RETROFIT AND RE-DESIGN OF A FOUR-STAGE CENTRIFUGAL COMPRESSOR

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ABSTRACT

Retrofit and redesign of multistage centrifugal compressors to meet the new operation requirements is highly demanded by many users. However, end users usually do not have full information of the compressor which prevents them to do the redesign/retrofit design. This paper presents comprehensive and convenient steps to redesign a four-stage centrifugal compressor with incomplete information to increase the volume flow rate by 20% while keeping diaphragm dimension constraints and applying new intercoolers. The original centrifugal compressor performance was first validated and analyzed by extracting the geometry parameters from CAD models, assembling the flow path and performing meanline calculation with AxSTREAM software suite. The effects of unknown data of the compressor such as the vaned diffuser of the first stage and the new heat exchanger on the compressor performance were studied. Performances of the recovered flow path were in good agreement with the provided performance data. From the meanline/streamline results, losses in each impeller were identified and guided redesign direction. Then the impellers were redesigned and 3D blades were profiled in AxSTREAM and stress analysis was performed to make sure blade stresses were under the material limit. The meanline results of the new design indicated that the mass flow rate increased by 20% and the power increased by about 15% at design point resulting in a more efficient design than the original compressor.

INTRODUCTION

Centrifugal compressor end users such as from process, petrochemical and air separation industries very often need to rerate compressors in order to increase the plant production, change fluid, etc. Current compressor geometry limitations and constraints should be considered for the rerate. James, Matthews, & Eads [1] gave general guidelines for the factors that need to be considered to purchase a new compressor or rerate an existing one. There are some options that users can modify in the compressor such as what Ludtke [2] mentioned; using wider impellers, increasing the rotational speed or decreasing the inlet temprature. The above literatures provide direction to the rerate process but no detail steps were introduced. Meanwhile, from the end user perpective, they usually do not have enough detail data of the compressor which prevent them from further analyzing the compressor performance and rerating it. This paper demonstrates the detailed steps of rerating a four-stage centrifugal compressor using the meanline method implemented in the commercial software AxSTREAM. The centrifugal compressor module of AxSTREAM was describled in Moroz, Govoruschenko, Pagur, & Romanenko [3].

The compressor that was retrofit and re-designed in this paper was a four-stage centrifugal compressor. The only data of the original compressor from the user were the CAD models reconstructed from laser scan, cross section drawing with limited estimated dimensions, labyrinth seal sketch, inlet and outlet temperature, pressure drop of intercooler and original performance map. In the original compressor were inlet guide vanes (IGV) and a vaned diffuser at the first stage. However, no data was provided at the beginning of the project for these elements. The goal was to increase the volume flow rate by 20% while keeping the original casing, diaphragm and shaft, and/or applying new intercoolers. This means that only impeller blade inlet/outlet angles and blade shape can be modified for this case since the inlet and outlet diameter of the impeller should be kept the same.

The performance map of the original compressor with the existing intercoolers was analyzed, validated and recovered with the aforementioned, limited information first. Then further redesign process started from the benchmark results. After analyzing the original compressor performance, the impeller was modified based on the meanline results and the performance of the new compressor was re-calculated to check the improvement. The detailed steps of the rerate process were:

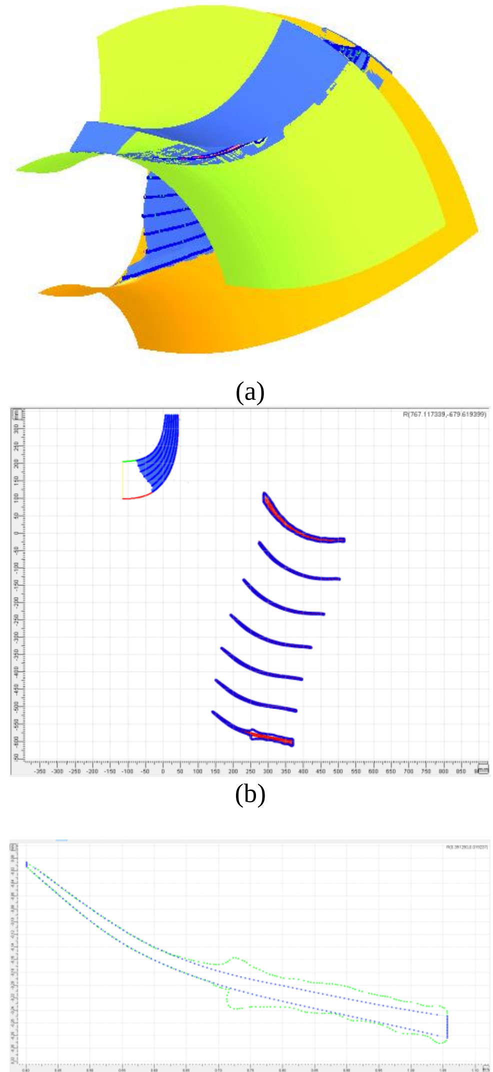
- Extracting impeller geometry parameters from scanned CAD models
- Assembling the flow path and evaluating the effect of the unknown data (IGV and vaned diffuser) on the compressor performance
- Analyzing the effect of the existing and the new intercoolers on the compressor performance
- Modifying the impeller and re-calculating the performance
- Stress analysis

NOMENCLATURE

MFR	--	mass flow rate
B	--	local blade angle
θ	--	blade wrap angle
σ	--	heat exchanger coefficient
ω	--	heat exchanger pressure loss coefficient
Δp	--	gas pressure drop in the intercooler
P_{in}	--	gas total pressure at intercooler inlet
T_{in}	--	gas total temperature at intercooler inlet
T_{out}	--	gas total temperature at intercooler outlet
$Thrf$	--	intercooler inlet water temperature

EXTRACTING GEOMETRY PARAMETERS

The impeller geometry parameters were first extracted by the AxSTREAM tool AxSLICE. The CAD models were reconstructed from scanned points. Because of the welds the surface close to hub and shroud was not smooth, manual smoothing was applied to those sliced, non-smooth sections near the hub or shroud, shown in Figure 1. The smoothing process was performed by applying splines to estimate the probable location of the profile points.



(a)

(b)

(c)

Figure 1 Extracting geometry in AxSLICE (a) hub and shroud definition (b) slicing sections (c) manual smoothing

ASSEMBLING THE FLOW PATH

After slicing the four stages impeller CAD models, the blade profiles can be imported into AxSTREAM. The dimensions of the volutes and vaneless diffusers were estimated from the cross-section drawing.

From this drawing, inlet guide vanes (IGV) were applied to this compressor and the diffuser of 1st stage was taken as a vaned diffuser. However, the data of these elements was missing. The IGVs were neglected for the analysis by assuming that these were fully open at design point. The vaned diffuser of the first stage was assumed as prismatic blade. Since its geometry was not known, the flow path was first analyzed with vaneless diffuser to obtain flow angle at impeller outlet which was estimated as the inlet metal angle of the vaned diffuser. The outlet metal angle of vaned diffuser was set by assuming a typical flow turn angle. The number of diffuser blades was unknown and was determined by checking how it affects the mass flow rate. The assembled flow path is shown in Figure 2.

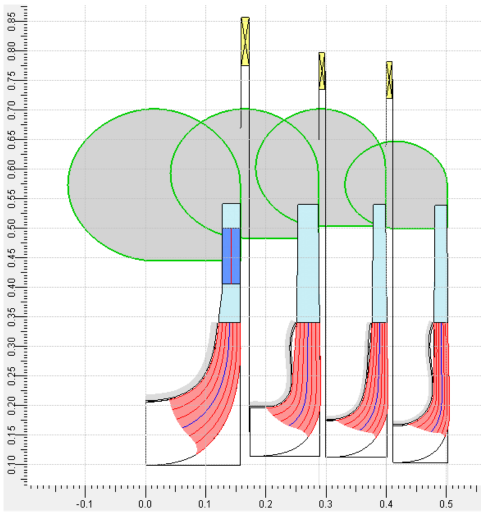


Figure 2 Assembled flow path

Design point inlet total pressure and temperature, outlet total temperature and shaft rotational speed were taken from the provided performance map and used for meanline calculation boundary conditions according to Table 1.

Table 1 boundary conditions for meanline calculation

Pt_in	total pressure at inlet	bar	0.981
Tt_in	total temperature at inlet	°C	20
Pt_out	total pressure at outlet	bar	9.81
	flow angle at inlet	deg	90
n_	shaft rotational speed	rpm	8856

The total pressure loss factor, heat exchanger coefficient and cooling fluid of the intercooler were the parameters used to investigate the intercooler for meanline analysis.

ANALYZING THE ORIGINAL COMPRESSOR PERFORMANCE

EFFECT OF VANED DIFFUSER

Since the vaned diffuser data was not provided, the geometry needed to be estimated and its effect on the performance should be compared with original performance. The inlet metal angle was set to the flow angle of the impeller and the turning angle was set to be about 10 to 12 degrees. Then the vaned diffuser blade profile was adjusted smoothly in the AxSTREAM radial blade profiling module. The 3D view, blade angle and wrap angle distributions are shown on Figure 3.

At design point the estimated mass flow rate was 51348 kg/hour and the power between 4200 and 4300 kW from the provided performance map. This data can be used as reference to check the mass flow deviation of the estimated vaned diffuser by the method above. Figure 4 shows the MFR varies with number of blades for vaned diffuser with geometry estimated. It can be seen that the MFR increases with a decrease of the number of vaned blades which means that the throat area of the diffuser is increasing with a decreasing number of blades.

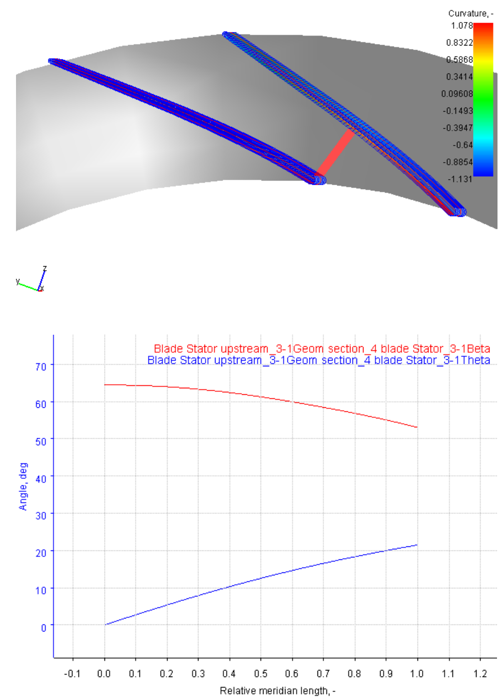


Figure 3 estimated van diffuser geometry (a) 3D view (b) Local blade angle β (top curve) and wrap angle θ (bottom curve) distribution

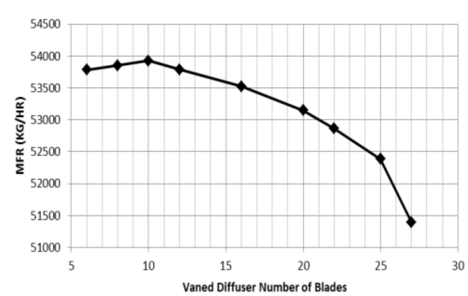


Figure 4 MFR versus vaned diffuser number of blades for old cooler and original impellers

Therefore, instead of fixing the number of vaned diffuser blades, the MFR for the old and the new heat exchanger for a different number of vaned diffuser blades was calculated in the following sections.

EFFECT OF INTERCOOLER

Using more efficient intercoolers (heat exchangers) decreases the downstream stage inlet temperature and can help increase the compressor mass flow rate and reduce its work. The main properties of intercoolers applied in AxSTREAM for meanline calculation are the total pressure loss factor ω, the heat exchanger coefficient σ and the intercooler inlet water temperature *Thrf*. The total pressure loss factor ω and the heat exchanger coefficient σ are defined as following:

$$\omega = \Delta p / P_{in} \tag{1}$$

$$\sigma = \frac{T_{in} - T_{out}}{T_{in} - Thrf} \tag{2}$$

Table 2 and Table 3 list the above properties for the old and new intercooler.

Table 2 old intercooler properties

	ω	σ	Thtf
stage 1	0.089	0.750	24.0
stage 2	0.056	0.730	26.0
stage 3	0.034	0.784	25.0

Table 3 new intercooler properties

	ω	σ	Thtf
stage 1	0.090	0.932	25.4
stage 2	0.056	0.917	27.9
stage 3	0.034	0.896	28.0

Table 4 gives the MFR for the old and the new heat exchanger for the original compressor for a different number of blades of the vaned diffuser and Figure 5 shows the MFR incremental percentage of the new heat exchanger relative to the old heat exchanger for each vaned diffuser studied. By applying the new heat exchanger, the MFR would increase but the increment would also depend on the vaned diffuser. On average, the MFR could vary 3 to 6% when the vaned diffuser number of blades is in the range of 12 to 18.

Table 4 Effects of heat exchanger on the MFR for original last stage

Number of vane blades	MFR (old cooler, kg/hr)	MFR (new cooler, kg/hr)	Increase (%)
18	53375.36	55130.64	3.3
16	53529.65	55764.99	4.2
14	53654.00	56531.90	5.4
12	53813.36	57351.02	6.6
10	53887.73	58210.54	8.0
8	53894.62	58941.90	9.4
6	53781.68	59263.35	10.2

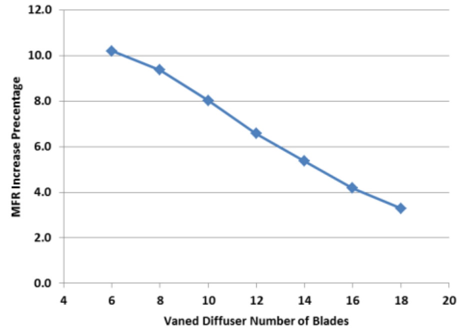
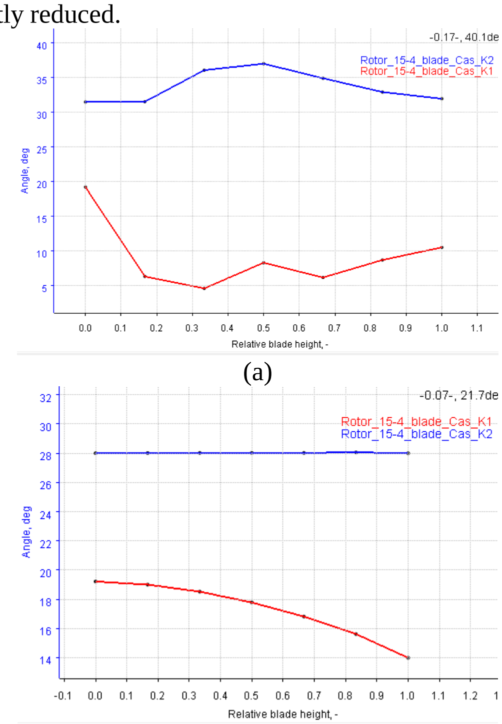


Figure 5 MFR increase percentage for new heat exchanger

REDESIGNING THE COMPRESSOR

MODIFYING THE LAST STAGE

It was found that the losses in the last stage of the original compressor were larger than in the first three stages so the last stage was modified first to see how it could reduce the losses. Figure 6 shows the inlet and outlet metal angle distribution along the blade height for the original and modified blade. The inlet and outlet metal angle adjustments were based on the flow angle that was calculated. After redistributing the metal angles and slightly changing the beta angle distribution and radius at inlet and outlet, the incidence angle at the last stage was significantly reduced.



(a)

(b)

Figure 6 Inlet (Red line, top) and Outlet (Blue line, bottom) metal angle distribution for the original blade (a) and for the modified impeller (b)

The effect of the modified stages and heat exchanger on the MFR is given in Table 5. It can be seen that the new heat exchanger still increases the MFR but the increment is lower than the one with the original modified impeller. This is because the MFR is increased for the modified last impeller relative to the original impeller (with the old intercooler.)

Table 5 Effects of heat exchanger on the MFR for modified last stage

number of vane blades	MFR (old cooler, kg/hr)	MFR (new cooler, kg/hr)	Increase (%)
14	55980.88	56868.84	1.6
12	56645.38	58071.38	2.5
10	56982.28	59375.70	4.2
8	57048.30	59891.38	5.0

Figure 7 gives the MFR for the four different cases. It can be seen that the MFR of the modified last stage increased compared to the original one with either the old or the new intercooler, and the new intercooler was also beneficial for both the original impeller and the modified impeller.

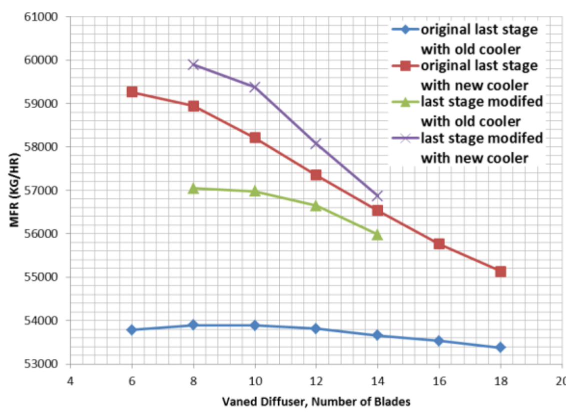


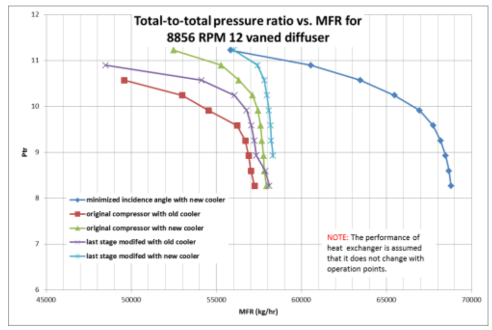
Figure 7 MFR of for old and new cooler under original and modified last stage

MINIMIZE INCIDENCE ANGLE OF FOUR STAGES

Since modifying the last stage increased the MFR, changing the incidence angle of the rest of the stages could also improve the performance. The "minimize incidence angle" feature was applied in the meanline calculation which adjusts the inlet metal angle of each stage according to the flow angles which in turn would decrease the losses and improve the performance at design point.

With the new blade inlet and outlet metal angles, the blades were profiled by redistributing the wrap angle and the local blade angle. The thickness distribution of the new blade was kept the same as for the original.

Figure 8 to Figure 10 give the maps of pressure ratio and power with mass flow rate at 12, 10 and 14 number of blades for the vaned diffuser for the original compressor, the last stage modified and the minimized incidence angle geometry with the old and the new heat exchanger at design rotational speed of 8856 rpm. For these performance maps calculation assumption that the performance of the heat exchanger does not change with the operation point was made since it was unknown at off-design points. From these results, the mass flow rate of the minimized incidence angle geometry increases by about 20% relative to the original compressor with old heat exchanger at design point while the power increases by about 15%. Moreover, the pressure ratio curve becomes flatter than the original compressor.



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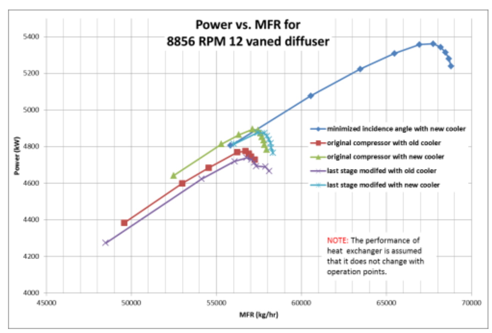

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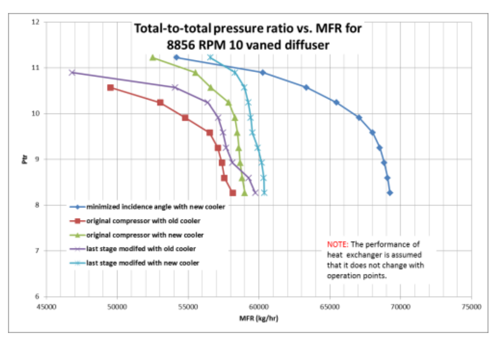
Figure 8 (a) pressure ratio and (b) power with MFR for 12 vaned diffuser blades



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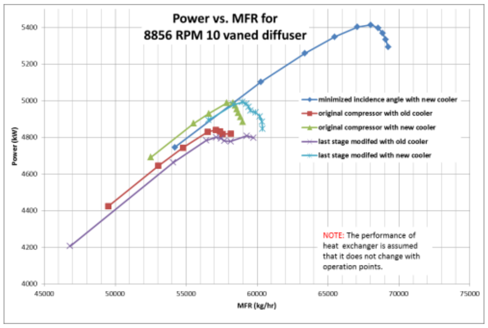

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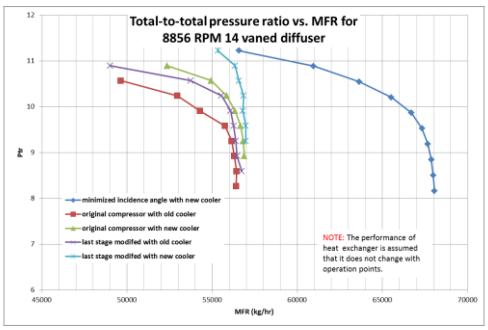
Figure 9 (a) pressure ratio and (b) power with MFR for 10 vaned diffuser



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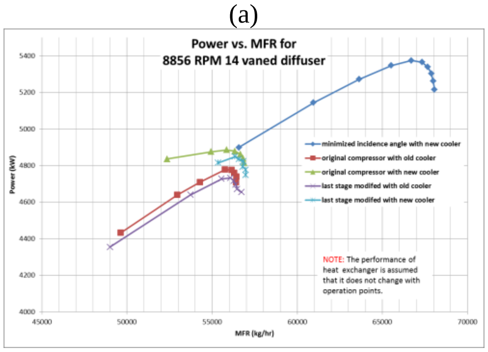


(b)

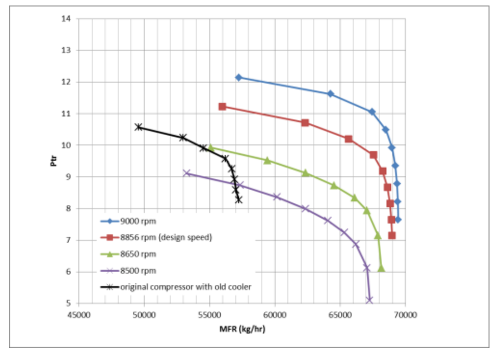
Figure 10 (a) pressure ratio and (b) power with MFR for 14 vaned diffuser blades

Before the number of blades of the vaned diffuser was determined, it was assumed as 12 and its effect on the MFR and power at design point were checked for the minimized incidence angle compressor. Table 6 gives the MFR and the power for the number of blades of the vaned diffuser from 10 to 14 for the minimized incidence angle geometry at design point. The difference of MFR for 10 and 14 is 0.035% and 0.052% relative to 12 vaned blades, and the power difference is 0.025% and 0.037% respectively. So after the incidence angle was minimized, the range of 10 to 14 number of vaned diffuser the performance is found to be very close.

As the power is increased by about 15% at design point when the MFR is increased by about 20% relative to the original compressor, it was not sure whether it exceeded the power limit of the driver for the incidence angle minimized compressor. Then speedlines of the incidence angle minimized compressor were calculated under the new heat exchanger also by assuming the heat exchanger performance was constant.

Table 6 Number of vaned blades effect on the MFR and power for minimized incidence angle geometry

Number of vane blades	12	10	14
MFR (kg/hr)	66606.81	66630.19	66572.15
Power (kW)	5375.14	5376.49	5373.15



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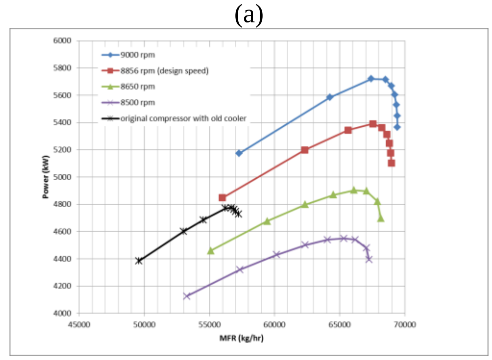

(b)

Figure 11 Off-design performance (a) pressure ratio (b) power for minimized incidence angle compressor

It can be seen from Figure 11 that the minimized incidence angle compressor can be operated at the lower speed to be in the power range of the original compressor with the increased mass flow but the pressure ratio is less than for the original compressor.

STRESS ANALYSIS

With the newly designed blades from the above procedures, the impeller was created by adding the hub and shroud disk which were adopted from the original impeller in CAD software. In order to check the stress distribution, 3D FEA analysis was performed for the four new impellers. The material for the impeller is 31CrMoV9, the elasticity module is 210,000

MPa, density is 7730 kg/m^3 and yield stress is about 860 MPa. Stress analysis was conducted for the four new impellers at 100% and 115% design rotational speed. Initially, the radius of the fillet was set as 2 mm for all four impellers.

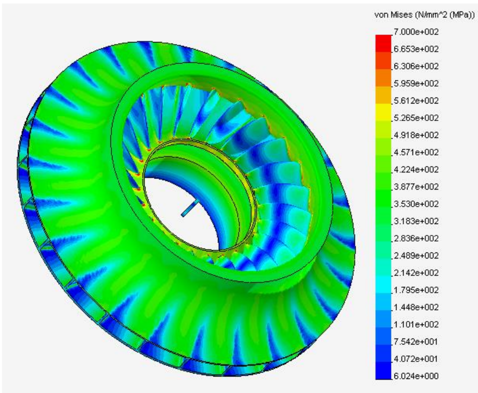

Figure 12 first stage overall stress distribution at 100% rotational speed

Overall stress distribution of the first stage impeller is shown in Figure 12. The stress was concentrated at the leading edge close to the hub and shroud. The stress distribution was similar for the other three stages. Table 7 lists the maximum stress value at leading edge near hub and shroud for each stage.

Table 7 Maximum stress value close to hub and shroud at leading edge at 100% rotational speed

	Max stress near LE close to hub (MPa)	Max stress near LE close to shroud (MPa)
stage 1	697.9	1065
stage 2	773.5	817.1
stage 3	640.8	714.5
stage 4	788.2	804.0

The impeller deformation is shown in Figure 13. Such stress and deformation distribution is caused by the relative shift of disks which contain most part of the impeller mass.

The front part of blade was squeezed between the disks and deformed into a wave shape. The bend of the blade LE causes the increased stress near hub and shroud. The regions of increased stress are small.

Since the disks, the number of blades and blade thickness were taken from the original impeller, changing fillet radius was an option to change stress distribution and reduce the maximum stress near the leading edge.

After several stress analysis iterations, radius of fillets between the blade and disks was found to be equal to 6 mm at the LE and smoothly transitions to 4 mm downstream for stage 1 and stage 4 while 6 mm is used for stage 2.

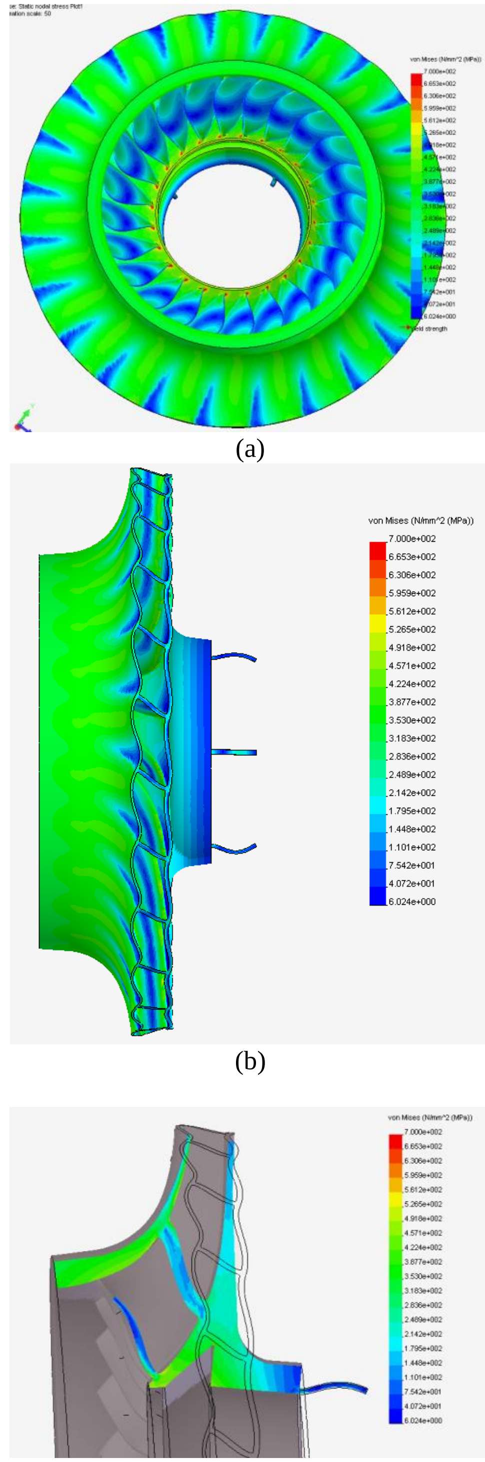

Figure 13 impeller deformed shape view (a) front (b) side and (c) cross section

With the above modifications, the maximum stress of the four impellers is below the material yield strength. The stress of the impellers with 115% of design speed was also checked and stress distribution is similar as before, see Figure 14.

The maximum stress values are given in Table 8. For the current material, the impeller should not operate at speeds well beyond the design rotation speed.

Table 8 Maximum stress value close to hub and shroud at leading edge with 115% rotational speed for modified impeller

	Max stress near LE close to hub (MPa)	Max stress near LE close to shroud (MPa)
stage 1	934.0	848.0
stage 2	959.7	1049.0
stage 3	944.7	963.2
stage 4	931.5	1018

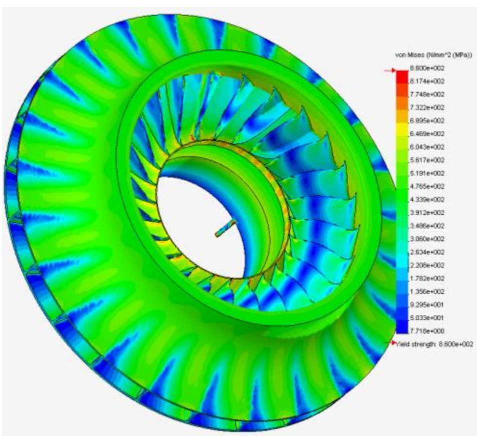

Figure 14 modified first stage overall stress distribution at 115% rotational speed

CONCLUSIONS

A four-stage centrifugal compressor with limited data from the user was retrofitted and redesigned in order to increase the volume flow rate by 20%. Detail rerate steps were described in this paper. The impeller geometry parameters were extracted from a scanned CAD model and the effect of the vaned diffuser and the heat exchanger effect on performance were studied. The four impellers were redesigned by keeping the inlet and outlet diameter constraints. Stress analysis at 100% rotational speed was conducted and the fillet radius distribution of impellers was modified to reduce the maximum stress under the selected material yield strength. The modified impeller stress distribution at 115% rotational speed was also checked. The conclusions of this rerate process were:

(1) When changing the heat exchanger from old to new, the MFR increased by about 5.4%.

(2) Redesign of the impellers of the 4 stages could significantly increase the MFR. Using modern technology, it was possible to design high efficient blading for a 40 year old compressor thus achieving project goal of increasing MFR by 20% while the power increase was no more than 15%.

Performance test of the new impellers is undergoing at this point. Comparison between test data and current calculation results will be presented in a future work.

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