DEVELOPMENT OF TEST RIG FOR THE STUDY OF THE INTERNAL BLADE COOLING

by

MOHD KHAIRUL IZHAM B. ABDUL RAHMAN

Dissertation submitted in partial fulfillment of

the requirements for the

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Universiti Teknologi PETRONAS

Bandar Seri Iskandar

31750 Tronoh

Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

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A project dissertation submitted to the

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Approved by:

____

Ir. Idris Ibrahim

Project Supervisor

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

June 2010

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own expert as specified in the references and acknowledgement, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

Mohd Khairul Izham Bin Abdul Rahman

ABSTRACT

Improved turbine blade internal cooling system by fully utilized the usage of the compressed air from compressor section can provide the dramatic improvement to the long life and strength of the blade, and hence increase up the gas turbine efficiency and performance. Lots of continuous study and research are being done by the energy company and manufacturers in order to get prove of the internal blade cooling system. Power turbine blade is facing extreme environment of high pressure and temperature which can damage and reduce its strength. By using the computerized simulation process, all the results are not very solid to prove the assumption or theories of the cooling system without the experimental data using a test rig in a laboratory. The same efforts are being done in UTP which is presented by starting with a first step in study the cooling system by develop a test rig or building a model of gas turbine engine which is will be used for the study for the study of the internal blade cooling system. This project is to provide the design specification and build a test rig of a gas turbine engine which is will be used for the study blade cooling, heat movement and temperature flow profile in the operating blade. With the presence of a test rig at end of this project, the experiments activities will be able to be conducted and provide the experimental data for accurate result and prove the data from computerized simulation process. Engineering methodologies are being applied in developing the test rig by using engineering standards including the economic and safety criteria. This report is to brief the work methodology and results of the project that have been conducted. In designing the test rig for the study of the blade internal cooling system, few equipment and method had been used and involved such as CNC machining and ANSYS software for the force and stress analysis. This report also provides the complete design of the test rig which is ready for fabrication.

ACKNOWLEDGEMENTS

Firstly, I would like to express my highly gratitude to The Almighty, for giving me the ability to conduct my project perfectly, without counting the tough and challenging moment that I have experienced during the event of carrying out several tasks.

To my supervisor, Ir. Idris Ibrahim, thank you for supporting my idea regarding this project, as soon as I had proposed it. I would like to extend a heartfelt thank you for showing me the ropes and patiently guiding me of once inexperienced student. Thank you also for your guidance and the infinite support given to me. Once again, I really appreciated your motivational advices, support and concerns in order to make this project a success.

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Also, I would like to express my gratitude and appreciation to mechanical engineering laboratory assistances, for guiding and helping me conducting experiment and handling machines. I would like to thank all those individuals who have contributed in one way or another I making this project success. Your cooperation is much appreciated.

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CHAPTER 1
INTRODUCTION

1.1 Background Of Study

Gas Turbine Working Principle

A schematic drawing of a gas turbine is shown in figure 1.1. The working principle behind the gas turbine is as follows. Ambient air is compressed in the compressor. This compressed air is directed to the combustion chamber. In the combustion chamber the compressed air is mixed with vaporized fuel and burned under constant pressure. This burning process results in a hot gas with high energy content. This hot gas is allowed to expand through the turbine where the energy in the gas is converted to a rotation of the turbine shaft. The turbine shaft powers both the compressor and a generator used to obtain electrical power from the gas turbine.

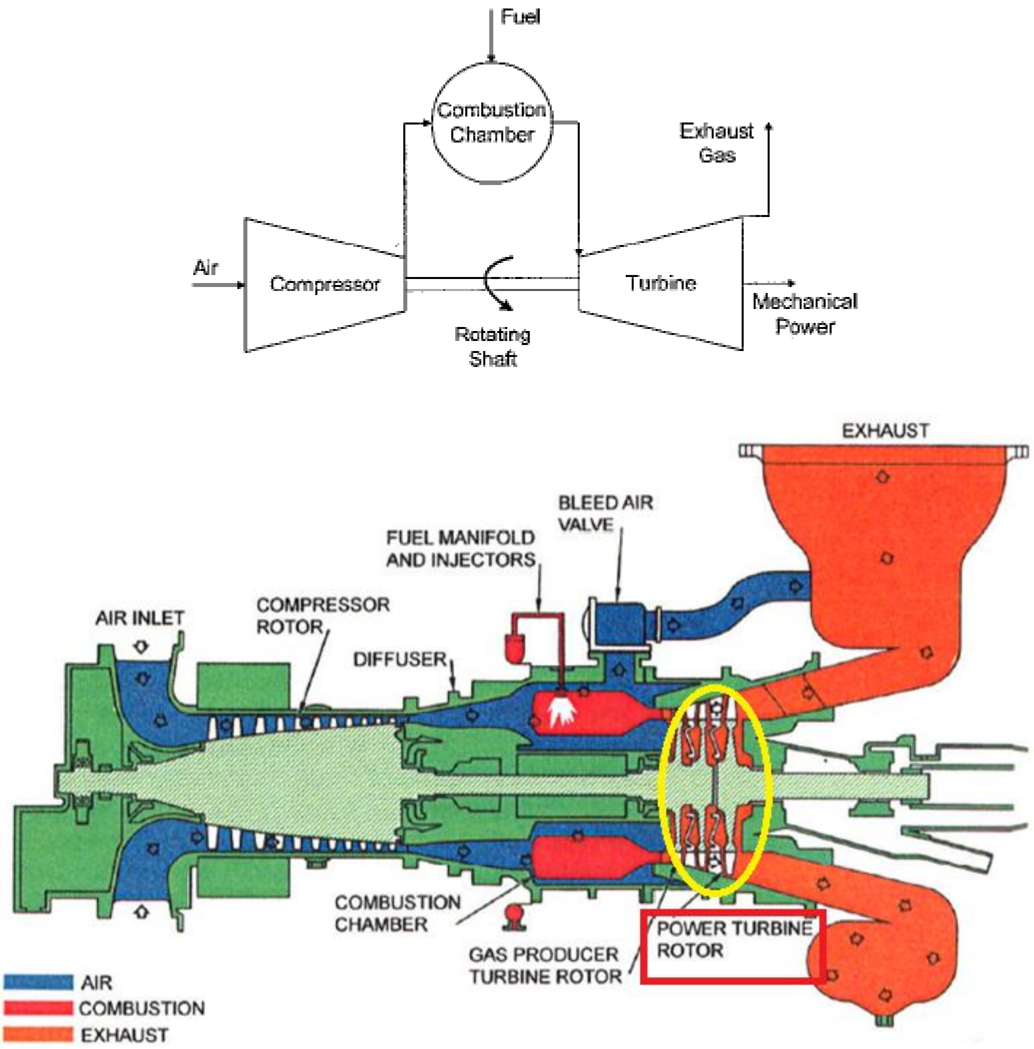


Figure 1.1: gas turbine working principle

As the Researchers continue to search for ways to increase the turbine inlet temperatures, to enhance specific power output and thermal efficiency of gas turbine engines, therefore, temperatures of turbine blades increase as well. The yellow circle in the figure 1.3 shows the area where the power turbine blade is located and as can see, it is located after the combustion chamber which means that it is facing the high temperature and high speed hot gas. The power turbine blade also rotates at very high speed up to 10000 rpm in high temperature environment which can damage the properties of the blade strength. To counterbalance this temperature rise, new active cooling of the turbine blades was developed and being study to investigate the heat movement and blade temperature profile in order to improve the efficiency and power generation capability of the gas turbine engine. For this purpose, the internal cooling method is being study in UTP by fabricating and designing the test rig which is will capable to provide the experimental data and results.

Figure 1.2: Turbine blade

Figure 1:3 Gas turbine blade damage due to rotation in high temperature and high pressure

1.2 Problem Statement

In order to investigate the cooling system on the blade, the experimental data is needed to be collected. The heat movement and temperature profile inside the blade need to be analyzed and studied. Due to the absence of the turbine engine in UTP, this project is about to build and develop the test rig for the real gas turbine engine. The test rig is generally used for designing components which closely simulate the working environment of the gas turbine engine. The test rig could be more efficiently used for other studies in the energy and mechanical field in investigating the behavior of gas turbine engine. It also can be improved further on for other studies in the future.

In the studying of the internal blade cooling system, the computerized simulation process do not providing us the experimental data or results for this patented cooling system. The first step to conduct the experiment on this matter is by having a model which can be study on it. In order to build the test rig, the design detail or the rig specification need to be provided.

1.2.1 Turbine Blade Failure

Blade internal cooling systems play a vital role in gas turbines, and the correct design is vital for safe and efficient whole-life operations. The blade internal cooling system is based upon the movement of heat and the profile of heat transfer from the combustion chamber to the power turbine blade. Due to high temperature of the hot gas from the combustor, the turbine blade needs to have the cooling system to avoid the reducing in the blade strength and structure failure.

The main cause of turbine blade failure is high cycle fatigue plus the high temperature that concentrates the stress on blade. . Fatigue failure is related to repeat cycling of the load in combustion chamber on a structural member. The fatigue life of a structural member or the number of load cycles it can survive is in general determined by the magnitude of the stress cycles. The exact relation between the magnitude of the stress and the fatigue life depends on the material properties of the structural member. In general higher stresses lead to a shorter fatigue life. For some materials fatigue only occurs if stresses exceed a certain minimum level for other materials there is no minimal stress level. If the stresses that are present on the turbine blade during operation and the material properties of the turbine blade are known then an estimation of the fatigue life of the turbine blade can be made.

Generally fatigue failure occurs as follows. After a number of load cycles a crack is initiated, high temperature put more stress concentration which the material structure expand, and the crack is propagates on it. This usually occurs at a point of relatively high stress concentration points with sharp geometrical discontinuities or points with relatively rough or soft surfaces. Once the crack is initiated it advances incrementally through the material with each stress cycle. In general this advance is very slow up to a certain point where it accelerates. The final failure occurs very rapidly. High cycle fatigue corresponds to failure after a high temperature and relatively large number of load cycles. High cycle fatigue occurs at stress levels well below the yield strength of the material where deformation is elastic. The failure of a structural member is not caused by excessive loading but by the repeated cycling of the load in high temperature operating condition.

In principle there are two ways in which the failure of turbine blades due to fatigue problems can be eliminated. These are a correct structural design and the blade internal cooling system. The correct design of a structural member can usually eliminate or dramatically reduce fatigue problems. For a turbine blade this is not always possible. The design of a blade is usually constrained by aerodynamic properties, weight, rotor length, etc. which can make the elimination of fatigue problems through design modifications very difficult if not impossible. The prevention of cyclic loading in gas turbines is virtually impossible in practice. The complex interior of the gas turbine makes the elimination of excitations difficult. There are numerous excitations that can occur within a gas turbine. These range from kee vibrations like intermittent stalls, surges and liquid slugs to forced vibrations like unbalance, vane passing, rotor stator rubbing and cavitations.

Modal analysis can be a powerful tool to assist in the identification and elimination of fatigue problems. The most obvious use of modal analysis is in determining the natural frequencies of the turbine blades and the heat movement inside the blade. Knowledge of these frequencies can be very useful in avoiding excessive excitations and thereby reducing the risk of fatigue failure.

A less obvious application of modal analysis is in the validation of computer generated models of the turbine blades. These models can be very useful to investigate turbine and turbine blade properties under running conditions. Finite element models can be used to predict the influence of design changes on the stresses and strains acting on the turbine blade under running conditions.

1.3 Objective and Scope of Study

The objective of this project in Final Year Project II (FYP II) is to provide the design specification of a test rig of a gas turbine engine which to be fabricated and used for the study of the internal blade cooling. Besides, the purpose of the test rig is:

- Provide forces and stress analysis on the components of rig which to be assembled.

The scope of this project comprises of several major aspect as:

- To provide the rig specification which meets the engineering factors, economic and safety criteria?
- To study the feasibility of the design which will be fabricated.

1.4 Feasibility of The Project

The project is feasible as it utilizes as an analysis of the design detail and specification of each component and parts of the rig. The stress and load analysis of the components will provide the life and running hours of the rig which can be used for the study of the internal blade cooling system and also other studies relates to the thermodynamics and turbomachinery equipments. The proposed project will be an improvement to the gas turbine performance and efficiency. It also provide opportunity to the UTP student to study about the gas turbine system and working principal since it is widely used in the petroleum, energy generation and aero derivative industry.

CHAPTER 2
LITERATURE REVIEW AND THEORY

2.1 Literature Review

The research on the gas turbine cooling system is keep continuously being done by the manufacturer and institution. For the Supply Company and manufacturer of the gas turbine engine, they are improving the strength of the power turbine blade by using chemical method, which using chemical to generate and growing up the blade. This lead to the high cost for the research and development process.

In SOLAR Turbine Corporation, they are using the film cooling method by using the potassium evaporative cooling system, in which used the potassium coating on the blade. In universities, the cost for purchasing a gas turbine engine, even the used one lead to the high cost and due to the confidential engineered technology of the company, so the option is by designing and developing the gas turbine prototype or test rig in order to continue the studies.

Designing a system to manage turbine blade cooling is not a new topic, however, it was found virtually no published experimental result for the patented system. There is lot of designs that was developed by manufactures and universities for the testing and experiment purpose. In Swiss Federal Institute of Technology Lausanne, the test rig for turbine blade cooling is done by designing and fabricating using the non rotating annular cascade concept where the radial-axial nozzle generates the flow condition in the test section of the annular wind tunnel. The device includes a nozzle allowing the adjustment of an axisymmetric steady flow with a swirl corresponding to the relative flow in an axial turbomachine. This test rig allows heavy measuring instrumentation of the blades but the costs of a rotating turbine or compressor row is high and facing technical difficulties of high power requirement to run the test rig.

This project is to develop a simple machine with fully utilize the original concept of gas turbine engine, simple parts assembly but provide accurate data and result. This test rig will be built by using original concept consist of rotation, compression and combustion. The test rig will used the rotating equipment which to model the real and same working principal

2.2 Test rig development

2.2.1Design attribute

Design attributes generally concern with geometric shape, size, machining and assembling process. This design attributes for the test rig developing and research, the iterative design procedure in combination with engineering analysis is came out, additional parameters are required to make the design procedure more effective. In completing this project, the Geometric entities of the test rig design will require frequent dimensional changes and modification after engineering analysis are identified and added to design attributes.

2.2.2 Manufacturing attributes

In UTP, the parts like shaft, skid and the blade housing will be fabricated in the manufacturing laboratory using lathe and CNC machine. The steel plate will be ordered or by using existing material available in the lab. Other complex part which cannot be produced in UTP will be fabricated at the outside manufacturer.

2.2.3 Parametric attributes

For the machining and documentation purpose, the AutoCAD and CATIA will be used for the plan and 3D drawings. The author will provide the softcopy drawings of AutoCAD for the CNC machine as it is easier and accurate for machining process. For the stress analysis on the structure failure of the blade, the author will use the ANSYS software to provide the accurate data.

2.2.4 Classification

The major components of gas turbine engine are turbine and compressor discs, blades, casings, shafts etc. They can be broadly classified as suggested below:

(a) Rotating components without a hole with respect to the axis of symmetry.

Example: Shaft type of components

(b) Rotating components with the hole with respect to the axis of symmetry.

Example: Discs, sealing rings,

(c) Housing type of components.

Example: Casings

(d) Components with complex profile (aerofoil shape).

Example: Blades

(e) Flange type of components.

Example: Disc blade attachment

This classification is conditional and covers all the components of the test rig. This classification can definitely be a guide line in grouping all the components. In some cases it may be possible to find that parts from the same family can have different design and manufacturing attributes. In such cases, sub groups could be formed. Also a group for sheet metal components could be formed. Shaft, turbine discs are classified as a rotating component with a hole with respect to the axis of symmetry.

2.2.5 Internal cooling mechanism

The gas turbine engine consists of few major parts such as compressor section and combustor chamber. For the test rig cooling mechanism, the real concept of the gas turbine engine system is being fully utilized. In a gas turbine engine, the air from the compressor site is supplied to the power turbine blade through the hollow shaft to cool down the blade in extreme environment. Same goes to this test rig, the atmosphere air is supplied using external air compressor to the blade through a hollow shaft which rotate at 1500 to 2000 rpm. Details of the mechanism and rig specification are discussed in the next section.

CHAPTER 3

METHODOLOGY

3.1 Rig Design Overview

Studying and Designing a system to manage blade cooling system is not a new topic. As NASA reported in their review on turbine blade cooling system, well over hundreds patent or simulation exist in industrial and academic institute pertaining to blade cooling. However, most of these are found virtually no published experimental data and results for these patented simulation system. Lots of patented system and advanced technology have been tried and used to improve the turbine blade cooling system which some of these are successful but involving and leading to the very high cost of money. The development of the test rig to simulate the real environment of the gas turbine engine system is a big effort in taking part of these studies.

The author of this work have focused his effort on designing a model of gas turbine engine which will provide the rotation to the blade in the high temperature condition. The rig design is focus on to rotate the 10 kg turbine blades in 650°C hot air housing, which are attached to a rotating shaft drive by electric motor.

In order to provide the rig specification, the following steps and process need to be done:

1. Finalized plan, design or detail drawing for fabrication
2. Analysis on the forces distribution, stress and allowable load on components
3. Analysis on the safety and economic criteria
4. Manufacture or fabrication
5. Assembling

3.2 Test Rig Development

3.2.1 Flow Chart of Test Rig Development

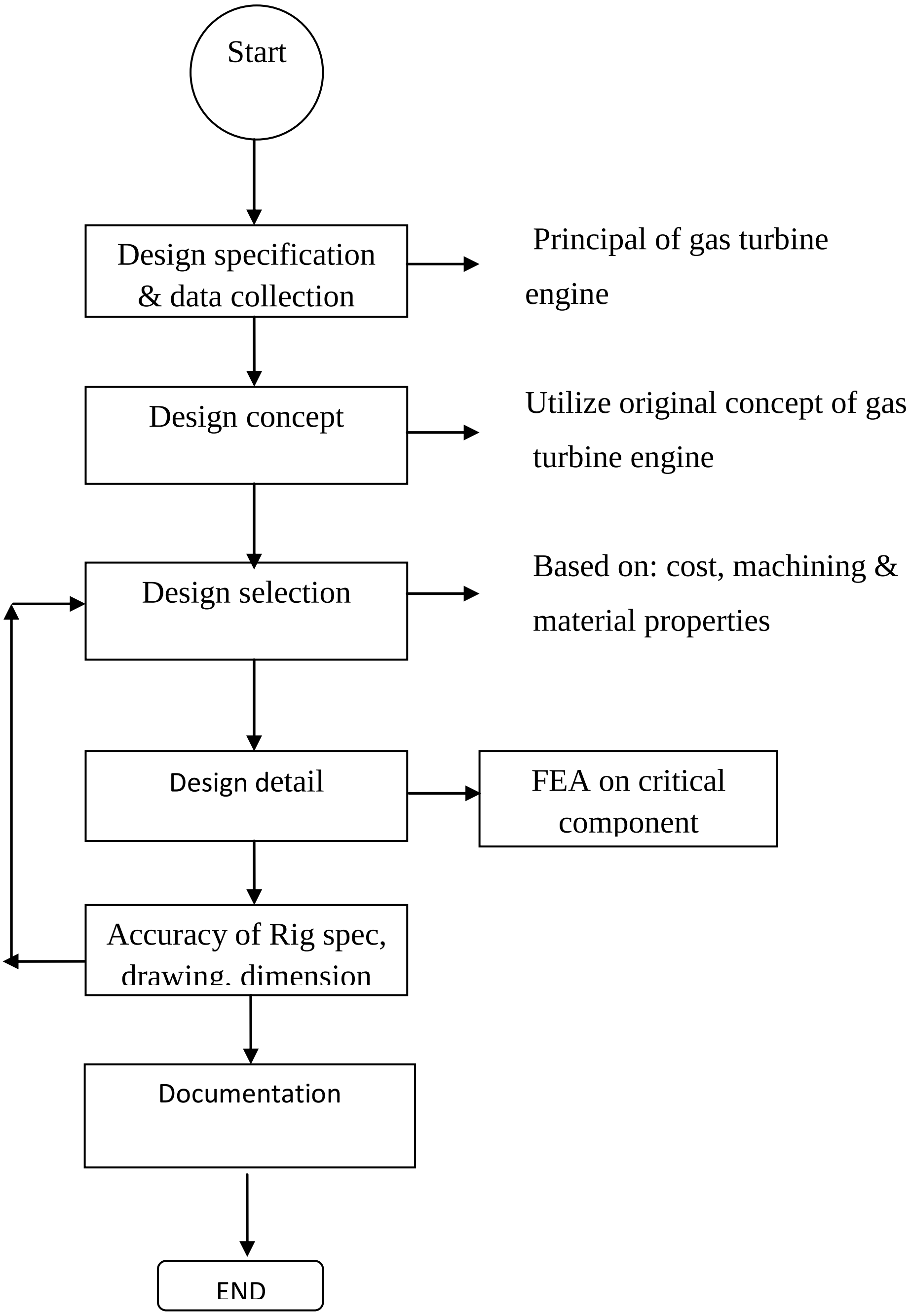


Figure 3.1: Flow chart of test rig development.

3.2.2 Rig specification

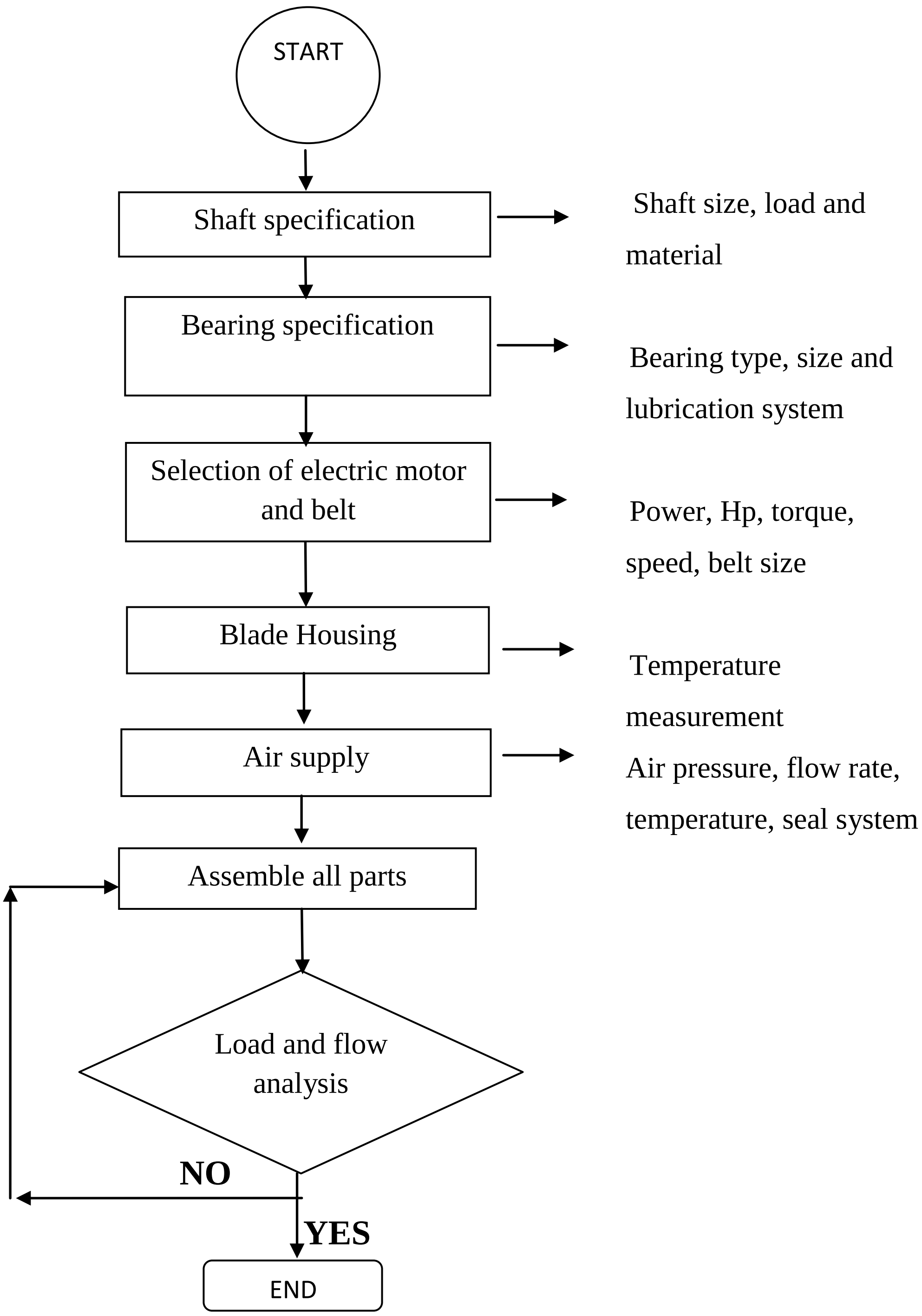
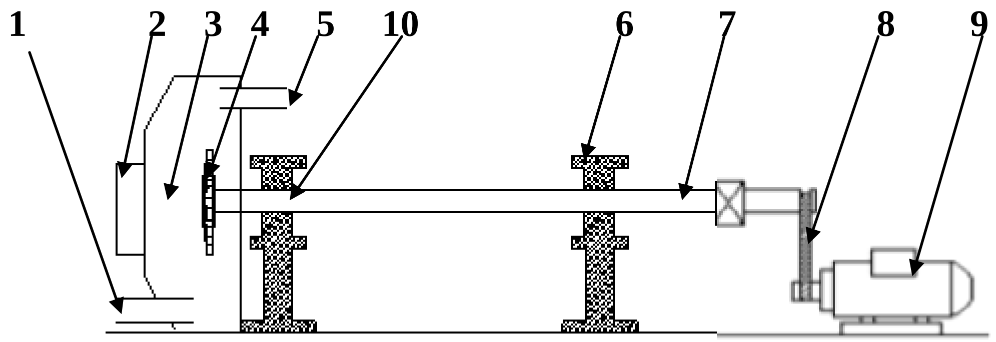


Figure 3.2: flow chart of rig specification development

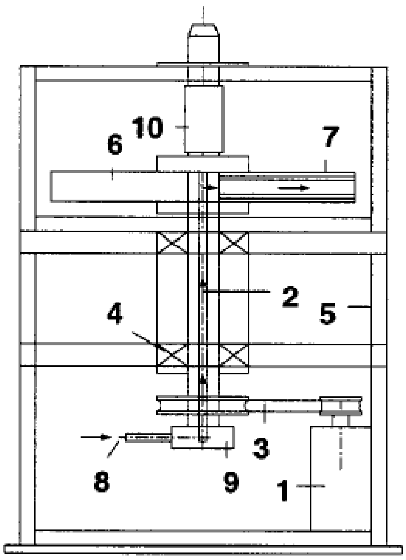
3.3 TEST RIG DESIGN CONCEPT

Specifications of rig were chosen to closely simulate the environment (temperature and heat transfer) and operating requirement (rotation) of large industrial engine.



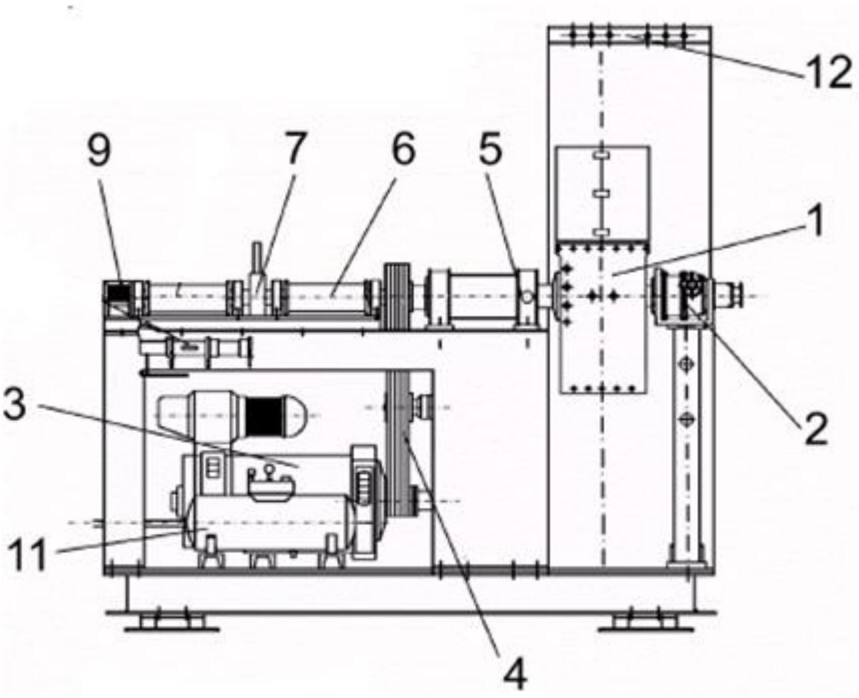
NO		NO	
1	Hot air-discharge	6	Mounting skid and bearing housing
2	Window	7	shaft
3	Hot air housing	8	belt
4	Turbine blade	9	Electric motor
5	Hot air-inlet	10	Roller bearing

Figure 3.3: test rig design concept



N		N	
1	Electric motor	6	Rotating arm
2	Hollow rotating shaft	7	Test section
3	Belt driven system	8	Compress air
4	Bearing	9	Rotary seal
5	Steel work table	10	Slip ring

Figure 3.4: top view of design concept



NO.		NO.	
1	rotating arm assembly	7	speed measurement
2	rotary seal	9	slip rings
3	electric motor	11	compressed-air tank
4	Belt	12	protective shell
5	imbalance control		
6	pressure gauge		

Figure 3.5: basic constructional details

Figure4.1.3 shows the basic constructional details of the experimental apparatus. The rotor assembly consists of a main drive shaft is supported on two bearings, respectively mounted on the main drive shaft, support the heated test sections, which model the flow geometry of a particular turbine blade cooling channel.

The left hand side of the main drive shaft (as indicated in Figure 3.5) has a centrally located blind hole to permit air to enter the shaft. The air then flows through the test section (turbine blade) trough the fir root of the blade.

The key element of the test rig is the rotating arm assembly containing the model. It is fixed to a shaft driven by a 10kW electric motor using belts. Ambient air is drawn through the model by a 22kW vacuum blower which is located downstream of the apparatus. The coolant passes through an orifice meter before entering the rotating system via a rotary seal. The

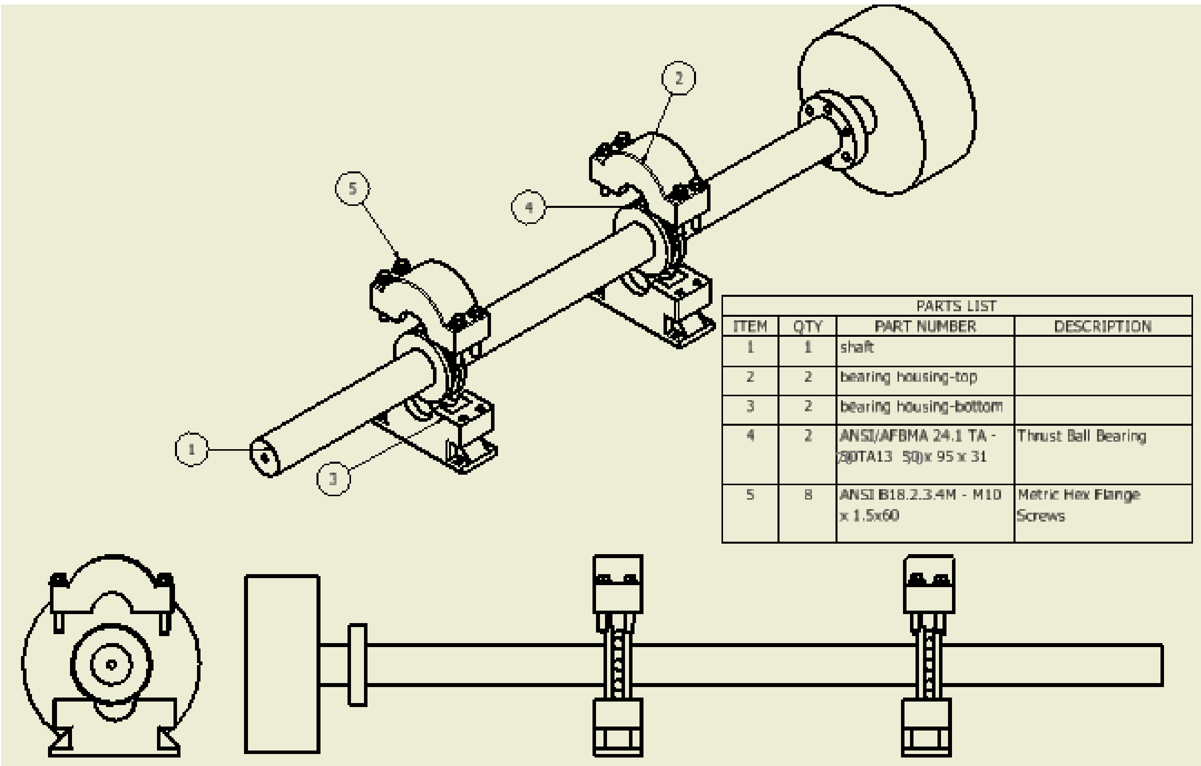


Figure 3.6: 3D view of rig system

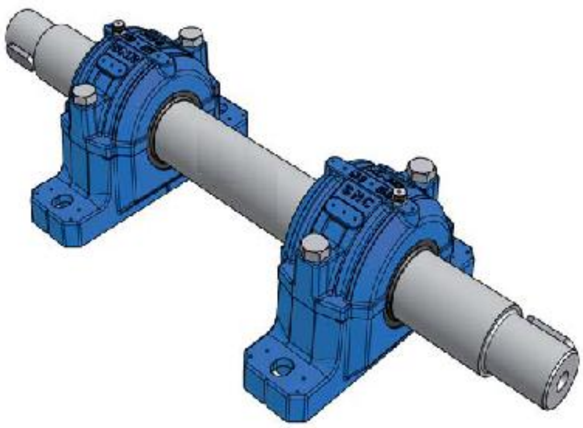


Figure 3.7: installation of shaft and bearing

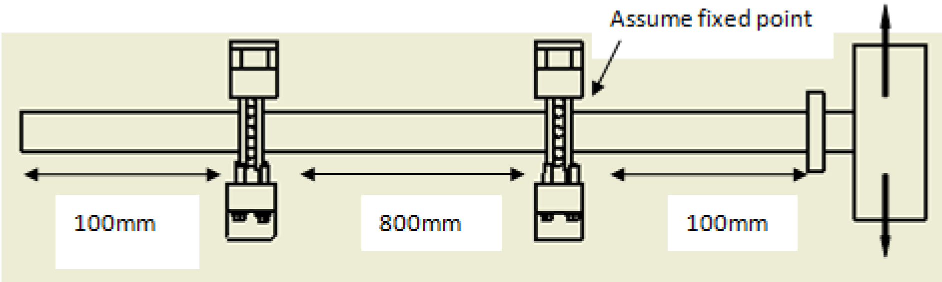
CHAPTER 4

RESULT AND DISCUSSION

4.1 Rotation

The 3.33 kg each blade is attached to the rotating shaft rotated by the electric motor at 1500 to 2000 Rpm. By assuming no friction on the bearing system and no vibration on the rotation, the rig can closely simulate the environment and operating requirements of a gas turbine engine.

Torsion and shear stress on disk blade



$$\sum M_y = 100\text{N}\,(0.1\text{m}) + 100\text{N}\,(0.1\text{m}) - \text{T} = 0$$

$$\text{T}=20\text{N.m}$$

$$\text{J}=\frac{\pi}{2}(0.035^4 - 0.005^4)$$

$$\text{J}=2.356\times10^{-6}$$

$$\tau_o=\text{T}\frac{TC_o}{J}=\frac{20N.m\times0.035}{2.356\times10^{-6}} = 297\text{Kpa}$$

$$\tau_i=\text{T}\frac{TC_i}{J}=\frac{20N.m\times0.005}{2.356\times10^{-6}} = 42\text{Kpa}$$

Figure 4.1: shaft dimension and torsional calculation

Disk blade attachment is the most critical part on the shaft, since the blade will be attached on it and operates at high speed of rotation, the force on the part need to be considered. To get the maximum load that the disk blade, by assuming one fixed point, this provide the maximum load on the disk point. Base on this calculation, the force exit on the disk blade attachment is 297Kpa of torsion load. In this calculation, the rotation speed of the shaft is been considered.

4.1.1 Injecting air into rotating shaft mechanism

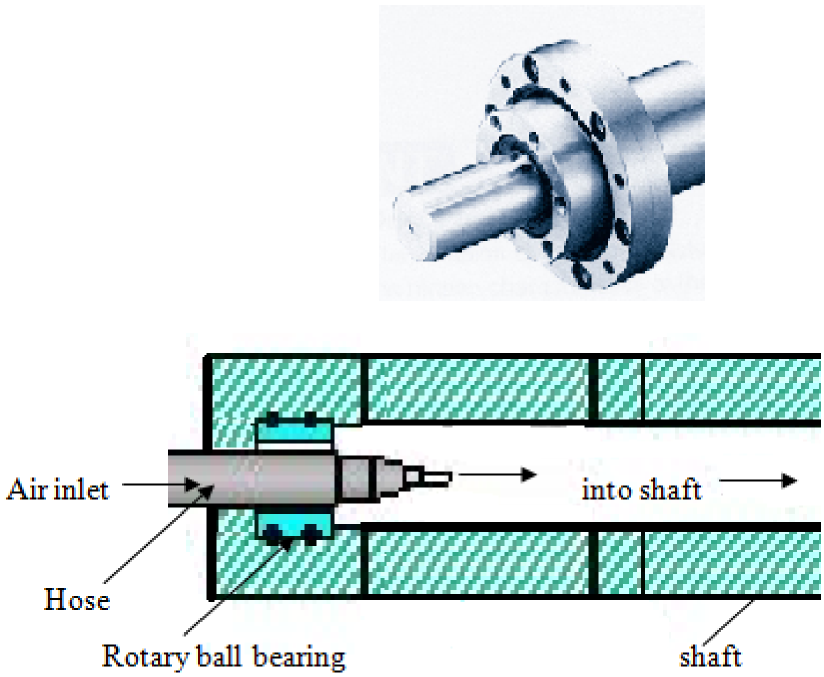


Figure 4.2: air injection mechanism

At the end of the shaft, the air from the air compressor is injected into the rotating shaft. The hose is attached to the shaft by using the rotaty ball bearing. The rotary ball bearing allows the shaft to rotate smoothly without giving friction and force to the injection hose. The injection hose will face backward force provided by pressure created during injection. The air pressure is 7 bar and create low force to the bearing.

4.1.2 Shaft

Shafts with circular cross-sections are used for transmission of power. Fatigue is among the most common cause of failure of such shafts. Fatigue failures start at the most vulnerable point in a dynamically stressed area particularly where there is a stress raiser. The stress raiser may be mechanical or metallurgical in nature, or sometimes a combination of the two. Thus, these rotating components are susceptible to fatigue by the nature of their operation and the fatigue failures are generally of the rotating-bending type. The common sites on shafts where fatigue cracks may initiate are the stress raisers occurring at the keyway root radius and where sharp changes in cross-sectional area of the shaft occur.

Shaft calculation due to rotational torque by belts:

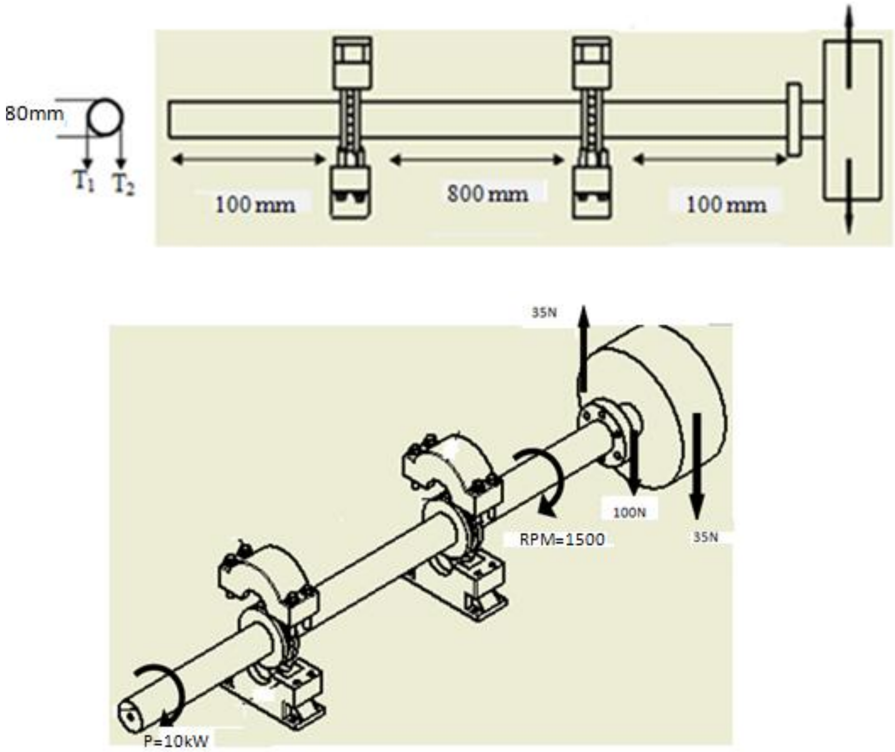


Figure 4.3: shaft diameter calculation

Belt pulley is keyed to the end of the shaft 100mm from the supporting bearing which kept at 800mm apart.

Power transmit, P =10kW shaft combined fatigue =1.5

Shaft speed = 1500 rpm shock factor = 1.25

Pulley diameter = 80mm belt pulley wrap angle= 180°

Belt tension ratio = 2.5 S_{ut} = 400Mpa S_{yt} = 240 Mpa

$T_{permissible} = 0.3\times S_{ut} = 0.3\times240 = 72$ Mpa

$T_{permissible} = 0.18\times S_{yt} = 0.18\times400 = 72$ Mpa whichever equation is minimum

$$\text{Power(P)} = \frac{2\pi NT}{60}$$

Torque (T) = $\frac{P\times60}{2\pi N} = \frac{10\times10^6\times60}{2\pi\times1500}$, T = 63661.97Nmm

Torque is transmitted by belt drive:

Torque = $(T_1\text{-}T_2)$ r, 63661.97Nmm = $(T_1\text{-}T_2)\times40$

$(T_1\text{-}T_2)$= 1591.54, $\frac{T_1}{T_2}$ =2.5, T_2=1061.03N, T_1= 2652.57N

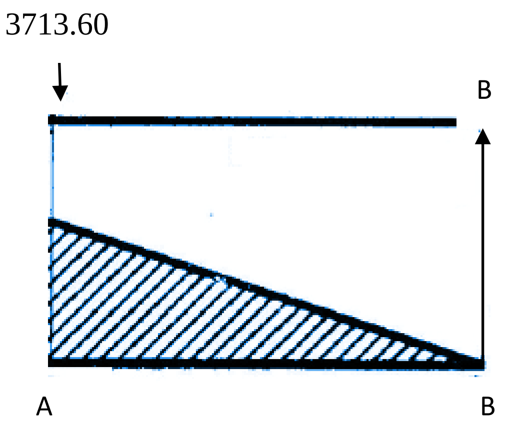


Figure 4.4: bending moment calculation

Since the belt tension act vertically downward, load at shaft become (T_1-T_2).

$\sum F_y$ $\quad F_A+F_B = 3713.60N$

$\sum M_B = 0,$

$3713.60 \times 100 = 371360$

Bending moment

$B.M_A = B.M_{MAX} = 371360Nmm$

4.1.3 Shaft Size

Shaft diameter

$$\sqrt{(K_t \times T)^2 + (K_B \times M)^2} = \frac{\pi}{16} d^3 \times 72$$

$$\sqrt{(1.25 \times 63661.97)^2 + (1.5 \times 371360)^2} = \frac{\pi}{16} d^3 \times 72$$

$d^3 = 39802.5$

D=35mm

The minimum diameter size of the shaft is 35mm. since this rig is using the hollow shaft for the air injection system, the shaft diameter need to be double for safety factor. The diameter of the shaft also base on the bearing size that are available in factory and supplier.

4.1.4 Shear Stress on Bolts

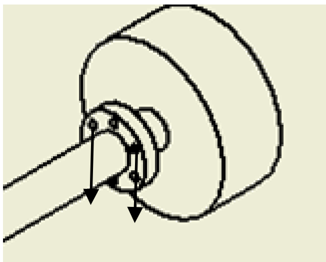


Figure 4.5: shear stress on bolts

Q = the torque of the shaft in Newton- meters

D = diameter of bolt circle in meters

N = number of bolts

d = diameter of bolt in centimeters

V = shear force acting on a bolt in Newtons

v = shear stress in Newtons per cm^2

$$V = \frac{2Q}{D \times N};\ \frac{63.66}{0.01 \times 6} = 2122\text{N}$$

$$v = \frac{V}{\frac{\pi \times d^2}{4}};\ \frac{2122}{\frac{\pi \times 1^2}{4}} = 2.7\text{kN}$$

4.1.5 Shaft analysis

Material Data

The following material behaviour assumptions apply to this analysis:

- Linear - stress is directly proportional to strain.
- Constant - all properties temperature-independent.
- Homogeneous - properties do not change throughout the volume of the part.
- Isotropic - material properties are identical in all directions.

Table 1: material data of shaft

Part3.ipt Statistics

Bounding Box Dimensions	220.0 mm 220.0 mm 1140 mm
Part Mass	44.42 kg
Part Volume	5.498e+006 mm^3
Mesh Relevance Setting	0
Nodes	21193
Elements	12298

Stainless Steel

Young's Modulus	1.93e+005 MPa
Poisson's Ratio	0.3
Mass Density	8.08e-006 kg/mm^3
Tensile Yield Strength	250.0 MPa
Tensile Ultimate Strength	0.0 MPa

Loads and Constraints

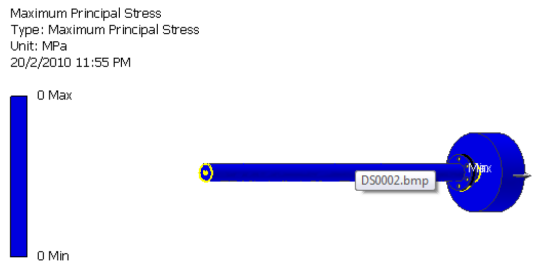
The following loads and constraints act on specific regions of the part. Regions were defined by selecting surfaces, cylinders, edges or vertices.

Table 2: load and constrain on shaft

Load and Constraint Definitions

Name	Type	Magnitude	Vector
Force 1	Surface Force	100.0 N	100.0 N 1.225e-014 N 0.0 N
Force 2	Surface Force	100.0 N	-100.0 N -1.225e-014 N 0.0 N
Fixed Constraint 1	Surface Fixed Constraint	0.0 mm	0.0 mm 0.0 mm 0.0 mm
Fixed Constraint 2	Edge Fixed Constraint	0.0 mm	0.0 mm 0.0 mm 0.0 mm

Maximum Principal Stress



4.1.6 Stress analysis

Though the shaft is subjected to combined bending, torsion and axial loads, in the absence of axial and bending load data from the designer/manufacturer, only torsional load on the shaft is considered for stress calculation.

The maximum shear stress at the surface of the shaft due to torsion is expressed as:

$$\tau_{\max} = 16Td_0/\pi\left(d_0^4 - d_i^4\right).$$

Where,

τ_{max}=maximum shear stress, do=outer diameter of the shaft,

di=inner diameter of the shaft, T=torsional moment at critical cross-section.

4.2 Skid and Pillow block Bearing Housing

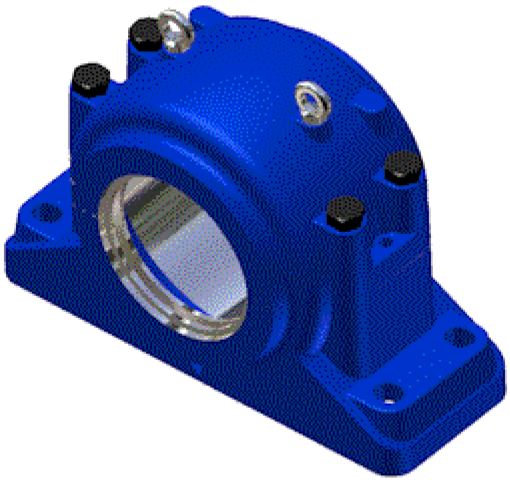
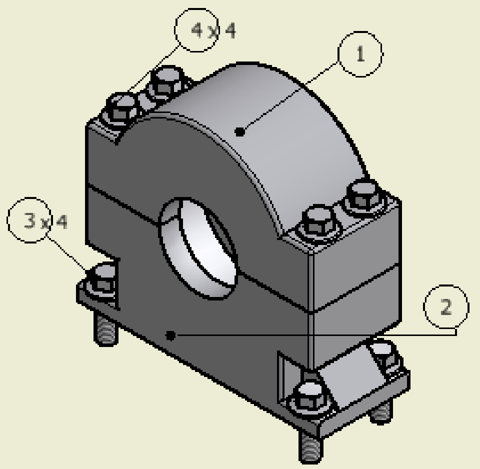


Figure 4.6: pillow block bearing housing

4.2.1 Basic design

The two-part bearing housings are made up of an upper and a lower section. This greatly simplifies mounting and maintenance of the units, as the bearing and sealing elements can be mounted on the shaft first and then simply inserted into the pre-positioned lower section of the housing.



PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	bearing housing-top	
2	1	bearing housing-bottom	
3	4	ANSI B18.2.3.4M - M10 x 1.5x30	Metric Hex Flange Screws
4	4	ANSI B18.2.3.4M - M10 x 1.5x60	Metric Hex Flange Screws

Figure 4.7: base view of bearing housing

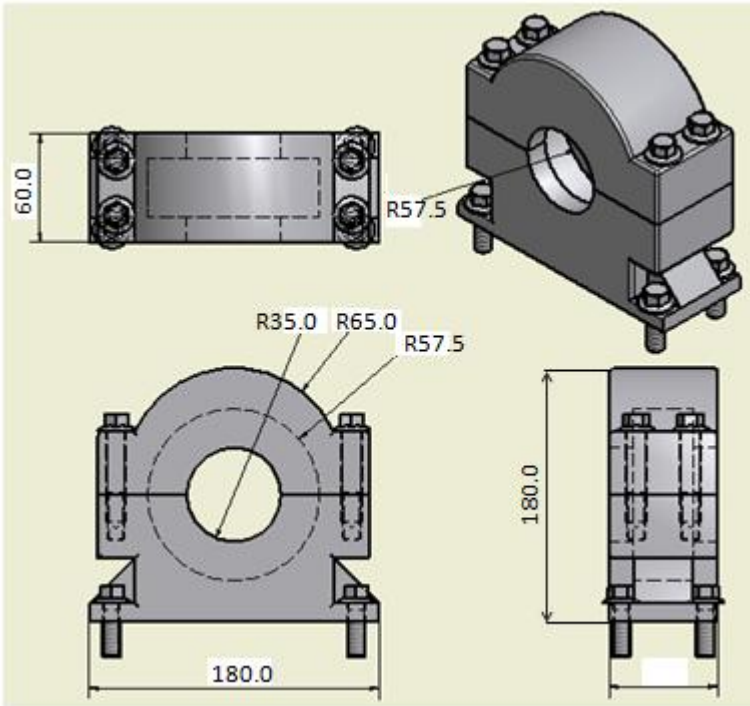


Figure 4.8: projected view of bearing housing

<u>4.2.2 Analisys of bearing housing- bottom</u>

Material Data

Table 3: material data for bearing housing- bottom

bearing housing-bottom.ipt Statistics	
180.0 mm Bounding Box Dimensions	160.0 mm 70.0 mm 60.0 mm
Part Mass	3.486 kg
Part Volume	4.435e+005 mm³
Mesh Relevance Setting	0
Nodes	6221
Elements	3082

Steel, Mild	
Young's Modulus	2.2e+005 MPa
Poisson's Ratio	0.275
Mass Density	7.86e-006 kg/mm³
Tensile Yield Strength	207.0 MPa
Tensile Ultimate Strength	345.0 MPa

Loads and Constraints

The following loads and constraints act on specific regions of the part

Table 4: load and constrain reaction on part-bottom

Constraint Reactions

Name	Force	Vector	Moment	Moment Vector
Fixed Constraint 1	100.0 N	-1.958e-008 N 100.0 N 7.374e-009 N	5.566e-003 N·mm	1.877e-005 N·mm -2.006e-006 N·mm 5.566e-003 N·mm

Results

The table below lists all structural results generated by the analysis. The following section provides figures showing each result contoured over the surface of the part.

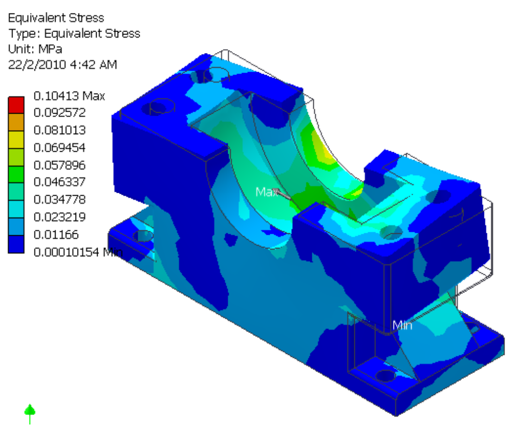
Safety factor was calculated by using the maximum equivalent stress failure theory for ductile materials. The stress limit was specified by the tensile yield strength of the material.

Table 5: load on parts-bottom

Structural Results

Name	Minimum	Maximum
Equivalent Stress	4.71e-005 MPa	4.184e-002 MPa
Maximum Principal Stress	-7.847e-003 MPa	1.635e-002 MPa
Minimum Principal Stress	-4.22e-002 MPa	1.681e-003 MPa
Deformation	0.0 mm	6.684e-006 mm
Safety Factor	15.0	N/A

Equivalent stress



max principal stress

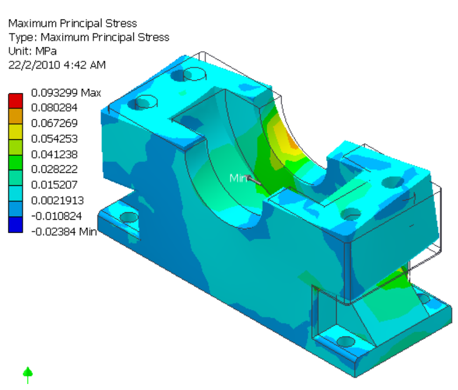


Figure 4.9: analysis of stress on parts- bottom

4.2.3 Analisys of bearing housing- top

Table 6: material data for bearing housing- top

bearing housing-top.ipt Statistics	
Bounding Box Dimensions	150.0 mm 70.0 mm 60.0 mm
Part Mass	2.544 kg
Part Volume	3.237e+005 mm³
Mesh Relevance Setting	0
Nodes	5929
Elements	3013

Steel, Mild	
Young's Modulus	2.2e+005 MPa
Poisson's Ratio	0.275
Mass Density	7.86e-006 kg/mm³
Tensile Yield Strength	207.0 MPa
Tensile Ultimate Strength	345.0 MPa

Loads and Constraints

The following loads and constraints act on specific regions of the part. Regions were defined by selecting surfaces, cylinders, edges or vertices.

Table 7: load and constrain reaction on part-top

Constraint Reactions

Name	Force	Vector	Moment	Moment Vector
Fixed Constraint 1	100.0 N	2.652e-008 N -100.0 N -3.099e-009 N	1.245e-003 N·mm	-1.092e-003 N·mm -1.495e-006 N·mm 5.968e-004 N·mm

Results

The table below lists all structural results generated by the analysis. The following section provides figures showing each result contoured over the surface of the part.

Safety factor was calculated by using the maximum equivalent stress failure theory for ductile materials. The stress limit was specified by the tensile yield strength of the material.

Table 8: result of load and stress on part-top

Structural Results

Name	Minimum	Maximum
Equivalent Stress	3.823e-004 MPa	0.1261 MPa
Maximum Principal Stress	-1.805e-002 MPa	0.1695 MPa
Minimum Principal Stress	-6.637e-002 MPa	4.384e-002 MPa
Deformation	0.0 mm	2.242e-005 mm
Safety Factor	15.0	N/A

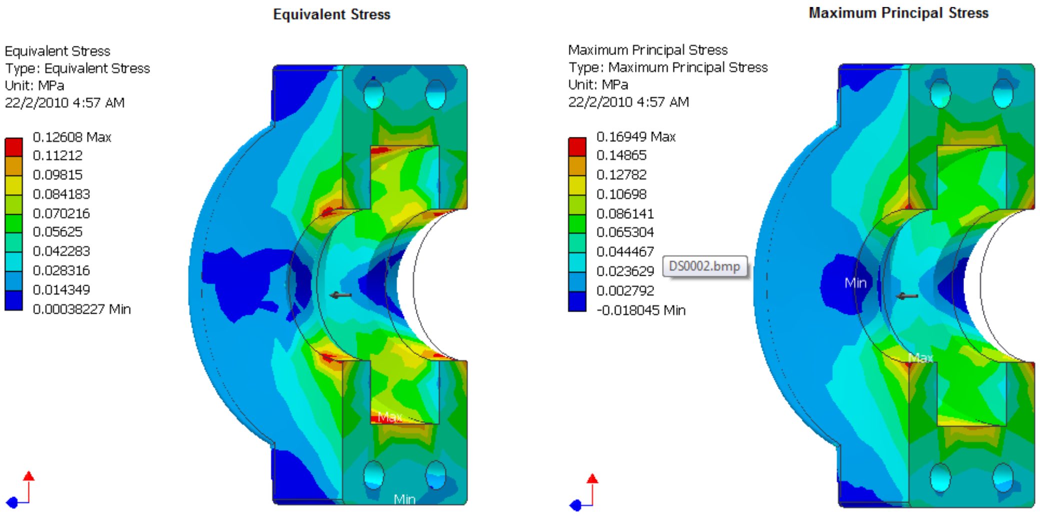


Figure 4.10: analysis of stress on parts-top

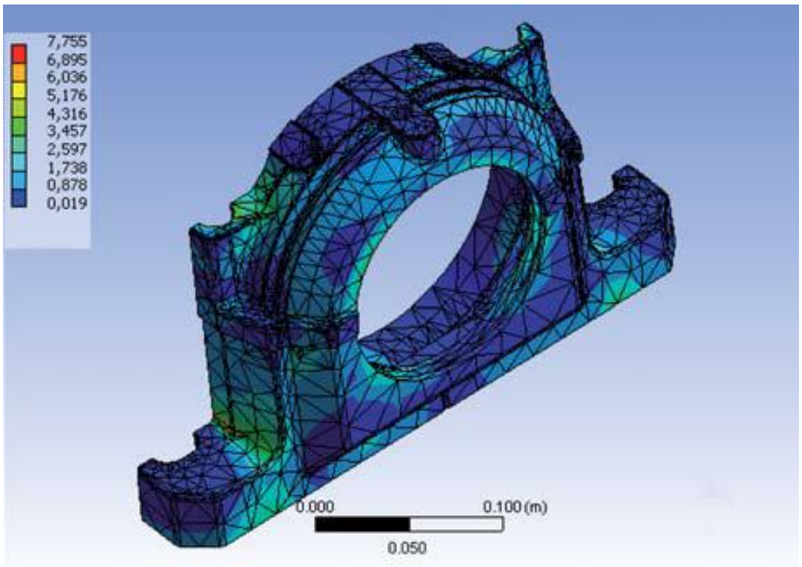


Figure 4.11: analysis of stress on pillow block bearing housing

The finite element method enables the optimum housing design to be determined in advance.

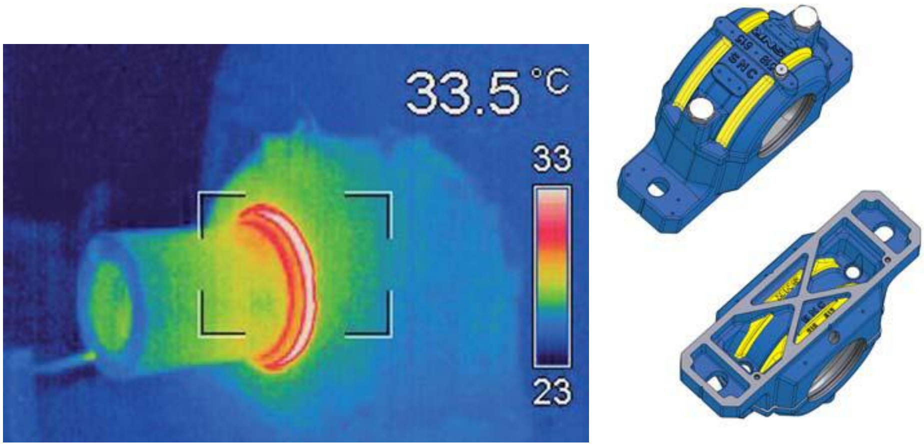


Figure 4.1.2: analysis of temperature effect on pillow block bearing housing

4.2.4 Strength properties and heat dissipation:

Temperature analysis shows the temperature created during the rotation due to the friction and the heat transfer from the hot air that injected to the blade. The circular ribbing on the housing body gives the housing excellent form stability and rigidity. Furthermore, this structural feature helps to optimize the vibration behaviour and heat dissipation of the units.

4.3 Bearing

Bearings are taken into account in the shaft calculation as border or transition conditions. Bearings are often considered stiff in the framework of the calculation with a beam model. There, the deflection is set to zero. For linearly elastic bearings, an additional equation per bearing and coordinate will be used. With bearings, the transition condition is a non-linear function of all displacements and rotational components. That is why, for a working point, it is possible to determine a stiffness leading to the same displacement. However, for critical speed calculation, the important factor is the tangent at the force-to-deflection characteristic in the working point.

A ball bearing is a type of rolling-element bearing which uses balls to maintain the separation between the moving parts of the bearing.

The purpose of a ball bearing is to reduce rotational friction and support radial and axial loads. It achieves this by using at least two races to contain the balls and transmit the loads through the balls. Usually one of the races is held fixed. As one of the bearing races rotates it causes the balls to rotate as well. Because the balls are rolling they have a much lower coefficient of friction than if two flat surfaces were rotating on each other.

Thrust ball bearing will be used for the test rig. The details part number and dimension are shown below.

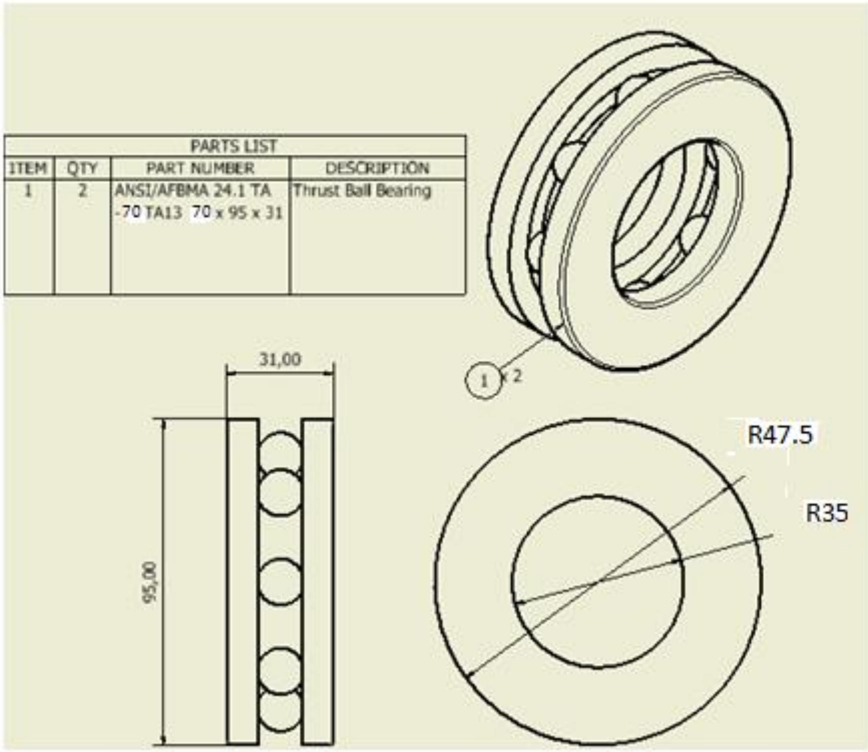


Figure 4.13: projected view of bearing

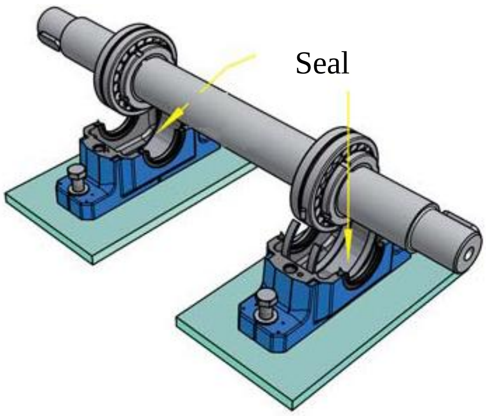


Figure 4.14: bearing attachment seal

4.3.1 Bearing lubrication

Seal system- Labyrinth seal

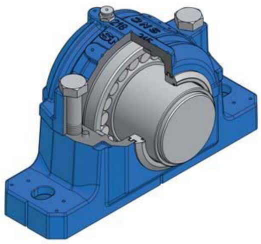


Figure 4.15: seal system

For adverse ambient conditions, the bearing housings can be fitted with labyrinth seals. The sealing ring and the sealing groove in the housing form a labyrinth with a narrow sealing gap. The great advantage of these seals is that the bearing arrangement can be operated at the permissible speed for the bearings used. The labyrinth ring is synchronised on the shaft by the installed round cord. The maximum misalignment of the shaft may not be greater than 0.3°. The operating temperature range for this seal is between -40°C and +200°C.

4.3.2 High temperature grease

The high temperature grease is used as the bearing lubrication since the rig will be operated in high temperature environment where the hot air that will be injected to the blade is 630 ^{o}C. The Grease has Dropping Point higher than 320^{o}C in the Temperature range -20^{o} C to 800^{o} C. It has excellent service life, superior stability and excellent load bearing properties.

Range: -20^{o} C (-4^{o} F) to 800^{o} C (1472^{o} F)

The Grease has:

- Lowest Coefficient of Friction
- Excellent Extreme Pressure (EP) properties due to Tungsten Disulfide (WS_2) particles. Load bearing property of Tungsten Disulfide is @ 300,000 psi.
- High resistance to water, rust and humid environment.
- Excellent mechanical stability under high shear.
- EP properties and strong resistance to abrasion means the grease is ideal for high load and impact load applications.

Figure 4.16: components of the bearing installation on pillow block bearing housing

4.4 Blade Attachment

The blade attachment is not directly connected to the shaft, it uses the coupling or plate attachment tightens by 5mm screw and nut. This design purpose is to make easier to install and uninstall and also considering other studied that need to change with other size of blade.

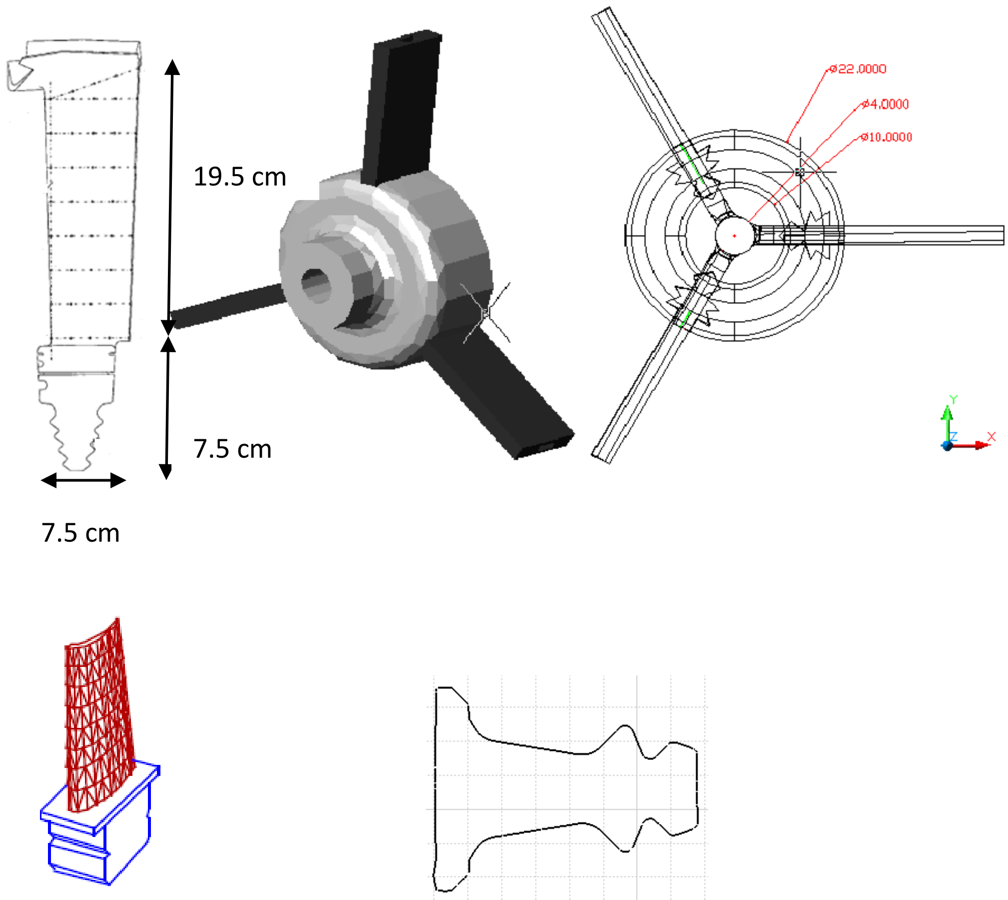
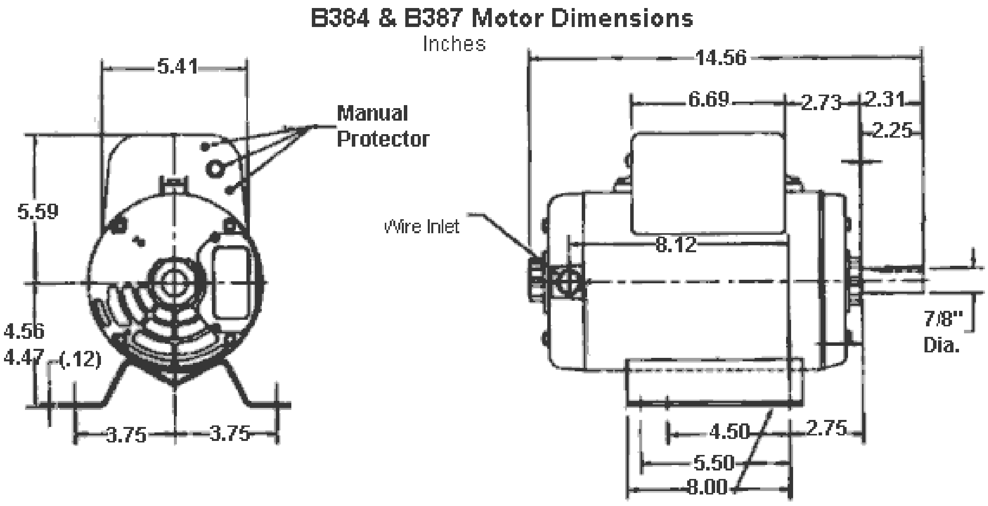


Figure 4.17: blade attachment on flange

4.5 Damper

Damper is used on the blade disk to prevent discomfort, damage, or outright structural failure caused by vibration. Damper is a disk that will be installed to support the blade during rotation that reduces the blade vibration.

4.6 Electric Motor



Electric Motor: 10kW	speed: 3500 Rpm	max Torque: 100 N.m
Belt: L: 115cm	Center –center: 50cm	Wrap angle: 180°
Input: 2000 Rpm	Output: 1000-1500Rpm	

Figure 4.18: detail of electric motor

4.7 Housing

The housing is generally acts as safety element which is used to cover the blade during the blade rotation at high Rpm. It is also to provide the high temperature environment to the blade.

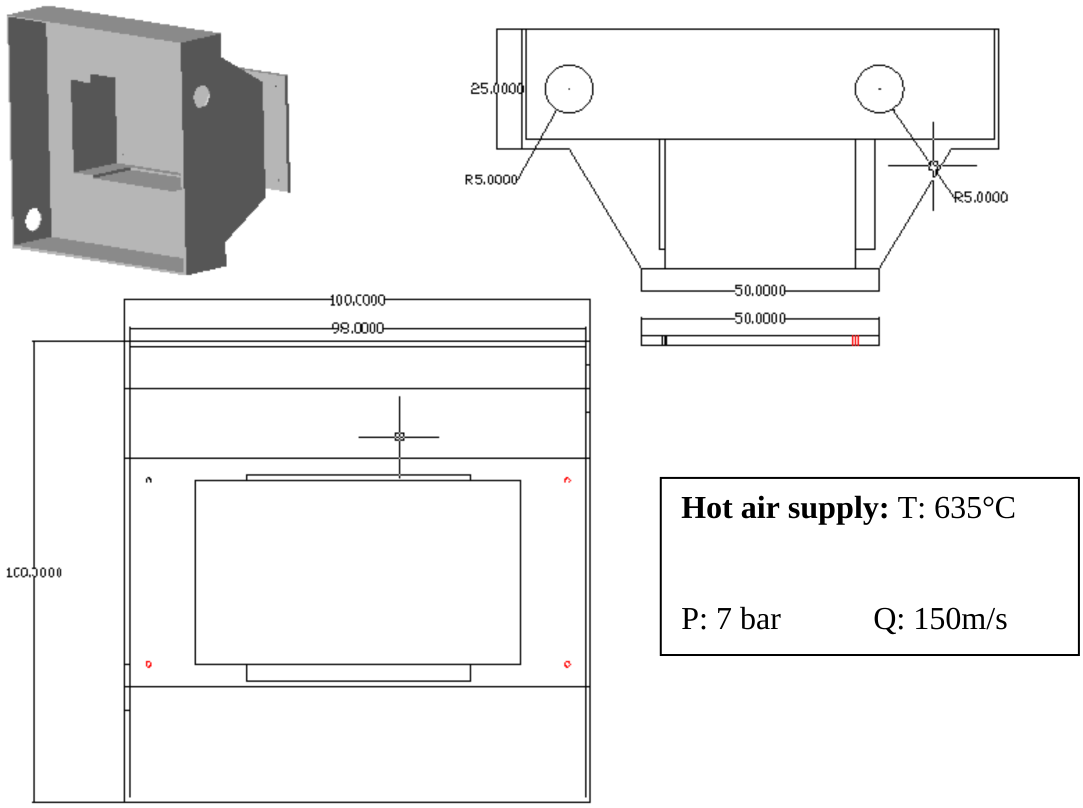


Figure 4.19: hot air and blade housing

4.8 Temperature and Heat Flow

The turbine blade is cooled by the ambient air or compressor discharge air (T3 air: 30 -70°C). This air will be injected to the turbine blade through the internal hollow rotating shaft. The air entering the fir root of the blade will used to cool the blade in the 650°C hot air condition supplied from the engine exhaust. The blade wall temperature will be measured using the infrared thermometer. The 70 bar pressure air is seals by the pneumatic valve and O-ring seal.

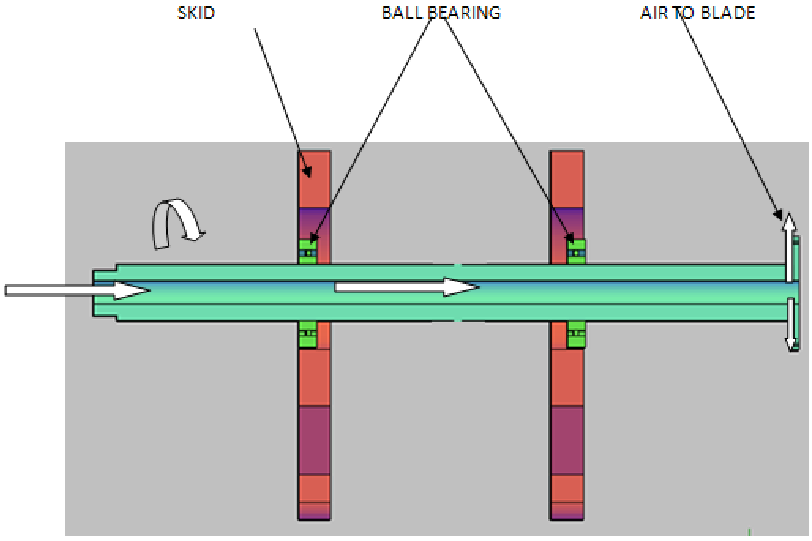
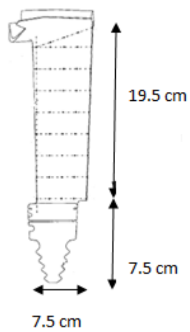
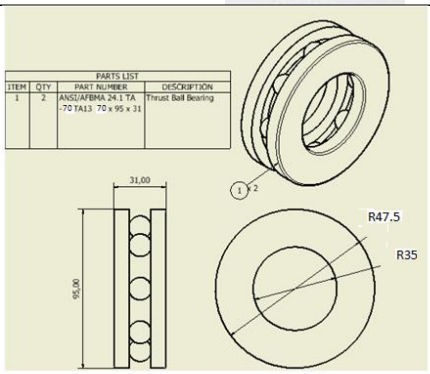
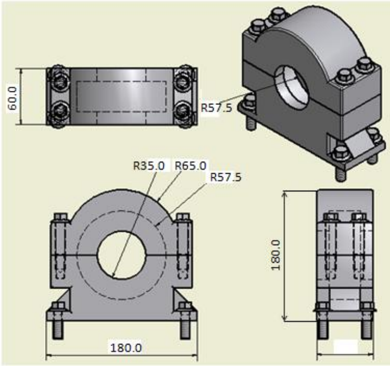
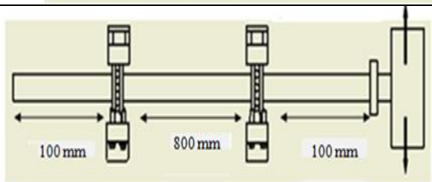
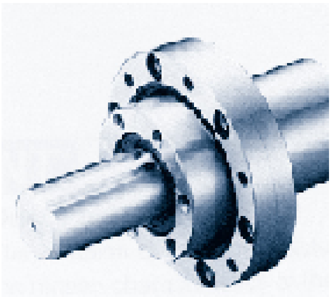
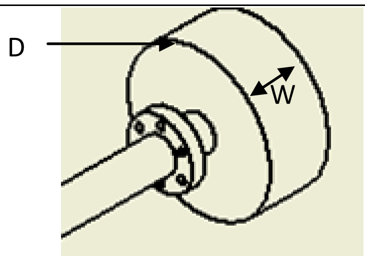
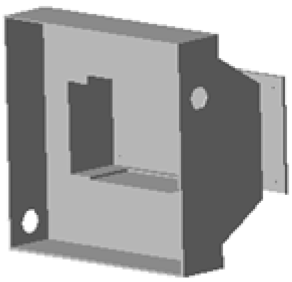


Figure 4.20: flow of air inside the shaft

4.9 Rig Parts and Specification

Table 9: list of parts and specification

ITEM		QUANTITY	SPECIFICATION
1	 	3	Turbine blade Height = 27cm Width= 6cm Length= 7.5cm Weight = 3.3kg/unit
2		2	Deep groove ball bearing ANSI/FBMA 24.1 TA 70TA13 70×95×31
3		2	Pillow block bearing housing 180x60x180 Shaft : r =35mm Bearing: inner r = 35mm ±2mm Outer r =57.5mm±2mm Pillow Block :r = 65mm

		8 8	Matrix hex flange screw ANSI B18.2.3.4M- M10x1.5x60 ANSI B18.2.3.4M- M10x1.5x30 Material: mild steel
4		1	Shaft L = 1000mm iD = 10mm oD= 70mm material: stainless steel
5		1	Rotary ball bearing ANSI/FBMA 24.1 TA 30TA13 30×40×30
6		1 6 6	Disk blade attachment D=220mm W=90mm Matrix hex flange screw ANSI B18.2.3.4M- M10x1.5x30 Hex Nut M10x1.0x10
7		1	Electric motor power: 10kW speed: 3500 Rpm max Torque: 100 N.m Output: 1000-1500Rpm
8		1	Blade housing Length=100cm Width=80cm Height=100cm

4.10 Discussion

In order to provide the specification of components of the rig system, few assumptions have been made:

- After the alignment of the shaft has been set, there is no vibration during shaft rotation. Vibration = 0
- The friction on the bearing is low and not affects the rotation.

The methodology used in designing and fabricating this test rig is possible to really solve the contradictions between design and manufacturing by adopting an integrated design to manufacturing. The method used in designing the test rig is based on the economical, safety and quality factors in order to get an accurate data for the studies. All the designing and fabrication work of the components will be fabricated by the suppliers and manufacturer.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusion

Gas turbine engine is widely used in oil and gas industry and other engineering fields as a driver to provide high rotational speed through the output shaft to operate other equipments like pumps and compressors. In a gas turbine engine, power turbine blade is located after the combustion chamber in high pressure and temperature environment, and operates at high rotation speed. This extreme environment can damage the blade and reduce its strength. In order to encounter this problem, a cooling system of the blade needs to be studied. The project on development of a test rig for the study of internal blade cooling is an encouraging effort to provide the equipment for the study and investigation not only for the internal blade cooling system in a gas turbine engine, but also for other energy subject, like heat transfer, thermodynamics and fluid dynamics. The objective of this project is to develop a test rig of gas turbine engine model that will be used for the study of the cooling system. Besides, with the presence of a test rig at the end of this project, the heat movement and temperature flow profile can be studied on the rig. For this project, the test rig development process covers all the engineering criteria which are important in providing the design specification. In order to complete the development of the test rig, engineering methodology have been applied and considered involving economic and safety criteria. The mechanical computerize and conventional machine and equipments have been used to do experiment and for the fabrication activities. All the design specification and material selection is based on the engineering standards including some calculation and analysis on the force and stresses .The finalized and complete design will be provide to the manufacturer to fabricate the rig system, hence, with the presence of the test rig, the experiment activities can be conducted in order to gain the experimental data to prove the computerizes simulation result on the internal blade cooling of a gas turbine engine. The designing and development process of the test rig have met the objective of this project.

5.2 Recommendation

To make the test rig running successfully and providing accurate data, the test rig need a design which is same and fully utilize the original concept and working condition of a real gas turbine engine. This development process is more efficient by having cooperation and help in term of advice from the manufacturer in providing the specification. In building the test rig, few problems have been countered such as the absence of full package CATIA software in UTP. It is highly recommending that UUTP need to have a full package of CATIA software which is very important and useful not only for this project, but also for other studies of other students in future.

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3. http://wwwedc.eng.cam.ac.uk/research/processmanagement/pm3/turbinedesign/
4. www.snr-bearing.com

Suppliers/ factory Contact person

1. Igus Malaysia Sdn Bhd- bearing supplier (Mr. chow:012-7093041)
2. Kean Seng Engineering Works (Mr. F.L.HO:016-5631170)
3. Pusing steel works (Mr. Wong: 012-5228649)

APPENDIX A
WORK ACTIVITIES FOR FYP I

week	Activities	problems
1	Selection of project topics	
2	Preliminary research work • gas turbine working principal • internal turbine blade cooling system • Discussion with SV and lab technician on design details.	Blade available in UTP is too big. Shaft design is depend on the blade size
3	Preliminary research work • Request and find turbine blade from turbine supplier. • design the plan detail for the test rig • identified each components type, design concept • prelim report	Still waiting the blade - All the dimensioning for other parts can't be decided due to the depending on blade size.
4	• Request and find turbine blade from turbine supplier. • research on the properties and price for the material • search for the steel manufacturer • design selection • submission of prelim report	Turbine supply company not agrees to give the blade due to confidential issue.
5	• design bearing and skid • material selection • take parameters of the engine in block 15	Some part cannot be machined in UTP
6	• design blade housing • revise the manufacture/ supplier catalog	
7	• Decide to use blade available in UTP • decide the component dimension	Need high power motor to turn big and heavy shaft and blade
8 to 9	• do survey on electric motor and air injection system • finalized bearing and shaft size	
10 to 12	• do survey on machine that able to get the fir root drawing and dimension	
13 to 14	• Completing the final report and oral presentation	

APPENDIX B
WORK ACTIVITIES FOR FYP II

Week	Activities	Problems
Semester break	• Provide detail drawing of the blade fir root or tail using CNC laser digitizer and CMM machine	• The machine only can provide drawing in point cloud format • Drawing only can be opened using full version of CATIA software which is can't be done using software in laboratory
1 t o 2	• Finalized design specification for fabrication • Conduct stress analysis using ANSYS software • Working on blade fir root dimension	• CNC machine not available in steel factory
2 to 4	• Survey the manufacturer and prices offered	• No company offers to fabricate and assemble parts
5 to 6	• Do survey on the disk blade attachment fabricator	• Need to provide casting to the fabricator
7 to 8	• Meet the bearing suppliers in KL • Do survey on the disk blade attachment fabrication • Survey the casting machine	• Suppliers agree to give the quotation on the bearing • No factory take custom casting process for only one product
9 to 10	• Submission of progress report II	•
10 to 12	• Contact suppliers for bearing quotation • Continue survey for other options • Poster submission • Fabricate prototype using rapid prototyping machine	• Suppliers do not give cooperation since this do not give big benefit for them • Prototype cannot be rotated due to brittle material
12 to 14	• Completing the documentation for the rig design • Submission of dissertation report and final presentation	•

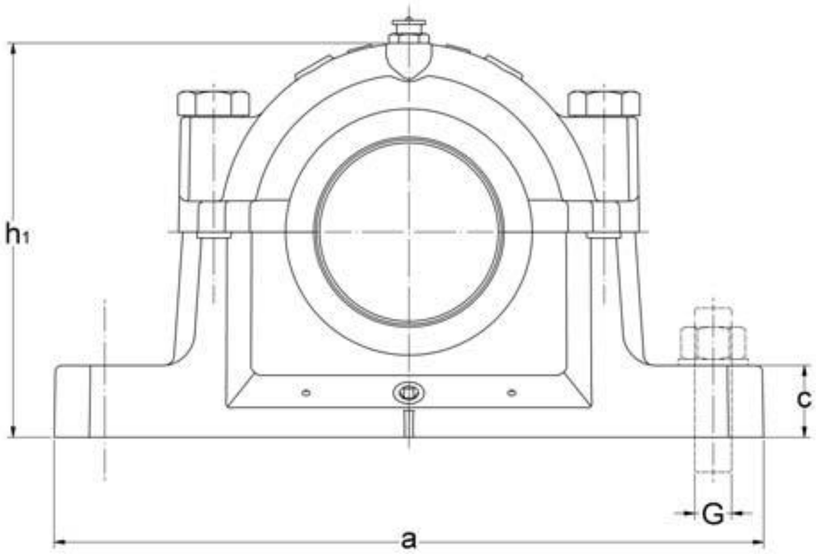
APPENDIX C
WORK PLAN FOR FYP I

No.	Detail/ week	1	2	3	4	5	6	7	8	9		10	11	12	13	14
1	Selection of project topic	■	■													
2	Preliminary research work on developing a test rig		■	■	■	■										
3	Submission of preliminary report				●											
4	Study and finalize the Design specification, detail, design selection of a test rig					■	■	■	■	■	Mid semester break					
5	Submission of progress report and seminar presentation								●							
6	Analysis on load and force distribution on rig components											■	■	■		
7	Finalize and provide rig spec for fabrication												■	■	■	
8	Submission of interim report															●
9	Oral presentation											Study week				

APPENDIX D
WORK PLAN FOR FYP II

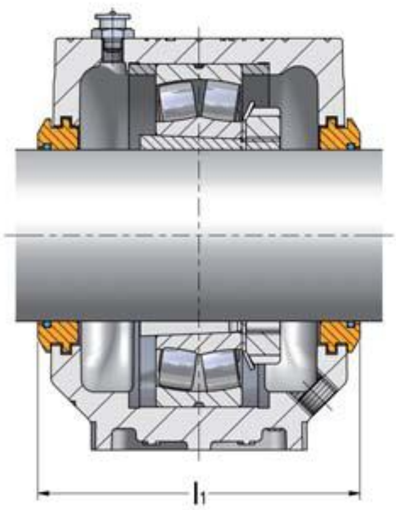
No.	Detail/week	1	2	3	4	5	6	7	8	9		10	11	12	13	14
1	Finalized design specification on dimension base on availability and cost for fabrication										Mid semester break					
2	Survey fabricators for fabrication the disk blade attachment and bearing															
3	Submission of progress report Survey CNC machine					●										
4	Do survey on the price (cost) and time offered by manufacturer for fabrication															
5	use reverse engineering for disk blade fabrication-casting															
6	Poster submission Survey casting machine															
7	Completing the documentation for rig design															
8	Submission of dissertation report															●
9	Final presentation													Study week		

APPENDIX-E
PILLOW BLOCK HOUSING FOR BEARING MOUNTING



	Housing dimensions [mm]																
D	a	b	c	g	h	l	m	G	u	v	h_1	m_1	n_2	m_2	n_1	n_3	Weight approx[1] [kg]
80	205	60	25	39	60	85	170	M12	15	20	107	160	44	188	34	50	3,2

APPENDIXE F
LABYRINTH SEAL



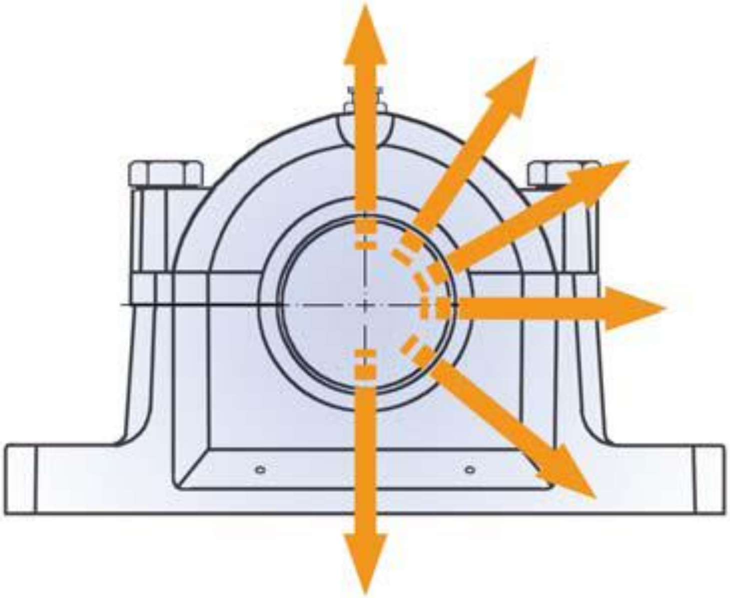
APPENDIXS G
SEAL SELECTION

Structural properties		SC..DS Double lip seal	SC..FS Felt strip seal	SC..SV V-ring seal	SC..LA Labyrinth seal	SC..TA Taconite seal
Operating temperature	°C	-40…+100	-40…+100	-40…+100	-40…+200	-40…+100
Circumferential speed	m/s	< 8	< 15	< 7[3]	> 15	< 10[4]
Possible misalignment	degrees	0,5…1	< 0,5	1…1,5	< 0,3	< 0,5
Relubrication		Unsuitable	Limited suitability	Suitable	Suitable	Ideally suited
Low friction		Suitable	Suitable[1]	Ideally suited	Ideally suited	Suitable[1]
Suitable for floating bearings		Ideally suited	Ideally suited	Unsuitable	Suitable	Suitable
Vertical installation		Suitable	Limited suitability	Ideally suited[2]	Unsuitable	Limited suitability
Sealing behaviour for:						
Splash water / moisture		Suitable	Limited suitability	Suitable	Unsuitable	Ideally suited
Ultra fine particles		Ideally suited	Suitable	Suitable	Suitable	Ideally suited
Fine particles		Ideally suited	Limited suitability	Suitable	Suitable	Ideally suited
Large particles		Suitable	Limited suitability	Limited suitability	Suitable	Ideally suited
Sharp-edge particles		Suitable	Suitable	Unsuitable	Ideally suited	Ideally suited
UV resistance		Suitable	Ideally suited	Unsuitable	Ideally suited	Ideally suited

Ideally suited — Limited suitability — Suitable — Unsuitable

[1] During running-in phase up to approx. 5m/s
[2] If V-ring is fitted inside on underside.
[3] Without additional supporting ring (axially secured: 7-12 m/s); axially and radially secured: >12 m/s)
[4] Depending on shaft diameter

APPENDIX H
LOADS AND TORQUE



Housing size SNC				Housing breaking loads in load direction 0° [kN]	55°	90°	120°	150°	180°	Connecting bolts (upper/lower section)[1] Property class 8.8	Max. Load capacity for both bolts in load direction 120° [kN]	150°	180°	Recommended tightening torque [Nm]	Foot bolts[1] Property class 8.8	Max. tightening torque [Nm]
205		505		180	160	95	70	60	80	M10x40	60	35	30	65	M12	87
206	305	506	605	200	170	100	80	67	85	M10x40	60	35	30	65	M12	87
207	306	507	606	224	190	121	85	80	95	M10x45	60	35	30	65	M12	87
208	307	508	607	265	220	132	95	85	115	M12x50	80	45	40	65	M12	87
209		509		280	235	140	100	90	120	M12x55	80	45	40	65	M12	87
210	308	510	608	315	265	160	121	110	140	M12x55	80	45	40	65	M12	87
211	309	511	609	355	280	170	125	118	145	M16x60	180	100	90	150	M16	210
212	310	512	610	355	300	180	132	125	160	M16x60	180	100	90	150	M16	210
213	311	513	611	400	345	210	150	132	170	M16x70	180	100	90	150	M16	210

APPENDIX I
SHAFT BENDING MOMENT DISTRIBUTION