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Optimal design of gearbox for application in knee mounted biomechanical energy harvester

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Abstract— This paper first discusses the various parameters which can affect the design of the gearbox for knee mounted energy harvester device and later it frames the optimization problem of mass function based on the dimensions of gearbox for the problem. Then based upon the unique characteristics of working conditions it identifies various dimensional, strength and operating constraints for the problem and in later part of the problem optimization problem is solved using Multi-Start approach of MATLAB Global Optimization toolbox and value of global optimum function is obtained considering all the local optimum solution of problem.

Index Terms— Biomechanical energy harvester, knee mounted device, human energy harvester, Optimization of gearbox, Gearbox design, Global optimization, MATLAB.

1 Introduction

Human power is an attractive energy source because of many reasons. First, muscle converts food into positive mechanical work with peak efficiency of approximately 25%, comparable to that of internal combustion engines [1]. Second, the work by human can be performed at a high rate, with 100 W mechanical easily sustainable by an average person. Finally food, the original source of the metabolic energy required by muscles, is nearly as rich an energy source as gasoline and approximately 100 fold greater than batteries of the same weight. Given these attractive properties, it is not surprising that around 250 inventions have focused on converting human mechanical power into electrical power. Compared with the many methods of harvesting human energy like piezoelectric, Electro-active polymer and electromagnetic generators, a light-weight electromagnetic generator is capable of efficiently converting mechanical power into electrical power in a form suitable for charging a battery [2]. Heel strike, and knee and ankle motions seem to be good candidates for energy harvesting devices, since a relatively large part of their total energy can be recovered through electromagnetic generators. Furthermore, these motions are almost all single-degree of-freedom movements, which simplify the employed harvester device design. Quantity of available power 36W and quality of power 85% of negative work makes knee joint an obvious choice for harvesting from joint motion.[3][4] Although the input speed and torque requirements for magnetic generators are not ideal for direct coupling to knee motion, we found them superior to the other alternatives because of the feasibility of designing efficient transmissions to convert the knee joint power into a suitable form [2].

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The device developed by Qingguo Li et al. as shown in figure 1 uses a one-way clutch to transmit only knee extensor motions, a spur gear transmission to amplify the angular velocity, an electromagnetic generator to convert mechanical energy to electrical energy and a customized orthopaedic knee brace to support the hardware and distributing the device reaction torque over a large leg surface area [3]. The model of this device has been explained in figure 1 and figure 2.The input shaft accepts the knee motion at 1:1 ratio through a simple hinge (uni-axis) knee brace. A one-way clutch on the input shaft couples the gear train with knee motion during knee extension, and decouples the gear train from knee motion during knee flexion. The gear train transfers the low speed (ωk) but high torque (τr) mechanical power into high speed (ωg) and low torque (τg) mechanical power suitable for power generation. A miniature brushless DC generator converts the mechanical energy in to electrical energy where E is the generated electrical potential, Rg is the generator terminal resistance and Rl is the external electrical load [4].

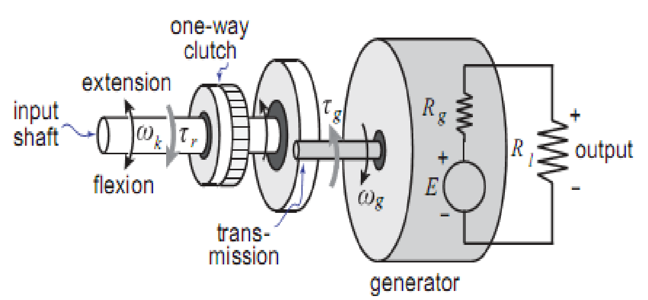

Figure 1 A simple model of knee mounted biomechanical energy harvester developed by Qingguo Li et al..

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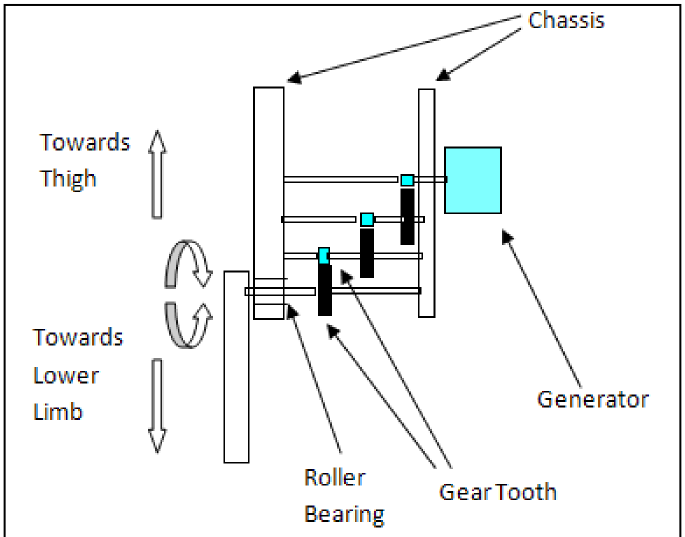

Figure 2 Model of knee mounted energy harvester considered for optimal design of gear box.

2. Initial Design

The electrical output of an electromagnetic generator is directly proportional to its angular speed and in this case of knee mounted energy harvester, it is proportional to gear ratio. The efficiency and electrical power output of the device are maximized at the highest gear ratios for a particular reaction torque indicating that we should choose the maximal gear ratio. Achieving higher gear ratios involves increasing gear diameter, decreasing gear diameter or both, however gear diameters can not be made arbitrarily large, due to size constraint, or arbitrarily small, due to strength requirements [2]. While these constraints could be partially circumvented by increasing the gear ratio through increasing the number of gear train stages, each additional pair of meshing teeth decreases transmission efficiency [7]. And each meshing pair also increases mass of the device which has to be kept low otherwise total human effort spent will become quite high. Considering above stated points, problem of achieving optimal design can be solved by minimizing the mass of the gearbox given it satisfies all dimensional, strength and minimum required gear ratio constraints.
Compound gear train is more compact than simple gear train and it is easy to achieve a high gear ratio through this gear train. So based on this compound gear train is chosen for gearbox [8].
A rotary magnetic-based generator typically rotates at a high speed (1000-10,000 rpm), while the human angular velocity for a typical joint is of the order of 20 rpm [2]. And from gait cycle, we know peak angular velocity at knee joint is 5 rad. /s (around 47 rpm) and average angular velocity of knee joint is around 2.65 rad. /s (around 25 rpm) [9] So considering 90 % of time energy to be harvested during knee extension phase angular velocity of knee joint comes out to be around 1rad./s. or 10 rpm and this leads us to a minimum gear ratio required to a order of around 100.

Now considering the cases of compound gear trains having minimum gear ratio 100 with keeping the number of stages to be minimal so for this starting with the case of gear train having two stages (as shown in figure 4) with some initial design approximation such as-

Gear ratio between stages = 10;
Module of teeth = 0.4 mm and 1 mm;

As minimum number of tooth on a gear are 14 for 20 degree pressure angle tooth to avoid interference between gear teeth [8]. So number of tooth is 14 and 140 for driven and driver tooth respectively.
Width of the teeth (b) = 10*m;
Where m is the module of gear tooth.
Service factor (C_s) = Our device operation lies in uniform type of working characteristics for driver and driven machines so value of service factor chosen is 1[8].
Now considering the peak torque value on knee joint as design torque value and corresponding angular velocity as design angular velocity and from gait profile of walking torque and corresponding value of angular velocity comes out to be around 15 N-m and 3 rad/s respectively .

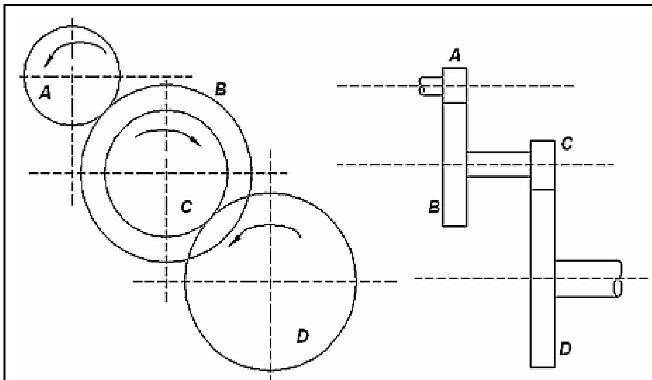

Figure 3 A two stage compound gear train [9]

As in between a meshing pair a small gear is more prone to failure than larger gear when both are made of same material so here we will concentrate only on small gear for stress strength constraints [8].
Velocity factor (C_v) given by Barth– For initial design angular velocity the velocity comes out to be less than 10 m/s which makes

$$C_v = 3/(3+v)$$

Where v is pitch velocity of teeth [8].
Ultimate strength of design material (S_{ut}) – 350 MPa.
Lewis form factor (Y) - For 14 number of tooth of 20 degree full depth involute system, it is 0.285.

Now estimating module based on beam strength through equation 1 we get value of module around 7 mm which is very large and with this the size and mass of system will become very large which we cannot afford [7].

$$m = \left[\frac{60 \times 10^6}{\pi} \left\{ \frac{(\mathrm{kW}) C_s (fs)}{z n C_v \left(\frac{b}{m} \right) \left(\frac{S_{ut}}{3} \right) Y} \right\} \right]^{1/3}$$

Where,
m – Module of gear tooth
kW – Power transmitted through the gear tooth
b – Width of gear tooth
C_s - Service factor
C_v – Velocity factor
fs - Factor of Safety

So next case is 3 - staged gear train and for this initial design approximations are-
Gear ratio between stages = 5;
Module of teeth = 0.5 mm;
So the tooth number becomes 14 and 70 for every meshing pair and all the other approximations are almost same except value of ultimate strength of material (S_{ut}) is chosen as 600 MPa.

Solving equation 1 for initial design approximation of module we get value of module around 1.5 mm, which makes it suitable to further optimization.

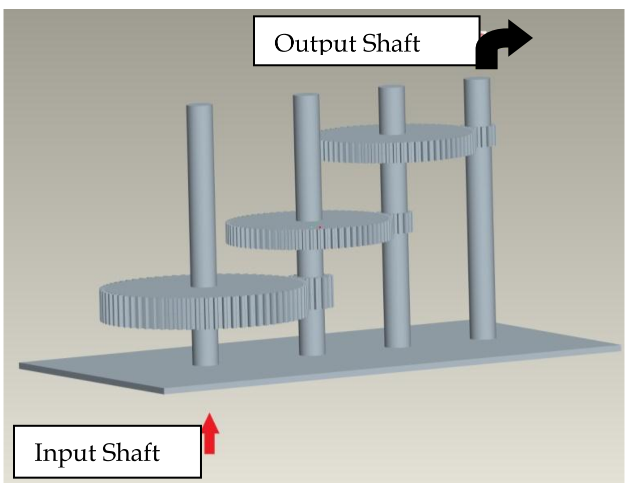

Figure 4 Initial design of 3-stage compound gearbox in Pro-Engineer software.

3. Optimization Problem

In this paper we will further discuss transmission part of the problem and first by defining the problem of optimization we will apply dimensional, strength and working constraints on the problem and will later solve the optimization problem with MATLAB Optimization toolbox.

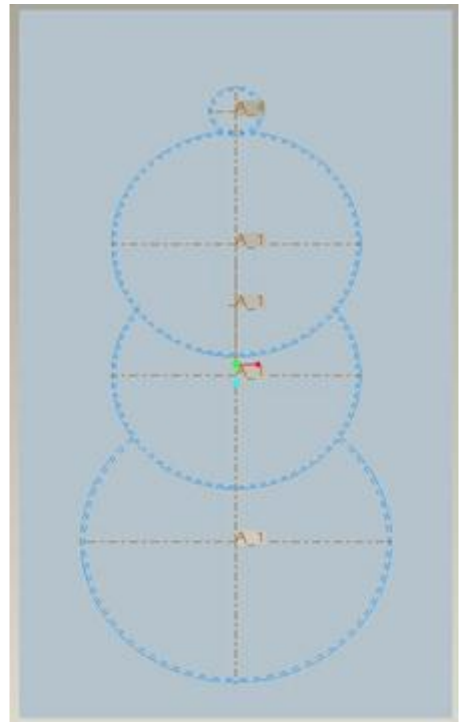

Figure 5 Top view of 3- stage compound gear train with initial design parameters.

3.1 Initial assumptions for Optimization

Some initial assumptions are made to accomplish the results-

1) All the four shafts are of same length and it is taken here 10cm. Diameter of sha is chosen as 10mm.
2) Material is same for all the shafts and gears and the value of ultimate tensile strength and density of material are 600 MPa and 7800kg/mm^3 respectively.

3.2 Defining Parameters

Variables of gear train are –

1) *Module* - Modules of 3 meshing pair is termed as m1 , m2 and m3 respectively.
2) *Number of gear tooth* – It is termed as $Z_{ij,}$

Where, i is number of meshing pair and
j number of meshing teeth.

So Z_{11} stands for number of tooth on first tooth of first meshing pair.

3) *Width of teeth* – It is termed as b1, b2 and b3 respectively for every meshing pair.

3.3 Optimization Function

Here the optimization function is mass of gear train which can be written as –

f=0.785*ϱ*(((m1*m1*z11*z11)- 100)*b1+((m1*m1*z12*z12-100)*b1)+((m2*m2*z21*z21)-100)*b2+((m2*m2*z22*z22)-100)*b2+((m3*m3*z31*z31)-100)*b3+((m3*m3*z32*z32)-100)*b3+(4*10000)); -(3)

Where "ϱ" is the density of material of gear train. This mass function has been calculated for 3 stage gear train as shown in figure 5.

3.4 Constraints

Here are constraints are of dimensional, strength, upper bound, lower bound of variables and operational type.

1) *Dimensional type of constraints* - A typical man has thigh width around 8 cm and here we have to mount our device [4]. So to make him comfortable width of system should not exceed 80 mm which can be directly understood by figure 3. So the dimensional constraints are –

m1*z11 <=80; -(4)
m1*z12 <=80; -(5)
m2*z21<=80; -(6)
m2*z22<=80; -(7)
m3*z31<=80; -(8)
m3*z32<=80; -(9)

2) *Tooth width constraints* – In practice, the optimum range of width which balances all factors like concentration of load and wearing of tooth surface is

8m < b < 12m. –(10)

Where b and m are width and module of tooth respectively. So constraints become

8*m1<=b1; -(11)
b1<=12*m1; -(12)
8*m2<=b2; -(13)
b2<=12*m2; -(14)
8*m3<=b3; -(15)
b3<=12*m3; -(16)

3) *Gear ratio constraints* - We want to increase the angular velocity with every stage and final gear ratio to be more than 100 so constraints become

z12<=z11; -(17)
z22<=z21; -(18)
z32<=z31; -(19)
100((z11*z21*z31)/ (z12*z22*z32))<= 0 ; -(20)

4) *Stress constraints* - In order to avoid failure of gear tooth due to bending,

$$P_{eff} <= S_b$$

Where S_b is bending strength of tooth and P_{eff} is the effective bending stress acting on gear tooth[8].
With a factor of safety (fs) equation becomes

S_b = (fs) * P_{eff}; -(21)
P_t = 2 * T / d ; -(22)

Effective bending stress(P_{eff}) is defined as[8]
P_{eff} = (C_s * P_t)/ C_v; -(23)

From figure 1 we know angular velocity and torque are 3 rad./s and 15 N-m respectively.

w1=3; -(24)
T1 = 15; -(25)
w2= (3*z11)/z12; -(26)
T2 = (15*z12)/z11;
w3= (w2*z21)/z22; -(27)
T3 = (T2*z22)/z21; -(28)
w4=(w3*z31)/z32; -(29)
T4= (T3*z32)/z31; -(30)

Where w, T and d are angular velocity, torque and diameters respectively. wi and Ti are angular velocity and torque of i^{th} shaft. Pitch diameters of gear are given by

d11 = m1*z11; -(31)
d12 = m1*z12; -(32)
d21 = m2*z21; -(33)
d22 = m2*z22; -(34)
d31 = m3*z31; -(35)
d32 = m3*z32; -(36)

Defining velocity and velocity factor

v1=(w1*d11)/2; -(37)
Cv1=3/(3+v1); -(38)
v2=(w2*d21)/2; -(39)
Cv2=3/(3+v2); -(40)
v3=(w3*d31)/2; -(41)
Cv3=3/(3+v3); -(42)

Finding Pt bending force

Pt1=2*T2/d12; -(43)
Pt2=2*T3/d22; -(44)
Pt3=2*T4/d32; -(45)

Calculating values of Peff

Peff1=Pt1/Cv1; -(46)
Peff2=Pt2/Cv2; -(47)
Peff3=Pt3/Cv3; -(48)

Applying stress-strength constraints

Pef1<= (200e6*(m1*b1*Y1)); -(49)
Pef2<=(200e6*(m2*b2*Y2)); -(50)
Pef3<=(200e6*(m3*b3*Y3)); -(51)

Modifying above constraint into function of only basic variables

z11+15*m1*z12*z12<=200e6*m1*m1*b1*Y1; -(52)
(15*z12*(2*z11+m2*z12*z21)<=
(200e6*m2*m2*z11*z11*z21*b2*Y2); -(53)

30*z22*(z12*z22+m3*z11*z21*z31)<=
2*200e6*m3*m3*z11*z31*z12*z22*b3*Y3; -(54)

5) *Initial design point* - Initial design points are stored in x0 variable. Order of the variable is represented by variable matrix x0 as shown in equation 55.

x0=[m1; m2; m3; z11; z12; z21; z22;z31; z32; b1; b2; b3] -(55)

And the value of initial design points, lower bounds and upper bounds of these variables is defined in equation 56, equation 57 and equation 58 respectively.

x0 = [1; 1; 1; 70; 14; 70; 14; 70; 14; 10; 8; 8]; -(56)

Lb = [0.5; 0.5; 0.5; 50; 14; 50; 14; 50; 14; 4; 4; 4]; -(57)

Ub = [2; 2; 2; 100; 25; 100; 25; 100; 25; 24; 24; 24]; -(58)

Lb and Ub – Represents lower bound and upper bound value respectively.

4 RESULTS AND CONCLUSION

Now solving the optimization problem for above 12 variables and 19 constraint through Multi-Start approach of MATLAB Global Optimization toolbox with 'active – set' algorithm for this multivariable constrained problem results in following optimum values of variables –

$m1^* = 0.5000$
$m2^* = 0.5000$
$m3^* = 0.5000$
$z11^* = 64.9823$
$z12^* = 14.0000$
$z21^* = 64.9822$
$z22^* = 14.0000$
$z31^* = 64.9823$
$z32^* = 14.0000$
$b1^* = 4.0000$
$b2^* = 4.0000$
$b3^* = 4.000$

And value of optimum function or weight of system comes out to be 0.3134 kg.

So through all the dimensional, operational and strength constraint we have developed a gearbox with optimized parameters for mass, which can bring better result out of knee mounted energy harvester device in terms of efficiency and portability of the device.

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