PROJECT PRELIMINARY REPORT

Submitted by:

ME11B098-VARUN KRISHNA R

ME11B121-SHARA RHAGHA WARDHAN B

ME11B123-AHMED SAMEER KHAN M

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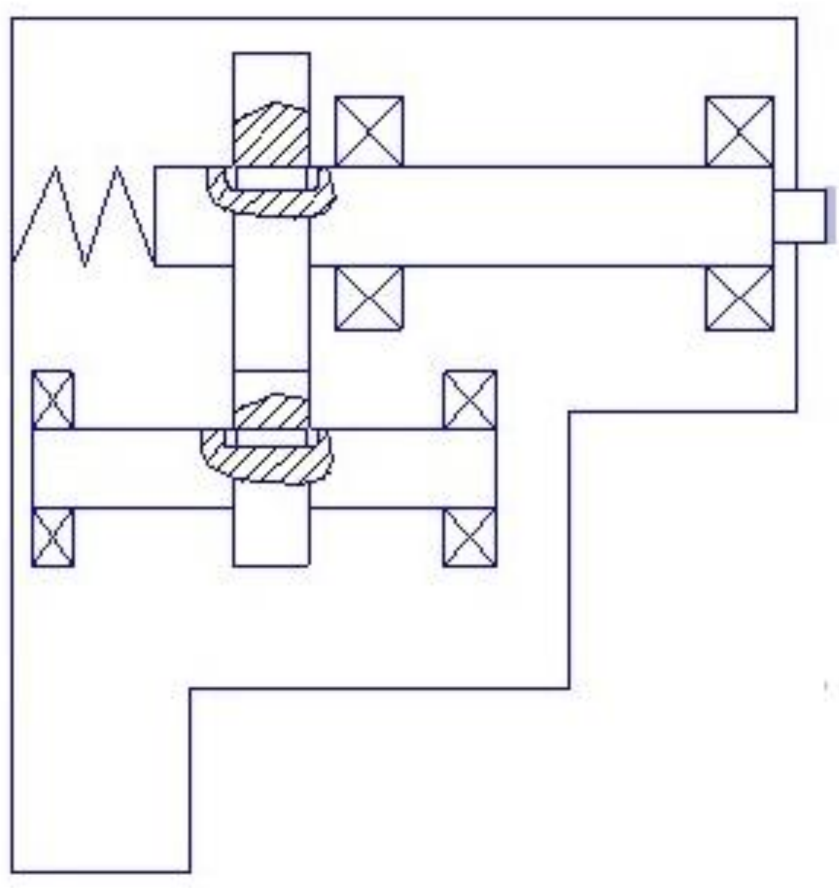
INTRODUCTION

Hand-drill machine is increasingly used in the industry for its ease of portability and handling. In the design given for the project, there are 2 shafts-input and output. The input and output shafts are tight fit on two roller bearings each. Each shaft has a gear through which the torque transmission occurs. The gears are attached to the shaft with the help of keys. A spring is attached to the output shaft to take care of the axial loads and vibrations that are transmitted to the shaft during the operation.

SCHEMATIC DIAGRAM OF THE HAND HELD DRILLING MACHINE



GEARS DESIGN ALGORITHM AND CALCULATIONS:

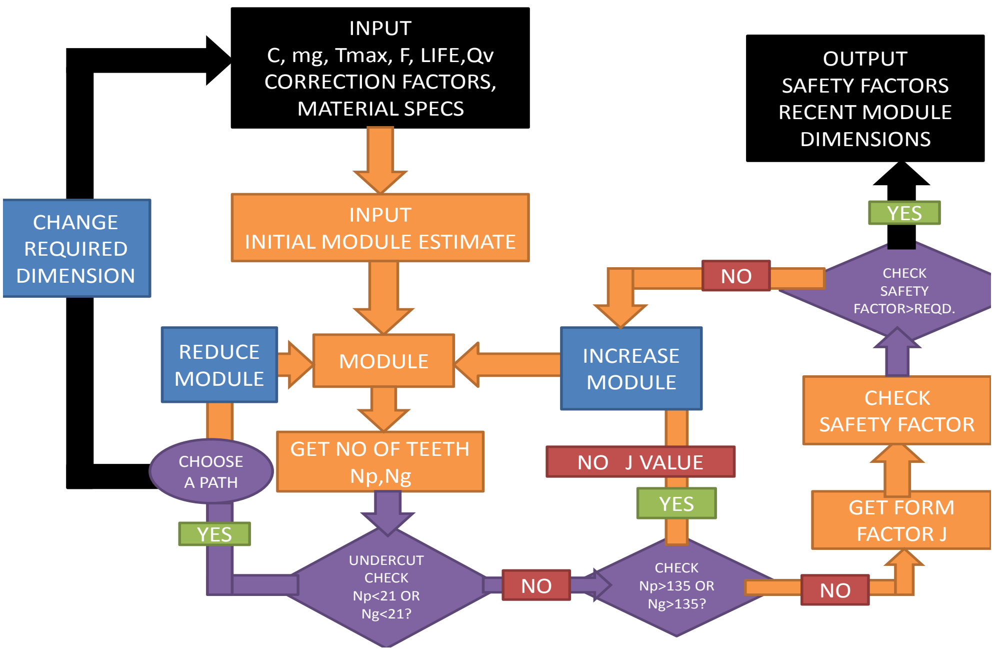
Table-1:Gear Design Calculations

COMMON INPUT		
CENTER DISTANCE	100	mm
GEAR RATIO	2	no unit
MAX RPM	700	rpm
MAX OUTPUT TORQUE	70	N-m
FACE WIDTH	30	mm
LIFE	5	YRS
	5	HRS/DAY
	5	DAYS/WEEK
Qv (quality factor)	8	
SAFETY FACTOR	2	no unit
(FOR BOTH BENDING AND COMPRESSION)		
COMMERCIAL APPLICATION		
RELIABILITY	95%	
TEMPERATURE	212	F

GEAR(DRIVEN)		
HARDNESS	400	BHN
THROUGH HARDENED		
AGMA CLASS A5 STEEL		
GRADE 1 STEEL		
STRENGTH IN BENDING	308	Mpa
STRENGTH IN SURFACE FATIGUE	1119	MPa
J	0.27	
BENDING STRESS	152	Mpa
Dgear	133	mm
OUTPUT FROM THE CODE		
SAFETY FACTOR	2.02	BENDING
	1.8	COMPRESSION
NO. OF TEETH	46	

PINION(DRIVER)		
HARDNESS	400	BHN
THROUGH HARDENED		
AGMA CLASS A5 STEEL		
GRADE 1 STEEL		
STRENGTH IN BENDING	311.9	Mpa
STRENGTH IN SURFACE FATIGUE	1101	MPa
J	0.244	
BENDING STRESS	168.6	Mpa
Dpinion	66.67	mm
OUTPUT FROM THE CODE		
SAFETY FACTOR	1.85	BENDING
	1.74	COMPRESSION
NO. OF TEETH	23	

COMMON OUTPUT		
MODULE(got from the output)	3	
COMPRESSIVE STRESS	834.2	Mpa
STRESS CORRECTION FACTORS		
K(I)	1	NON IDLER GEAR
K(B)	1	SOLID GEARS
K(S)	1	USUALLY 1
SHOCK FOR DRIVEN(SHOCKINESS OF LOAD)	HEAVY	
SHOCK FOR DRIVING(SHOCKINESS OF LOAD)	UNIFORM	
K(a)-SHOCKINESS FACTOR	1.75	
K(v)-VIBRATION LOAD FACTOR	0.794	
K(m)-LOAD DISTRIBUTION DUE TO AXIAL MIS	1.6	
SURFACE FATIGUE FACTORS		
ELASTIC COEFFICIENT(Cp FOR STEEL)	191	



SHAFT DESIGN ALGORITHM AND CALCULATIONS

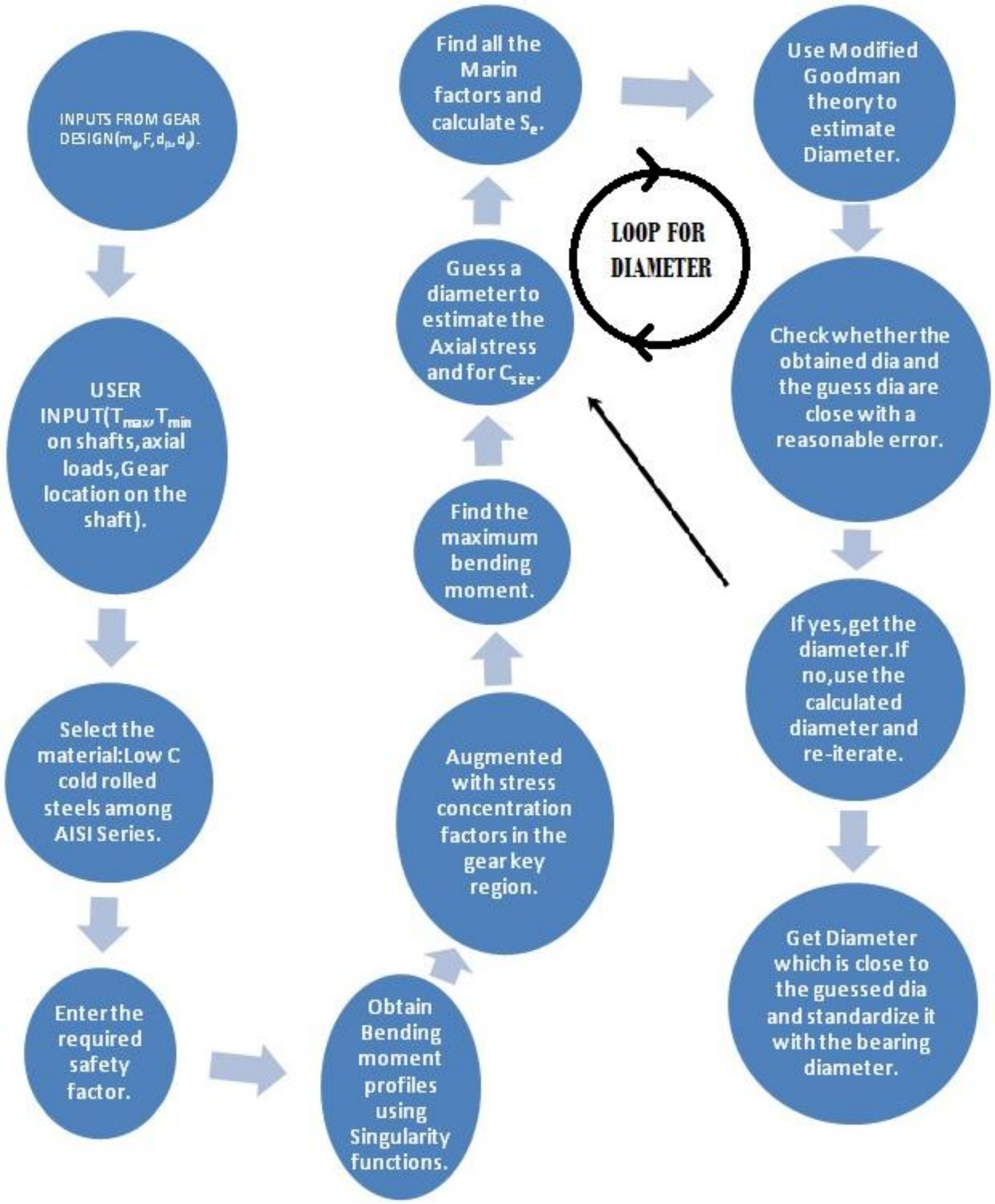


Table 2: Shaft Calculations

OUTPUT SHAFT		
Tmax	70	Nm
Tmin	46	Nm
Axial load	5	kN
Location of Gear	80	mm
(w.r.t both the bearings)	-20	mm
AISI STEEL 1010		
SAFETY FACTOR	2.5	
Marin factors		
C(load)	0.7	AXIAL
C(surf)	0.9444	COLD ROLLED
C(temp)	1	100 DEGREE CELSIUS
C(reliabilty)	0.897	90% reliability
C(size)	0.8827	Based on Guess dia.
Notch Radius	0.25	mm
Kt	2.5	
Kts	2.5	
Kf	1.75	
Kfs	1.855	
Kfm	1.75	
Kfsm	1.855	
OUTPUT SHAFT dia	21.57	mm
MODIFIED TO 25mm FOR BEARING DESIGN		

INPUT SHAFT		
Tmax	70	Nm
Tmin	46	Nm
Axial load	0	kN
Location of Gear	40	mm
	20	mm
AISI STEEL 1010		
SAFETY FACTOR	2.5	
Marin factors		
C(load)	1	BENDING
C(surf)	0.9444	COLD ROLLED
C(temp)	1	100 DEGREE CELSIUS
C(reliabilty)	0.897	90% reliability
C(size)	0.906	Based on Guess dia.
Notch Radius	0.25	mm
Kt	2.5	
Kts	2.5	
Kf	1.75	
Kfs	1.855	
Kfm	1.75	
Kfsm	1.855	
INPUT SHAFT dia	16.5	mm
MODIFIED TO 17mm BY BEARING LIFE		

BOLTED JOINTS-CALCULATIONS:

Assuming there are 9 bolted joints around the frame. They are under SHEAR LOADING.

From the FBD, the net axial force is 5000 N. Net transverse load(from the bearing) is equal to 2100N. Hence, the net force acting on each bolted joint is 602.56 N.

Assuming Mild steel with Ultimate Shear Strength as 340 N/mm^2 , and factor of safety = 3 , we calculate the minimum required diameter as **2.6 mm.**

Hence we use bolted joints of nominal diameter **5mm.**

BEARING DESIGN CALCULATIONS AND ALGORITHM:

From the FBD of the shaft, we estimate reaction forces. We assumed a tight fit at the bearing and take a coefficient of friction = 0.7. From this we estimate the axial load taken by the bearings and calculate life. Table-1 is used for finding V,X and Y factors specified by SKF.

F_{axial}=**980** N and F_{radial}=**1400** N (Taking μ as 0.7 and $F_a = \mu \cdot F_r$). Taking the diameter of the bearing to be **25mm**(the next highest specified value of Bearing w.r.t Shaft diameter),we get the values of C and C_0(**17252 N and 11577 N respectively**) from the SKF Bearing table.

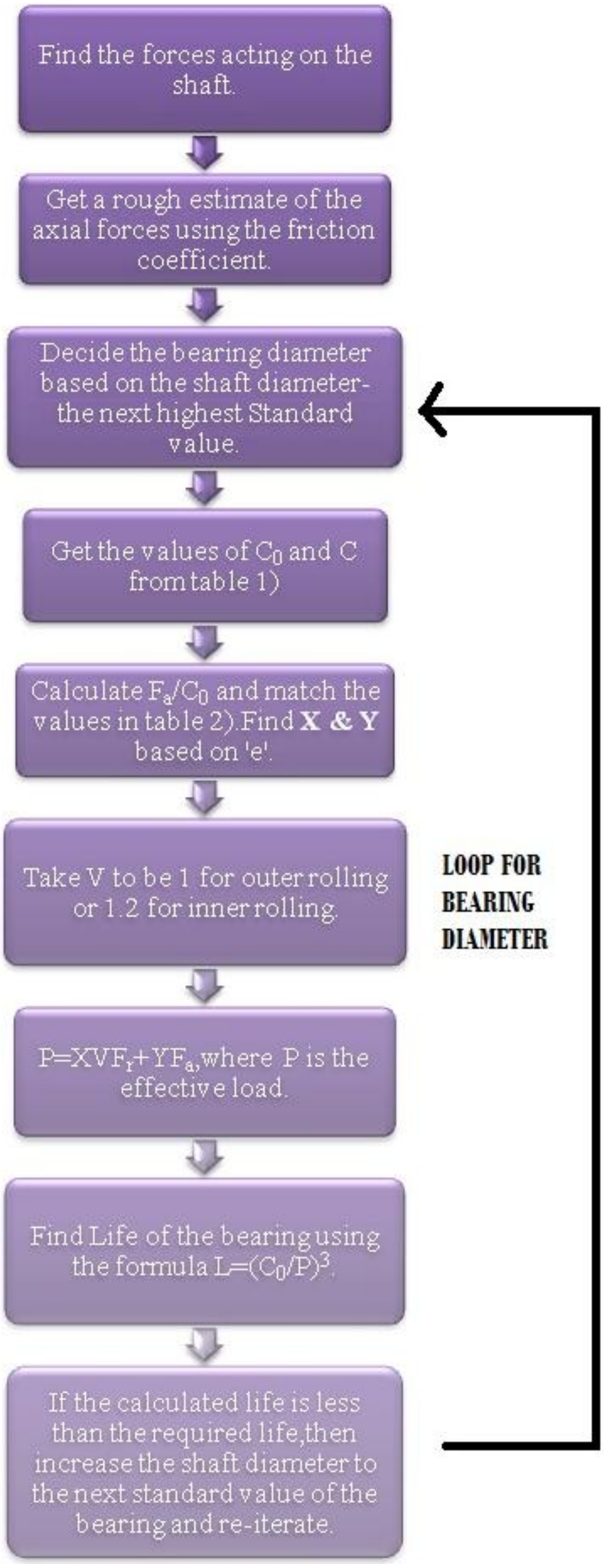
We take V=1 for rotation of outer race.From the table 1),$\mathbf{F_{axial}/C_0=0.084}$ and the corresponding **e=0.28**.[$F_{axial}/V.F_{radial}$ >e]→ **X=0.56 and Y=1.55** .

$$\mathbf{P} = (F_{radial} \times V \times X)+(F_{axial} \times Y)$$

$$= (1400 \text{ X } 0.56)+(980 \text{ X } 1.55) = \mathbf{2303\ N}.$$

Then, Life=$(C/P)^3$ from which we get Life=**420.4 million cycles>273 million cycles(working cycles).**

BEARING ALGORITHM



SPRING DESIGN ALGORITHM AND CALCULATIONS

Algorithm of spring

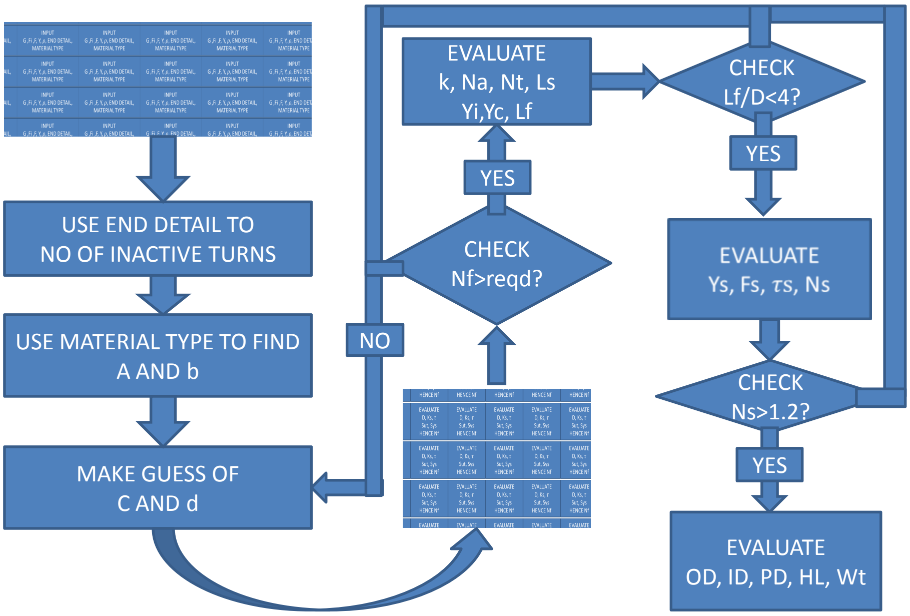


TABLE 3:SPRING CALCULATIONS

STEADY AXIAL LOAD	F	3750	N
PRELOAD WHILE ASSEMBLY	Fi	2500	N
MAXIMUM DEFLECTION	Ymax	10	mm
SPRING WIRE DIAMETER	d	4.5	mm
SPRING INDEX	C	4	
SPRING MEAN COIL DIAMETER	D	18	mm
INITIAL COMPRESSION OF SPRING	Yi	20.4	mm
MATERIAL SHEAR MODULUS	G	80	GPa
MATERIAL IS COLD DRAWN			
COEFFICIENT	A	1753.3	MPa
EXPONENT	b	-0.1822	
END DETAIL	SQUARED		
TURNS CORRECTION	Nc	2	
MINIMUMSAFETY FACTOR REQUIREI	Nf	1.3	
STATIC FACTOR	Ks	1.125	
YIELDING SHEAR STRESS	tou	2121.2	MPa
ULTIMATE YIELD STRENGTH	Sut	4692.9	MPa
SHEAR YIELD STRENGTH	Sys	2815.7	MPa
SAFETY FACTOR AGAINST YIELDING	Nf_yield	1.3274	
HENCE DESIGN IS SAFE AGAINST YIELDING			

SPRING CONSTANT w/o rounding turns	K	125	kN/m
ACTIVE NUMBER OF TURNS w/o rounding	Na_w_r	5.625	
ACTIVE NUMBER OF TURNS	Na	5.75	
SPRING CONSTANT(rounding turns)	K	122.28	kN/m
TOTAL NUMBER OF TURNS	Nt	7.75	
SHUT LENGTH OF SPRING	Ls	34.9	mm
INITIAL DEFLECTION	Yi	20.4	mm
CLASH ALLOWANCE		15	%
CLASH LENGTH ALLOWANCE	y_clash	1.5	mm
FREE LENGTH OF SPRING	Lf	66.8	mm
Lf/D RATIO	Lf/D	3.7122	
SINCE Lf/D<4, DESIGN IS SAFE AGAINST BUCKLING			
DEFLECTION WHEN SPRING IS SHUT	y_shut	31.9	mm
FORCE IN SHUT LENGTH	F_shut	3.91	kN
SHUT HEIGHT STRESS	tou_shut	2209.6	MPa
SAFETY FACTOR AGAINST SHUT HEIGHT YIELDING	Nf_SHUT	1.2743	
HENCE DESIGN IS SAFE AGAINST SHUT HEIGHT YIELDING			
OTHER PARAMETERS			
OUTER DIAMETER OF SPRING		22.5	mm
INNER DIAMETER OF SPRING		13.5	mm
MINIMUM HOLE DIAMETER THAT		23.4	mm
SHOULD BE USED WITH THIS SPRING			
MAXIMUM PIN DIAMETER THAT		12.6	mm
SHOULD BE USED WITH THIS SPRING			
WEIGHT OF THE SPRING	Wt	0.0541	kg
ASSEMBLED LENGTH OF THE SPRING	La	46.375	mm

CHOICE OF COUPLING

Hooke coupling can be used to couple the input shaft to the motor because of:

- No axial misalignment. This is necessary as gear meshing should be proper.
- Sufficient torque capacity.

KEY DESIGN ALGORITHM AND CALCULATIONS

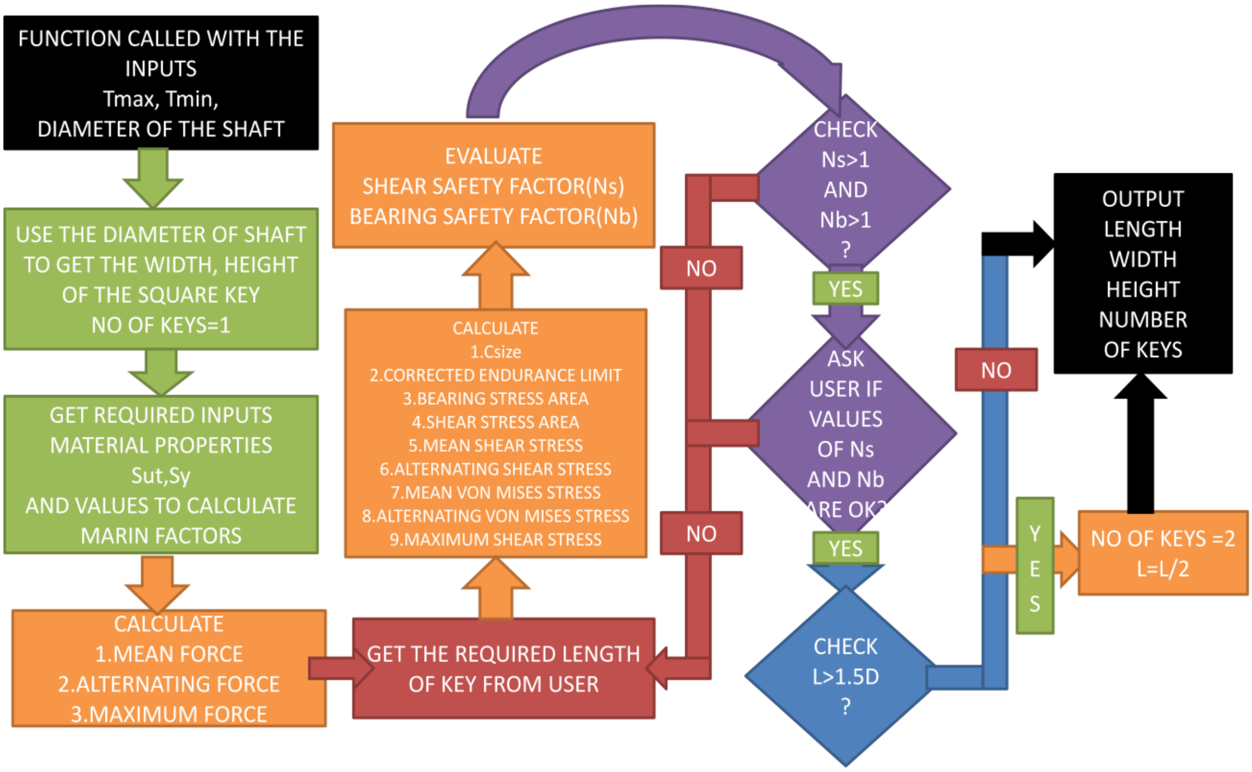


Table 4:KEY CALCULATIONS

		KEY FOR OUTPUT SHAFT		**KEY FOR INPUT SHAFT**	
MATERIAL		1010 STEEL		1010 STEEL	
DIAMETER OF THE SHAFT	D	25	mm	17	mm
ULTIMATE TENSILE STRENGTH	Sut	365	MPa	365	MPa
YIELD STRESS	Sy	303	MPa	303	MPa
UNCORRECTED ENDURANCE LIMIT	Se'	182.5	MPa	182.5	MPa
Cload		1	BENDING	1	BENDING
Csize		1	guessed based on dia	1	guessed based on dia
Ctemp		1	<212F	1	<212F
Csurf		0.9444	COLD ROLLED	0.9444	COLD ROLLED
Creliab		0.897	90%	0.897	90%
CORRECTED ENDURANCE LIMIT	Se	154.605	Mpa	154.605	Mpa
ALTERNATING SHEAR STRESS		9.23	MPa	12.834	MPa
VON MISES STRESS(alternating)		16	MPa	22.23	MPa
MEAN SHEAR STRESS		44.61	MPa	62.03	MPa
VON MISES STRESS(mean)		77.28	MPa	107.44	MPa
FACTOR OF SAFETY AGAINST SHEAR		3.17		2.3	
FACTOR OF SAFETY AGAINST BEARING STRESS		2.46		2.02	
WIDTH	W	8	mm	5	mm
HEIGHT	H	7	mm	5	mm
LENGTH	L	13	mm	22	mm
		LENGTH IS LESS THAN 1.5D		LENGTH IS LESS THAN 1.5D	

METHODS FOR CROSS-CHECKING THE OUTPUTS FROM THE CODE--INTERNAL CONSISTENCY, MATERIAL STRENGTHS, FACTORS OF SAFETY

The entire code is split into 5 modules namely Gear design, Shaft design, Bearing design, Key design, Spring design. Each module has its own sub-functions. During runtime, the modules are called in sequence.

1. In the gears module, the method to cross-check is the **direct output** of the module(which is changed suitably in each iteration) and the corresponding number of teeth.
   A. In each iteration, we can see how the code changes the module and the corresponding consequence(i.e. if we have undercutting, or too many teeth or insufficient safety factors).
   B. The materials we have used are AGMA Class A1-A5 steels.
   C. If the factors of safety are too less, or if we have consistent issue of undercutting(which can be observed in output) each time the module is changed, we allow a choice for re-entry of dimensions and material specifications.
2. In shaft design, we can **output the diameter in each iteration** to **show convergence.** The diameter is obtained deterministically based on the minimum required factor of safety provided by user.
   A. The material to be used is chosen by the user among a list of Cold-rolled low Carbon steels. AISI 1010 →AISI 1050
3. We standardise the above diameter and use the appropriate bearing corresponding to this. The only thing to **check** is the **L_{10} life of the bearing,** which should be **above the expected life** of the hand-drill. This is compared with the expected life and presented as a ratio.
4. Key design is straightforward. The cross-sectional dimensions are determined from the diameter of the shaft. Given an input of desired length, we output factor of safety. The user can decide to change the length for better factors of safety. The material is chosen among the same AISI steels.
5. For spring design, the parameters are **output in each step** so that we can see the working of the code. The user is allowed to intervene and change some parameters to get better factors of safety and/or diameters.

ANALYSIS OF INTEGRATED DESIGN-STRATEGIES TO ENSURE CONSISTENT RESULTS FROM EACH SUB-MODULE OF THE OVERALL DESIGN CODE

1. The gear design is the first step. We can re-run the code with many combinations of gear ratio, center distance, face width, material specifications to get required factors of safety.

2. The obtained parameters(gear ratio, pinion/gear diameters) are fed to shaft design code. We ensure that the diameters of the shaft are **sufficiently less than the diameters of the gears** on them.

3. Moreover, for bearing design, the shaft diameters are standardised to the next higher standard value available. We must check if the corresponding bearing has sufficient life. Else we must **change the shaft diameter** to a higher value.

4. For key design, we ensure that the length of the key is less than 1.5 times the diameter of the shaft. If not, we should use two keys.

5. In spring design, the diameter of the coil should be close to the diameter of the shaft. It would be pointless using a large diameter spring for a smaller shaft and vice versa.

Hence, if we do not get sufficient factors of safety of the spring for the expected diameters(near the shaft diameter), we may have to change shaft diameters accordingly. And consequently this affects the key design and the bearing design

Summary

1. We observe that we can't always have any kind of gear ratio or center distance suiting a factor of safety. Constraints on the availability of Form Factor(for a certain range of teeth), available material hardness, standardised modules all influence the design.

2. Obtained dimensions should be standardised with reference to some standard tables. For instance, the cross-sectional dimensions of keys and bearing bores have standard values. We have **restricted freedom** on certain parameters when it comes to design.

3. Also, the output of each module must be cross-checked so that we have a **reasonable size** as dictated by the application.

4. In each code, we allow for user intervention whenever the code tends to find itself in an infinite loop or when the dimensional changes are not suited to the user's preference. At the same time, there is a need to prompt the user if the target factor of safety cannot be achieved.

References:

1. Machine Design, Robert.L.Norton, Third Edition.

2. D.O.M.E Class Slides..