AN-NAJAH NATIONAL UNIVERSITY

FACULTY OF ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

WIND PUMP

Graduation Project Submitted in Partial Fulfillment of the Requirements for the Degree of B.Sc in Mechanical Engineering.

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الإهداء

الحمد لله رب العالمين الذي أنعم علينا بنعم كثيرة ورزقنا من المعرفة والعلم لتكون مشكاة تضيء لنا الدرب وتعيننا على نوائب الدهر والصلاة والسلام على سيدنا محمد خاتم الأنبياء والمرسلين.

نهدي هذا العمل إلى من سهرن الليالي الطوال صاحبات الحنان أمهاتنا الحبيبات

إلى الماس الذي لا ينكسر نبع العطاء المتواصل إلى الوالد حيا وما بعد الحياة

إلى صناع المجد والكرامة حاملي رايات الشرف والبطولة شهدائنا الأبرار

إلى قناديل الدرب والشموع التي لا تنطفئ أساتذتنا الكرام

والى كل من ساهم في إنجاح برنامجنا التعليمي

إليكم جميعا نهدي هذا العمل المتواضع.

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ABSTRACT

The main object of the project is to make watering process easier using the wind pump. The project may be important to farmers who have problems with watering their trees and plants in there far lands. The main expectation is to solve the problem by making a prototype of wind pump

This project was made a way too far from nowadays but never really came to Palestine although it is important to it, so the project will be making one without the needs of a foreign one by using simple equipments .

CHAPTER ONE: INTRODUCTION

The project of a water wind pump design is based on the hydraulic power science and mechanical design using two main components, these are wind mill (wind pump) and the water pump.

This project will help to solve the problem of watering the crops and plants from wells, where farmers used to bring water from the well by buckets using their hands and walk to the tree or the crops, which takes long time and hard process job.

The aims of this project are to use the power of winds to raise water from the will for the irrigation process so that the process would be easier and faster .

1.1 Hydraulics [1]

Hydraulics is a topic in applied science and engineering dealing with the mechanical properties of liquids. At a very basic level hydraulics is the liquid version of pneumatics. Fluid mechanics provides the theoretical foundation for hydraulics, which focuses on the engineering uses of fluid properties. In fluid power, hydraulics is used for the generation, control, and transmission of power by the use of pressurized liquids. Hydraulic topics range through most science and engineering disciplines, and cover concepts such as pipe flow, dam design, fluidics and fluid control circuitry, pumps, turbines, hydropower, computational fluid dynamics, flow measurement, river channel behavior and erosion.

1.2 Wind Pumps [2]

A wind pump is a windmill used for pumping water, either as a source of fresh water from wells, or for draining low-lying areas of land. Once a common fixture on farms in semi-arid areas, wind pumps are still used today where electric power is not available or expensive.

Figure 1.1 Wind Powered Water Pump

Windmills were used to pump water since at least the 9th century in what is now Afghanistan, Iran and Pakistan. The use of windmills became widespread across the Muslim world and later spread to China and India as well. Windmills were later used extensively in Europe, particularly in the Netherlands and the East Anglia area of Great Britain, from the late Middle Ages onwards, to drain land for agricultural or building purposes. Early immigrants to the New World brought with them the technology of windmills from Europe.

To construct a wind pump, the shaft of the fan needs to be matched to the pump. With non-electric wind pumps, high solidity shafts are best used in conjunction with positive displacement (piston) pumps, because, single-acting piston pumps need about three times as much torque to start them as to keep them going. Low solidity rotors, on the other hand, are best used with centrifugal pumps, water ladder pumps and chain and washer pumps, where the torque needed by the pump for starting is less than that needed for running at design speed. Low solidity rotors are best used if they are intended to drive an electricity generator; which in turn can drive the pump.

CHAPTER TWO: PUMPS[3]

2.1 Introduction

A pump is a device used to move fluids, such as liquids, gases or slurries. A pump displaces a volume by physical or mechanical action. Pumps fall into three major groups: direct lift, displacement, and gravity pumps. Their names describe the method for moving a fluid.

There is more than twenty different kinds of pumps each one for a different purpose, but in this project tow kinds of pumps are to be tested the first one is the rotary vane pump and the second one is the piston pump.

2.2 Pumps Types

With the development of industry and progress in scientific research, the pumps were developed, each depending on the purpose of use, and here are some of those types.

2.2.1 Positive displacement pump

A positive displacement pump causes a fluid to move by trapping a fixed amount of it and then forcing (displacing) that trapped volume into the discharge pipe.

Some positive displacement pumps work using an expanding cavity on the suction side and a decreasing cavity on the discharge side. Liquid flows into the pump as the cavity on the suction side expands and the liquid flows out of the discharge as the cavity collapses. The volume is constant given each cycle of operation.

Positive displacement pump behavior:

Positive displacement pumps, unlike centrifugal or roto-dynamic pumps, will in theory produce the same flow at a given speed (rpm) no matter what the discharge pressure. Thus, positive displacement pumps are "constant flow machines". However due to a slight increase in internal leakage as the pressure increases, a truly constant flow rate cannot be achieved.

A positive displacement pump must not be operated against a closed valve on the discharge side of the pump, because it has no shut-off head like centrifugal pumps.

positive displacement pump operating against a closed discharge valve will continue to produce flow and the pressure in the discharge line will increase, until the line bursts or the pump is severely damaged, or both.

Positive displacement types

A positive displacement pump can be further classified according to the mechanism used to move the fluid:

Rotary-type positive displacement: internal gear, screw, shuttle block, flexible vane or sliding vane, circumferential piston as shown in figure 1.2, helical twisted roots.

Reciprocating-type positive displacement: piston or diaphragm pumps.

Linear-type positive displacement: rope pumps and chain pumps.

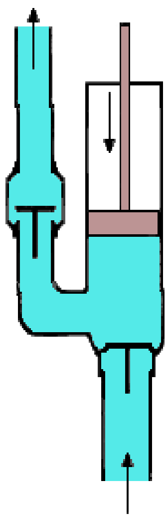


Figure 2.1 Piston Pump

2.2.2 Impulse pumps

Impulse pumps use pressure created by gas (usually air). In some impulse pumps the gas trapped in the liquid (usually water), is released and accumulated somewhere in the pump, creating a pressure which can push part of the liquid upwards.

Impulse pumps types

Hydraulic ram pumps: uses pressure built up internally from released gas in liquid flow.

Pulsar pumps - run with natural resources, by kinetic energy only. as shown below.

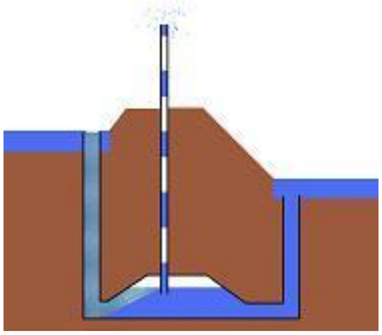


Figure 2. 2 The Pulsar Pump

Airlift pumps- run on air inserted into pipe, pushing up the water, when bubbles move upward, or on pressure inside pipe pushing water up.

2.2.3 Velocity pumps

Rotodynamic pumps (centrifugal pumps figure 2.3) are a type of velocity pump in which kinetic energy is added to the fluid by increasing the flow velocity. This increase in energy is converted to a gain in potential energy (pressure) when the velocity is reduced prior to or as the flow exits the pump into the discharge pipe. This conversion of kinetic energy to pressure can be explained by the First law of thermodynamics or more specifically by Bernoulli's principle.

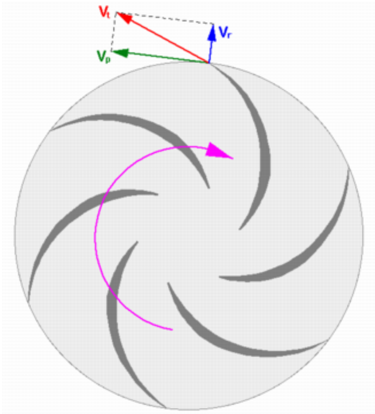


Figure 2.3 A Centrifugal Pump uses a Spinning Impeller

One practical difference between dynamic and positive displacement pumps is their ability to operate under closed valve conditions. Positive displacement pumps physically displace the fluid; hence closing a valve downstream of a positive displacement pump will result in a continual build up in pressure resulting in mechanical failure of either pipeline or pump. Dynamic pumps differ in that they can be safely operated under closed valve conditions (for short periods of time).

CHAPTER THREE: METHODOLOGY

3.1 Introduction

In order to build a fine wind pump the work should be divided in two main parts. The first one is to study the location of the well that the wind pump is to be placed in and know how deep the well is down. The other part is the mechanical design of the wind pump and the simulation of all parts then the analysis of the parts.

3.2 Basic Instructions

- Find out the depth of the well and that is 6m.
- The fan is to stand a minimum of 4 meters above anything that could abstract the wind.
- The wind pump can be directly installed over the water source (the well).
- Connect the entire mechanism (fan, shaft, bearings, crank, rocker, coupler, and column).
- the pump is to be installed.

The drive mechanism transfers the energy produced by the wind to the pump that pumps the water. Then comes hooking up all the piping and tank storage unit.

3.3 Wind Pump Mechanism Building Process

3.3.1 Main parts

1- Fan : consists of eight blades.

2- Shaft.

3- Crank.

4- Bearings.

5- Coupler.

6- Rocker.

7- The Column.

8- Pump (piston pump).

9- Plastic tubes.

10- Tank to collect water that is pumped from the well.

3.3.2 Working procedure

1. first of all building the fan of the mill. That is 3.8m diameter and made of eight rods of steel each one is 1.85 m long and then connect the 70*60 cm plastic sheets to the rods in an angle of 45º to take the maximum energy from the winds to get a great efficiency of the mills fan rotational speed. as shown in figure 3.1

Figure 3.1 Constructed Fan

2. Installing the fan of the mill on a 4m long column of steel by using a bearing.

3. after that the energy have to be transported from the fan to the pump by connecting the fan with the pump by using a crank.

In this project two kinds of pumps were used in order to pump the water from the well. The first one was the rotary vane pump as shown in figure 3.2, the problem in using this pump was that it needs high speed relatively. to solve this problem the pump was replaced by a two piston pump as shown in figure 3.3.

Figure 3.2 Rotary Van Pump

Figure3. 3 Pump Installation

3.3.3 The Used piston pump

The used pump consists of two pistons pump, the pump is handmade one because the available piston pumps in the markets are very cost.

Figure3.4 The Handmade Piston Pump

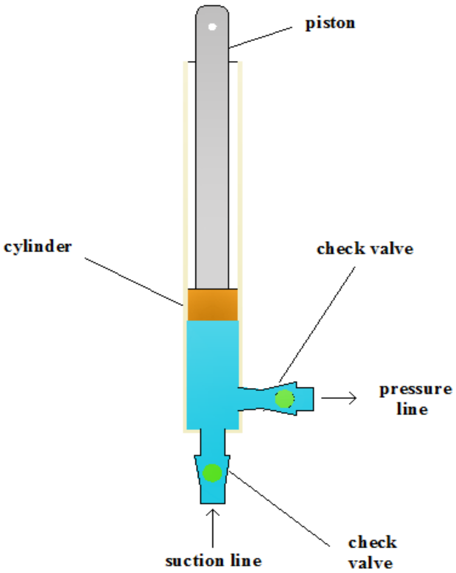


Figure 3.5 Piston Pump Parts

CHAPTER FOUR: FLUIDE ANALISYS

4.1 Introduction

the fluid analyses was established according to the head that the used pump was able to reach, and therefore the force needed to move the piston could be calculated. the flow rate of the pump was calculated by measuring the dimensions of the pump cylinder, knowing that the pump consists of tow cylinders.

if the change in the internal volume of the cylinder is known in each extension of the cylinder, then the theoretical volumetric displacement of the pump will be known, and the theoretical flow rate will be known too, by finding out the number of cycles the piston makes per minutes.

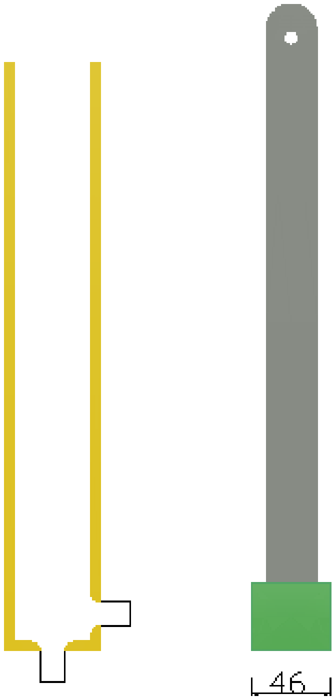


Figure 4.1 Piston Dimensions (mm)

4.2 Volumetric Displacement Calculation[4]

The volumetric displacement (***VD***) is the volume of water that the pump can rise by each cycle of work. and this quantity depends on the change of the internal volume of cylinder by each extension and on the number of piston in the pump.

$$VD_{th} = A \times S \times n \quad (4\text{-}1)$$

where: VD_{th} is the theoretical volumetric displacement.

A is the inner cross sectional area of the cylinder.

S is the stroke of the piston.

n is the number of pistons.

$A = \pi r^2 = \pi(2.3)^2 = 16.6$ cm^2

$S = 24$ cm

$n = 2$ pistons

$VD_{th} = (16.6)(24)(2) = 796.8$ cm^3

Because every pump has an efficiency then there is a losses in the theoretical volumetric displacement. to know the actual volumetric displacement of the pump VD_{act} there is an experiment should be done. in this experiment the volume that the pump can rise from the well was measured in six trails, and the results are shown below in table (4-1).from these results the volumetric efficiency (***VD%)*** of the pump could be calculated.

$$VD\% = \frac{VD_{act}}{VD_{th}} \tag{4-2}$$

Table 4.1 Volumetric Efficiency

VD_{th} (liters)	VD_{act} (liters)	$VD\%$
0.7968	0.565	71%
0.7968	0.533	67%
0.7968	0.525	66%
0.7968	0.597	75%
0.7968	0.621	78%
0.7968	0.585	74%

Sample of calculation

Trail (1)

$VD\% = 0.565/0.7968 = 71\%$

Average volumetric efficiency $VD\% = 71.75\%$

4.3 Flow Rate Calculations[4]

The flow rate (***Q***) depends on the fans revolutions per minute and also depends on the volumetric displacement of the pump.

Theoretical flow rate $Q_{th} = VD*N$ (4-3)

Where: ***VD*** is the theoretical volumetric displacement (liters).

N is the angular speed of the fan (rev/min).

To find the rotational speed ***N*** the following experiment must be done.

the speed was measured in four days, three times a day, and the results was tabulated in the following table:

Table 4.2 Experimental Rotational Speed N (rpm)

DAY	7:00 AM	3:00 PM	7:00 PM	Average (rpm)
	N (rpm)	***N*** (rpm)	***N*** (rpm)	
first day	9	36	26	23.666667
second day	6	28	21	18.333333
third day	11	29	23	21
forth day	6	22.5	24	17.5
			N_{avg} = 20.11 rpm	

$Q_{Theoretical} = VD * N_{avg} = 0.7968 * 20 = 16$ liter/min

Q_{actual} = volumetric efficiency * theoretical flow rate

Q_{actual} = 71.75% * 16 (liter/min) = 11.48 liter/min

As the fan placed in a suitable location, suppose it works eight hours a day on average, then the amount of water that will be pumped from the well is calculated by:

$V=Q_{act}*t$ (4-4)

Where ***t*** is the working period in minutes

$t = 8*60 = 480$ minutes

$V = 11.48 * 480 = 5.5104\ m^3/day$

This amount of water pumped per day is enough to be used to irrigate 450 young olive trees, furthermore farmers do not irrigate their crops daily.

4.4 Aerodynamics Calculations

This section deals with the forces that acts on the fan blades due to the flow of air that causes it to rotate. as the air hit the blades there is a torque arises on the shaft of the fan as shown in figure 4.2 , due to lack of equipment needed to measure the torque this torque was measured experimentally by suspending a load on the crank and increasing this load until the fan could not rotate. with this load and Knowing the distance between the suspending point and the center of the shaft, the resultant torque could be calculated.

this experiment was Implemented in a windy day, in order to find out the maximum force that affects the fan blades.

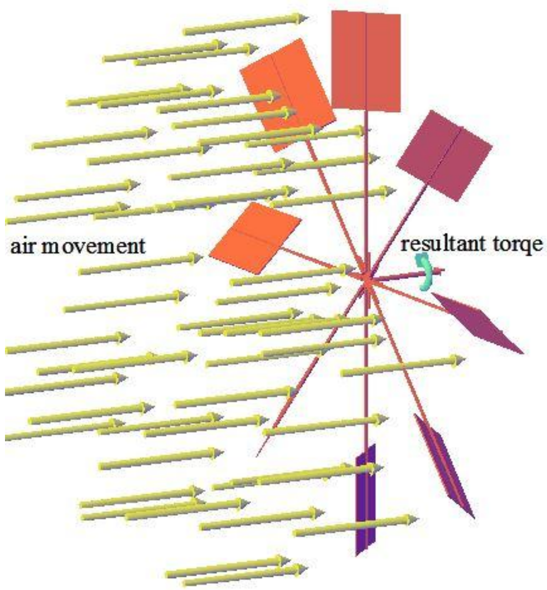


Figure 4.2 Air Movement and Resultant Torque.

the measured torque was 98 N.m, knowing that this torque produced by eight blades, the produced torque by each blades is the result of dividing the total torque by eight.

T_{blade} = 98/8 = 12.25 N.m

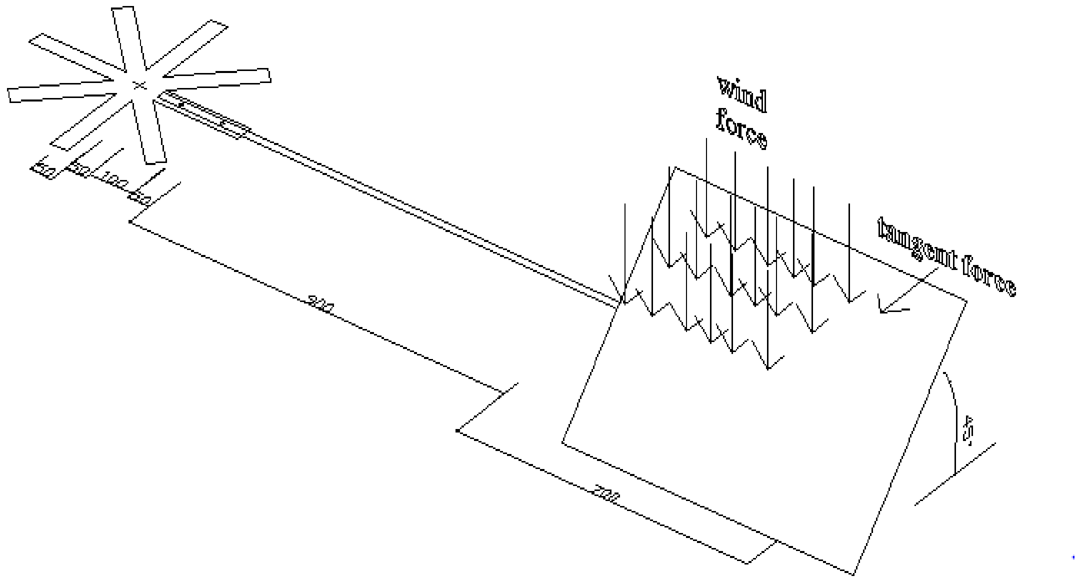


Figure 4.3 Wind Force and Dimensions (mm).

if this torque divided by the distance from the center of the plade to the center of the shaft, the tangent force that produce this torque can be calculated.

$$T_{plade} = F_{tangent} * d \qquad (4\text{-}5)$$

$$F_{tangent} = T_{plade} / d$$

where d is the distance from the center of the blade to the center of the shaft.

$F_{tangent} = 12.25/1.85 = 6.7$ N

as shown in figure 4.3 the blade oblique by angle of 45^{o} so the bending force is exactly as the tangent force; but before calculating the bending force it must be taken into consideration that the actual tangent force is bigger than the calculated one because it losses some work against the air resistance. as this work couldn`t be calculated, the tangent force value was increased by the half in order to correct its value.

$F_{tangent(correct)} = 1.5 * F_{tangent}$

$F_{tangent(correct)} = 1.5*6.7 = 10.05$ N

$F_{bending} = 10.05$ N , this force produce bending load on the rod that support the blade.

4.5 Forces on the Pistons of the Pump[4]

To find this force it important to explain the principle of the piston pump. when a force acts on the piston in the suction stroke, a negative pressure (vacuum pressure P_v) is resulted, when the absolute value of this pressure equal the pressure of the same head of liquid from the level of water to the level to which water needed pumped, the water rises from the source. and in compression stroke the water rises from the cylinder to the reservoir (tank).

$P_v = \rho * g * h$

where ρ is the density of the water (Kg/m^3)

g is gravity acceleration (m/sec^2)

h the head of the pump above water surface (m).

$\rho = 1000\ Kg/m^3$

$g = 9.81\ m/sec^2$

$h = 6\ m$

$P_v = 1000*9.81*6 = 58860\ Pa$

The pressure from the piston is $P_{theoretical} = 58860\ Pa$

But this value is smaller than the actual pressure because of pressure drop in valves and tubes. to find true pressure from the piston, the pressure drop must be added to the theoretical value. as a result of the used valves the pressure drop is high somehow, assumed to be half the theoretical pressure.

$P = P_{theoretical} * 1.5$

$P = 58860*1.5 = 88290\ Pa$

$F_{on\ the\ piston} = P * A$

where A is the piston area (m^2)

$A = 0.001619\ m^2$

$F_{on\ the\ piston} = 88290*0.001619 = 146\ N$

this force must be applied on each pistons. so the force that the coupler must affects on the rocker of the pump must produce a moment equal to the moment from the pistons forces as shown in figure 4.6.

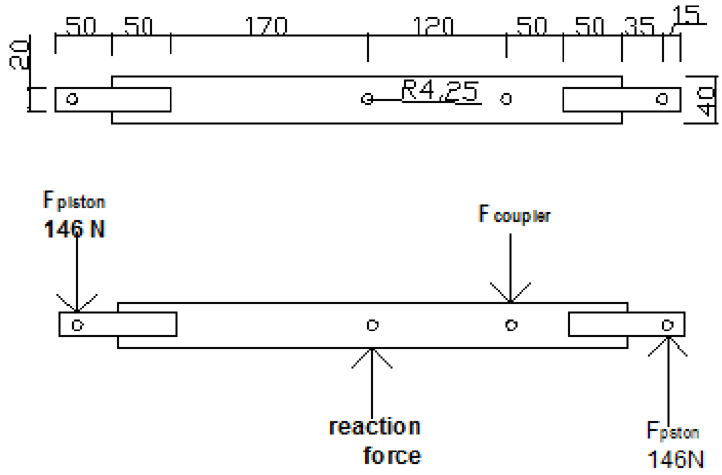


Figure 4.6 Piston Force

from static analysis $\boldsymbol{F_{coupler}}$ = 632 N

CHAPTER FIVE: DESIGN ALALYSIS

5.1 Introduction

The technique that has been followed in this chapter is that the safety factor was calculated in each part of the prototype and according to this factor it was decided to re-design that part to fit the circumstances or not.

As almost all the parts in the project exposed to fatigue loading, so It is important to clarify some concepts and terminology used in the analysis. and here are some of these terms.

Moment (*M*): is the tendency of a static force to twist or rotate an object; see the article torque for details. This is an important, basic concept in engineering and physics. A moment is valued mathematically as the product of the force and the moment arm. The moment arm is the perpendicular distance from the point of rotation, to the line of action of the force. the effect of the moment is a normal stress. [5]

Torque (*T*): it is the same as moment but it is result from a moving force, and the effect of it is a shear stress. [5]

Stress (σ): is a measure of the internal forces acting within a deformable body Quantitatively, it is a measure of the average force per unit area of a surface within the body on which internal forces act. [5]

Shear Stress (τ): is defined as the component of stress coplanar with a material cross section. Shear stress arises from the force vector component parallel to the cross section. Normal stress, on the other hand, arises from the force vector component perpendicular to the material cross section on which it acts. [5]

Von Mises Stress or **Equivalent Tensile Stress** (σ'), a scalar stress value that can be computed from the stress state. In this case, a material is said to start yielding when its Von Mises stress reaches a critical value known as the yield strength S_y. The Von Mises stress is used to predict yielding of materials under any loading condition from results of simple uniaxial tensile tests. The von Mises stress satisfies the property that two stress states with equal distortion energy have equal von Mises stress. [5]

Endurance Limit (S_e): are all expressions used to describe a property of materials, the amplitude or range of cyclic stress that can be applied to the material without causing fatigue failure. [6]

The Modified Endurance Limit (S_e') : it is the endurance limit that takes into consideration the conditions of the part. [6]

Goodman Relation: is an equation used to quantify the interaction of mean and alternating stresses on the fatigue life of a material.

Ultimate tensile strength (S_{ut}): or tensile strength or ultimate strength, is the maximum stress that a material can withstand while being stretched or pulled before necking, which is when the specimen's cross-section starts to significantly contract. [5]

Yield Strength (S_y): it is defined in engineering and materials science as the stress at which a material begins to deform plastically. at the yield point the material will deform elastically and will return to its original shape when the applied stress is removed. Once the yield point is passed, some fraction of the deformation will be permanent and non-reversible. [5]

Second Moment Of Area (I): is a property of a cross-section that can be used to predict the resistance of a beam to bending and deflection around an axis that lies in the cross-sectional plane. The stress in, and deflection of, a beam under load depends not only on the load but also on the geometry of the beam's cross-section: larger values of second moment cause smaller values of stress and deflection. This is why beams with larger second moments of area, such as I-beams, are used in building construction in preference to other beams with the same cross-sectional area. [5]

Second Polar Moment Of Area (J): is a property of a cross-section that can be used to predict the resistance of a beam to twist around an axis that pass through the centroid of the cross-sectional plane. [5]

modulus of elasticity (E): is the mathematical description of an object or substance's tendency to be deformed elastically when a force is applied to it. The elastic modulus of an object is defined as the slope of its stress–strain curve in the elastic deformation region. [5]

5.2 Calculations [6] [7]

5.2.1 The Rocker

As shown below.

Figure 5.1 Rocker

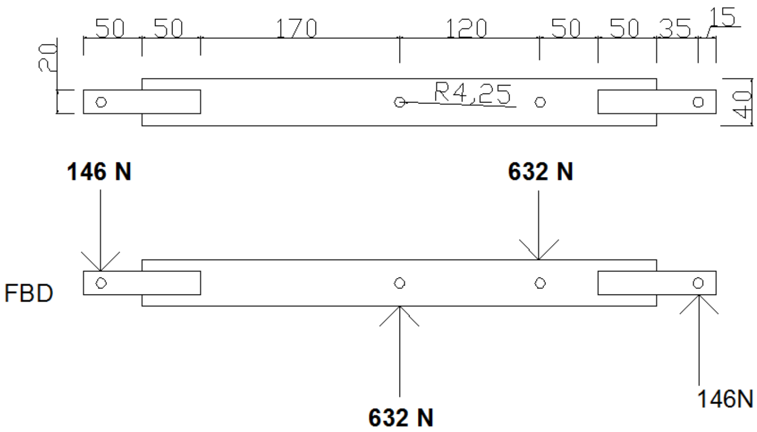


Figure 5.2 Rocker Dimensions(mm) and loads(N)

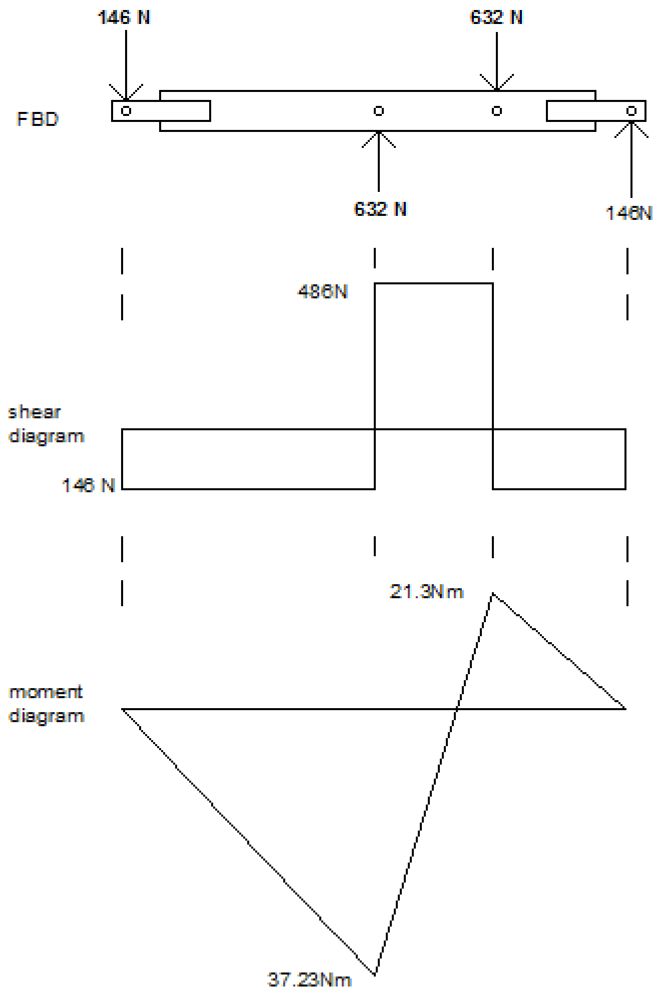


Figure 5.3 Shear and Moment Diagram

The rocker is 8x40mm rectangular cross section bar. it is made of AISI 1010 (low carbon steel).

from table A-1

$\boldsymbol{S_y}$ =180 MPa.

$\boldsymbol{S_{ut}}$ = 320 MPa

from the moment diagram the section b-b is the critical section in the bar.

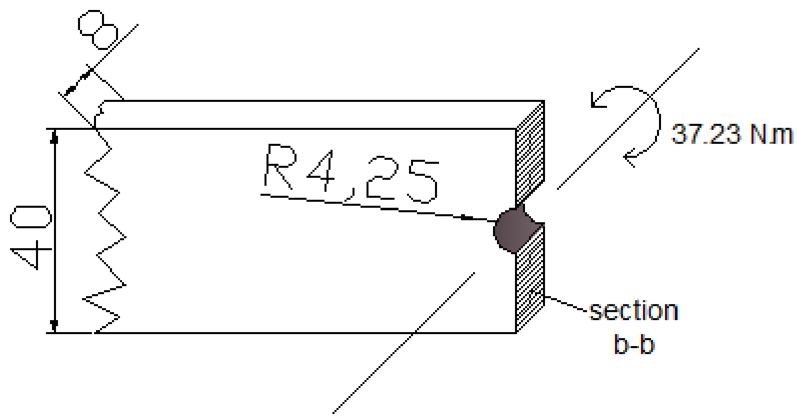


Figure 5.4 Section b-b.

$$\sigma_o = \frac{MC}{I} \tag{5-1}$$

where: σ_o is the normal stress on the section.

M the moment on the section.

C the farthest distance from the neutral axis.

I is the second moment of area.

M = 37.23 N.m $\qquad$ C = 0.02 m

$$I=\frac{1}{12}\,b(h2^3-h1^3) \tag{5-2}$$

b = 8 mm

h_1 = 40mm

h_2 = 8.5mm

$I = \frac{1}{12}\,8(40^3-8.5^3) = 42257.25\ \text{mm}^4$

$\sigma = \frac{37.23*0.02}{42257.25}*10^{12} = 17.6$ MPa.

this stress is the nominal stress. as a result of the hole in the bar there is a stress concentration so the true stress is the result of substitution in the following equation.

$$\sigma=K_f*\sigma_o \tag{5-3}$$

$$K_f = 1+q(K_t-1) \tag{5-4}$$

where K_f is the stress concentration factor.

K_t is a reduced value for K_f

q = is notch sensitivity

from figure (5.5)

$q = 0.75$

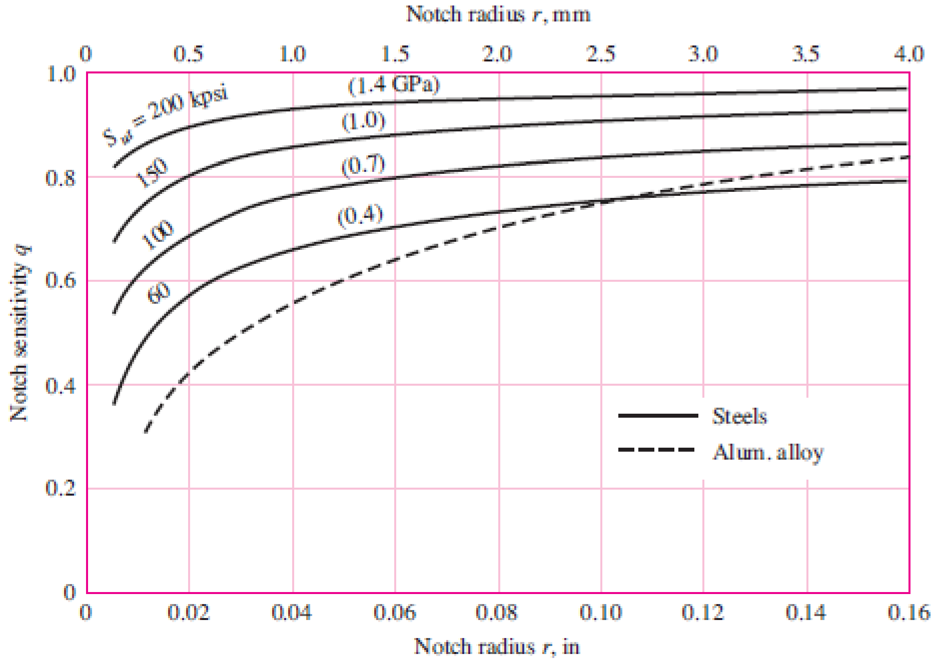


Figure 5.5 Notch Sensitivity *q*

from figure 5.6

$d/w = 8.5/40 = 0.2125$

$K_t = 1$

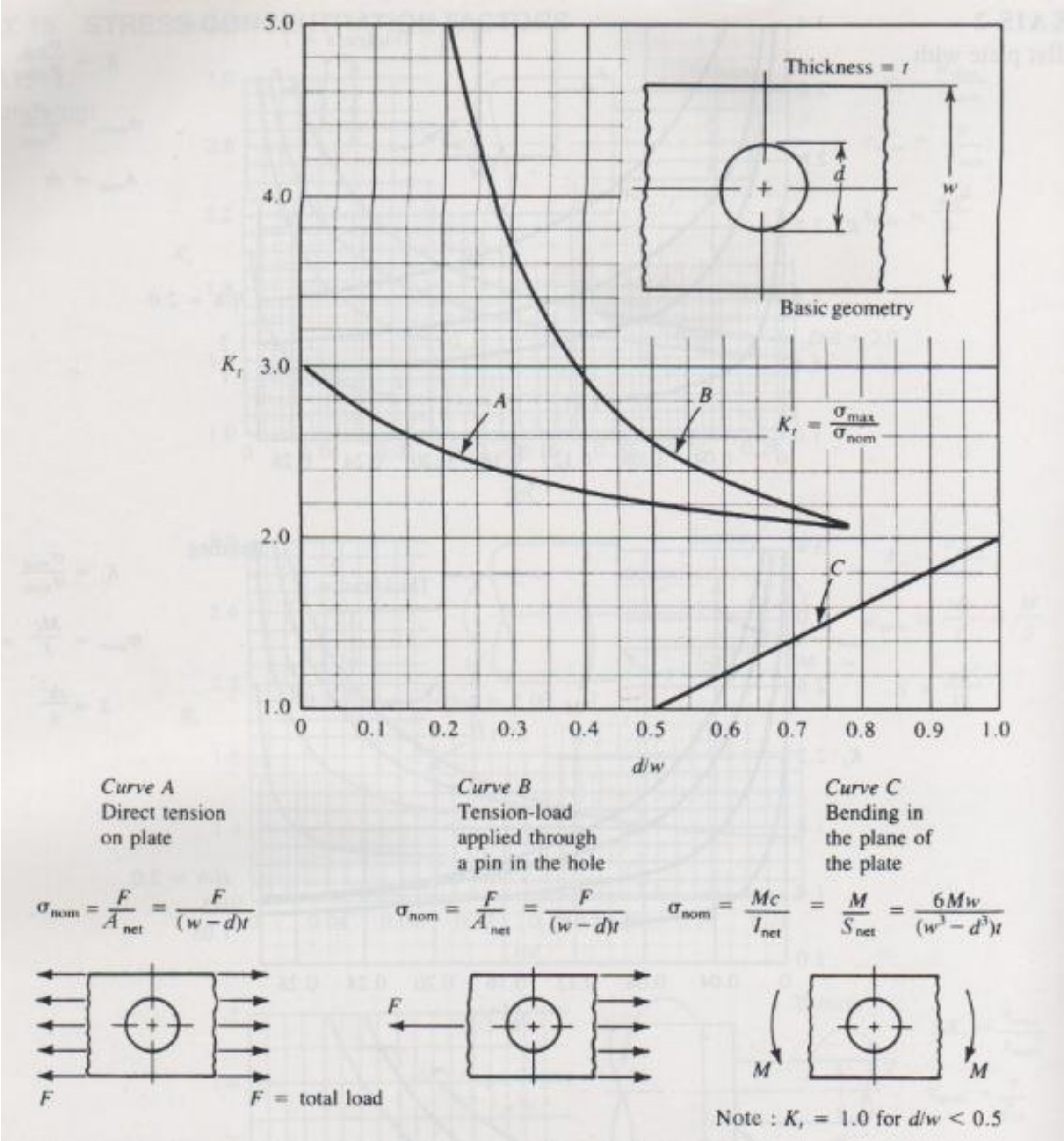


Figure 5.6 K_t For a Rectangular Bar With Central Hole[8]

K_f = 1+[0.75* (1-1)] = 1

σ = 1*17.6 = 17.6 MPa.

by modified good-man criteria for fatigue failure due to completely reversed stresses

$$\frac{1}{n} = \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} \tag{5-5}$$

$$\sigma_a = \frac{|\sigma_{max} - \sigma_{min}|}{2} \tag{5-6}$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \qquad (5\text{-}7)$$

$$Se = K_a\, K_b\, K_c\, K_d\, K_e\, K_f\, Se' \qquad (5\text{-}8)$$

where S_e is the modified endurance limit.

K_a is the surface condition modification factor.

K_b is the size modification factor.

K_c is the load modification factor.

K_d is the temperature modification factor.

K_e is the reliability factor .

K_f miscellaneous-effect modification factor.

$S_e{}'$ is rotary-beam test specimen endurance limit.

as the moment subjected to the bar is completely reversed then the stress is completely reversed and in the case of completely reversed stress the maximum stress is the +ve value of the calculated stress, and the minimum value is the -ve value too.

$$\sigma_m = \frac{17.6-17.6}{2} = 0 \qquad (5\text{-}9)$$

$$\sigma_a = \frac{17.6+17.6}{2} = 17.6 \text{ MPa}. \qquad (5\text{-}10)$$

K_a surface factor:

$$K_a = aS_{ut}{}^{b} \qquad (5\text{-}11)$$

for the used steel (plane carbon steel) and fro table (A-2)

a = 57.7 b = -0.718

$K_a = 57.7 * 320^{-0.718} = 0.91$

K_b size factor :

$$K_b = \begin{cases} \left(\frac{de}{7.62}\right)^{-0.107} & 2.79 \le d_e \le 51\ mm \\ 1.51 de^{-0.157} & 51 < d_e \le 254\ mm \end{cases} \qquad (5\text{-}12)$$

$$d_e = 0.808(hb)^{0.5} \qquad (5\text{-}13)$$

h = 40 mm b = 8 mm

$\boldsymbol{d_e}$ = 0.808*(40*8)$^{0.5}$ = 14.45 mm

$\boldsymbol{K_b}$ = 0.93

$\boldsymbol{K_c}$ loading factor:

$\boldsymbol{K_c}$ = 1 for bending. (5-14)

$\boldsymbol{K_d}$ temperature factor.

$\boldsymbol{K_d}$ = 1 ,from table (A-3) at 20^o on average.

$\boldsymbol{K_e}$ reliability factor

from table (A-4) $\boldsymbol{K_e}$ = 0.814 for 99% needed reliability.

$\boldsymbol{K_f}$ miscellaneous-effect modification factor.

as the stress concentration factor was taken into consideration ,it is reasonable to take a value of $\boldsymbol{K_f}$ = 1

S_e' is rotary-beam test specimen endurance limit.

$\boldsymbol{S_e' = 0.5S_{ut}}$; $\boldsymbol{S_{ut}} \leq$ 1400MPa (5-15)

$\boldsymbol{S_e'}$ = 0.5*320 = 160 MPa.

$\boldsymbol{S_e = K_a\, K_b\, K_c\, K_d\, K_e\, K_f\, S_e'}$

$\boldsymbol{S_e}$ = 0.91*0.93*1*1*0.814*1*160 = 110.222 MPa.

by modified good-man criteria

$$\frac{\mathbf{1}}{\boldsymbol{n}} = \frac{\boldsymbol{\sigma_a}}{\boldsymbol{S_e}} + \frac{\boldsymbol{\sigma_m}}{\boldsymbol{S_{ut}}} \qquad (5\text{-}16)$$

where n is the safety factor.

$\boldsymbol{\sigma_a}$ = 17.6 MPa.

$\boldsymbol{\sigma_m}$ = 0

$$\boldsymbol{n} = \frac{\boldsymbol{S_e}}{\boldsymbol{\sigma_a}} = \frac{110.222}{17.6} = 6.25$$

it is an acceptable value, it may be reselect another bar with less dimensions but the available dimensions in the market is small to carry such loads. so there is no need for redesign the bar.

5.2.2 Welded part in the rocker

The figure below shows the welded part from front side of the rocker. at both ends of the rocker there is tow similar welded parts, the force on each one is half of the force on each piston 73 N.

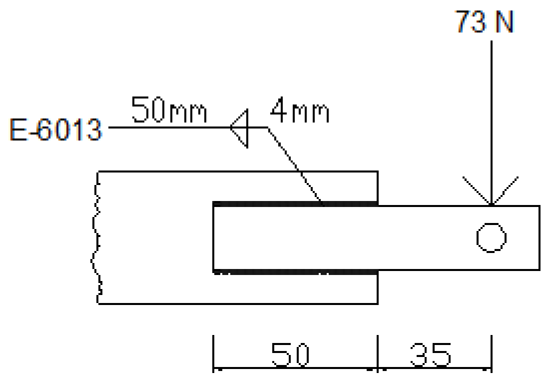


Figure 5.7 Welded Part in the Rocker

An E6013 electrode was used , from table (A-5) S_y = 345 MPa ,S_{ut} = 427 MPa by using modified good-man criteria for sheer stress

$$n = \frac{S_e S_{ut}}{\tau_a\, S_{ut} + \tau_m\, S_e} \tag{5-17}$$

where: S_e is the modified endurance limit.

τ_a is the shear stress amplitude.

τ_m is the mean shear stress.

$S_e = K_a\, K_b\, K_c\, K_d\, K_e\, K_f\, S_e'$

$S_e' = 0.5 S_{ut}$; $S_{ut} \leq 1400$MPa

S_e' = 0.5*427 = 213.5 MPa.

K_a surface factor:

$K_a = a S_{ut}^{\ b}$

for the used electrode and from table (A-2)

a = 272 b = -0.995

$K_a = 272*427^{-0.995} = 0.6565$

K_b size factor :

$$K_b = \begin{cases} \left(\frac{d}{7.62}\right)^{-0.107} & 2.79 \le d_e \le 51\ mm \\ 1.51d^{-0.157} & 51 < d_e \le 254\ mm \end{cases}$$

But as the sheer is almost uniform on the weld area; K_b can be 0.8

K_c loading factor:

K_c = 0.59 for torsion.

K_d temperature factor.

K_d = 1 ,from table (A-3) at 20^o on average.

K_e reliability factor :

for 99% reliability needed for the bar K_e = 0.814 from table (A-4)

K_f miscellaneous-effect modification factor.

as the stress concentration factor was taken into consideration ,it is reasonable to take K_f =1

Se = 0.6565*0.8*0.59*1*0.814*1*213.5 = 53.85 MPa.

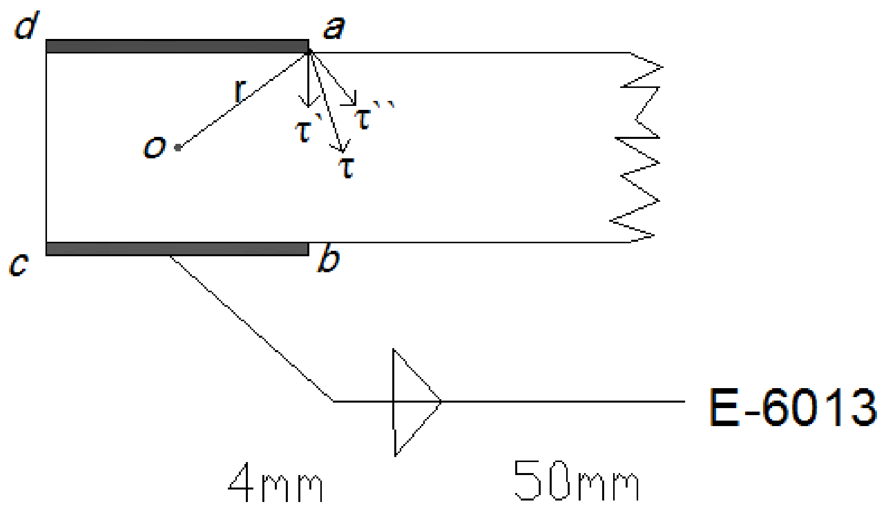


Figure 5.8 Shear Stress on Weld

$$\tau` = \frac{V}{A} \quad (5\text{-}18)$$

where $\tau`$ is the primary sheer stress from the direct sheer force on the welding.

V is the direct sheer force on the welding area.

A is the throat area of the welds.

from table (A-6) $\boldsymbol{A}$ = 1.41$\boldsymbol{hd}$

where: $\boldsymbol{h}$ is leg size of the welding.

$\boldsymbol{d}$ is the length of the weld line.

$\boldsymbol{h}$ = 4 mm

$\boldsymbol{d}$ = 50 mm

$\boldsymbol{A}$ = 1.41*4*50 = 282 mm^2

V = 438 N.

$\boldsymbol{A}$ = 282 mm^2

$\boldsymbol{\tau}^{`} = \frac{219}{282}$ = 0.777 MPa.

$$\boldsymbol{\tau}^{``} = \frac{\boldsymbol{M*r}}{\hat{J}} \qquad (5\text{-}19)$$

where: $\boldsymbol{\tau}^{``}$ is the secondary sheer stress of the welds.

$\boldsymbol{M}$ is the moment at the centroid of the weld result from the force applied to the member.

$\boldsymbol{r}$ is the distance from the centroid of the welds to the point of interest.

$\hat{\boldsymbol{J}}$ is the second polar moment of area of the welds about the centroid of the welds.

$$\boldsymbol{M = F*L} \qquad (5\text{-}20)$$

where: $\boldsymbol{L}$ is the distance from the point impact force to the centroid of welds.

$\boldsymbol{L}$ = 0.06 m

$\boldsymbol{F}$ = 146/2 = 73N

$\boldsymbol{M}$ = 73*0.06 = 4.38N.m

$\boldsymbol{r} = \sqrt{\boldsymbol{x^2 + y^2}} = \sqrt{2.5^2 + 1^2}$ = 26.92mm

$$\hat{\boldsymbol{J}} = \mathbf{0.707}*\ \hat{\boldsymbol{J}}_{\boldsymbol{u}} \qquad (5\text{-}21)$$

$\hat{\boldsymbol{J}}_{\boldsymbol{u}}$ is the unit second polar moment of area of the welds.

$\hat{\boldsymbol{J}}_{\boldsymbol{u}} = \frac{\boldsymbol{d(3b^2+d^2)}}{\boldsymbol{6}}$ from table (A-6)

$\hat{\boldsymbol{J}}_{\boldsymbol{u}}$ = 30833.3 mm^4

$\hat{J}$ = 0.707*30833.3 = 21799.1mm^4

$\tau'' = \frac{4.38*26.92}{21799.16} * 10^9 = 5.41$MPa

$\tau''_{vertical} = \tau'' \cos 21.8 = 5.41(\cos 21.8) = 5.023$ MPa

$\tau''_{horizontal} = \tau'' \sin 21.8 = 5.41(\sin 21.8) = 2.01$ MPa

$\tau_{eq\ vertical} = \tau' + \tau''_{v} = 0.777 + 5.023 = 5.8$ MPa

$\tau_{eq\ horizontal} = \tau''_{horizontal} = 2.01$ MPa

$$\tau = \sqrt{(\tau_{eq\ horizontal})^2 + (\tau_{eq\ vertical})^2}$$

$\tau = 6.14$ MPa

$$\tau_a = K_{fs} * \tau \tag{5-22}$$

$K_{fs} = 2.7$ from table (A-7)

$\tau_a = 2.7(6.14) = 16.578$ MPa

$n_f = S_e / \tau_a = 53.85 / 16.578 = 3.2$ acceptable value.

5.2.3 Bearing stress

5.2.3.1 Bearing stress in the member

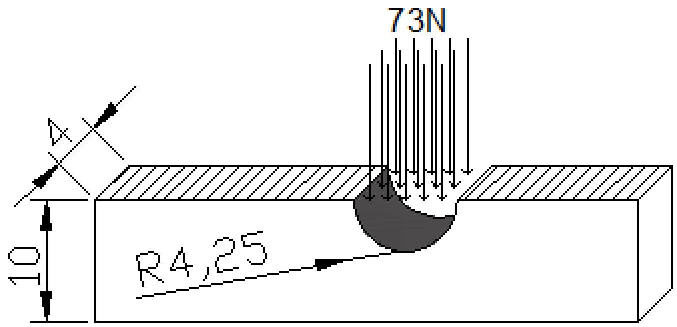


Figure 5.9 Bearing Stress in Member

$$\sigma = \frac{F}{A} \tag{5-23}$$

A = the projection area = $t \times d$ = 4(8.5) = 34 mm^2

$\boldsymbol{\sigma}$ = 73/34 = 2.14 MPa

As $\boldsymbol{\sigma}$ is very small with respect to the yield strength so it is possible to ignore the effect of fatigue failure.

5.2.3.2 Bearing stress in the pin

The pin is made of AISI 1010 steel

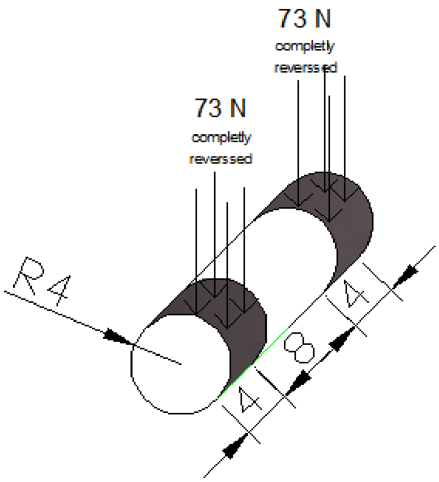


Figure 5.10 The Pin

$\boldsymbol{A}$ = 4(8) = 32 mm^2

$\boldsymbol{\sigma}$ = 73/32 = 2.28 MPa

as $\boldsymbol{\sigma}$ is very small compared with $\boldsymbol{S_y}$;Then the effect of the fatigue can be neglected.

5.2.4 The Coupler

Figure 5.11 The Coupler

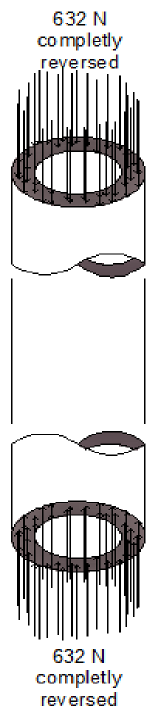


Figure 5.12 The Loads on the Coupler

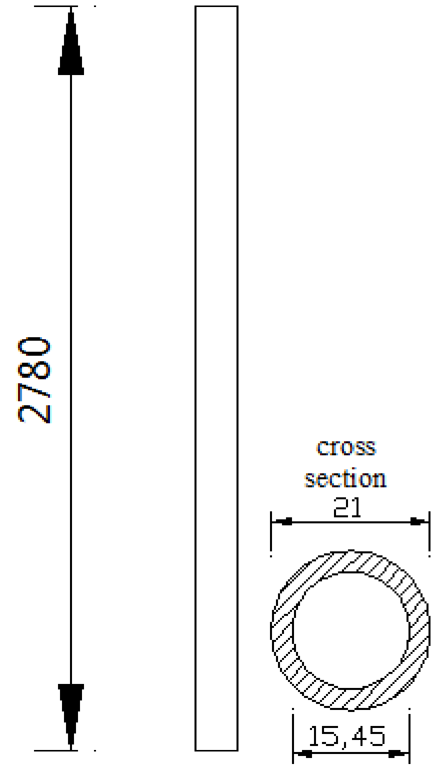


Figure 5.13 Coupler Dimensions (mm)

It is the part in the prototype that connect the rocker with the crank, made of AISI 1010 steel. as the outer diameter of the coupler is very small compared to its length, so the buckling in the coupler must taken into consideration.

S_y = 180 MPa

S_{ut} = 320 MPa

The load at which the buckling starts "P_{cr}"

$$P_{cr} = \frac{C\,\pi^2\,E\,I}{L^2} \tag{5-24}$$

where C is the end condition constant.

E is the modulus of elasticity.

I is the moment of area of the cross section.

L is the length of the column.

C = 1 from table (A-8)

$$I = \frac{\pi}{64}\,(d_o^{\,4} - d_i^{\,4}) \tag{5-25}$$

$$I = \frac{\pi}{64}\,((21)^4 - (15.45)^4) = 6749.6$$

E = 207 GPa from table (A-9)

L = 2.78 m

P_{cr} = 1784.2 N

$$n = P_{cr}/F = 1784.2/632 = 2.82 \tag{5-26}$$

where n is the safety factor against buckling

For fatigue failure, for the column the force applied to it is completely reversed.

Using Goodman Criteria :

$$\frac{1}{n} = \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}}$$

But $S_m = 0$

$n = S_e/S_a$

$S_e = K_a\,K_b\,K_c\,K_d\,K_e\,K_f\,S_e'$

S_e' = (0.5) S_{ut} when ($S_{ut} \leq 1400$ MPa)

S_e' = (0.5)(320) = 160 MPa

$\boldsymbol{K_a = a\, S_{ut}^{\,b}}$

From table (A-2) we have ($\boldsymbol{a}$ = 275) , ($\boldsymbol{b}$ = -0.995)

$\boldsymbol{K_a}$ = $275(320)^{-0.995}$ = 1.03

$\boldsymbol{K_b}$ = 1 for axial loading there no size effect

$\boldsymbol{K_c}$ = 0.85 for axial loading

$\boldsymbol{K_d}$ = 1 from table (A-3)

$\boldsymbol{K_e}$ =0.814 for 99% needed reliability from table (A-4)

$\boldsymbol{K_f}$ = 1 as there no stress concentration effects

$\boldsymbol{S_e}$ = 140.08 MPa

$\boldsymbol{\sigma_a = F/A} = 438.6/\pi\boldsymbol{(r_o^2 - r_i^2)} = 632/158.884 = 3.97$ MPa

as $\boldsymbol{\sigma_a}$ is very small compared with $\boldsymbol{S_e}$;So it is very safe against fatigue failure. and no need to compute the factor of safety

5.2.5 The Welds in the coupler

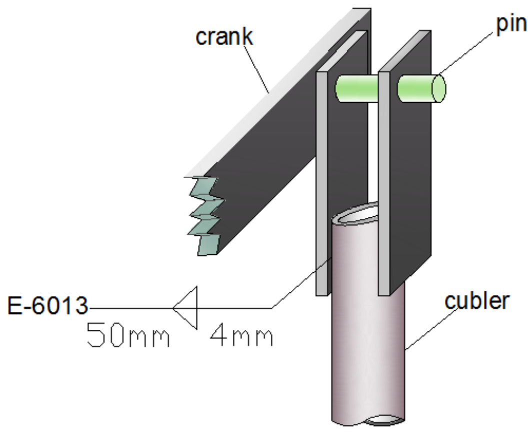


Figure 5.14 Weld in the coupler

$$\boldsymbol{\tau' = V/A} \qquad (5\text{-}27)$$

$\boldsymbol{A}$ = the throat area

$\boldsymbol{A}$ = **2(1.41)** $\boldsymbol{hd}$ from table (5-6)

$\boldsymbol{A}$ = 2(1.41)(4)(5) = 56.4 mm^2

Using Goodman criteria :

$$\boldsymbol{n = \frac{S_e}{\acute{\tau}_a}} \qquad \text{(as the load is completely reversed)}$$

$\boldsymbol{S_e = K_a\, K_b\, K_c\, K_d\, K_e\, K_f\, S_e'}$

$S'_e = \mathbf{0.5} * S_{ut}$ when ($S_{ut} \leq 1400$ MPa)

$S'_e = (0.5)(427) = 213.5$ MPa

$K_a = a\, S_{ut}^{\,b}$ From table (A-2) we have (a = 272) , (b = -0.995)

$K_a = 272(427)^{-0.995} = 0.65$

$K_b = 1$ for uniform shear stress at the throat

$K_c = 0.59$ (5-28)

$K_d = 1$ from table (A-3)

$K_e = 0.814$ for 99% needed reliability

$K_f = 1$ as stress concentration is to be taken into consideration later.

$S_e = 66.6$ MPa

$\tau'_a = K_{fs} \times \tau'$

$K_{fs} = 2.7$ from table (A-7) for the end of parallel fillet weld

$\tau'_a = 2.7(316/56.4) = 15.13$ MPa

$n = S_e / \tau'_a = 66.6/15.13 = 4.4$

the factor of safety is acceptable.

5.2.6 The Pin of the crank

The pin is shown in figure (5-14), The critical section is the bar of the pin where the pin connected to the crank. the load on the pin is 632 N completely reversed.

$M_a = F * L$ (5-29)

where L is the distance from axis of the coupler to the base of the pin.

$M_a = 632\ (0.026) = 16.432$ N.m

$$\sigma_{max} = \frac{MC}{I}$$

$C = d/2 = 6$ mm (d = 12 mm)

$I = \pi d^4/64 = \pi(12)^4/64 = 1017.87\ mm^4$

$\sigma = 16.432(0.006)(10)^{12} / (1017.87) = 96.7$ MPa

$\tau = V/A = 632/\pi(6)^2 = 5.58$ MPa

σ_{max} is completely reversed

for fatigue failure use the Goodman criteria :

$$\frac{1}{n} = \frac{\sigma' a}{Se} + \frac{\sigma' m}{Sut}$$

$S_e = K_a\, K_b\, K_c\, K_d\, K_e\, K_f\, S'_e$

$S'_e = 0.5 * S_{ut}$ when ($S_{ut} \leq 1400$ MPa)

$S'_e = (0.5)(320) = 160$ MPa

$K_a = a\, S_{ut}^{\,b}$

From table (6-2) for machined or cold drawn having (a = 4.51) , (b = 0.265)

$K_a = 4.51(320)^{0.265} = 0.978$

K_b :

$d_e = 0.37d$ (5-30)

$d_e = 0.37(12) = 4.44$ mm

for $2.79 \leq d_e \leq 51$ mm

$$K_b = \left(\frac{d}{7.62}\right)^{-0.107} = \left(\frac{4.44}{7.62}\right)^{-0.107} = 1.05$$

$K_c = 1$ for bending

$K_d = 1$ from table (5-3)

$K_e = 0.814$ for 99% needed reliability, from table (5-4).

$K_f = 1$ as there no stress concentration effects.

$S_e = 134.9$ MPa

the van misses stress σ':

$$\sigma' = \frac{1}{\sqrt{2}}[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^{\;2} + \tau_{yz}^{\;2} + \tau_{zx}^{\;2})]^{1/2} \qquad (5\text{-}31)$$

$\sigma' = 69.05$

then $\sigma'_a = 69.05$ MPa

$n_f = S_e / \sigma'_a = 134.9/69.05 = 1.95$

acceptable against fatigue failure.

5.2.7 The Crank

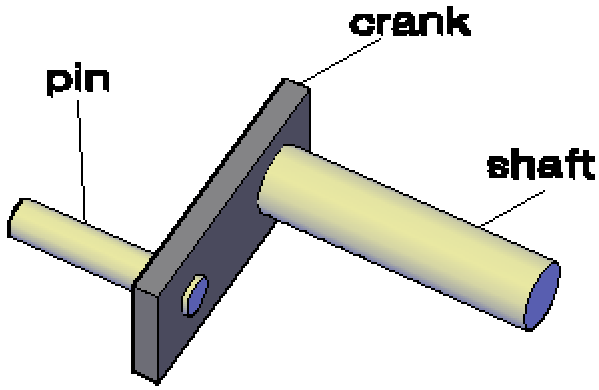


Figure 5.15 The Crank

The crank is a rectangular cross section bar (40*8mm), made of AISI 1010 steel, exactly the same as the rocker, and because of no stress concentration on the crank, it normally to get a higher safety factor in the crank, so there is no need for stress analysis.

And because the safety factor is high in this case it can be replaced with another crank with less cross sectional area. But the available size of this bar is very smaller than $(40\times8\text{mm})$ so the one in use is acceptable.

5.2.8 Welding between the crank and the shaft.

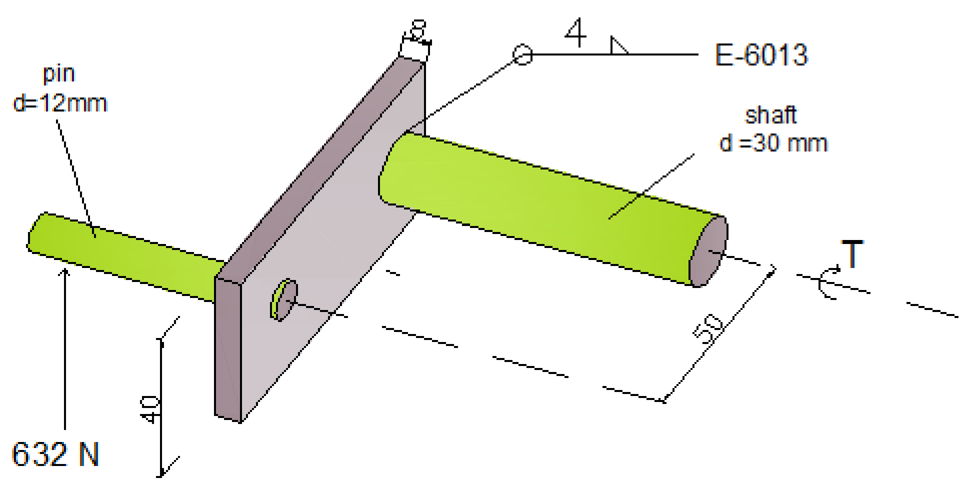


Figure 5.16 The Shaft and the Crank Assembly

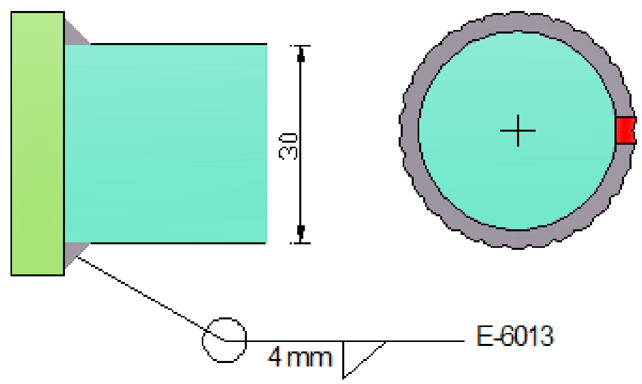


Figure 5.17 Welding Between Shaft and Crank

As shown in the previous figure (5-17) the red region is the most critical region, because the resultant shear stress at this area is the sum of the primary sheer stress from the direct sheer force on the welding(τ') and the secondary shear stress (τ''). the welding is suspended to a torsion load and shear stress. The torque is equal to 31.6 N.m.

r = 17 mm

$\tau' = V/A$

$A = 1.414\, h*r$ from table (A-6)

$A = 1.414(4)(17) = 96.152\ mm^2$

Then:

$\tau' = V/A = 632/96.152 = 6.57$ MPa

$$\tau'' = M\,r\,/\,J \qquad (5\text{-}32)$$

$J = 0.707(h)J_u$

From table (A-6) for circular shape:

$J_u = 2\,\pi\,r^3 = 2\,\pi\,(17)^3 = 30869.29\ mm^3$

Then:

$J = 0.707(4)(30896.29) = 87298.3\ mm^4$

$M = T$ that the shaft transmit

M = 31.6 N.m

$\tau'' = 31.6(10)^3(17)/87298.3 = 6.15$ MPa the resultant shear stress at the red region.

$\tau'_{resultant} = \tau' + \tau'' = 12.75$ MPa

$\tau_{a\ res} = 12.75$ MPa

$\tau_{m\ res} = 0$

$S_e = K_a\, K_b\, K_c\, K_d\, K_e\, K_f\, S'_e$

$S'_e = 0.5\, S_{ut}$ when ($S_{ut} \leq 1400$ MPa)

$S'_e = (0.5)(427) = 213.5$ MPa

$K_a = a\, S_{ut}^{\,b}$

From table (A-2) $a = 272$, $b = -0.995$

$K_a = 0.65$

K_b for the section in the welding :

$d_e = 0.37\, d = 0.37(30) = 11.1$ mm

for $2.79 \leq d_e \leq 51$ mm

$$K_b = \left(\frac{d}{7.62}\right)^{-0.107} = \left(\frac{11.1}{7.62}\right)^{-0.107} = 0.95$$

$K_c = 0.59$

$K_d = 1$ from table (A-3)

$K_e = 0.814$ for 99% needed reliability from table (A-4).

$K_f = 1$ as the stress concentration is taken in consideration.

From table (A-7) for end of parallel fillet weld $K_{ts} = 2.7$

Now:

$S_e = 63.31$ MPa

Now using Goodman criteria for strain :

$$\frac{1}{n} = \frac{\tau_a}{S_e}$$

For τ_a having :

$\tau_{a\ \text{corrected}} = K_{ts} \times \tau_a = 2.7 * 12.75 = 34.425$ MPa

then :

$$\frac{1}{n} = \frac{34.425}{63.31}$$

n = 1.84 (acceptable)

5.2.9 The Shaft

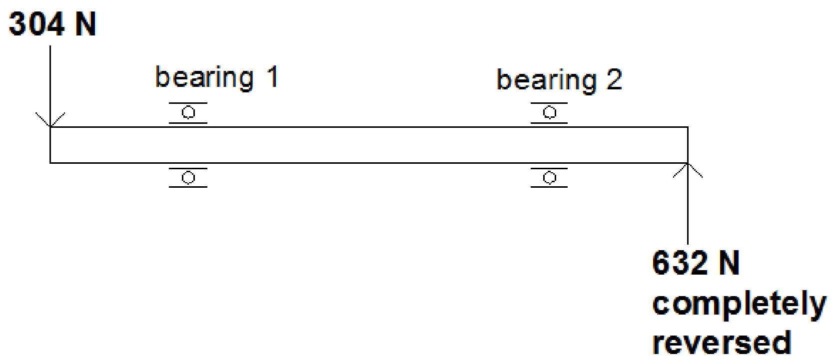


Figure 5.18 The Shaft

The shaft is a solid circular bar made of AISI 1010 low carbon steel. the 632 N is the force from the coupler and the 304 N is the weight of the fan As the 630 N is completely reversed and the 304 N is constant, then the reactions forces and the shear and the moment diagram must shown in two situation. the fist one is the first half of the shaft cycle when the force 630 N upward. and the other one in the second half of the shaft cycle, where the force 630 N is downward.

Situation (1)

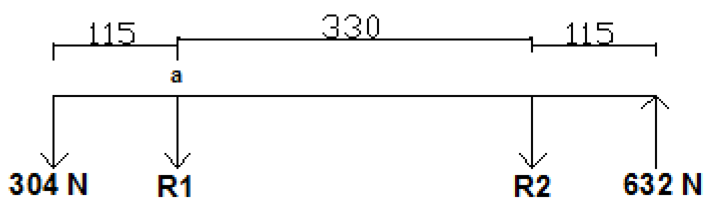


Figure 5.19 FBD Diagram Situation (1)

$\sum \boldsymbol{Ma} = 0$

$\boldsymbol{R_2}(330) - 632(445) - 304(115) = 0$

$\boldsymbol{R_2} = 958$ N

$+\uparrow \boldsymbol{Fy} = 0$

$\boldsymbol{R1} = 630$ N

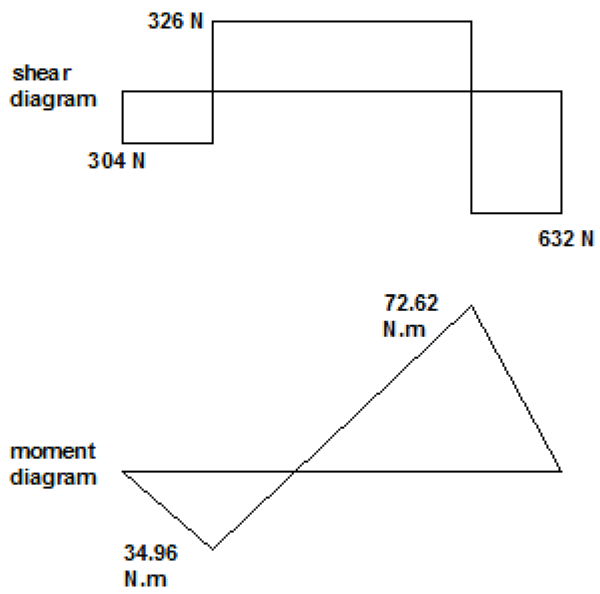


Figure 5.20 Shear and Moment Diagram for Situation (1)

Situation (2)

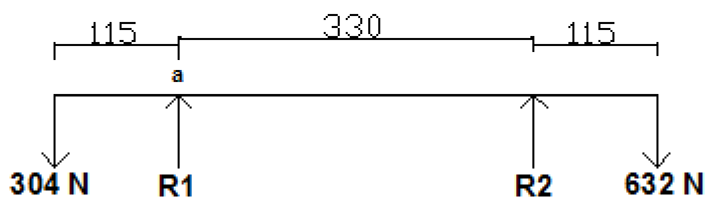


Figure 5.21 FBD Diagram Situation (2)

$\sum \boldsymbol{Ma} = 0$

$\boldsymbol{R_2}\,(330) - 632(445) + (304)(115) = 0$

$\boldsymbol{R_2} = 746$ N

$\boldsymbol{R_1} = 190$ N

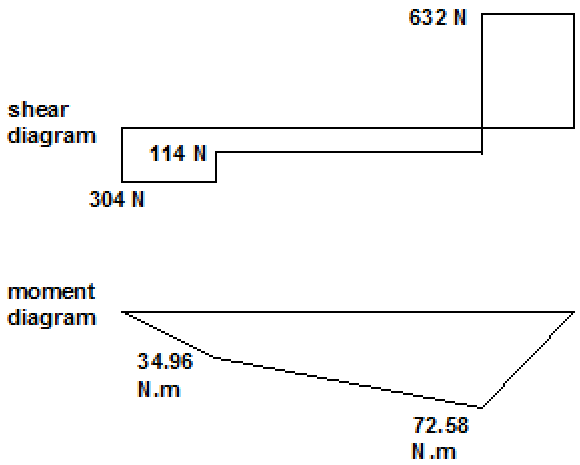


Figure 5.22 Shear and Moment Diagram for Situation (2)

Form the moment diagrams in the two situations the it is possible to regard that the moment at bearing 2 is completely reversed moment with an amplitude of 72.62 N.m, the moment at bearing 1 can be ignored since it less than that at bearing 2.

$Ma,_{b2}$ = 72.62 N.m

$Mm,_{b2}$ =0

The torque is = 31.6 N.m

$$T_a = \frac{|Tmax - Tmin|}{2} = 0$$

$$T_m = \frac{|Tmax + Tmin|}{2} = 31.6 \text{ N.m}$$

$$\sigma = \frac{MC}{I}$$

$$\tau = \frac{Tr}{J}$$

r = 15 mm

C = 15 mm

$I = (\pi/64)\ (d^4)$

$J = (\pi/32)\ (d^4)$

I = 39760.8 mm^4

J = 79521.6 mm^4

$$\sigma_a = \frac{M_a C}{I} = \frac{72.62\ (0.015)}{39760.8} = 27.4 \text{ MPa}$$

$\sigma_m = 0$

$\tau = \frac{Tr}{J} = \frac{31.6(0.015)}{79521.6} = 6\text{ MPa} = \tau_m$

$\tau_a = 0$

the Von Misses stress σ':

$$\sigma' = \frac{1}{\sqrt{2}}[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)]^{1/2} \quad (5\text{-}33)$$

$\sigma'_a = \frac{27.4}{\sqrt{2}} = 19.374\text{ MPa}$

$\sigma'_m = \frac{6}{\sqrt{2}} = 4.24\text{ MPa}$

using modified Goodman criteria:

$\frac{1}{n} = \frac{\sigma'_a}{S_e} + \frac{\sigma'_m}{S_{ut}}$

$S_{ut} = 320$ MPa

$S_y = 180$ MPa

$S_e = K_a K_b K_c K_d K_e K_f S'_e$

$S'_e = 0.5\, S_{ut}$ when ($S_{ut} \leq 1400$ MPa)

$S'_e = (0.5)(320) = 160$ MPa

$K_a = a\, S_{ut}^{\,b}$

From table (A-2) $a = 4.51$, $b = -0.265$

$K_a = 4.51(320)^{-0.265} = 0.97$

$$K_b = \begin{cases} \left(\frac{d}{7.62}\right)^{-0.107} & 2.79 \leq de \leq 51\ mm \\ 1.51d^{-0.157} & 51 < de \leq 254\ mm \end{cases}$$

$d_e = (0.37)d$

$d_e = (0.37)(30)^2 = 333$ mm

$K_b = 1.51(333)^{-0.157} = 0.606$

$K_c = 1$ for bending.

$K_d = 1$ from table (A-4)

$K_e = 0.814$ for 99% reliability

$K_f = 1$ as there no stress concentration effects

S_e = 76.5 MPa

$$\frac{1}{n} = \frac{19.374}{76.5} + \frac{4.24}{320} = 0.28$$

n = 3.57 acceptable safety factor

The shaft rotates in a low speed so the critical speed of the shaft is not important in this situation.

5.2.10 Bearing selection[7]

The procedures used for selecting bearing was based on the way used in Mechanical Design of Machine Elements and Machines Jack A. Collins 2nd Edition.

First perform a complete force analysis, and employ appropriate shaft design equations to determine a tentative strength-based shaft diameter at the bearing site.

From the force analysis, calculate the radial load F_r and axial thrust load F_a to be supported by the proposed bearing. these loads are the reaction forces on the shaft at the bearing positions.

Determine the design life requirement, for the bearing.(the bearing design is for infinite life)

Access the severity of any shock or impact associated with the application so that an impact factor, I_f may be determined from table A-11

Tentatively select the type of bearing to be used from table A-12

Calculate the dynamic equivalent radial load P_e from the following empirical relationship:

$$P_e = X_d F_r + Y_d F_a \qquad (5\text{-}34)$$

Where : X_d : dynamic radial load factor, based on bearing geometry from table A-13

Y_d : dynamic axial load factor, based on bearing geometry from table A-13

The combinations X_{d1}, Y_{d1}, and X_{d2}, Y_{d2} should both be calculated, and whichever combination gives the larger value for P_e should be used.

Execution:

From the force analysis of the shaft the maximum radial load is equal to

F_r = 958N

F_a = 8*F_t = 8*10.05 = 80.4 N.

The design life requirement is infinite life.

Using table A-11, since a light impact is specified as (1.2-1.5) , so midrange impact factor may be calculated as I_f = (1.2+1.5)/2= 1.35

From table A-12 it is appropriate to use a single raw deep groove ball bearing .

Dynamic equivalent radial load may be calculated from the following the equation.

$$P_e = X_d F_r + Y_d F_a$$

from table A-13 for a single row radial bearing

$X_{d1} = 1$ $Y_{d1} = 0$

$X_{d2} = 0.55$ $Y_{d2} = 1.45$

So:

P_{e1} = 1 (958) + 0 (80.4) = 958 N

P_{e2} = 0.55 (958) + 1.45 (80.4) = 643.5 N

Since $P_{e1} > P_{e2}$ so $P_e = P_{e1}$ = 958 N

the bearing should be selected from table A-14 (single row deep groove ball bearing) taking in consideration at that approximate fatigue load must exceeds or equal to the modified dynamic equivalent radial load ($P_f \geq P_e$ x I_f)

$P_e I_f$ = 958x 1.35 = 1293.3 N = 1.2933KN

So the number of appropriate bearing can be used is 6309 at which P_f = 1.34 KN larger than 1.293 and the bore diameter is 45 mm.

as the bore diameter is larger than the shaft diameter, then the shaft diameter must be increased completely or partially at the bearing position.

5.2.11 Welding between the shaft and the blades

The connection between the shaft and the bars is exactly the same between the shaft and the crank. But in this case the shear that is distributed uniformly is less than that at the welding between the shaft and the crank, because the secondary shear stress equal to zero. So it is safe having a safety factor larger than 1.26

5.2.12 The bar that holds the blade

there are four bars at the end of the shaft, Arranged on top of each other so that the deviation from the axis of the shaft is an angle of 45°. The bars that support the plades are rods (40*8mm) made of AISI 1010 low carbon steel.

S_y = 180 MPa

S_{ut} = 320 MPa

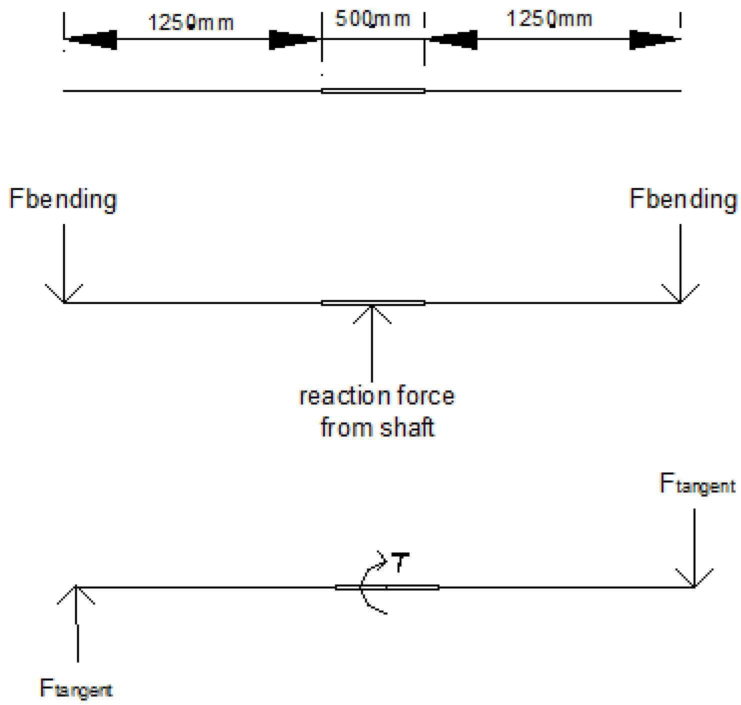


Figure 5.23 The Bar Which Holds the Plade (Simplified Drawing)

from the previous figure, it shown that the bending moment is maximum at the middle of the bar, caused by the bending forces. as shown below in figure 5.24

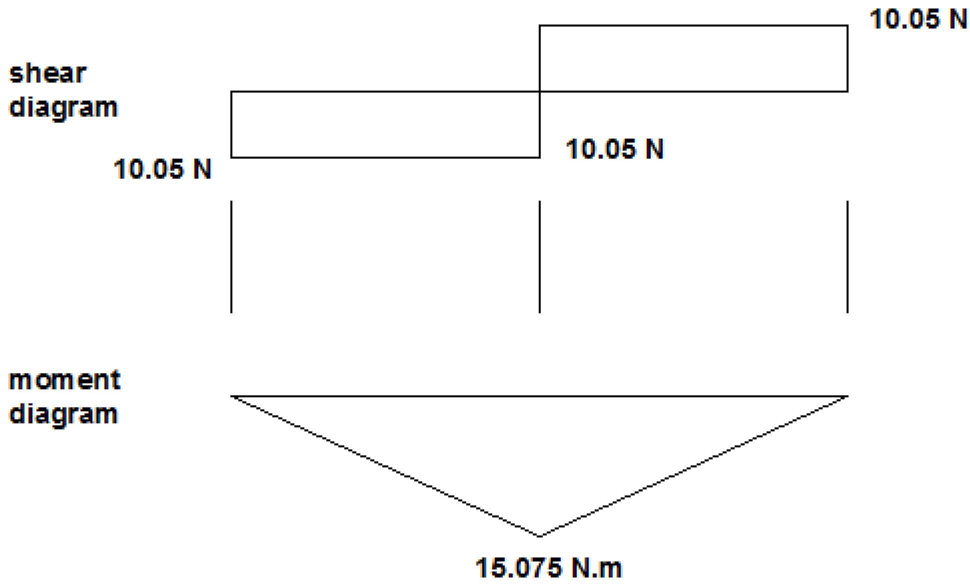


Figure 5.24 Shear and Moment Diagrams

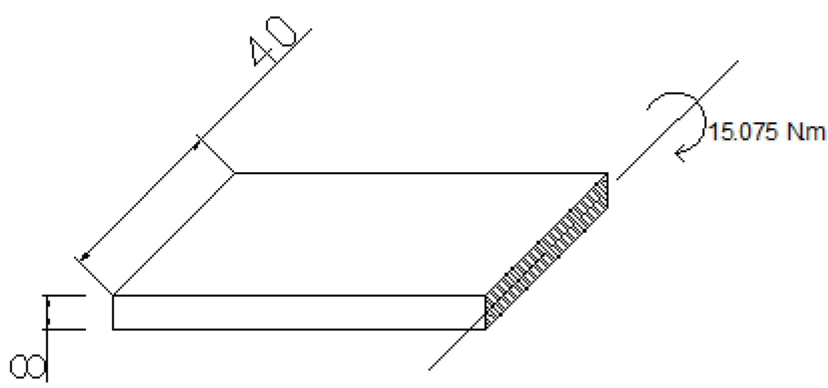


Figure 5.25 Middle Section in the Bar

$\sigma_{max} = \dfrac{\boldsymbol{MC}}{\boldsymbol{I}}$

$\boldsymbol{I} = \frac{1}{12}\ \boldsymbol{bh^3}$

$\boldsymbol{h}$ = 8 mm.

$\boldsymbol{b}$ = 40 mm.

$\boldsymbol{I}$ = 1706.7 mm^4

$\boldsymbol{C}$ = 4 mm

$$\sigma_{max} = \frac{15.075 * 0.004}{1706.7} = 35.33 \text{ MPa}$$

$$n_y = \frac{S_y}{\sigma_{max}} = 180 / 35.33 = 5.09 \text{ acceptable value.}$$

where n_y is the safety factor against yielding. also the tangent force causes a bending moment on the bar, but the resultant stress is less than that caused by the bending force, as the moment of area I is larger in this case.

5.2.13 Rod`s bolts for blades

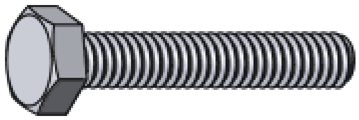


Figure 5.25 M16x1 Bolt

Each rods is fixed by two bolts. the used bolts are M6×1fine series of medium carbon, from table A-10 the tensile stress area A_t = 20.1 mm^2 , from table A-11 the minimum proof strength F_b = 225 MPa. from the static analysis as shown below in figures 5.26 and 5.27 the loads on the first bolt is 101.5 N tension and 145.7 N sear load. while the load on the second bolt is only 155.75 N shear load.

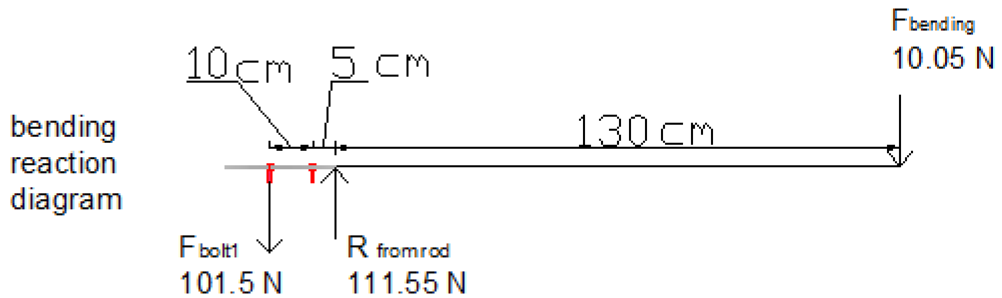


Figure 5.26 Bending Force Reaction

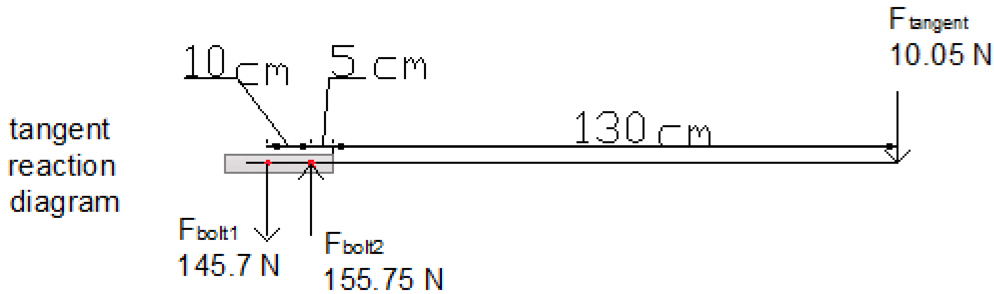


Figure 5.27 Tangent Force Reaction

Bolt 1:

$$\boldsymbol{\sigma} = \frac{\boldsymbol{F}}{\boldsymbol{A_t}}$$

$$\boldsymbol{\sigma} = \frac{101.5}{20.1} = 5.05 \text{ MPa.}$$

$$\boldsymbol{\tau} = \frac{\boldsymbol{F}}{\boldsymbol{A_t}}$$

$$\boldsymbol{\tau} = \frac{145.7}{20.1} = 7.25 \text{ MPa.}$$

$\boldsymbol{\sigma'}$ = 13.05 MPa.

Bolt 2:

$$\boldsymbol{\sigma} = \frac{\boldsymbol{F}}{\boldsymbol{A_t}}$$

$$\boldsymbol{\sigma} = 0$$

$$\boldsymbol{\tau} = \frac{\boldsymbol{F}}{\boldsymbol{A_t}}$$

$$\boldsymbol{\tau} = \frac{155.75}{20.1} = 7.74 \text{ MPa.}$$

so the factor of safety to be calculated for the first bolt because the Von misses stress is larger than that in the second bolt.

$$\boldsymbol{n} = \frac{\boldsymbol{F_b}}{\boldsymbol{\sigma}`}$$

$$\boldsymbol{n} = \frac{225}{13.05} = 17.2$$;it is very safe against failure.

5.2.14 the Supporting Rod for Blades

the used rod is 20x20 mm of 1.5mm thickness made of AISI 1010, from static analysis in figures 5.26, 5.27 it is clear that the tangent force and the bending force affect the rod by a bending moment, but maximum moment from both forces is in two different positions. the maximum bending moment from the tangent force is at 1.35 m far from the middle of the blat; in the section where the second bolt is placed.

$M_{tangent} = FxD$

$M_{tangent} = 10.05*1.35 = 13.567$ N.m

the resultant stress

$$\sigma = \frac{MC}{I}$$

$C = .01$ m

$I = 5059.25\ \text{mm}^4$

$$\sigma = \frac{13.567*.01}{5059.25} = 26.81\ \text{MPa.}$$

$$n = \frac{s_y}{\sigma} = \frac{180}{26.81} = 6.7$$

at this section there is no effect for the bending force because of the used bolts, the maximum bending moment from the bending force is at 1.3 m far from the middle of the plat; at the end of the bar that supports the blades. at this section the two forces affects by two equals bending moments.

$M = FxD$

$M = 10.05*1.3 = 13.065$ N.m

the resultant stresses from both moments

$$\sigma = 2*\frac{MC}{I}$$

$I = 5059.25\ \text{mm}^4$

$$\sigma = 2*\frac{13.065*0.01}{5059.25} = 51.64\ \text{MPa.}$$

$\boldsymbol{n} = \frac{s_y}{\sigma} = \frac{180}{28.82} = 3.49$ Somewhat high against yielding.

5.2.15 The Bearing supporting part

it is a rectangular solid bar 40x140 mm, Since the piece is installed tightly and large-sized compared to other parts. And the forces that affect it is small compared with the geometric dimensions. it possible to ignore stress analysis.

5.2.16 The Column

The column is made of AISI 1010, the force that affect it is the bending force of all blades. as shown in figure 5.28 the force affect the column by bending moment, it`s maximum value at the base of the column.

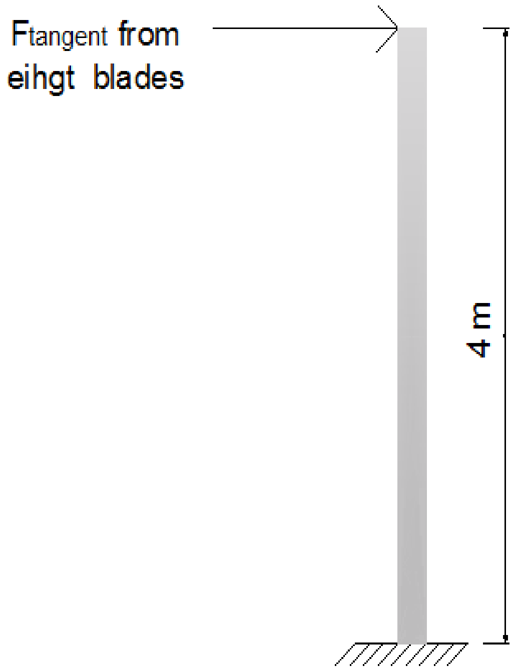


Figure 5.28 The Column

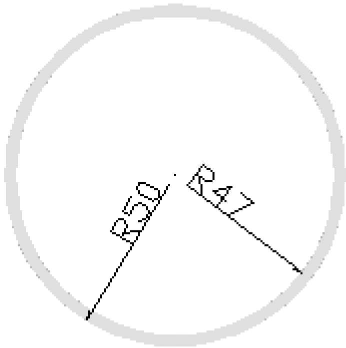


Figure 5.29 The Column cross section (mm)

$\boldsymbol{\sigma} = \frac{\boldsymbol{MC}}{\boldsymbol{I}}$

$\boldsymbol{C}$ = .05 m

$\boldsymbol{I}$ = 1076246 mm^4

$\boldsymbol{M} = \boldsymbol{8 * F_t * D}$

$\boldsymbol{M}$= 8*10.05*4 = 321.6 N.m

$\boldsymbol{\sigma} = \frac{321.6 * .05}{1076246} =$ 10.76 MPa.

$\boldsymbol{n} = \frac{\boldsymbol{s_y}}{\boldsymbol{\sigma}} = \frac{180}{10.76}$ = 16.7 acceptable value for safety against yielding.

5.2.17 The Plastic sheets

The used sheets are 70x60cm plastic sheets of 1cm thickness, and no need for calculation for this part.

CONCLUSION AND RECOMMENDATIONS

From the previous calculations the main safety factor of the project is 1.84 and this is an acceptable value, so the project could be implemented and published to reach a commercial level, specially that the project doesn`t coast so much, since the total coast was 250 $.

APPENDIX

Table A-1 Yield And Tensile Strength Of Some Steels [6]

1 UNS No.	2 SAE and/or AISI No.	3 Processing	4 Tensile Strength, MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation in 2 in, %	7 Reduction in Area, %	8 Brinell Hardness
G10060	1006	HR	300 (43)	170 (24)	30	55	86
		CD	330 (48)	280 (41)	20	45	95
G10100	1010	HR	320 (47)	180 (26)	28	50	95
		CD	370 (53)	300 (44)	20	40	105
G10150	1015	HR	340 (50)	190 (27.5)	28	50	101
		CD	390 (56)	320 (47)	18	40	111
G10180	1018	HR	400 (58)	220 (32)	25	50	116
		CD	440 (64)	370 (54)	15	40	126
G10200	1020	HR	380 (55)	210 (30)	25	50	111
		CD	470 (68)	390 (57)	15	40	131
G10300	1030	HR	470 (68)	260 (37.5)	20	42	137
		CD	520 (76)	440 (64)	12	35	149
G10350	1035	HR	500 (72)	270 (39.5)	18	40	143
		CD	550 (80)	460 (67)	12	35	163
G10400	1040	HR	520 (76)	290 (42)	18	40	149
		CD	590 (85)	490 (71)	12	35	170
G10450	1045	HR	570 (82)	310 (45)	16	40	163
		CD	630 (91)	530 (77)	12	35	179
G10500	1050	HR	620 (90)	340 (49.5)	15	35	179
		CD	690 (100)	580 (84)	10	30	197
G10600	1060	HR	680 (98)	370 (54)	12	30	201
G10800	1080	HR	770 (112)	420 (61.5)	10	25	229
G10950	1095	HR	830 (120)	460 (66)	10	25	248

Table A-2 Parameters for Marin Surface Modification [6]

Surface Finish	Factor *a* S_{ut}, kpsi	Factor *a* S_{ut}, MPa	Exponent *b*
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

Table A-3 Temperature Factor K_d [6]

Temperature, °C	S_T/S_{RT}	Temperature, °F	S_T/S_{RT}
20	1.000	70	1.000
50	1.010	100	1.008
100	1.020	200	1.020
150	1.025	300	1.024
200	1.020	400	1.018
250	1.000	500	0.995
300	0.975	600	0.963
350	0.943	700	0.927
400	0.900	800	0.872
450	0.843	900	0.797
500	0.768	1000	0.698
550	0.672	1100	0.567
600	0.549		

Table A-4 Reliability Factors ke [6]

Reliability, %	Transformation Variate z_a	Reliability Factor k_e
50	0	1.000
90	1.288	0.897
95	1.645	0.868
99	2.326	0.814
99.9	3.091	0.753
99.99	3.719	0.702
99.999	4.265	0.659
99.9999	4.753	0.620

Table A-5 Minimum Weld-Metal [6]

AWS Electrode Number*	Tensile Strength kpsi (MPa)	Yield Strength, kpsi (MPa)	Percent Elongation
E60xx	62 (427)	50 (345)	17–25
E70xx	70 (482)	57 (393)	22
E80xx	80 (551)	67 (462)	19
E90xx	90 (620)	77 (531)	14–17
E100xx	100 (689)	87 (600)	13–16
E120xx	120 (827)	107 (737)	14

Table A-6 Throat Area of Welds [6]

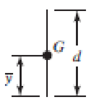
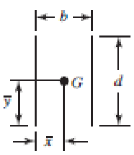
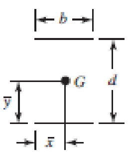
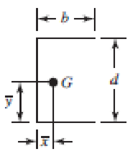
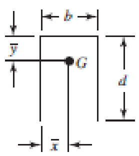
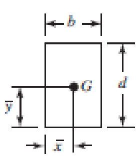
Weld	Throat Area	Location of *G*	Unit Second Moment of Area
	$A = 0.707hd$	$\bar{x} = 0$ $\bar{y} = d/2$	$I_u = \dfrac{d^3}{12}$
	$A = 1.414hd$	$\bar{x} = b/2$ $\bar{y} = d/2$	$I_u = \dfrac{d^3}{6}$
	$A = 1.414hd$	$\bar{x} = b/2$ $\bar{y} = d/2$	$I_u = \dfrac{bd^2}{2}$
	$A = 0.707h(2b + d)$	$\bar{x} = \dfrac{b^2}{2b+d}$ $\bar{y} = d/2$	$I_u = \dfrac{d^2}{12}(6b+d)$
	$A = 0.707h(b + 2d)$	$\bar{x} = b/2$ $\bar{y} = \dfrac{d^2}{b+2d}$	$I_u = \dfrac{2d^3}{3} - 2d^2\bar{y} + (b+2d)\bar{y}^2$
	$A = 1.414h(b + d)$	$\bar{x} = b/2$ $\bar{y} = d/2$	$I_u = \dfrac{d^2}{6}(3b+d)$

Table A-6 (continued)

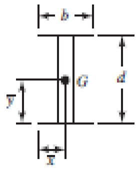
	$A = 1.414h(b + d)$	$\bar{x} = b/2$ $\bar{y} = d/2$	$I_u = \dfrac{d^2}{6}(3b + d)$
	$A = 1.414\pi hr$		$I_u = \pi r^3$

Table A-7 Fatigue Stress Concentration Factor K_{fs} [6]

Type of Weld	K_{fs}
Reinforced butt weld	1.2
Toe of transverse fillet weld	1.5
End of parallel fillet weld	2.7
T-butt joint with sharp corners	2.0

Table A-8 The End Condition Constant C [6]

	End-Condition Constant C		
Column End Conditions	Theoretical Value	Conservative Value	Recommended Value*
Fixed-free	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
Rounded-rounded	1	1	1
Fixed-rounded	2	1	1.2
Fixed-fixed	4	1	1.2

Table A-9 Physical Constants of Materials [6]

Material	Modulus of Elasticity E		Modulus of Rigidity G		Poisson's Ratio ν	Unit Weight w		
	Mpsi	GPa	Mpsi	GPa		lbf/in^3	lbf/ft^3	kN/m^3
Aluminum (all alloys)	10.4	71.7	3.9	26.9	0.333	0.098	169	26.6
Beryllium copper	18.0	124.0	7.0	48.3	0.285	0.297	513	80.6
Brass	15.4	106.0	5.82	40.1	0.324	0.309	534	83.8
Carbon steel	30.0	207.0	11.5	79.3	0.292	0.282	487	76.5
Cast iron (gray)	14.5	100.0	6.0	41.4	0.211	0.260	450	70.6
Copper	17.2	119.0	6.49	44.7	0.326	0.322	556	87.3
Douglas fir	1.6	11.0	0.6	4.1	0.33	0.016	28	4.3
Glass	6.7	46.2	2.7	18.6	0.245	0.094	162	25.4
Inconel	31.0	214.0	11.0	75.8	0.290	0.307	530	83.3
Lead	5.3	36.5	1.9	13.1	0.425	0.411	710	111.5
Magnesium	6.5	44.8	2.4	16.5	0.350	0.065	112	17.6
Molybdenum	48.0	331.0	17.0	117.0	0.307	0.368	636	100.0
Monel metal	26.0	179.0	9.5	65.5	0.320	0.319	551	86.6
Nickel silver	18.5	127.0	7.0	48.3	0.322	0.316	546	85.8
Nickel steel	30.0	207.0	11.5	79.3	0.291	0.280	484	76.0
Phosphor bronze	16.1	111.0	6.0	41.4	0.349	0.295	510	80.1
Stainless steel (18-8)	27.6	190.0	10.6	73.1	0.305	0.280	484	76.0
Titanium alloys	16.5	114.0	6.2	42.4	0.340	0.160	276	43.4

Table A-10 Diameters and Areas of Coarse-Pitch and Fine-Pitch Metric Threads. [6]

Nominal Major Diameter d mm	Coarse-Pitch Series			Fine-Pitch Series		
	Pitch p mm	Tensile-Stress Area A_t mm²	Minor-Diameter Area A_r mm²	Pitch p mm	Tensile-Stress Area A_t mm²	Minor-Diameter Area A_r mm²
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259
24	3	353	324	2	384	365
30	3.5	561	519	2	621	596
36	4	817	759	2	915	884
42	4.5	1120	1050	2	1260	1230
48	5	1470	1380	2	1670	1630
56	5.5	2030	1910	2	2300	2250
64	6	2680	2520	2	3030	2980
72	6	3460	3280	2	3860	3800
80	6	4340	4140	1.5	4850	4800
90	6	5590	5360	2	6100	6020
100	6	6990	6740	2	7560	7470

Table A-11 Estimated Impact Factors for Various Applications[7]

Type of Application	Impact Factor *IF*
Uniform load, no impact	1.0–1.2
Precision gearing	1.1–1.2
Commercial gearing	1.1–1.3
Toothed belts	1.1–1.3
Light impact	1.2–1.5
Vee belts	1.2–2.5
Moderate impact	1.5–2.0
Flat belts	1.5–4.5
Heavy impact	2.0–5.0

Table A-12 Bearing Characteristics That May Influence The Selection of Bearing Type[7]

Bearing Type	Radial Capacity	Thrust Capacity	Limiting Speed	Radial Stiffness	Axial Stiffness
Deep-groove ball	moderate	moderate—both directions	high	moderate	low
Maximum-capacity ball	moderate (plus)	moderate—one direction	high	moderate (plus)	low (plus)
Angular contact ball	moderate	moderate (plus)—one direction	high (minus)	moderate	moderate
Cylindrical roller	high	none	moderate (plus)	high	none
Spherical roller	high	moderate—both directions	moderate	high (minus)	moderate
Needle roller	moderate to high	none	moderate to very high	moderate to high	none
Single-row tapered roller	high (minus)	moderate (plus)—one direction	moderate	high (minus)	moderate
Double-row tapered roller	high	moderate—both directions	moderate	high	moderate

Table A-13 Approximate Radial Load Factor For Selected Bearing Types[7]

Bearing Type	Dynamic				Static			
	X_{d_1}	Y_{d_1}	X_{d_2}	Y_{d_2}	X_{s_1}	Y_{s_1}	X_{s_2}	Y_{s_2}
Single-row radial ball bearing	1	0	0.55	1.45	1	0	0.6	0.5
Single-row angular contact ball bearing (shallow angle)	1	0	0.45	1.2	1	0	0.5	0.45
Single-row angular contact ball bearing (steep angle)	1	0	0.4	0.75	1	0	0.5	0.35
Double-row radial ball bearing	1	0	0.55	1.45	1	0	0.6	0.5
Double-row angular contact ball bearing (shallow angle)	1	1.55	0.7	1.9	1	0	1	0.9
Double-row angular contact ball bearing (steep angle)	1	0.75	0.6	1.25	1	0	1	0.65
Self-aligning single-row ball bearing[2]	1	0	0.4	$0.4\cot\alpha$	1	0	0.5	$0.2\cot\alpha$
Self-aligning double-row ball bearing	1	$0.4\cot\alpha$	0.65	$0.65\cot\alpha$	1	0	1	$0.45\cot\alpha$
Straight roller bearing[2] ($\alpha = 0$); (cannot take thrust)	1	0	—	—	1	0	1	0
Single-row roller bearing[3] ($\alpha \neq 0$)	1	0	0.4	$0.4\cot\alpha$	1	0	0.5	$0.2\cot\alpha$
Double-row roller bearing ($\alpha \neq 0$)	1	$0.45\cot\alpha$	0.65	$0.65\cot\alpha$	1	0	1	$0.45\cot\alpha$
Self-aligning single-row roller bearing	1	0	0.4	$0.4\cot\alpha$	1	0	0.5	$0.2\cot\alpha$
Self-aligning double-row roller bearing	1	$0.45\cot\alpha$	0.65	$0.65\cot\alpha$	1	0	1	$0.45\cot\alpha$

Table A-14 Dimentions and Load Ratings for Selected Single Row Radial-Deep Groove Ball Bearings[7]

Bearing Number	Bore		Outside Diameter		Width		Max Fillet Radius[1]		Min Shaft Abutment Diameter[2]		Basic Load Rating[3] C_d		C_s		Approximate Fatigue Load P_f		Limiting Speed,[5]
	mm	in	mm	in	mm	in	mm	in	mm	in	kN	10^3 lbf	kN	10^3 lbf	kN	10^3 lbf	With Grease/ Oil
6000	10	0.3937	26	1.0236	8	0.3150	0.3	0.012	12	0.472	4.62	1.04	1.96	0.44	0.08	0.019	30/36
6200			30	1.1811	9	0.3543	0.6	0.024	14	0.551	5.07	1.14	2.36	0.53	0.10	0.023	24/30
6300			35	1.3780	11	0.4331	0.6	0.024	14	0.551	8.06	1.81	3.40	0.76	0.14	0.032	20/26
6002	15	0.5906	32	1.2598	9	0.3543	0.3	0.012	17	0.669	5.59	1.26	2.85	0.64	0.12	0.027	22/28
6202			35	1.3780	11	0.4331	0.6	0.024	19	0.748	7.80	1.75	3.75	0.84	0.16	0.036	19/24
6302			42	1.6535	13	0.5118	1	0.039	20	0.787	11.40	2.56	5.40	1.21	0.23	0.051	17/20
6004	20	0.7874	42	1.6535	12	0.4724	0.6	0.024	24	0.945	9.36	2.10	5.00	1.12	0.21	0.048	17/20
6204			47	1.8504	14	0.5512	1	0.039	25	0.984	12.70	2.86	6.55	1.47	0.28	0.063	15/18
6304			52	2.0472	15	0.5906	1	0.039	26.5	1.043	15.90	3.57	7.80	1.75	0.34	0.075	13/16
6005	25	0.9843	47	1.8504	12	0.4724	0.6	0.024	29	1.142	11.20	2.52	6.55	1.47	0.28	0.062	15/18
6205			52	2.0472	15	0.5906	1	0.039	30	1.181	14.00	3.15	7.80	1.75	0.34	0.075	12/15
6305			62	2.4409	17	0.6693	1	0.039	31.5	1.240	22.50	5.06	11.60	2.61	0.49	0.110	11/14
6006	30	1.1811	55	2.1654	13	0.5118	1	0.039	35	1.378	13.30	2.99	8.30	1.87	0.36	0.080	12/15
6206			62	2.4409	16	0.6299	1	0.039	35	1.378	19.50	4.38	11.20	2.52	0.48	0.107	10/13
6306			72	2.8346	19	0.7480	1	0.039	36.5	1.437	28.10	6.32	16.00	3.60	0.67	0.151	9/11
6007	35	1.3780	62	2.4409	14	0.5512	1	0.039	40	1.575	15.90	3.57	10.20	2.29	0.44	0.099	10/13
6207			72	2.8346	17	0.6693	1	0.039	41.5	1.634	25.50	5.73	15.30	3.44	0.66	0.147	9/11
6307			80	3.1496	21	0.8268	1.5	0.059	43	1.693	33.20	7.46	19.00	4.27	0.82	0.183	8.5/10
6008	40	1.5748	68	2.6772	15	0.5906	1	0.039	45	1.772	16.80	3.78	11.60	2.61	0.49	0.110	9.5/12
6208			80	3.1496	18	0.7087	1	0.039	46.5	1.831	30.70	6.90	19.00	4.27	0.80	0.180	8.5/10
6308			90	3.5433	23	0.9055	1.5	0.059	48	1.890	41.00	9.22	24.00	5.40	1.02	0.229	7.5/9
6009	45	1.7717	75	2.9528	16	0.6299	1	0.039	50	1.969	20.80	4.68	14.60	3.28	0.64	0.144	9/11
6209			85	3.3465	19	0.7480	1	0.039	51.5	2.028	33.20	7.46	21.60	4.86	0.92	0.206	7.5/9
6309			100	3.9370	25	0.9843	1.5	0.059	53	2.087	52.70	11.90	31.50	7.08	1.34	0.301	6.7/8
6010	50	1.9685	80	3.1496	16	0.6299	1	0.039	55	2.165	21.60	4.86	16.00	3.60	0.71	0.160	8.5/10
6210			90	3.5433	20	0.7874	1	0.039	56.5	2.224	35.10	7.89	23.20	5.22	0.98	0.220	7/8.5
6310			110	4.3307	27	1.0630	2	0.079	59	2.323	61.80	13.90	38.00	8.54	1.60	0.360	6.3/7.5
6011	55	2.1654	90	3.5433	18	0.7087	1	0.039	61.5	2.421	28.10	6.32	21.20	4.77	0.90	0.202	7.5/9
6211			100	3.9370	21	0.8268	1.5	0.059	63	2.480	43.60	9.80	29.00	6.52	1.25	0.281	6.3/7.5
6311			120	4.7244	29	1.1417	2	0.079	64	2.520	71.50	16.10	45.00	10.10	1.90	0.427	5.6/6.7
6012	60	2.3622	95	3.7402	18	0.7087	1	0.039	66.5	2.618	29.60	6.65	23.20	5.22	0.98	0.220	6.7/8
6212			110	4.3307	22	0.8661	1.5	0.059	68	2.677	47.50	10.70	32.50	7.31	1.40	0.315	6/7
6312			130	5.1181	31	1.2205	2	0.079	71	2.795	81.90	18.40	52.00	11.70	2.20	0.495	5/6
6016	80	3.1496	125	4.9213	22	0.8661	1	0.039	86.5	3.406	47.50	10.70	40.00	8.99	1.66	0.373	5.3/6.3
6216			140	5.5118	26	1.0236	2	0.079	89	3.504	70.20	15.80	55.00	12.40	2.20	0.495	4.5/5.3
6316			170	6.6929	39	1.5354	2	0.079	91	3.583	124.00	27.90	86.50	19.50	3.25	0.731	3.8/4.5
6020	100	3.9370	150	5.9055	24	0.9449	1.5	0.059	108	4.252	60.50	13.60	54.00	12.10	2.04	0.459	4.3/5
6220			180	7.0866	34	1.3386	2	0.079	111	4.370	124.00	27.90	93.00	20.90	3.35	0.753	3.4/4
6320			215	8.4646	47	1.8504	2.5	0.098	113	4.449	174.00	39.10	140.00	31.50	4.75	1.070	3/3.6
6030	150	5.9055	225	8.8583	35	1.3780	2	0.079	161	6.339	125.00	28.10	125.00	28.10	3.90	0.877	2.6/3.2
6230			270	10.6299	45	1.7717	2.5	0.098	163	6.417	174.00	39.10	166.00	37.30	4.90	1.100	2/2.6
6330			320	12.5984	65	2.5591	3	0.118	166	6.535	276.00	62.10	285.00	64.10	7.80	1.750	1.9/2.4
6040	200	7.8740	310	12.2047	51	2.0079	2	0.079	211	8.307	216.00	48.60	245.00	55.10	6.40	1.440	1.9/2.4
6240			360	14.1732	58	2.2835	3	0.118	216	8.504	270.00	60.70	310.00	69.70	7.80	1.750	1.7/2
6340			420	16.5354	80	3.1496	4	0.157	220	8.661	377.00	84.80	465.00	105.0	11.2	2.520	1.5/1.8

1 Maximum allowable fillet radius at shaft (and housing) abutment.
2 For housing dimensions see manufacturer's catalogs.
3 C_d C_s
4 Equivalent radial load below which infinite life may be expected; analogous to fatigue endurance limit; contact SKF® for more accurate values.
5 *Absolute* rotational speed of the *inner race relative to the outer race*.

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