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Research Article

Structural and Thermal Analyses of F Class Gas Turbine Compressor Blade

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ABSTRACT

Gas turbines are used extensively for aircraft propulsion, land-based power generation, and industrial applications. They are consist of many parts. One of the important parts is the blade and disc. Blade and disc are individual components that make up the compressor section. That is why all effects on these components directly affect the unit itself. For that reason engineering calculations on such critical parts are important. Some power plants and gas turbines failed regarding wrong engineering calculations. One of them happened two years ago on GE (General Electric) 9FB gas turbine in Turkey. Some turbine blades have broken and created high-cost damage on the unit. Some engineering calculations have been failed regarding side running conditions. For that reason, this study has been performed to protect and verify the engineering value of AEN 94.3A F-type gas turbine side running conditions. In this study, the structural and thermal analysis of the final stage compressor rotor blade and disc, which are currently used on-site, was performed by using the ANSYS program. Initially, basic knowledge of blade and disc design drawings were reviewed and design steps of the existing 3D (Three dimensions) blade and disc configuration were described. For that reason, a 3D model of the existing compressor blade and the disc has been done in the SolidWorks design program. Later on, this model was transferred to the ANSYS program and analyzed. In the analysis, the parameters formed in the blade geometry were determined. By creating a design geometry with the selected parameters, the stress of the existing blade under operating conditions was examined. All external parameters in this study were taken from an F-type gas turbine operation under real field conditions. The stresses obtained from different regions on the blade were examined. After the thermal and structural analysis, obtained results have been compared with side engineering measurements. By comparison, it was observed and verified that the unit normal side running condition is safe and engineering calculations are sufficient.

In this study before starting the analysis, general information about gas turbines has been presented. Gas turbines have been briefly introduced and then, the basic operation principles of turbines have been explained. As it is known, gas turbines are shaped based on thermodynamic principles. In this study, engineering thermodynamics in gas turbines is briefly explained as well.

Keywords: *Gas turbine, Structural and thermal analysis, Turbine blade,*

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F Sınıfı Gaz Türbini Kompresor Kanadının Yapısal ve Termal Analizinin Yapılması

ÖZ

Gaz türbinleri, uçak motorlari, zemin esasli enerji üretimi ve endüstriyel uygulamalar için yaygın olarak kullanılmaktadır. Birçok parçadan oluşurlar. Önemli parçalardan biri kanat ve disktir. Kanat ve disk, kompresör bölümünü oluşturan ayrı bileşenlerdir. Bu nedenle bu bileşenler üzerindeki tüm etkiler doğrudan ünitenin kendisini etkiler. Bundan dolayi bu tür kritik parçalar üzerinde mühendislik hesaplamaları önemlidir. Bazı santrallerde gaz türbinlerinde yanlış mühendislik hesaplarından dolayı arızalar meydana gelmektedir. Bu arızalardan biri iki yıl önce Türkiye'de GE (General Electric) 9FB gaz türbininde yaşandı. Bazı türbin kanatları kırılmış ve ünitede yüksek maliyetli hasar meydana gelmiştir. Bazı mühendislik hesaplamaları saha çalışma koşullarına uymadığından dolayı başarısız oldu. Bu çalışma, AEN 94.3A F tipi gaz türbininin saha çalışma mühendislik değerlerini kontrol etmek ve doğrulamak için yapılmıştır. Bu çalışmada, halihazırda sahada kullanılan son kademe kompresör rotor kanadı ve diskinin yapısal ve termal analizi ANSYS programı kullanılarak yapılmıştır. İlk olarak, kanat ve disk tasarım çizimlerinin temel bilgileri gözden geçirilmiş ve 3D (Üç boyutlu) konfigürasyonunun tasarımları yapılmıştır. SolidWorks tasarım programında mevcut kompresör kanadı ve diskinin 3 boyutlu modeli yapılmıştır. Daha sonra bu model ANSYS programına aktarılarak analiz edilmiştir. Analizde kanat geometrisinde oluşan parametreler incelenmiştir. Seçilen parametreler ile mevcut kanadın ve diskin işletme koşullarındaki gerilmeleri incelenmiştir. Bu çalışmadaki tüm çalışma parametreleri, gerçek saha koşullarında çalışan F tipi gaz türbini çalışma değerlerinden alınmıştır. Kanat üzerinde farklı bölgelerden elde edilen gerilmeler incelenmiştir. Elde edilen bu değerler sahada ölçülen mühendislik değerleriyle karşılaştırılmıştır. Bu karşılaştırma sonucunda ünitenin normal saha çalışma koşulunun güvenli olduğu ve mühendislik hesaplarının yeterli olduğu gözlemlenmiş ve doğrulanmıştır.

Bu çalışmada analize başlamadan önce gaz türbinleri hakkında genel bilgiler verilmiştir. Gaz türbinleri kısaca tanıtılmış, ardından türbinlerin temel çalışma prensipleri anlatılmıştır. Bilindiği gibi gaz türbinleri termodinamik esaslara göre şekillendirilmektedir. Bu çalışmada gaz türbinlerinde mühendislik termodinamiği de kısaca anlatılmıştır.

Anahtar sözcükler: Gaz türbini, Yapısal ve termal analiz, Türbin kanadı

I. INTRODUCTION

Gas turbines are an indispensable part of the modern power generation systems that are used to generate energy. They obtain their power by utilizing the energy of burnt gases and the air which is at high temperature and pressure by expanding through the several stages of fixed and rotating blades. Fixed and rotating blades increase the pressure and temperature. By increasing the pressure and temperature and as well as speed a centrifugal or axial compressor is needed. These types of compressors are suffıcıent for such an individual process. The last stage of a turbine is called as hot section side which the hot air from the combustion chamber is faced. The turbine hot section side is coupled to the turbine shaft. After the compression, the working hot fluid expands in a turbine, then assuming that there are no losses in each component, the power developed by the turbine can be increased by increasing the volume of working fluid at constant pressure or increasing the pressure at constant volume. If the working fluid passes from the combustion chamber, the pressure and temperature rise. Working fluid converts mechanical energy to electric energy by rotating coupled generators. The exhaust gas that loses its energy on the turbine can be sent to the atmosphere or can be used for heating and boiling water for the steam turbine. This system is called a combined cycle. The others which exhaust the gas to atmosphere is called as a simple cycle [1-4].

There are many manufacturers of these machines, as well as many types. The most important manufacturers are General Electric, Siemens, Mitsubishi, Ansaldo, and Alstom. These manufacturers produce gas turbines with various features according to their power and capacity. The most produced types are the heavy-duty (high capacity) ones called F-type. This type of unit is used in industry for producing electricity.

The first studies on gas turbines were done by Rowen and Undrill from General Electric. In these studies, basic control systems and modeling of combined cycle power plants were discussed. Later on, these studies have been developed by various companies and have survived to the present day. Today, the development studies of these processes are carried on further [5-6].

The reason why gas turbines are widely used in national networks is that the frequency of electricity can be kept constant. Frequency control is difficult in electrical energy obtained from other power plants like hydraulic, wind, solar, etc. For that reason, gas turbines are widely used all over the world.

Failure of a gas turbine can create a significant negative effect on power generation. For that reason, the engineering calculation should be verified regarding side running conditions. If side values do not inspect regarding design calculation the unit may fail. A couple of years ago GE 9FB gas turbines have been failed in some sites in different countries. When GE did root cause analyses they found that material and engineering calculations for stress analyses and material were wrong regarding side running conditions. Most of the turbine blades have been broken from the connection point of the disc. This issue created high-cost damage to the unit. This study is about the verification of engineering calculation AEN 94.3A Ansaldo F-type gas turbine. The calculation has been done on the last stage compressor blade and disc. Airflow reaches the maximum level of temperature, pressure, and vibration at the last stage. That is why some engineering calculation has been performed by using real side conditions.

Blade and disc are the heart of gas turbines engines and serve as a medium of transfer of energy from the gases to the turbine rotor. Blades are rounded around the disc shown in Figure 1. Each stage of the compressor has a different size and shape of blade and disc. All discs tidied with each other via tie roads. These tie roads are strong and long enough for connecting all discs. AEN 94.3A gas turbine consists of 15 stages. All stages can be recoupled from each other by losing tie roads. This process creates great benefits for maintenance activities. Disc center shown in Figures 3,4, and 5 has a hole for using cold air regarding thermal expansions. Cooled air passes from this hole for cooling the turbine blades as well as the disc itself [2-6].

In the analysis of gas turbine blade and disc, discretizing a freestanding blade and disc and using appropriate element relation is more advantageous than the continuum approach, in that, it is simple to carry out the analytical work. During this study structural analyses have been performed on the blade and disc, and thermal analyses have been performed on the disc. The center of the disc is a hole and cooling air passes from here. That is why two different temperature is affecting the disc. The analyses are made to determine the positive or negative situation that may occur in the part due to external factors such as force, angular or radial velocity, and temperature acting on the part. ANSYS program is used for thermal and structural analyses [7-18].

When the literature is scanned, it is possible to come across some studies on structural analysis and shape efficiency in gas turbines. In these studies, researchers worked on many different analysis methods. Some of them used the Matrix Method, and some of them used the Finite Difference Method and the Finite Element Method (FEM) [16-21]. In their study, Meng and Zhangqi tried an analytical procedure by making simplifications in large-scale wind turbine structure. In this context, they used the matrix method and developed computer code to complete the details of the frequencies and mode shapes of the vibration beam. Finally, they stated that the obtained FEM analysis results were in agreement with the experimental study [19]. Krishnakanth et al. carried out the design and structural and thermal analysis of the gas turbine blade in their study. They used Ansys software as the finite element software and determined the most suitable one among 3 different materials by performing steady-state analysis. They also stated that they observed maximum elongation at the tip of the blade and minimum elongation at the root of the blade [20]. In another study, a detailed report on the developments in the design and structural analysis of commercial jet engine fan blades is presented. It also includes the main technical problems related to fan blades in terms of high structural integrity, stability and durability, and solutions to these problems [21]. In another study, Kauss et al tried to predict the stress-strain state of a turbine blade model by performing thermal analysis and structural mechanical analysis. In the results, the two molybdenum-based alloys Mo–17.5Si–8B and Mo–9Si–8B were compared with the nickel-based superalloy CMSX-4, and they indicated that the molybdenum-based alloys showed much better resistance to deformation. As a result, it was emphasized that Mo-9Si-8B alloy is a very advantageous material for high pressure turbine blades [22]. Recently, the most common and most useful program for the finite element method is ANSYS software. This program is calculating all requests easily and gives the exact solution. That is why it is used by many design companies and gas turbine manufacturers [16-22]. This study has been performed to protect and verify the engineering value of AEN 94.3A F-type gas turbine side running conditions. In this study, the structural and thermal analysis of the final stage compressor rotor blade and disc, which are currently used on-site, was performed by using the ANSYS program.

II. DETERMINATION OF STRUCTURAL ANALYSIS METHOD

The turbine compressor usually sits at the front of the engine. There are two main types of compressors, the centrifugal compressor, and the axial compressors. Both of them draw air and compress it before it is fed into the combustion chamber. The compressor consists of discs and blades. Discs are compelled by each other via tied roads and blades are fixed on the disc for coupling together. Both rotate like a shaft.

Discs and blades faced high temperature, pressure, and speed. At the last stage of the compressor, these values reach the maximum level. For that reason, most of the failure gets at the last stage. That is why this study focused on the last stage analyses.

Before starting analyses, blade and disc modals have been inspected for the 3D (three dimensions) modal. Blade and disc current modal profiles are generated by using the SolidWorks CAD program. Key points are joined by drawing Spline curves used to obtain a smooth contour. The contour (2D) models are then converted into an area and then volume (3D) models were generated by extrusion, see below in Figure 1.

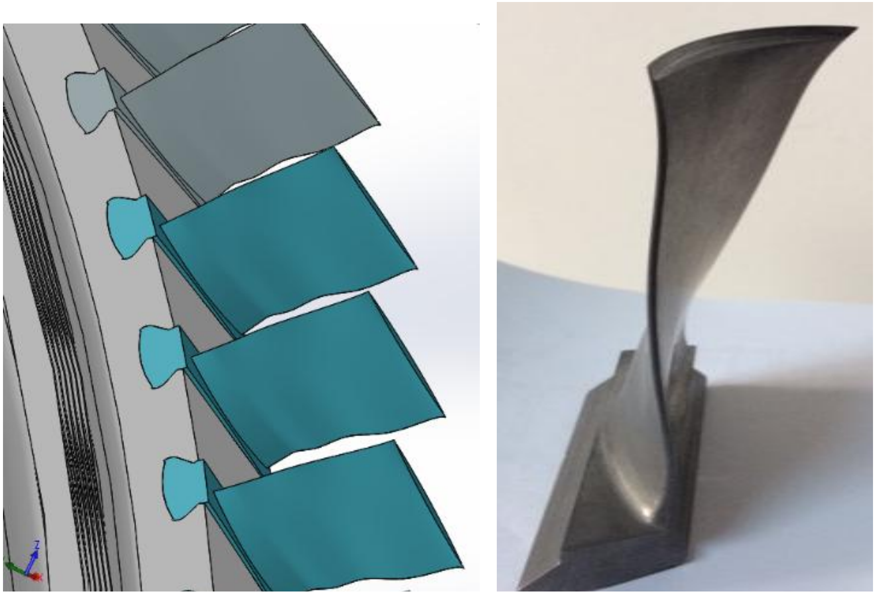

Figure 1. *The geometry of the blade and disc*

During the present paper, the effect of increasing the complexity and surface area of the air passages has also been studied. The thermal and structural analysis has been performed to investigate the effect of real side conditions on the blade and disc. The study has been performed assuming steady-state conditions by using ANSYS software.

The methodology used for performing the study followed a three-stage path. Design, thermal analysis of disc, and structural analysis of disc and blade. As it is seen in Figure 2, before starting analysis all side values of rotating direction applied force, temperature, and fixing points have been defined on the disc and blade

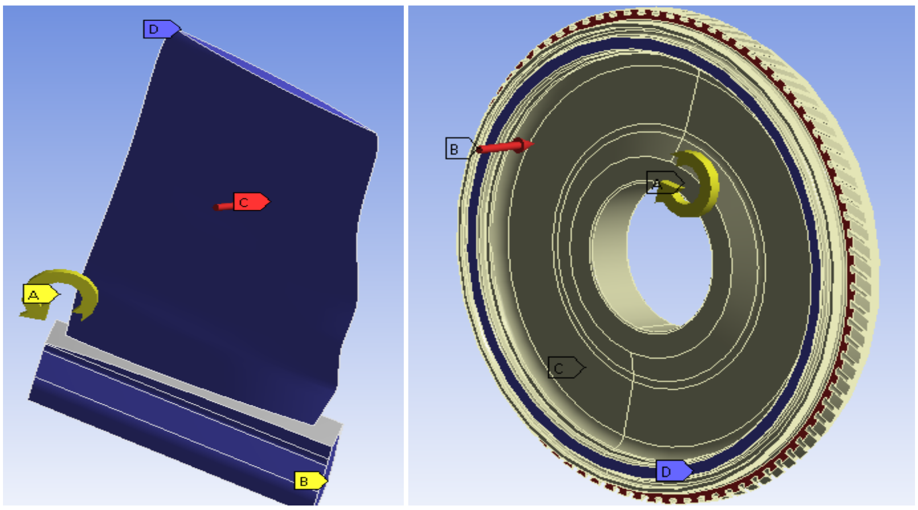

Figure 2. *Applied side values on the and blade disc*

The thermal analysis has been performed on the disc because cooling air passes from the center of the disc and on the top of the disc hot air follows. This means that there are temperature differences. Thermal analysis has been performed using the Steady State module on the ANSYS program. To initiate the method fine mesh has been generated on the 3 Dimensional models. On the other hand, thermal analysis has not been performed on the blade because the following air temperature is the same at every surface of the blade.

The structural analysis has been performed on the disc and blade by using the Static Structural modal of the ANSYS program. The speed of the unit is 3000 rpm and generally, all F-Type units have the same speed and acting force is 22 bar, the temperature is about 500^0C. In Figure 2, the output effect of the unit has been shown on the surface of the blade and disc. After analysis, the result of both thermal and structural analysis have been compared with the actual measurement of the side-real calculations. These processes have been verified that the calculation result is convenient for the unit to run in safe conditions.

A.1 Determination of The Field Values

The values, which are used during the analysis, are taken from the DCS (Discrimental Control System) system while the unit is operating at different MWs (Mega Watts). The DCS system is the Siemens T3000 facility operating program, and in this program, the values are taken from the instruments in the field and transferred directly to the program in the control room. Operators in the control room monitor these values instantly. Generally, temperature, pressure, power, valve opening positions, flow values , and other parameters from the field are transferred to the DCS system digitally. DCS screenshot can be seen in Figure 3. As it is seen, all values related to the unit can be monitored and controlled from the DCS program. In this way, the control of the facility becomes very easy and safe.

The values that we need from DCS, have been recorded for about one month. These values have been taken under different power for our analysis safety. In Table 1, below, field values have been followed and recorded for about a month. On the table, thermodynamic real values have been calculated theoretically by using the Excell program.

During the analysis, the compressor outlet temperature was considered as the average while performing the thermal analysis of the last stage compressor disc. In the structural analysis, the temperature, speed, and pressure parameters were discussed and maximum values have been taken for safety.

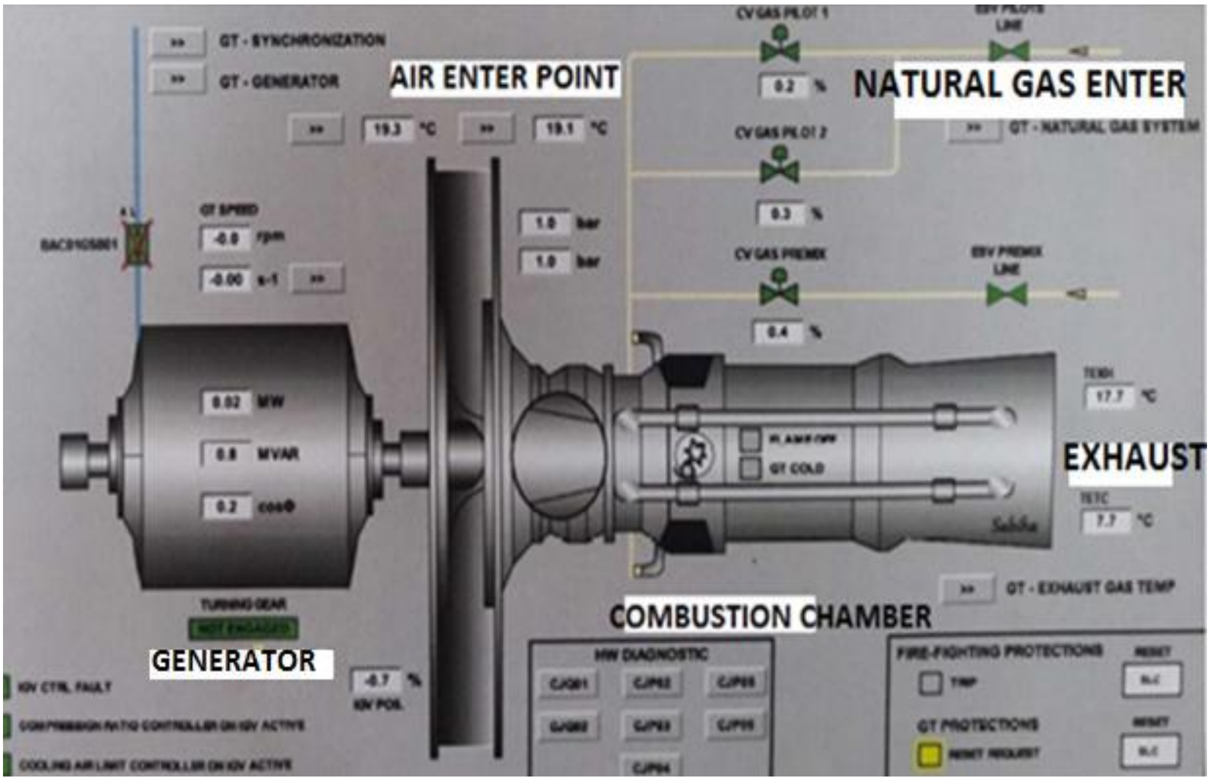

***Figure 3**. Turbine DCS view*

A couple of years ago GE 9FB F-type gas turbine has been failed in some plants. Unfortunately, this failure has created great cost loss for users and GE. When GE did root cause analyses, it was observed that some engineering calculations on material and thermal expansion of the part did not match regarding the side-real running conditions. Figure 4 shows the broken blades from the turbine side. If the thermal and structural extensions are not the same with the cases means that blades may touch the cases and create tip grinding and after a while break on the proper zone. That is why this study has been performed to make sure that AEN 94.3A F-type gas turbine side running values are safe, reliable related to design parameters. During the analysis, it was observed that the unit parameters are sufficient enough for side condition

***Figure 4**. Broken blades on F-type unit*

Table 1. Gas Turbine Thermodynamics Value

F CLASS GAS TURBINE THERMODYNAMIC VALUEs DURING NORMAL RUNNING CONDITION

Test No	Relative Humidity %	Envirement Tempreture T1 °C	Compressor Exit TemperatureT2 °C	Combustion Chamber Temperature T3 °C	Exhaust Temperature T4 °C	Envirement Pressure P1 bar	Compressor Pressure P2	Combustion Chamber Exit Pressure P3	Exhaust Pressure P4	Envirement Enthalpy h1	Compressor Enthalpy h2	Combustion Chamber Enthalpy h3	Exhaust Enthalpy h4	Work done in Compressor W12	Work done in Combustion Chamber	Work done in Turbine W34	Net Work Wnet	Efficiency Wnet/Q23	Pressure Ratio r	Power P(MW)	Net Power after Correction(MW)
1	40.84	1.53	362.12	1212	557	0.962	18.24	0.239	0.434	2	370.42	1345.14	583.25	-372.42	974.72	761.89	1134.31	1.16	0.55	256	263
2	48.43	1.66	367	1227	556	0.963	18.51	0.239	0.376	2	373.15	1362.14	580.47	-375.15	988.99	781.67	1156.82	1.17	0.64	261	268
3	47.62	5.67	337	1236	540	0.96	16.32	0.199	0.266	5.08	345.04	1373.04	563.11	-350.12	1028	809.93	1160.05	1.13	0.75	216	219
4	93.36	5.18	371.9	1234	561	0.956	17.05	0.219	0.311	5.01	380.04	1371.06	586.13	-385.05	991.02	784.93	1169.98	1.18	0.70	242	244
5	66.73	2.16	382.97	1228	564	0.977	18.8	0.252	0.417	2	393.04	1362.04	589.14	-395.04	969	772.9	1167.94	1.21	0.60	260	273
6	70.59	3.56	386.04	1215	559	0.982	16.8	0.222	0.365	3.12	397.01	1348.2	584.89	-400.13	951.19	763.31	1163.44	1.22	0.61	228	239
7	52.94	3.88	404	1237	559	0.978	18.66	0.224	0.417	4.01	415.03	1376.04	584.89	-419.04	961.01	791.15	1210.19	1.26	0.54	265	276
8	74.59	8.78	359.83	1232	568	0.972	16.09	0.164	0.162	8.15	367.03	1371.4	594.86	-375.18	1004.37	776.54	1151.72	1.15	1.01	216	221
9	36.27	17.58	391.56	1240	574	0.97	15.13	0.191	0.252	17.09	403.05	1378	600.89	-420.14	974.95	777.11	1197.25	1.23	0.76	207	209
10	80.89	9.3	401.71	1231	567	0.967	16.32	0.216	0.412	9.02	404.01	1361.01	595	-413.03	957	766.01	1179.04	1.23	0.52	226	230
11	38.15	18.56	363.66	1227	576	0.966	12.09	0.165	0.189	18.42	373.05	1361.14	603	-391.47	988.09	758.14	1149.61	1.16	0.87	142	146
12	74.73	7.25	416.05	1225	566	0.969	18.61	0.215	0.364	7.05	430.04	1360	591.26	-437.09	929.96	768.74	1205.83	1.30	0.59	266	274
13	88.25	4.75	414.43	1230	565	0.965	18.77	0.213	0.37	5.02	429.02	1366.66	590.67	-434.04	937.64	775.99	1210.03	1.29	0.58	270	278
14	83.36	9.37	369.89	1231	520	0.963	12.42	0.194	0.261	9.12	377.06	1367	541.16	-386.18	989.94	825.84	1212.02	1.22	0.74	161	162
15	61.22	9.35	372.85	1227	567	0.968	14.5	0.183	0.229	9.1	380.04	1361.04	595	-389.14	981	766.04	1155.18	1.18	0.80	191	195
16	32.74	16.11	390.55	1229	572	0.969	14.85	0.182	0.232	16.5	401.06	1365	598.34	-417.56	963.94	766.66	1184.22	1.23	0.78	202	203
17	71.12	9.34	379.05	1230	567	0.976	15.42	0.19	0.237	9.08	389.05	1366	595	-398.13	976.95	771	1169.13	1.20	0.80	208	214
18	84.24	5.63	390.34	1237	562	0.973	17	0.206	0.298	5.23	402.05	1374.44	587.36	-407.28	972.39	787.08	1194.36	1.23	0.69	238	247
19	42.48	16.53	391.42	1234	573	0.973	15.05	0.177	0.225	16.35	401.08	1371.99	598	-417.43	970.91	773.99	1191.42	1.23	0.79	206	206
20	90.23	7.79	390.56	1239	565	0.973	16.65	0.214	0.316	8.05	400.09	1377.87	590.67	-408.14	977.78	787.2	1195.34	1.22	0.68	232	239
21	55.61	13.35	394.88	1224	571	0.973	15.88	0.215	0.312	13.25	405.01	1358	597.2	-418.26	952.99	760.8	1179.06	1.24	0.69	221	224
22	50.31	16.64	383.58	1226	574	0.971	14.45	0.189	0.235	16.08	392.04	1361	598.87	-408.12	968.96	762.13	1170.25	1.21	0.80	193	194
23	41.48	11.52	388.38	1228	568	0.967	15.73	0.189	0.245	11.25	398.04	1364.74	593.56	-409.29	966.7	771.18	1180.47	1.22	0.77	214	218
24	54.84	13.37	414.48	1234	572	0.957	17.48	0.213	0.316	13.2	422.06	1371.56	598.64	-435.26	949.5	772.92	1208.18	1.27	0.67	254	254
25	38.17	16.53	411.29	1236	573	0.96	16.88	0.187	0.229	16.08	423.01	1373	599.56	-439.09	949.99	773.44	1212.53	1.28	0.82	242	246
26	53.16	9.2	394.54	1227	569	0.967	17.62	0.165	0.187	9.05	403.15	1364.74	595.74	-412.2	961.59	769	1181.2	1.23	0.88	248	254
27	53.19	11.14	396.63	1237	568	0.973	16.04	0.208	0.298	11.08	409.14	1374	593.56	-420.22	964.86	780.44	1200.66	1.24	0.70	222	228
28	33.08	17.09	408.98	1239	574	0.97	16.66	0.203	0.257	17.56	419.16	1378	600.89	-436.72	958.84	777.11	1213.83	1.27	0.79	235	236
29	23.04	24.87	423.73	1226	580	0.964	16.35	0.217	0.315	24.07	435.18	1361.75	607.27	-459.25	926.57	754.48	1213.73	1.31	0.69	229	243

A.2. Materials Used In The Blade and Disc

Gas turbine materials have developed rapidly beyond the conventional ferrous alloys consisting of steel and stainless steel of various compositions. Several types of nickel and cobalt-base alloys have been developed and widely used in blades and discs. These alloys are high resistance to temperature and corrosion. Chromium additions have been used for temperature strength and oxidation resistance. For the hot section side, a protective coating has been used to enhance hot erosion-corrosion and as well as protect from high temperature. This type of coating is called Thermal Barrier Coating (TBC).

The present paper deals with the stresses that act on the blade and disc due to high angular speeds and the second are thermal stresses that arise due to temperature differences in disc material. In the first stage of rotating the blade and disc, the temperature effect is not very high. However, as the stage progresses, temperature and other factors have great effects on the blade and disc. On the other hand, this temperature does not have a great effect on the elongation of the material due to the material structure generally. Because this zone is not a hot section side of the unit. The first stage blades have high centripetal stresses due to their relatively long length. The high pressure and temperature cause the formation of centripetal stresses at the last stage. Because the blades are short and small. Taking these effects into account, blade and disc materials are generally chosen from Titanium-alloy based materials. The turbine blade and disc are manufactured from Inconel 718 alloy. Having high mechanical strength, Inconel 718 material is also resistant to high temperatures. It has certain advantages in terms of fatigue characteristics. In general, palladium and ruthenium are added to this material for increasing its resistance to corrosion. The technical specifications of the alloy are available in Table 2.

Table 2. *Inconel 718 Alloy Material Characteristics*

Percentage of materials:	Al 0.2 - 0.8 C Max 0.08, Fe Max 17, Ni Max 50-55, S Max 0.015, Ti 0,65 – 1.15, Cr 17-21, Mo 2.8-3.3, Mn Max 0.35
Density	8,22 g/cm^3
Elastic Module	204.9 kN/ mm^2
Maximum Strength	1375 MPa
Yield Strength	725 MPa
Tensile Strength	1035 MPa
Elongation	(9 µm/m), (20°C), (9.4 µm/m), (250°C)
Specific Heat	0. 435 J/g°C
Thermal conductivity	11.4 W/mK
Melting Temperature	1370-1430 ^{0}C

A.3. Structural Analysis (ANSYS Analysis)

The determination of blade and disc geometry during the design phase is based on much-repeated analysis. It is very difficult to perform these analyzes by using only three-dimensional finite elements. That is why special programs must be used. In general, it is obligatory to carry out the above studies in terms of determining the dimensions during the initial design. However, in our study, a new design was not being applied since the current design was examined regarding field running conditions of an existing part. The study was continued on the current design. First of all, parts with existing two-dimensional technical drawings were drawn using a three-dimensional CAD program (SolidWorks). In this way, it has been examined how accurately it can be represented by an axisymmetric analysis while performing the analysis. By doing this, the effects of working conditions in real field conditions on the blade and disc were examined. The other main aim of this study is to analyze the blade and disc geometry of a gas turbine under real field operating conditions.

During the blade analysis, the only centripetal force was considered to shorten the analysis times. This force was applied from the part where the blade was attached. In this case, the effect of the blades on the disc geometry is only due to the stresses of the centripetal force at the base of the blade. That is why the ANSYS program was carried out for analysis. The ANSYS program is especially effective in obtaining healthy results for such analysis. Especially in the aviation industry, this program has a wide range of uses.

A.4. Generating Geometry for Structural Analysis

Before the analysis, the geometry must be created by using a CAD program. In this study, SolidWorks was used to create geometry. Below in Figure 5 can be seen the geometry created in the SolidWorks CAD program. The blade has at about 100.3 mm high and 86 blades surround the disc. In the analyses, just one blade on the last stage has been detected, because all parameters are in steady-state condition and the rotor running at a stable speed.

Analysis has been started with a disc in which blades are attached. In the disc part, first of all, structural and then thermal analysis was performed. During the analysis, it was examined whether the existing elongations created any negative effect on the component or not. As it is known, there is a certain distance between the rotor and the stator. This distance is called clearances. If this clearance is smaller, means that our engine efficiency is high. Due to the low distance between stator and rotor, air loss is less and for that reason, efficiency is increasing. Therefore, the wrong elongation may occur in the blades and the disc creates direct contact of the rotor with the stator. This may create damage to the surface of the rotor and stator.

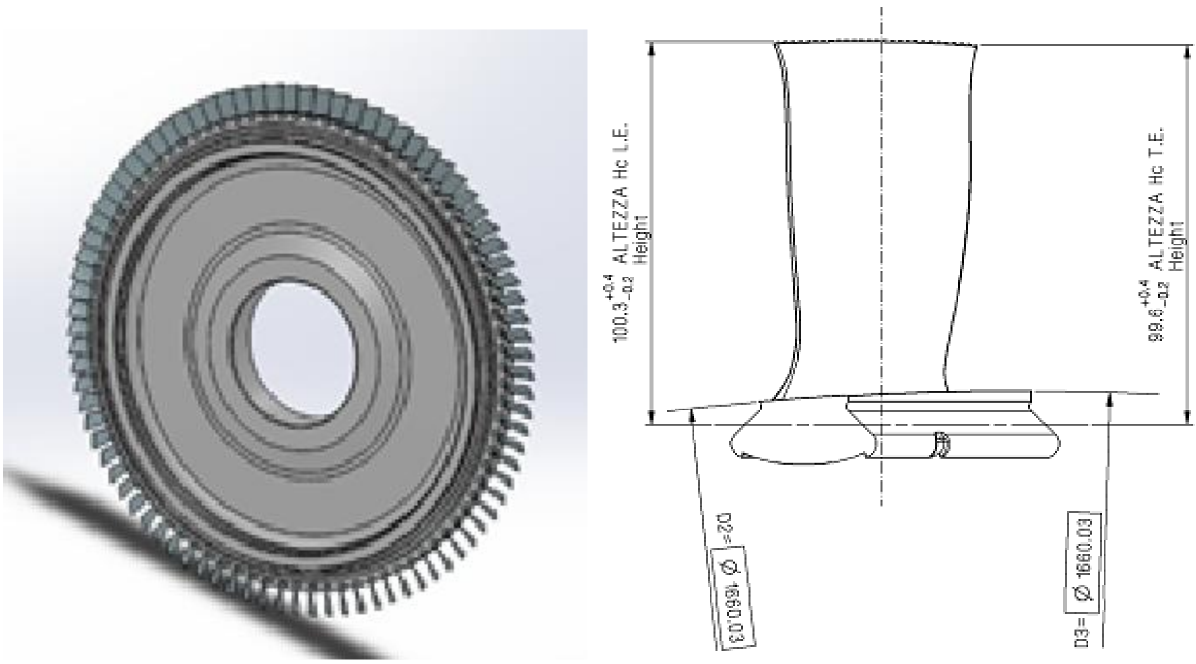

Figure 5. *Disc and Blade CAD Geometry*

III. STRUCTURAL AND THERMAL ANALYSIS OF THE COMPRESSOR DISC

First of all, for starting the analysis, the Static Structural command in the ANSYS program has been run. The disc and blade geometry is opened from the CAD modal. Before starting the structural and thermal analysis, the geometry must be Meshed by using the ANSYS program. The purpose of the Mesh is to break a complex volume into small segments for better simulation. As a definition of the Mesh, it is made from cells and points. It can have any shape and size and is used to solve Partial Differential Equations. The higher the mesh quality, the better our structural and thermal analysis results. The Mesh done in the ANSYS program is shown in Figure 6.

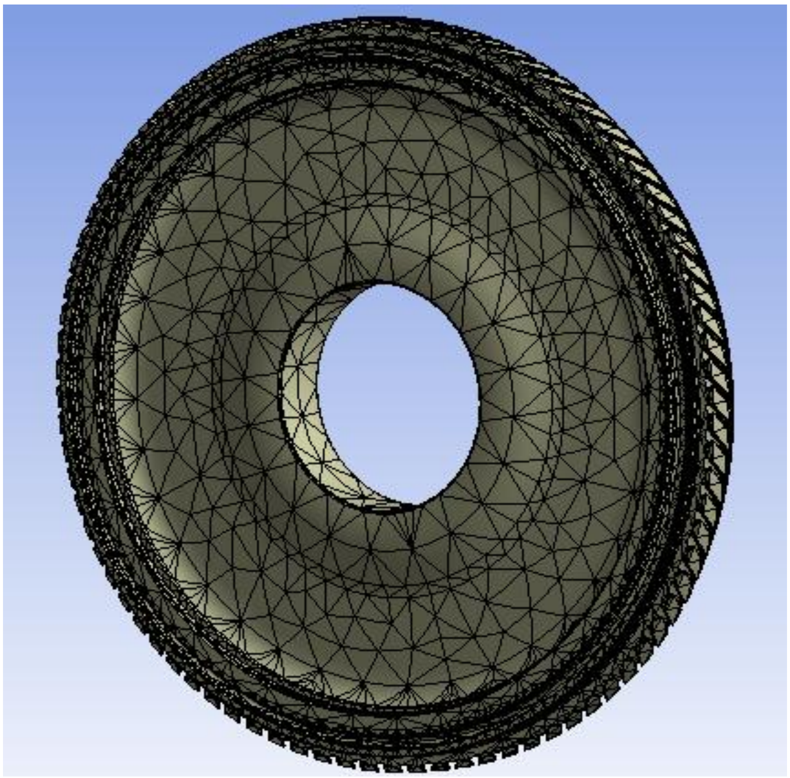

Figure 6. *Disc Mesh*

The technical specifications of our Mesh geometry are given in Table 3.

Table 3. *Model Specifications*

Points	**450880**
Mass	**1000,9 kg**
Area	**1**
Volume	**2,1664e8 mm^3**
Connection points	**450880**
Pars	**259102**
Type of parts	**PLANE 1**

A.3.1. Performing Structural Analysis

After meshing the geometry, it is possible to move on to the structural analysis. As mentioned before, Mesh quality must be very good to create out successful analysis. Mesh analyses were checked and predicted that the quality of the Mesh is sufficient and can be moved on to structural analysis. In the structural analysis, the parameters, which affect the disc, were determined one by one. The values of these parameters are taken from Table 1 obtained from the real field conditions taken from DCS.

Ambient pressure, rotation direction and speed, disc fixation point, and disc support point were determined for starting the structural analysis. The rotation value is 3000 rpm, which corresponds to 314 rad/s. Although the normal ambient pressure is 17 bar, instead 22 bar is taken for safety reasons. In Figure 7 can be seen the parameters affecting the disc points.

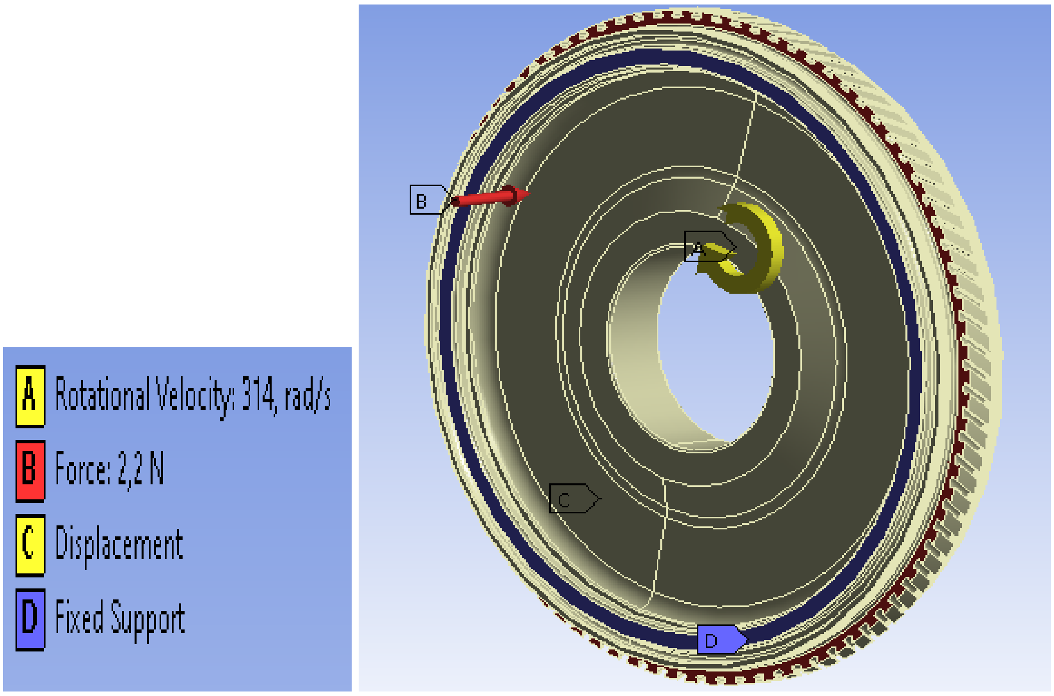

Figure 7. *Structural Analyses Parameters*

After the parameters were correctly assigned to the disc, the structural analysis was started by using the ANSYS program. In the structural analysis, the total deformation, stress, and elastic elongation parameters of the part were examined. As can be seen in Figure 8, the total deformation occurs on the side of the sharp edge and some cooling holes.

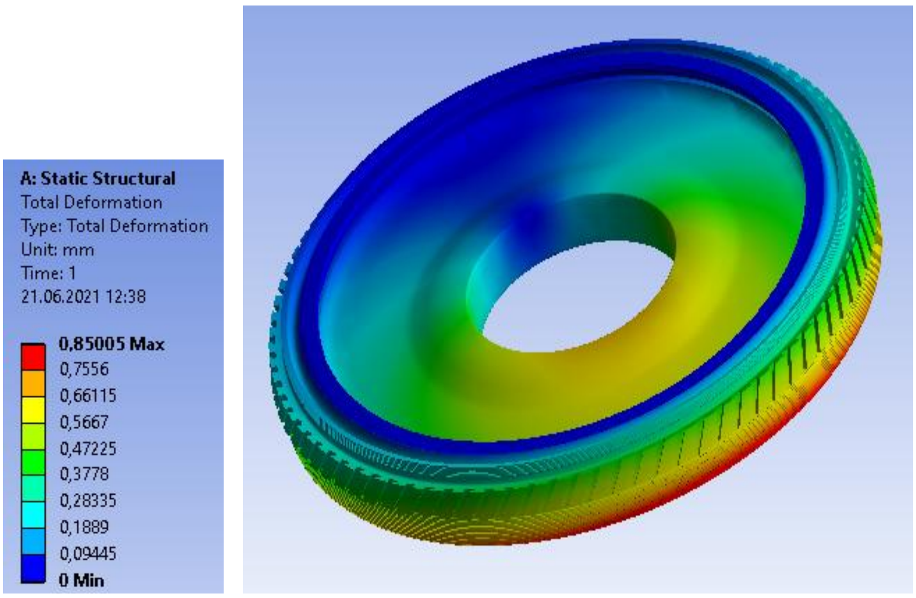

Figure 8. *Total Deformation*

As can be seen from the above result, the variables applied on the blade cause deformation of only 0.85005 mm. As it is stated in the conclusion part, these values do not create any effect on the unit.

The equivalent stresses that were occurred on the part after the total deformation was also analyzed. As can be seen in Figure 9 below, the stresses on the part are at a minimum level.

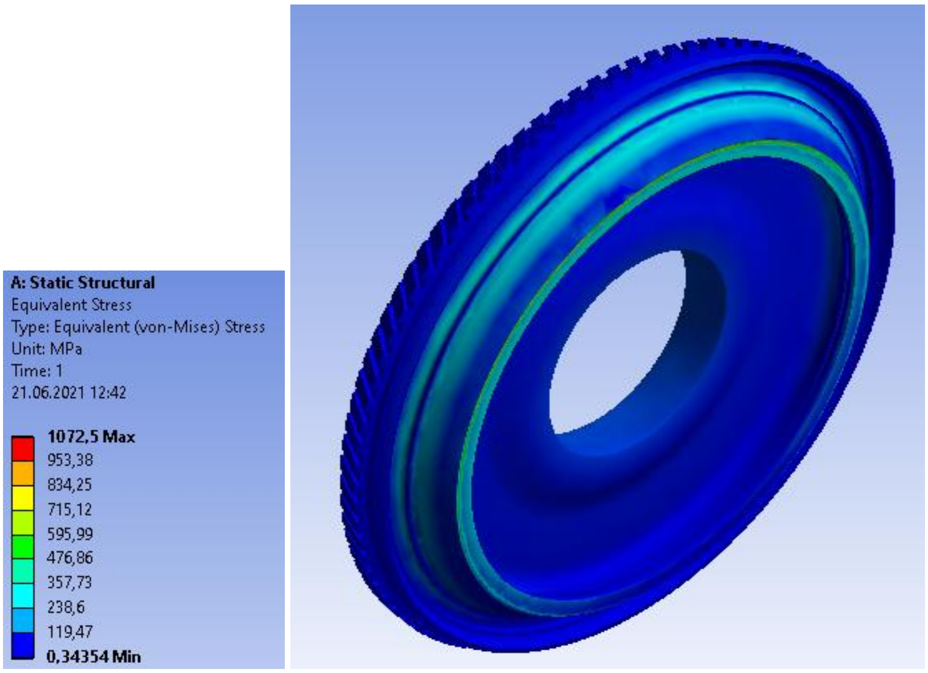

Figure 9. *Stress on the disc*

Elastic tensions that may occur on the part are shown in Figure 10. As can be seen from here, the value got is at the minimum level.

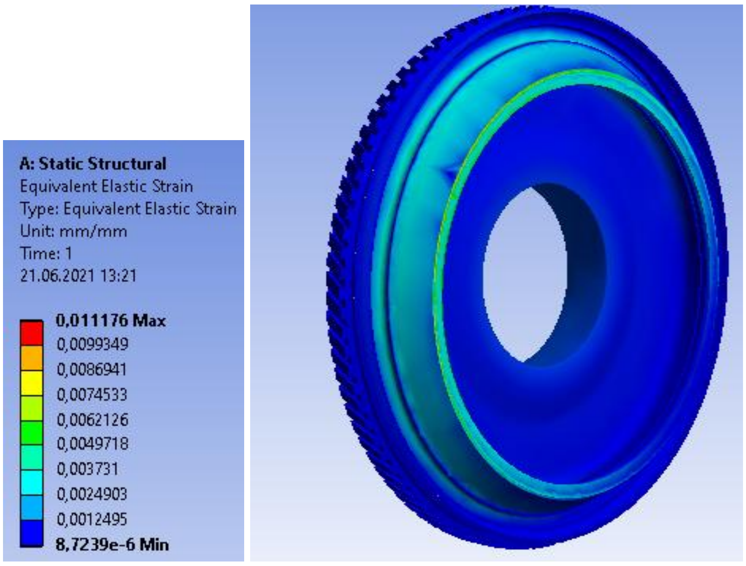

Figure 10. *Elastic Strain*

A.3.2. Performing Thermal Analysis

Heat transfer in fluid occurs utilizing conduction and also convection. Forced convection is the domain phenomenon in the disc and as well as in the turbine blade. Relatively cold air from the 6th stage of the compressor is used to cool the final stage of the disc. Therefore, thermal analysis of the disc is required and the result of this analysis has a direct effect on the disc. Two types of temperature values were used as parameters while performing thermal analysis. The first of these is the main cooling air and this value is taken as 150 ^{0}C, the other is the temperature coming to the outside of the compressor and it is taken as 500 ^{0}C. Figure 11 shows the regions affected by these two values.

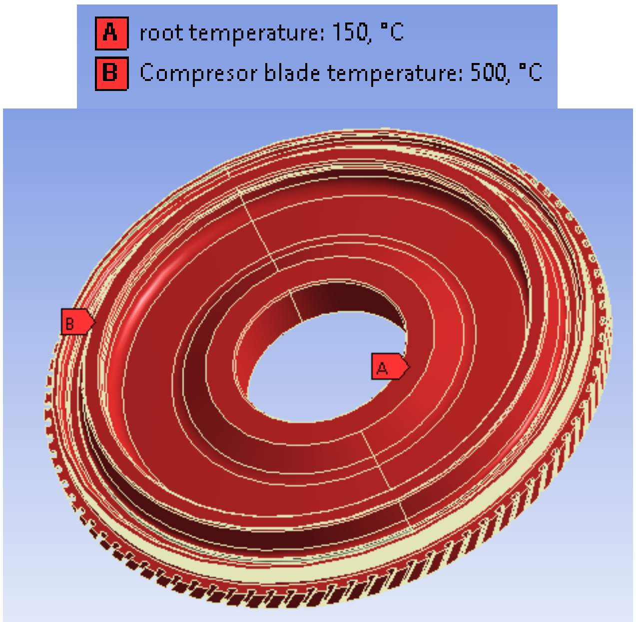

Figure 11. *Temperature Effect Locations*

While performing thermal analysis, three parameters were examined based on the situation. These are total heat exchange, heat exchange direction, and temperature. The effect regions of these parameters were analyzed during our study. In Figure 12, below, you can find the total heat exchange values and the affected area. As it can be understood from here, the heat exchange has a minimal effect on the disc. This shows how safe our discs have been designed.

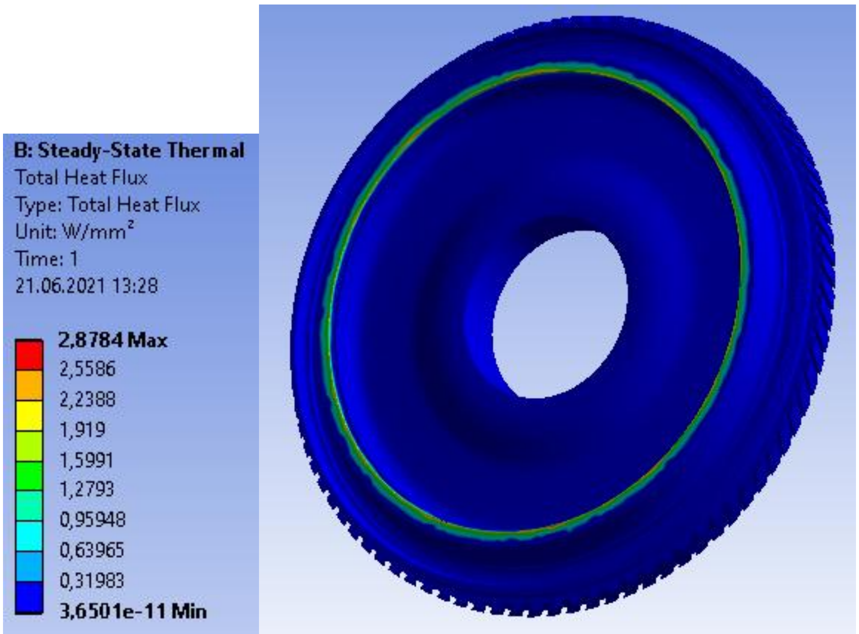

Figure 12 *Total Heat Exchange*

As can be seen from Figure 12, the greatest effect occurs in the leading edge region. This value is minimal level.

Another variable that is examined in thermal analysis is the direction of the heat exchange zone. In this case, the regions, where the temperature varies the most, were examined. The heat exchange direction and region are shown in Figure 13.

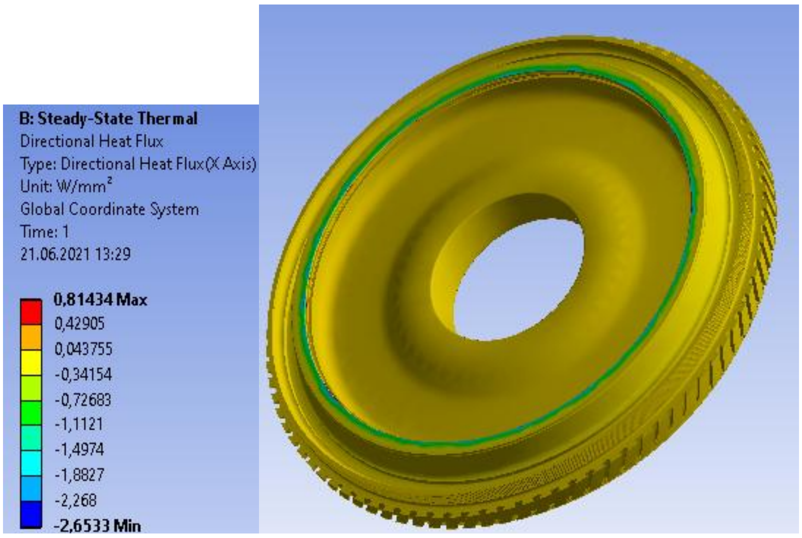

Figure 13. *Heat Exchange Zone*

The last parameter examined in thermal analysis is the effect of temperature on the disc. As it is known, the increasing temperature can cause certain deformations on the disc. The effect of temperature on the disc is shown in Figure 14 below. As can be understood from here, the disc is not affected much by the temperature parameter.

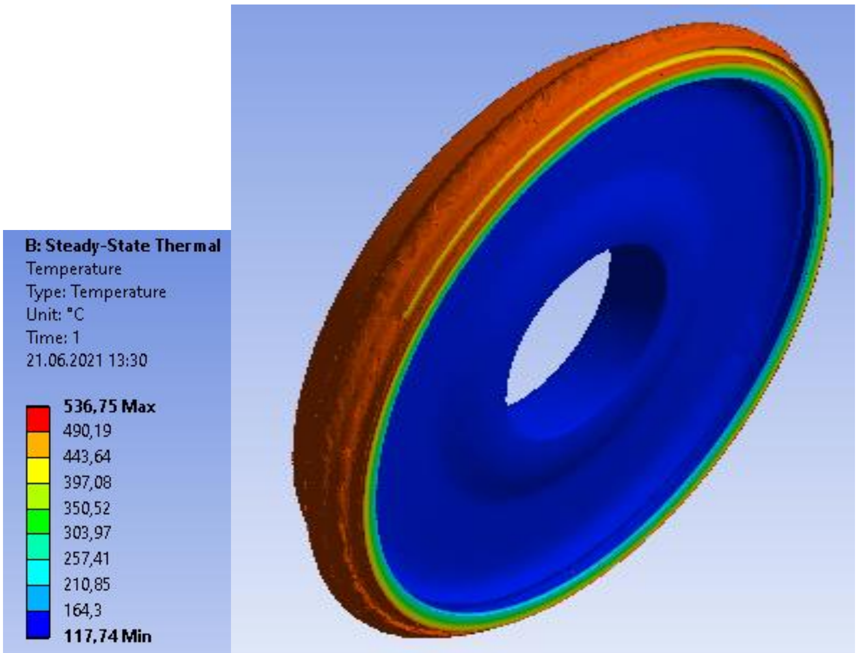

Figure 14. *Temperature Effect*

IV. <u>STRUCTURAL ANALYSIS OF THE COMPRESSOR BLADE</u>

As was done in the disc analysis, the Static Structural command runs in the ANSYS program. Geometry is opened in the CAD from the Model command. To perform the analysis, the three-dimensional geometry of the part must be opened in the CAD form. Before starting the structural analysis, the geometry must have meshed from the ANSYS program. The purpose of the mesh, as mentioned earlier, is to break up a complex volume into small parts to be simulated. It can have almost any shape in any size. It is used to solve Partial Differential Equations. The higher the mesh quality, the better our structural analysis results are. The Mesh result is as in Figure 15.

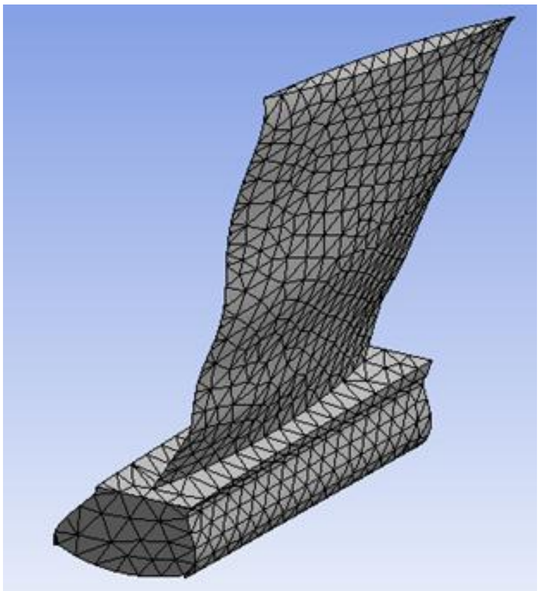

Figure 15. *Mesh Form of the Blade*

The technical specifications of the Mesh done are given in Table 4.

Table 4. *Mesh Specs*

Points	**10140**
Mass	**0,24271 kg**
Area	**1**
Volume	**52534 mm^3**
Elements	**Nearly 5625**
Types	**PLANE 1**

On the other hand, only structural analysis was held on the blades. Thermal analysis of the blade is not required. Because it is only the ambient temperature that affects the blade. The blade does not have any cooling temperature from the outside.

A.4.1. Performing Structural Analysis

Structural analysis was performed on the turbine blade to analyze the stress, strain, and deformation on the compressor blade. The yield criteria were taken into account to relate the stress state with the uniaxial stress state. The analysis for the blade needs to be repeated step by step as done in the disc part. After Mehs is done on the geometry, it is possible to move on to structural analysis. As mentioned before, Mesh quality must be very good to carry out a successful analysis study. After analysis of the meshwork, it was predicted that the quality is so good and can be moved on to structural analysis. The parameters affecting the blade and analysis were determined one by one for starting the structural analysis. The values of these parameters are taken from Table 1. which is obtained from the real field study. The pressure and temperature values affecting the compressor's last stage blade are taken from this table as well.

For structural analysis, ambient pressure, rotation direction and speed, blade fixing point, and blade support point values and regions were determined. The rotation value is 3000 rpm, which corresponds to 314 rad/s. Although the normal ambient pressure is 17 bar and 22 bar is taken for safety reasons. In Figure 16 you can see the parameters acting on the blade.

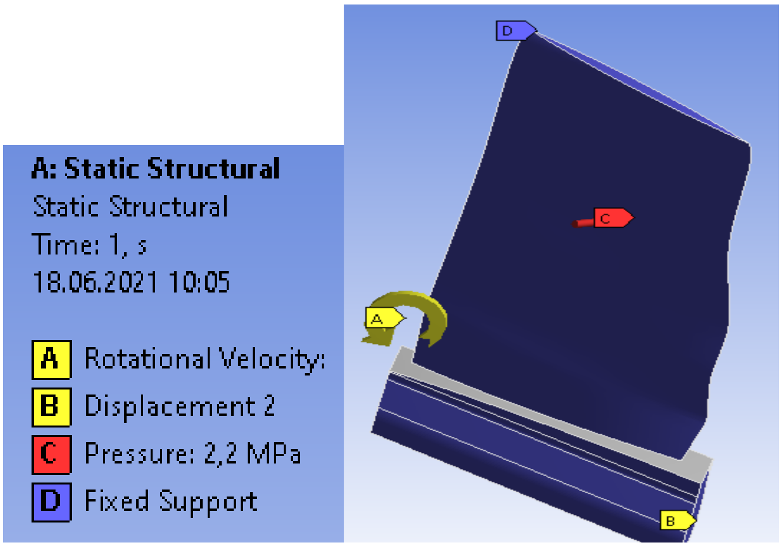

Figure 16. *Parameters on the Blade*

After the parameters affecting the blade were assigned correctly, the structural analysis was started. Rotation speed and direction, force, blade fixing points, and blade sliding surface parameters affect the blade analysis. In the structural analysis, the total deformation, stress, and elastic elongation parameters of the part were examined.

The total deformation is the total change of shape in the given working part. As can be viewed in Figure 17, total deformation is at a minimum level. This value does not affect the blade. Therefore, the blade does not undergo any deformation.

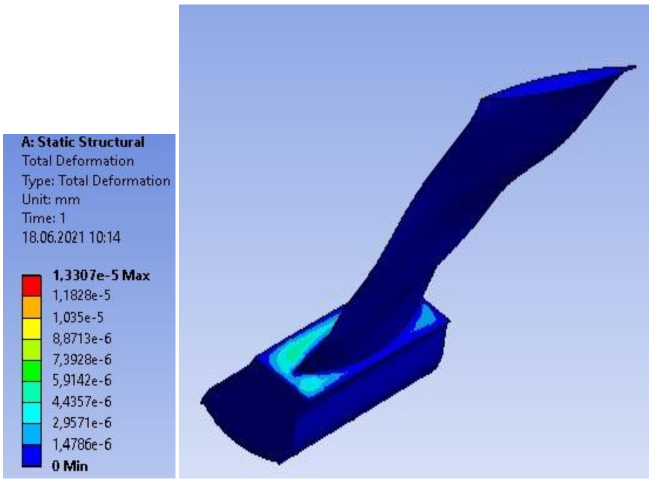

Figure 17. Total Deformation

Another analysis that needs to be done is the equivalent stresses that occur on the blade. Equivalent stress is widely used for representing a material status of ductile material. It is used in engineering to a scalar value to determine if the material has yielded or failed. The root of the blade has more strength when compared to the free end of the blade. When the loads started to apply the pressure started to affect slowly. The effect happens on the corners, bottom, and middle. The stresses were analyzed after total deformation analysis. As can be seen in Figure 18 below, the stress in the part is at a minimum level.

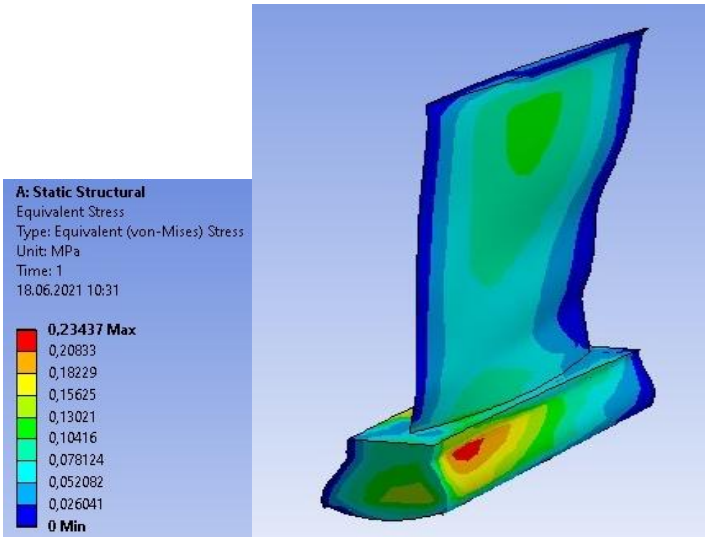

Figure 18. *Stresses on the Blade*

The maximum stress occurs on the blade only at the fixing points. The other sides are safe. All values show that the blade is safe regarding real side conditions.

The equivalent elastic strain is defined as the limit for the values strain which the part rebounds and comes back to the original shape when the load is removed. The elastic strain that may occur in the part can be seen in Figure 19. Regarding analysis, the strain condition is safe on the blade.

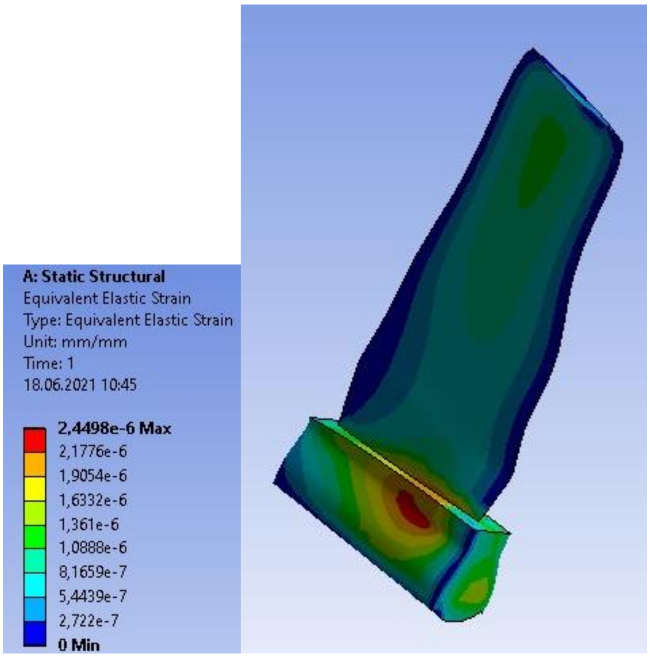

Figure 19. *Elastic Strain On the Blade*

V. CONCLUSION

The structural and thermal analyses were performed on the compressor blade and disc to verify the real side running conditions on the machine. The analysis results can be evaluated by measuring the distances between the stator and rotor in the turbine. In these results, it can be verified by checking whether the elongations occurred in our disc and blades touching the stator or not. If these elongations are smaller than the distance between our stator and rotor, the accuracy of the study is sufficient.

As seen in Figure 20 below, the standard measurement parameters of class F-type gas turbines, the values are recorded by measuring from two points while the turbine is stopped.

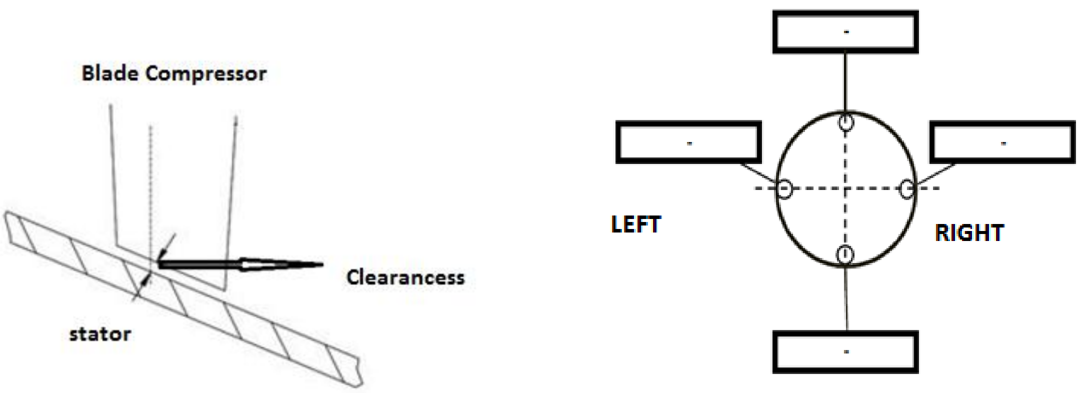

Figure 20. *Clearance Check*

This measurement process is carried out for all stages. The field measurements share with the engineering department of the turbine manufacturer, and it is checked whether the measurement values are within the desired limits. If the engineering confirms that these values are within the desired limits, the field values are recorded and presented to the customer as a report. These values are also taken as a reference for future maintenance. The values obtained from the field for this study are shown in Figure 21 below. The last stage values of the compressor were taken as reference. Because last stage blade and disc analysis were performed.

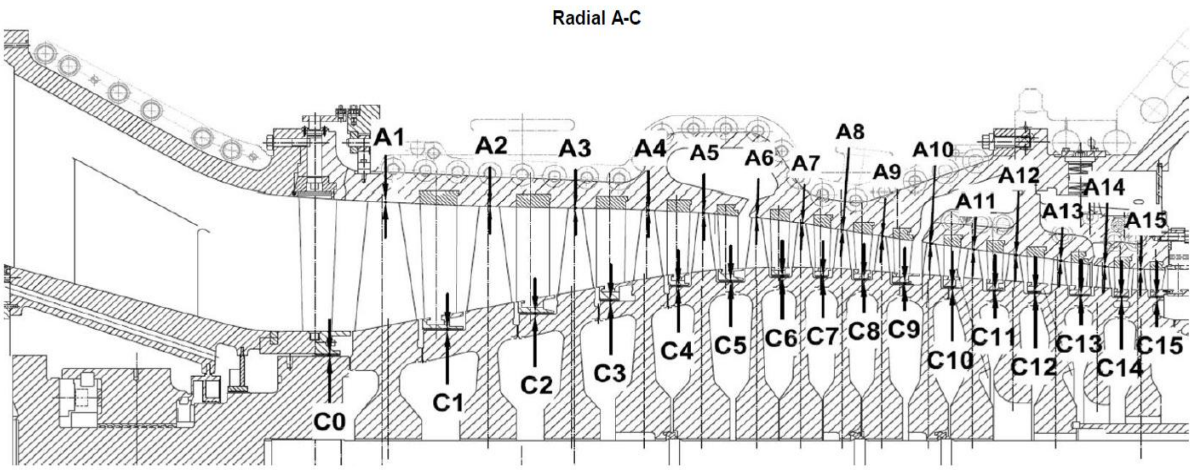

Figure 21. *Clearance Check Points*

As can be seen in Table 5. the measurement points have been taken under normal field conditions indicated in the table one by one. In this figure, **A** represents the gaps between the compressor and the stator, **C** represents the gaps between the stator fixed blades and the compressor. Our reference here is the **A15**. Because it is the last stage of the compressor. Table 5. shows the values received from the field.

Table 5. *Field Record*

Measurement point	Left		Right		Rated value	Measurement point	Left		Right		Rated value
	Input	Output	Input	Output			Input	Output	Input	Output	
A1	1,55 mm	-	1,55 mm	-	- mm	C0	1,15 mm	0,90 mm	1,20 mm	1,30 mm	- mm
A2	2,35 mm	1,80 mm	2,00 mm	1,55 mm	- mm	C1	1,40 mm	1,05 mm	1,15 mm	1,10 mm	- mm
A3	1,90 mm	-	1,55 mm	-	- mm	C2	0,75 mm	0,95 mm	0,50 mm	0,80 mm	- mm
A4	2,15 mm	-	1,70 mm	-	- mm	C3	0,95 mm	0,85 mm	0,95 mm	0,95 mm	- mm
A5	2,40 mm	3,50 mm	1,80 mm	3,05 mm	- mm	C4	1,30 mm	1,40 mm	1,35 mm	1,35 mm	- mm
A6	2,70 mm	4,00 mm	2,10 mm	3,45 mm	- mm	C5	1,70 mm	1,90 mm	1,30 mm	1,25 mm	- mm
A7	2,50 mm	3,50 mm	1,15 mm	2,85 mm	- mm	C6	1,30 mm	1,50 mm	1,15 mm	1,20 mm	- mm
A8	2,00 mm	2,65 mm	1,20 mm	2,15 mm	- mm	C7	1,70 mm	1,40 mm	1,05 mm	1,15 mm	- mm
A9	1,80 mm	-	0,85 mm	-	- mm	C8	1,55 mm	1,40 mm	0,75 mm	0,60 mm	- mm
A10	1,70 mm	-	3,85 mm	-	- mm	C9	1,15 mm	1,10 mm	0,75 mm	0,55 mm	- mm
A11	1,40 mm	-	3,60 mm	-	- mm	C10	0,65 mm	0,45 mm	2,85 mm	2,65 mm	- mm
A12	1,60 mm	-	3,75 mm	-	- mm	C11	0,85 mm	0,75 mm	3,20 mm	3,10 mm	- mm
A13	1,70 mm	-	3,80 mm	-	- mm	C12	0,60 mm	0,55 mm	2,80 mm	2,75 mm	- mm
A14	1,65 mm	-	3,80 mm	-	- mm	C13	0,60 mm	0,65 mm	2,80 mm	2,40 mm	- mm
A15	2,00 mm	-	4,00 mm	-	- mm	C14	0,60 mm	0,75 mm	2,40 mm	2,50 mm	- mm
						C15	1,10 mm	1,10 mm	3,05 mm	3,35 mm	- mm
						C16 (*)	-	-	-	-	- mm
						C17 (*)	-	-	-	-	- mm

(*) Value calculated by diameter measurements

As can see from Table 5 the measurement values on the A15 compressor and the stator are 2 mm on the left and 4 mm on the right. In our previous analysis, a maximum elongation of 0.0000001457 mm occurs in our blade during normal operating conditions. In disc, this elongation is 0.85mm. The total elongation on both blade and disc is 0.8500001457 mm. This is much smaller than our field measurement value. As can be seen from these results, the analysis results that have been done are correct and within the desired limits.

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