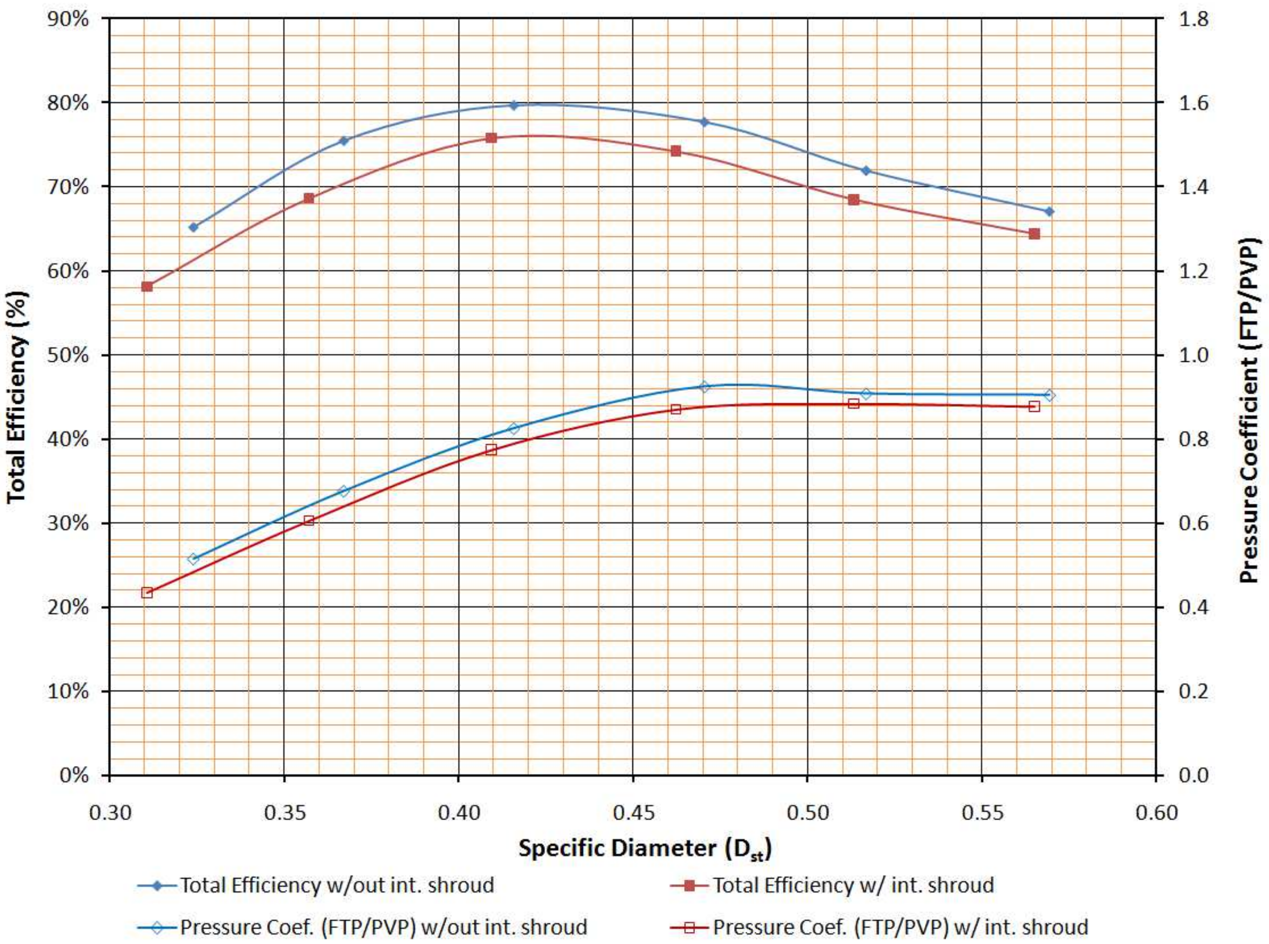

Figure 1 – Historical data of 565.2 mm diameter Garden City Fan 222BF impeller, showing efficiency improvement resulting from removal of intermediate shroud.

The efficiency of the 565.2 mm diameter fan impeller tested without the intermediate shroud was 4 percentage points greater than the same impeller with the intermediate shroud through the entire operating range of the fan.

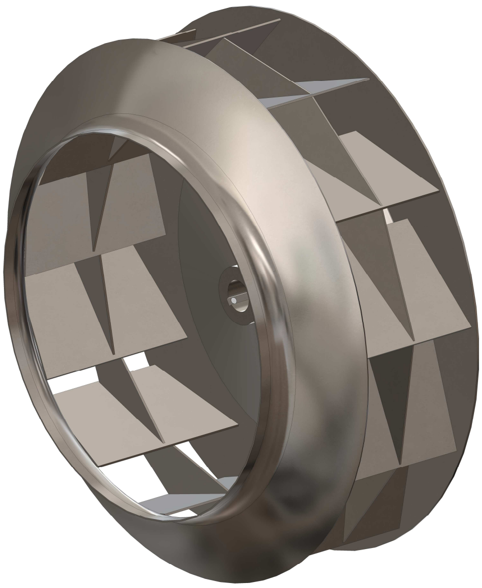

Figure 2 – Example of Garden City BF impeller used in historical testing.

The performance of the two fan tests is presented in non-dimensional form in Figure 1, with the total efficiency (% TE) and the pressure coefficient (FTP/PVP) both plotted against specific diameter (D_{st}) of the fan. Specific diameter is a constant value for all flow and pressure points on a parabolic system curve for a given fan diameter. The chart may therefore be used to assess the relative difference in total efficiency, fan total pressure and fan speed for a given fan diameter at a required operating point.

The data in Figure 1 indicates that a 5 to 6 percent savings in operating power plus a relatively small decrease in operating speed of the fan may be obtained by removing the intermediate shroud. The reduction in power was the primary justification for investigating the possibility of removing the intermediate shroud. The authors' objective was therefore to remove the intermediate shroud whilst maintaining the required performance of 3363 Pa total pressure at 4.60 m^3/sec volume flow rate with a 508 mm diameter impeller.

Figure 3 – Test setup to compare performance impact of intermediate shroud; setup per AMCA 210-07, Figure 12, Installation Type B.

A 381 mm diameter impeller model with, and a second impeller without, an intermediate shroud were tested in accordance with the requirements of AMCA Standard 210-07, Installation Type B [4]. A photograph of the test setup is shown in Figure 3. The results are presented in Figure 4 and Figure 5, scaled, using the fan laws, to predict the 508 mm diameter fan performance.

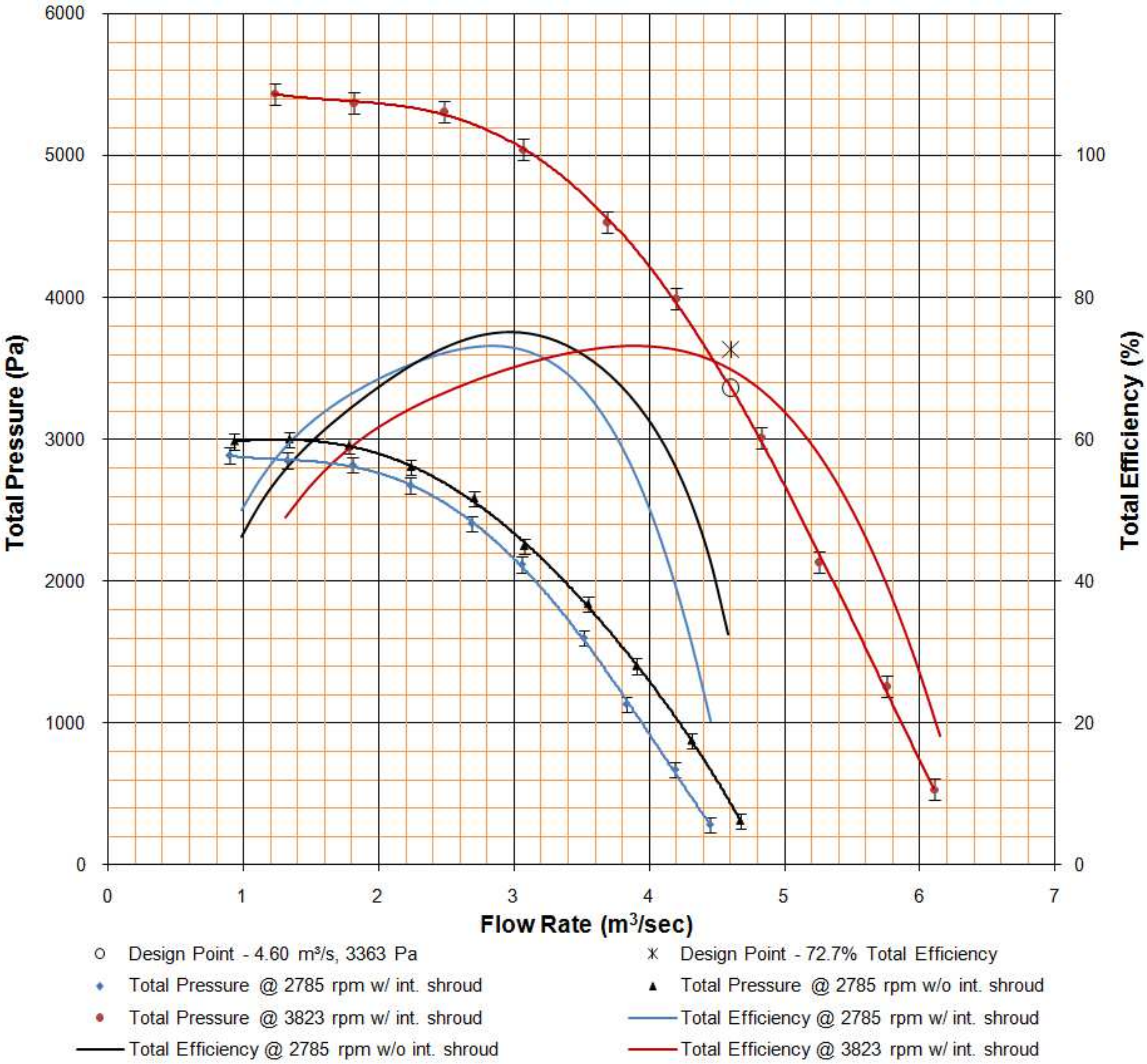

Figure 4 – Comparison of ø508 mm backward curved single-surface impeller with and without intermediate shroud at 2785 rpm (maximum speed of impeller without intermediate shroud). Impeller with intermediate shroud shown at 3823 rpm, meeting flow and pressure design point, but missing efficiency design point. All data is scaled, via fan laws, from ø381 mm diameter impeller.

Figure 4 shows the results, scaled via fan laws, for the 508 mm fan with and without an intermediate shroud at the maximum speed of the impeller without an intermediate shroud (2785 rpm), built of standard ASTM A572 Grade 50 steel construction. Also shown in Figure 4 are the results for the fan with intermediate shroud scaled to an operating speed of 3823 rpm to meet the design point flow and pressure required. However, the desired design point total efficiency of 72.7%, also shown in Figure 4, is not met by the impeller with the intermediate shroud.

Figure 5 shows the results, scaled via fan laws, for the 508 mm fan with and without the intermediate shroud, running at speeds required to attain the required design point pressure and flow. The speed required with the intermediate shroud is 3823 rpm, identical to Figure 4. The speed required without the intermediate shroud is 3698 rpm, which provides the desired total efficiency at the duty point. The fan without the intermediate shroud can attain the required duty point flow and pressure at 96.7% of the speed and 96.1% of the power required of the same fan with an intermediate shroud.

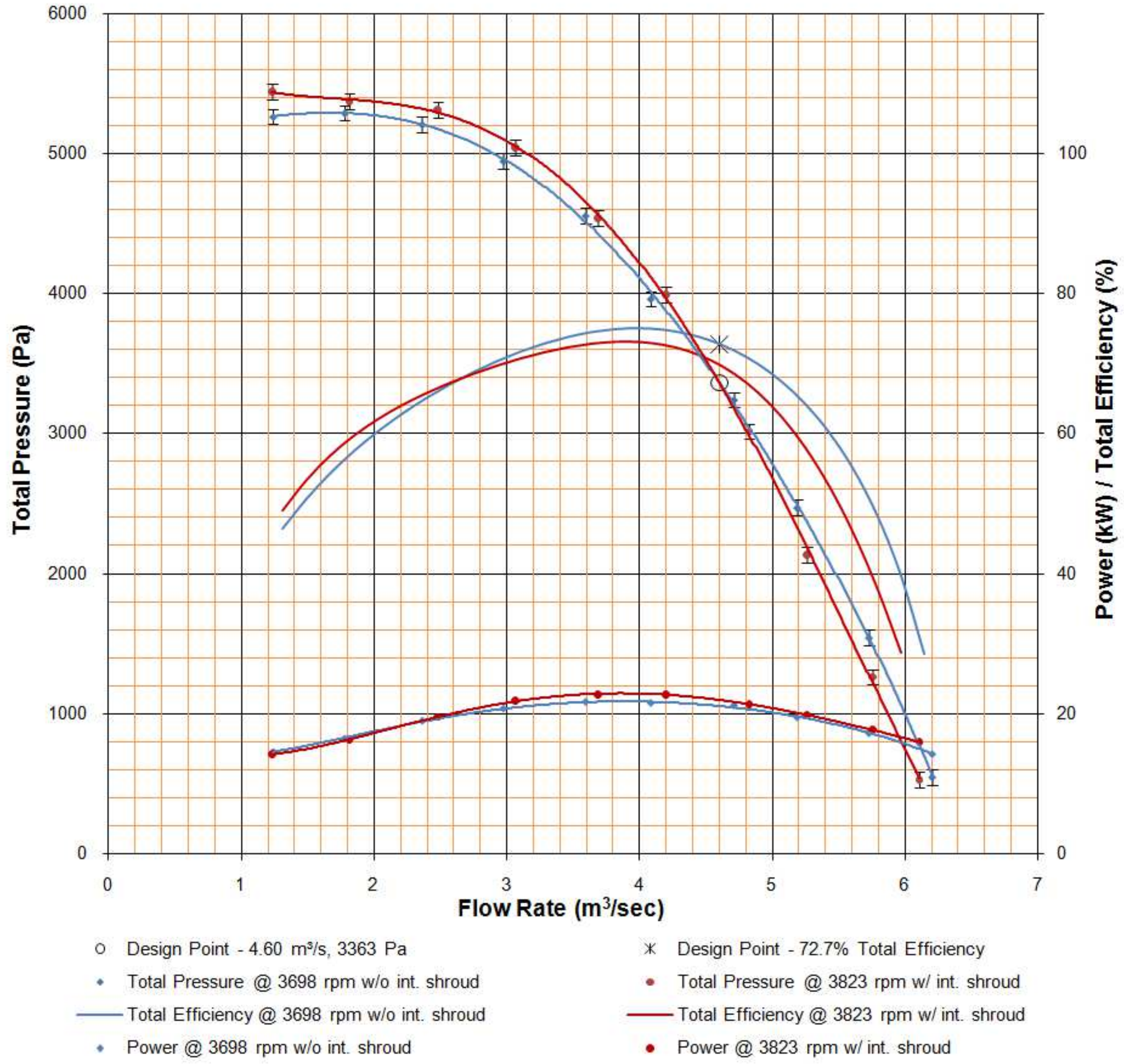

Figure 5 – Comparison of 508 mm backward curved single-surface impeller at design point for flow and pressure. Impeller with intermediate shroud at 3823 rpm. Impeller without intermediate shroud at 3698 rpm. All data is scaled, via fan laws, from ø381 mm diameter impeller.

MECHANICAL DESIGN

While removal of the intermediate shroud results in an aerodynamic benefit, it is mechanically challenging. The nature of the challenge may be understood by conceptualizing the blade as a beam. The beam is rigidly supported at both ends with a uniform, distributed load that arises as a consequence of centrifugal force associated with the fan's rotation. The maximum bending moment is a function of the square of the beam's length, Equation 1. Using Equation 2, the maximum stress in the beam is proportional to the maximum bending moment.

$$M_{\max} = \frac{wL^2}{12} \tag{1}$$

$$\sigma_{\max} = \frac{M_{\max} \cdot c}{I} \tag{2}$$

M_{max}	Maximum bending moment (N·m)
L	Beam length (m)
w	Load per unit length (N/m)
σ_{max}	Maximum stress in the beam (N/m^2)
I	Moment of inertia (m^4)
c	Distance from neutral surface (m)

Removal of the intermediate shroud doubles the length of the beam and in so doing increases the maximum stress by a factor of four. For the 508 mm diameter impeller, the rated maximum speed of the impeller without intermediate shroud, made of ASTM A572 Grade 50 steel, is 2785 rpm. Using the Solidworks Simulation 2011 software package, finite element analysis of the impeller, using solid curvature-based mesh at the maximum run speed gave satisfactory results, Figure 6. The maximum acceptable stress level has been established based on over 25 years of service history.

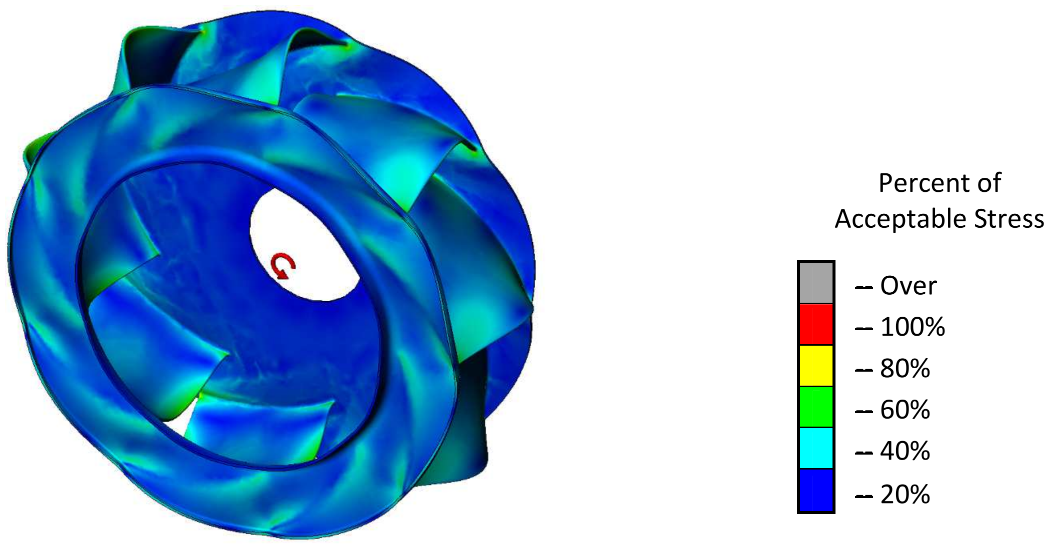

Figure 6 – Von Mises stress as percentage of yield strength; impeller without intermediate shroud, ASTM A572 Grade 50 steel at 2785 rpm.

Analysis of the same impeller at the desired run speed (3698 rpm) gave unsatisfactory results, Figure 7. All four corners of the blades have predicted stress levels beyond the material yield strength, indicating plastic deformation.

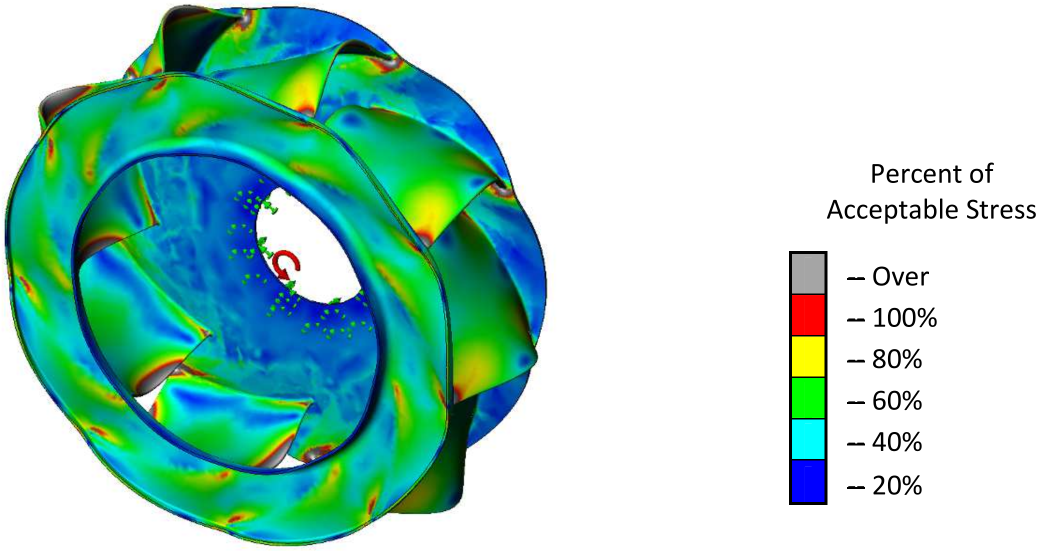

Figure 7 – Von Mises stress as a percentage of yield strength, impeller without intermediate shroud, ASTM A572 Grade 50 steel at 3698 rpm.

Similar finite element analysis was run on the ASTM A572 Grade 50 steel impeller with the intermediate shroud at the predicted run speed (3823 rpm) required to match the performance of the impeller without the intermediate shroud. With the intermediate shroud in place, allowable stress was not exceeded, Figure 8.

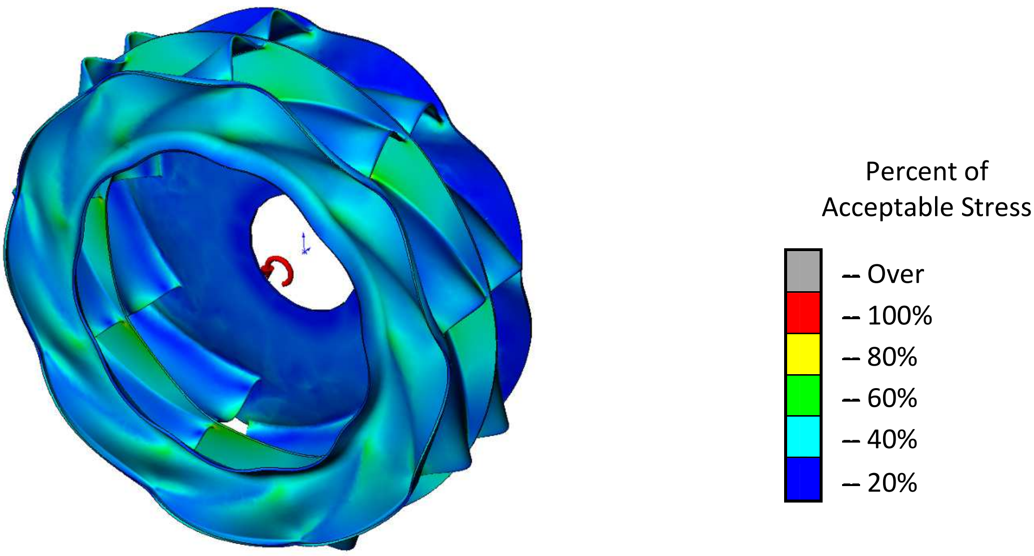

Figure 8 - Von Mises stress as a percentage of yield strength, impeller with intermediate shroud ASTM A572 Grade 50 steel at 3823 rpm.

A way to reach the required speed without the intermediate shroud is to use a different type of steel with sufficient strength to withstand the additional centrifugal force. While there are stronger materials available, Grade 80 steel provided the best combination of weldability, formability and availability as presented in Table 1. Availability can be a particular problem with more exotic grades of steel. Grade 80 steel has sufficient availability, though some thicknesses are not available. While Grade 80 steel is 27.4% more expensive then Grade 50 steel on a weight basis, elimination of the intermediate shroud results in an increase in material cost of 11.9%. Additionally, elimination of the intermediate shroud reduced the labor needed to produce the impeller because of the elimination of welding associated with fitting an intermediate shroud. As a consequence the total manufacturing cost of a Grade 80 impeller without intermediate shroud was estimated to be 42.8% less than that of a Grade 50 impeller with an intermediate shroud, Table 2. This saving was achieved despite Grade 80 steel being more expensive then the Grade 50 steel.

Table 1 – Comparison of materials considered for impeller construction.

	Yield Strength (MPa)	Yield Strength (MPa)	Weldability	Formability	Availability
Plain Carbon Steel	221	400	Excellent	Excellent	Excellent
A572 Grade 50	345	448	Good	Good	Excellent
A656 Grade 80	552	655	Good	Good	Good
Domex 100	689	758	Good	Fair	Poor

Table 2 – Price comparison of A572 Grade 50 Steel with A656 Grade 80 Steel.

	Intermediate Shroud Required?	Impeller Mass (kg)	Material Cost per unit weight (as % of Gr. 50 cost)	Total Material Cost (as % of Gr. 50 cost)	Labor Hours (Fabrication & Welding)	Labor Cost (as % of Gr. 50 cost)	Finished Impeller Cost (as % of Gr. 50 cost)
A572 Grade 50	Yes	18.6	100%	100%	5.1	100%	100%
A656 Grade 80	No	16.3	127.4%	111.9%	2.6	51.9%	57.2%

Finite element analysis of the impeller without intermediate shroud and using Grade 80 steel material properties indicated that the impeller would remain below the material's allowable maximum stress (Figure 9). Though stress levels are acceptable, the authors recognized that removing the intermediate shroud will reduce the impeller stiffness. Since removing the intermediate shroud will reduce impeller stiffness, the potentially reduced natural frequency was regarded by the authors as problematic. A natural frequency or harmonic of a natural frequency may be close to fan running speed. A modal finite element analysis was therefore undertaken.

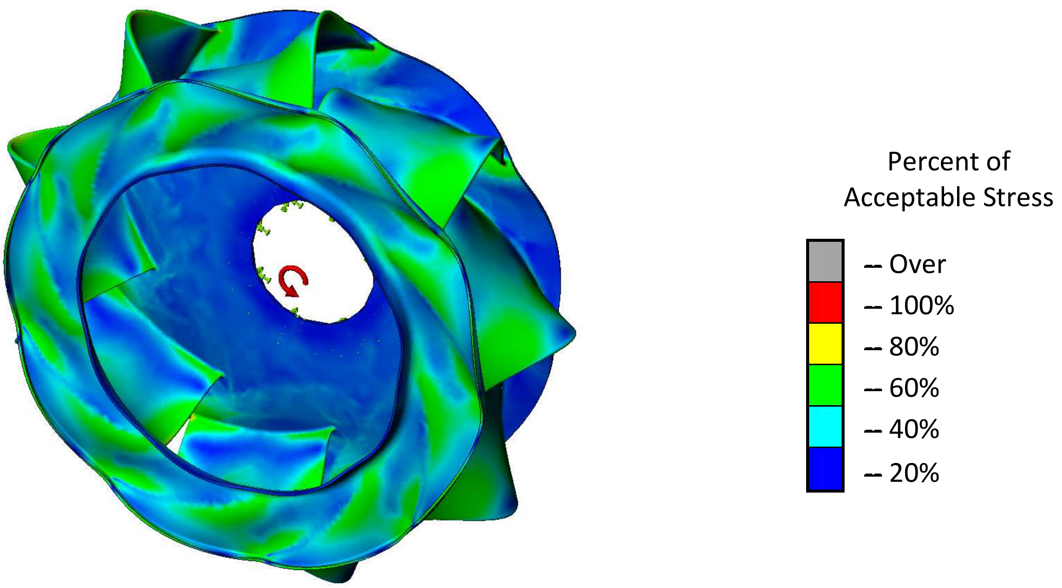

Figure 9 – Von Mises stress as a percentage of yield strength, impeller without intermediate shroud, ASTM A 656 Grade 80 steel at 3698 rpm.

MODAL ANALYSIS

A modal finite element analysis was undertaken using Solidworks Simulation 2011, comparing the natural frequencies predicted for the Grade 50 impeller with intermediate shroud to those predicted in the Grade 80 impeller without an intermediate shroud. The boundary conditions for the two analyses were the same, to enable the change in natural frequencies to be assessed.

Results of the modal finite element analysis (Table 3) give both impellers' natural frequencies (in a static condition). The results include two first diametral modes, as both the Grade 50 and Grade 80 impellers having nine blades. As a consiquence of the impellers asymetry, they exhibited a first diametral mode at two diferent locations around their center axis, with slightly different associated natural frequencies. If the impellers had been designed with an even number of blades, there would have been only one 1st diametral mode. The second predicted mode was the impeller unbrella mode. Second diametral mode and first torsional modes were at frequencies high enough to be reasonably ignored.

The results presented in Table 3 indicate that the first diametral mode increased by 14% and 11% with removal of the intermediate shroud. While it may seem counter-intuitive that the removal of the intermediate shroud would stiffen the impeller, in this case, the reduced mass helps to offset the reduction in stiffness. It is also likely that the intermediate shroud provided little resistance to the first diametral mode. Additionally, due to limited availability of gauges of Grade 80 steel sheet, both blades and backplate are 0.1875" (4.76 mm) thick, as compared to .1345" (3.42 mm) thick for the Grade 50 impeller.

The umbrella mode natural frequency reduces by 3% when the intermidiate shroud is removed, Table 3. The reduction indicates that the intermidiate shroud did provide some stiffness. The significance of the reduction may be better appreciated when considered within the context of multiple of running speed, Table 4. Without the intermediate shroud, running speed needed to achive the fan duty point is reduced from 3823 rpm (63.7 Hz) to 3698 rpm (61.6 Hz). The umbrella mode is therefore 4.1 times running frequency both with and without an intermidiate shroud. As the multiple of running speed has not changed, the authors have confidence that there will be no in-service issues associated with the umbrella mode.

In contrast to the unbrella mode, the first diametral modes changes form 2.1 and 2.2 times running frequency to 2.5 and 2.6 times running frequency. A first diametral mode of 2.2 times running frequency is marginally acceptable, however a running frequency of 2.1 is not acceptable. It is too close to a multiple of 2.0 (the second harmonic of running speed). The second harmonic can be excited by coupling misalignment, and consequently must be avoided. Removal of the intermediate shroud increased the multiple of running speed to aceptable values. As such removal of the intermidiate shroud has improved the dynamic performance of the impeller. Plots of the exaggerated first diametral mode and umbrella mode shapes of the Grade 80 impeller are presented in Figure 10.

Table 3 – Resonant frequencies of impeller with and without intermediate shroud.

Mode Shape	Grade 50 With Intermediate Shroud Nat. Freq. (Hz)	Grade 80 Without Intermidiate Shroud Nat. Freq. (Hz)	Percentage Change
1st Diametral	136.7	156.3	+ 14%
1st Diametral	142.8	158.3	+ 11%
Umbrella	260.6	253.8	- 3%

Table 4 – Resonant frequencies as a percentage of fan running speed.

	Grade 50 With Intermediate Shroud			Grade 80 Without Intermidiate Shroud		
Mode Shape	Nat. Freq. (Hz)	Running Frequency (Hz)	Harmonic	Nat. Freq. (Hz)	Running Frequency (Hz)	Harmonic
1st Diametral	136.7	63.7	2.1	156.3	61.6	2.5
1st Diameteral	142.8	63.7	2.2	158.3	61.6	2.6
Umbrella	260.6	63.7	4.1	253.8	61.6	4.1

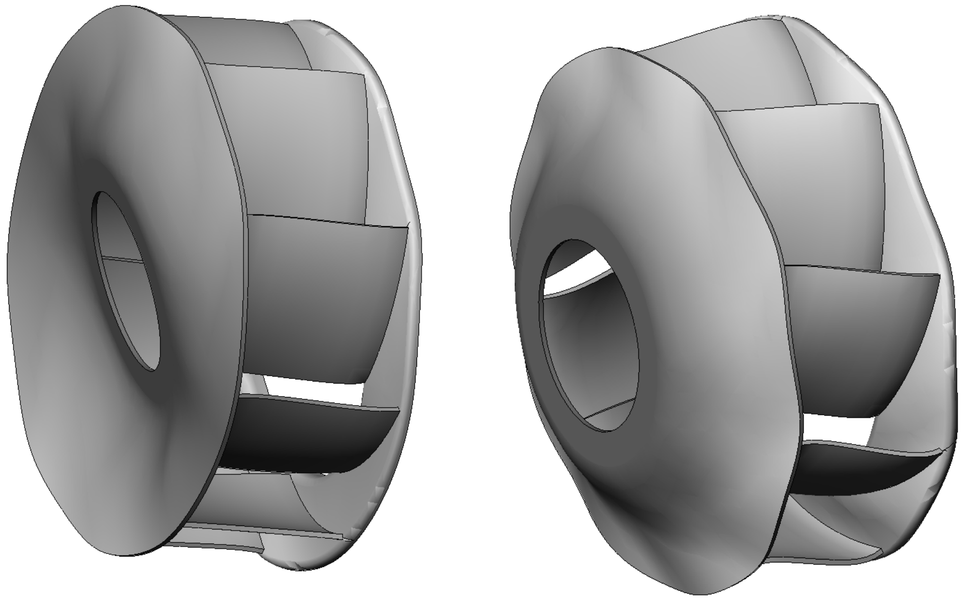

Figure 10 – First and second mode shape of the Grade 80 material impeller at rest; left, 1st diametrial mode: 156.3 Hz ,right, umbrella mode: 253.8 Hz .

AERODYNAMIC & MECHANICAL VALIDATION

To confirm the structural integrity of the impeller, over-speed tests were conducted at 125% of run speed (4623 rpm). The speed was held for 15 minutes in an over-speed chamber. After testing, the impeller was examined and showed no signs of cracking or failure, validating the finite element structural analysis results.

To confirm the Grade 80 impeller performance predicted using model test data and the fan laws, a 508 mm Grade 80 impeller without an intermediate shroud was tested according to AMCA 210-07, Installation B and compared to a Grade 50 impeller with an intermediate shroud. The results for the original Grade 50 impeller with intermediate shroud and the results for the new Grade 80 impeller without intermediate shroud are presented in Figure 11. Fan speeds were adjusted for both the Grade 50 and Grade 80 impeller to achieve the design point (3363 Pa, 4.60 m^3/sec). The resulting data indicates that both fans outperformed the scaled data. The difference in efficiency is about 3 points and the Grade 80 fan slightly outperforms the required efficiency (72.7%).

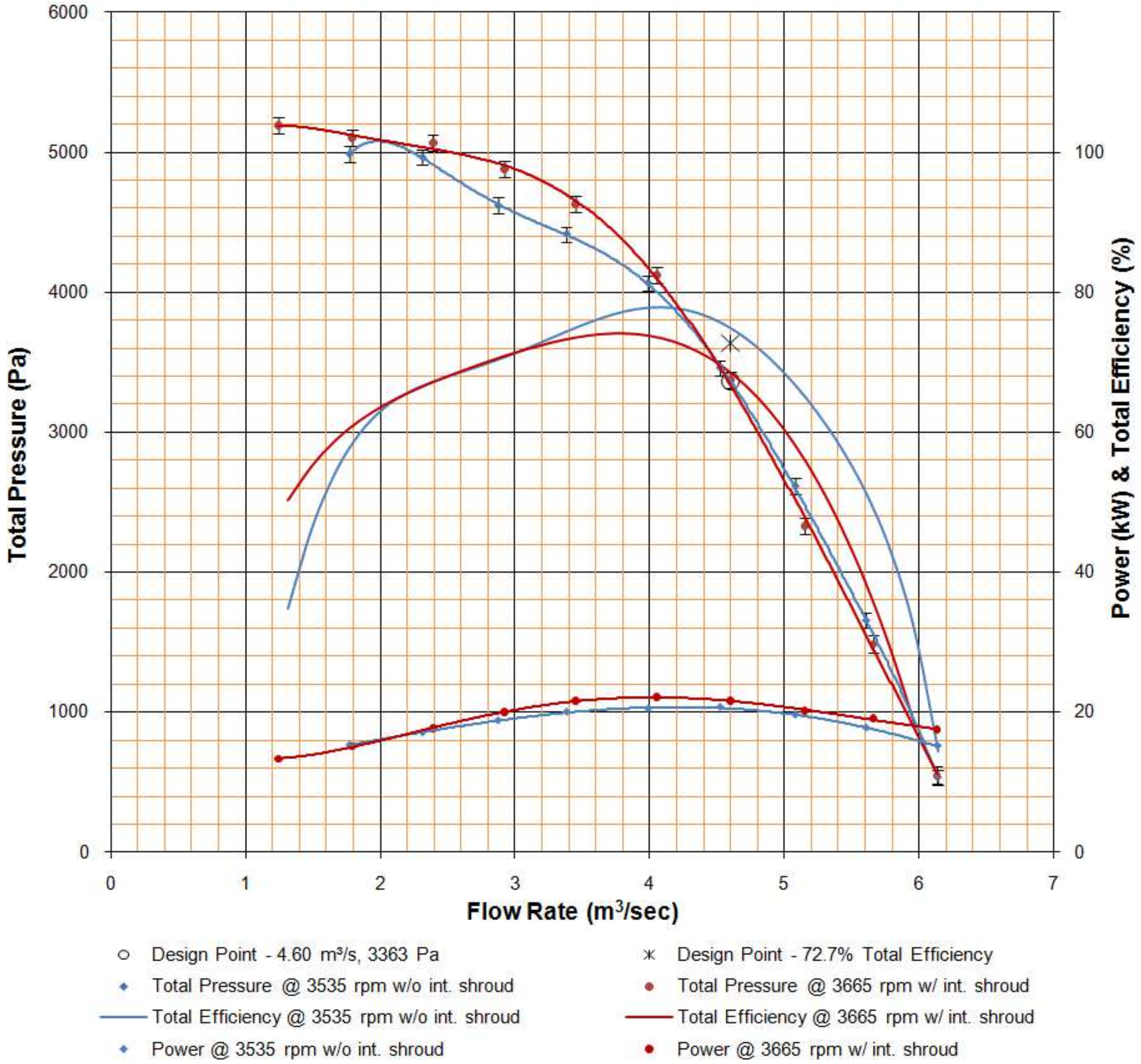

Figure 11 – Test results comparing ø508 mm backward curved single-surface impellers at design point ($4.60m^3/sec$, 3363 Pa, 72.7% total efficiency); with intermediate shroud at 3665 rpm, without intermediate shroud at 3535 rpm.

CONCLUSIONS

A duty point of 3363 Pa at 4.60 m^3/sec was required in a backward curved fan with a maximum impeller diameter of 508 mm. Standard construction of the impeller required an intermediate shroud. Previously unpublished data on the performance of a historic Garden City Fan backward inclined impeller indicated that an intermediate shroud reduces the impeller efficiency. Historic performance tests had shown that on other types of centrifugal fans, the efficiency loss could be as much as 5-6%. Since efficiency was a prime concern in this application, the authors chose to remove the intermediate shroud. Tests on the studied impeller indicated a 3.3% reduction in required speed and 3.9% reduction in required power.

The maximum speed of the impeller without the intermediate shroud, using standard construction, was insufficient to reach the required speed of 3698 rpm. Other available materials were examined and ASTM A656 Grade 80 steel was found to be most suitable for the application. Finite element analysis of the impeller indicated that the Grade 80 steel could run at the required speed and remain within allowable stress limits.

An over-speed test was performed that confirmed that the Grade 80 impeller without intermediate shroud was able to operate at the target rotational speed. Manufacturing cost of the impeller was reduced by 42% despite the higher cost of Grade 80 steel when compared to the original Grade 50 steel. Cost was lower as a consequence of the elimination of the material cost associated with the intermediate shroud and labor cost associated with welding the intermediate shroud.

Grade 80 steel has the potential to be used in a wide range of fabricated steel centrifugal impellers to increase the maximum speed attainable without the need for an intermediate shroud. The elimination of an intermediate shroud improves aerodynamic efficiency by between 2% and 6%. The actual improvement is dependent on the impeller geometry before and after the intermediate shroud is removed. While in some cases, the improvement may be nominal, there will be cases where the improvement could provide significant savings in operating costs or potentially enable motor size to be reduced.

For the specific impeller considered, the authors concluded that manufacturing cost savings and the aerodynamic improvements made worthwhile elimination of the intermediate shroud. To accommodate the removal of the intermediate shroud, a switch from Grade 50 to Grade 80 steel proved to be an effective solution for the constant duty point studied.

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