Scilab Textbook Companion for Machine Design by T. H. Wentzell, P. E[1]

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Book Description

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Scilab numbering policy used in this document and the relation to the above book.

Exa Example (Solved example)

Eqn Equation (Particular equation of the above book)

AP Appendix to Example(Scilab Code that is an Appednix to a particular Example of the above book)

For example, Exa 3.51 means solved example 3.51 of this book. Sec 2.3 means a scilab code whose theory is explained in Section 2.3 of the book.

Contents

List of Scilab Codes

Chapter 1

What Is Mechanical Design

Scilab code Exa 1.2 Sample Engineering Calculation

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-1.2    Page 13 ')    //Example 1.2

Sy=61000                          //[psi] Tensile strength of AISI 1020 cold drawn steel from Appendix 4 Page no 470
SF=2;                             //[] safety factor
F=300;                            //[lb] Weight of the ball
L=36;                             //[in] Length of round bar
Sy=61000;                         //[psi] Tensile strength from Appendix 4
M=F*L;                            //[in*lb] Bending moment Appendix 2

Sall=Sy/SF;                       //[psi] Allowable stress
Z=M/Sall;                         //[in^3] Section modulus for bending Sall=M/Z
D=(32*Z/%pi)^(1/3);               //[in] Diameter of bar
```

```
//Use 13/8 in bar
D1=1.625;

mprintf('\n\n Diameter of Bar is %f in',D1);

//Checking Deflection
I=%pi*D1^4/64;                    //[in^4] Moment of
    inertia Appendix 3
E=30*10^6;                        //[lb/in^2] Modulus of
    elasticity
Delta=F*L^3/(3*E*I);              //[in] Deflection

//Note- In the book I=0.342 in^4 is used instead of
    I=0.3422814 in^4

mprintf('\n The deflection of bar is %f in',Delta);


//Note: The deviation of answer from the answer
    given in the book is due to round off error.(In
    the book values are rounded while in scilab
    actual values are taken)
```

Chapter 2

Force Work and Power

Scilab code Exa 2.1 Torque

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-2.1    Page 26 ')     //Example 2.1

T=1080*12;                          //[in*lb] Torque in axle
d=30;                               //[in] Diameter of tire
F=T/(d/2);                          //[lb] Force exerted on
   the road surface

mprintf('\n\n The tire exerts %f lb force on the
   road surface',F);
```

Scilab code Exa 2.2 Work and Power

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-2.2    Page 28 ')     //Example 2.2
```

```
G=3.6;                              //Diffential ratio
N=3500/G;                           //[rpm] Axle rotational speed
d=30;                               //[in] Diameter of tire
dist=N/(60)*(%pi*d)                 //[in] Distance traveled in 1 sec
dist=dist/12;                       //[ft] Distance traveled in 1 sec
t=1;                                //[sec] Time period
F=864;                              //[lb] Force exerted by tire on road surface

W=F*dist;                           //[ft*lb] Workdone in 1 sec
P=F*dist/t;                         //[ft*lb/sec] Power
hp=P/550;                           //[hp] Power in horse power 1hp=550 ft*lb/sec

mprintf('\n\n Power to do work %f hp',hp);

//Comparing it to motor output:

Tm=300*12;                           //[in*lb] Engine torque
Nm=3500;                             //[rpm] Engine speed
Pm=Tm*Nm/63000;

mprintf('\n Motor output %f hp',Pm);
mprintf('\n The power output equaled the power at tire/road surface.');

//Note: The deviation of answer from the answer given in the book is due to round off error.(In the book values are rounded while in scilab actual values are taken)
```

Scilab code Exa 2.3 Force Pressure Relationship

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-2.3   Page 29 ')    //Example 2.3

T=300*12;                    //[in*lb] Engine torque
d=8;                         //[in] Crankshaft effective
    diameter

F=T/(d/2);                   //[lb] Force exerted by
    piston

A=%pi*(2^2)/4;               //[in^2] Area of cross
    section of piston
P=F/A;                       //[lb/in^2] Pressure in
    cylinder

mprintf('\n\n Pressure inside cylinder %f lb/in^2',P
    );
```

Chapter 3

Stress and Deformation

Scilab code Exa 3.1 `Stress and Deflection under Compressive Axial Load`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-3.1    Page No-41 \n');

F=20000;                //[lb] Load applied to steel
   bar
L=6;                    //[in] Length of steel bar
d=1;                    //[in] Diameter of steel bar
A=%pi*(d^2)/4;          //[in^2] Area of cross
   section of steel bar
E=30*10^6;              //[lb/in^2] Modulus of
   elasticity for AISI 1020 hot-rolled steel
Sy=30000;               //[lb/in^2] Yield limit

S=F/A;                  //[lb/in^2] Stress in bar
mprintf('\na. Stress in bar=%f lb/in^2.',S);

delta=F*L/(A*E);        //[in] Change in length of
   bar
mprintf('\nb. bar shorten by %f in.',delta);
```

```

if Sy>S then
    mprintf('\nc. The stress of %f psi is less than Sy of %f psi, so it will\n  return to its original size because the yield limit was not exceeded.',S,Sy);
else
    mprintf('The bar will not return to its original length')
end

//Note: The deviation of answer from the answer given in the book is due to round off error.(In the book values are rounded while in scilab actual values are taken)
```

Scilab code Exa 3.2 `Stress and Deflection due to Bending`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-3.2 Page No.43\n');

b=2;                        //[in] Width of beam
h=2;                        //[in] Height of beam
I=(b*h^3)/12;               //[in^4] Moment of inertia
F=3000;                     //[lb] Load applied to beam
L=36;                       //[in] Length of beam
c=1;                        //[in] Distance of outer most fiber from neutral axis
E=30*10^6;                  //[lb/in^2] Modulus of elasticity
Sy=30000;                   //[lb/in^2] Yield strength
Su=55000;                   //[lb/in^2] Ultimate
```

```
strength
SF=2;                        //[] Safety factor based on ultimate stress

M=F*L/4;                     //[lb*in] Bending moment
S=(M/I)*c;                   //[lb/in^2] Bending stress

//Note-In the book I=1.33 in^4 is used instead of I=1.3333333 in^2

mprintf('\na. The maximum stress in beam is %f lb/in^2',S);

delta=-F*L^3/(48*E*I);        //[in] Maximum deflection in this beam

mprintf('\nb. The maximum deflection in this beam is %f in.',delta);

if Sy>S then
    mprintf('\nc. Yes, the stress of %f lb/in^2 is less than the yield of Sy=%f lb/in^2.',S,Sy);
else
    mprintf('\nc. No, the stress of %f lb/in^2 is greater than the yield of Sy=%f lb/in^2',S,Sy);
end

Sall=Su/SF;                   //[lb/in^2] Allowable stress

if Sall>S then
    mprintf('\nd. It is acceptable because allowable stress is greater than the acttual stress of %f lb/in^2.',S);
else
    mprintf('\nd. Design is not acceptable because allowable stress is less than the actual
```

```
        stress of %f lb/in^2.',S)
end

//Note: The deviation of answer from the answer
   given in the book is due to round off error.(In
   the book values are rounded while in scilab
   actual values are taken)
```

Scilab code Exa 3.3 Shear Stress

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-3.3 Page No.45\n');

Su=80*10^3;            //[lb/in^2] Ultimate strength
d=0.5;                 //[in] Diameter of pin
As=%pi*d^2/4;          //[in^2] Area of cross section
   of pin
F=20*10^3;             //[lb] Load acting

Ss=F/(2*As);           //[lb/in^2] Shear stress

if 0.5*Su>=Ss & 0.6*Su>=Ss then
    mprintf('Pin would not fail');
else
    mprintf('\n Actual stress is too high and the
        pin would fail.');
end
```

Scilab code Exa 3.4 Torsional Shear Stress

```
clc;
```

```
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-3.4 Page No.46\n');

hp=10;              //[hp] Power transmitted
rpm=1750;           //[rpm] Turning speed
d=0.5;              //[in] Diameter of shaft
L=12;               //[in] Length of shaft
G=11.5*10^6         //[lb/in^2] shear modulus of elasticity
Su=62000;           //[lb/in^2]

T=63000*hp/rpm;     //[in*lb] Torque transmitted
Z=%pi*d^3/16;       //[in^3] Polar section modulus
Ss=T/Z;             //[lb/in^2] Torsional shear stress

//Note- In the book Z=0.025 in^3 is used instead of Z=0.0245437 in^3

mprintf('\na. Stress in the shaft is %f lb/in^2.',Ss)

J=%pi*d^4/32;       //[in^4] Polar moment of inertia
theta=T*L/(J*G);    //[radians]

//Note- In the book J=0.0061 in^4 is used instead of J=0.0061359 in^4

mprintf('\nb. The angular deflection of shaft would be %f radians',theta);

SF=3;                //[] Safety factor based on ultimate strength

Zd=T/(0.5*Su/SF);    //[in^3] Polar section modulus required for SF=3
Dd=(16*Zd/%pi)^(1/3); //[in] Diameter of shaft
```

```
    required Z=%pi*d^3/16

mprintf('\nc. Diameter of shaft required is %f in.',
    Dd);
```

Scilab code Exa 3.5 Critical Load in Pinned End Column

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-3.5 Page No.53\n');

L=30;                    //[in] Length of link
d=5/8;                   //[in] Diameter of link
I=%pi*d^4/64;            //[in^4] Moment of
    inertia
A=%pi*d^2/4;             //[in^2] Area of cross
    section
E=30*10^6;               //[lb/in^2] Modulus of
    elasticity

r=sqrt(I/A);             //[in] Radius of gyration

mprintf('\n The radius of gyration %f in.',r);

K=1;                     //[] End support
    condition factor

Le=K*L;                  //[in] Effective length

mprintf('\n Effective length is %f in',Le);

SR=Le/r;                 //[] Slenderness ratio

mprintf('\n Slenderness ratio is %f.',SR)
```

```
Sy=42000;                  //[lb/in^2] Yield strength

Cc=sqrt(2*%pi^2*E/Sy);     //[] Column constant

mprintf('The column constant is %f.',Cc);

if SR>Cc then
    mprintf('\n Slenderness ratio is greater than column constant, so use the euler formula')
end

I=%pi*d^4/64;              //[in^4] Moment of inertia

mprintf('\n The moment of inertia is %f in^4',I);

Pc=%pi^2*E*I/Le^2;         //[lb] Critical force

//Note- In the book I=0.0075 in^4 is used instead of I=0.0074901 in^4

mprintf('\n The critical force is %f lb.',Pc);
```

Scilab code Exa 3.6 Critical Load in Fixed End Column

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-3.6 Page No.55\n');

L=60;                     //[in] Length of column
Sy=36000;                 //[lb/in^2] Yield strength
SF=2;                     //[] Safty factor
E=30*10^6;                //[lb/in^2] Modulus of elasticity
```

```
A=2.26;                   //[in^2] Area of cross
    section (Appendix 5.4)
I=0.764;                  //[in^4] Moment of inertia
    (Appendix 5.4)

r=sqrt(I/A);              //[in] Radius of gyration

K=0.65;                   //[] End support condition
    factor from Figure 3.8
Le=K*L;                   //[in] Effective length

mprintf('\n The effective length is %f in.',Le);

SR=Le/r;                  //[] Slenderness ratio

mprintf('\n The slenderness ratio is %f.',SR);

Cc=sqrt(2*%pi^2*E/Sy);    //[] Column constant

mprintf('\n The column constant is %f.',Cc);

if Cc>SR then
    mprintf('\n The column constant is greater than
        slenderness ratio, so use the Johnson formula
        .');
end

F=(A*Sy/SF)*(1-Sy*SR^2/(4*%pi^2*E));

mprintf('\n The acceptable load for a safty factor
    of 2 is %f lb.',F);
```

Chapter 4

Combined Stress and Failure Theories

Scilab code Exa 4.1 Design of Short Column with Eccentric Load

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-4.1    Page 66 ')

D=2;                  //[in] Dia. of short column
F=10000;             //[lb] Load applied
L=15;                 //[in] Length of column
e=2;                  //[in] Offset of load

A=(%pi*D^2)/4;        //[in^2] Area of cross section
    of column
SA=F/A;               //[lb/in^2] Axial Stress

Z=(%pi*D^3)/32;       //[in^4] Section modulus for
    bending
M=F*e;                //[lb*in] Bending moment
SB=M/Z;               //[lb/in^2] Bemding stress

```

```
S=-SA-SB;                  //S=(+-)SA+(+-)SB Max. stress

//The bending stress and axial stress are added on inner side of column

mprintf('\n\n Maximum stress in column is %f lb/in^2.\n',S)
```

Scilab code Exa 4.2 Coplanar Shear

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-4.2     Page 67 ')

F1=800;                   //[lb] Vertical force
F2=600;                   //[lb] Horizontal force
D=0.5;                    //[in] Pin diameter
A=(%pi*D^2)/4;           //[in^2] Area of cross section of pin

F=sqrt(F1^2+F2^2);        //[lb] Resultant force on pin
S=F/A;                    //[lb/in^2] Shear stress in pin

//If forces were not perpendicular, they would be added vectorially.
mprintf('\n\n Shear stress in pin is %f lb/in^2.',S);
```

Scilab code Exa 4.3 Combined Torsion and Shear

```
clc;
clear;
```

```
mprintf('MACHINE DESIGN\n Timothy H. Wentzell, P.E.\n Example 4.3   Page no 68');

P=50;               //[hp] Power transmitted
N=300;              //[rpm] Speed
D=10;               //[in] Effective pitch diameter of sprocket
d=1;                //[in] Diameter of shaft from figure 4.3
Z=(%pi*d^3)/16;     //[in^3] Section modulus of shaft
A=(%pi*d^2)/4;      //[in^2] Area of cross section

T=(63000/N)*P;     //[lb*in] Torque required to transmit power
F=T/(D/2);          //[lb] Driving force in chain

Ss=F/A;             //[lb/in^2] Shear stress in shaft

St=T/Z;             //[lb/in^2] Torsional stress in shaft

S=Ss+St;            //[lb/in^2] Resultant stress

//Note-There is mistake in addition of Ss and St.

//This value would be compared to shear stress allowable for shaft material

mprintf('\n\n The combined stress in 1 inch diameter shaft is %f lb/in^2.',S);
```

Scilab code Exa 4.4 Combined Normal and Shear Stress

```
clc;
clear;
```

```
mprintf('MACHINE DESIGN\n Timothy H. Wentzell, P.E.\
   n Example 4.4  Page no 71')

P=20;               //[hp] Power transmitted by chain
   drive
n=500;              //[rpm] speed
d=8;                //[in] Pitch diameter of sprocket
fos=2;
D=1.25;             //[in] Diameter of shaft
L=12;               //[in] Distance between two
   supporting bearings
Z1=%pi*D^3/16;      //[in^3] Section modulus for
   torsion
Z2=%pi*D^3/32;      //[in^3] Section modulus for
   bending

T=63000*P/n;        //[in*lb] Torque on shaft

F=T/(d/2);          //[lb] Force in chain

M=F*L/4;            //[in*lb] Bending moment in shaft

Ss=T/Z1;            //[lb/in^2] Torsional shear stress

Sb=M/Z2;            //[lb/in^2] Bending normal stress

//Note- In the book Sb=9860 lb/in^2 is used instead
   of Sb=9856.7075 lb/in^2

S=(Sb/2)+sqrt(Ss^2+(Sb/2)^2);  //[lb/in^2] Combined
   max. stress

Sy=30000;           //[lb/in^2]From APPENDIX 4 Page no
   -470 for AISI 1020 and Hot-rolled steel
FOS=(Sy/2)/S;       //[]Actual factor of safty

if S < Sy/2 then    //Strength is greater than
   combined stress so design is safe
```

```
    mprintf('\n\n Design is acceptable and Combined
        stress is %f lb/in^2',S);
else
    mprintf('\n\n Design is not acceptable');
end
```

Chapter 5

Repeated Loading

Scilab code Exa 5.1 `Design of a Shaft using the Soderberg Method`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-5.1 Page No.93\n');

SF=2;                    //[] Safety factor
F=500;                   //[lb] Load
L=40;                    //[in] Length of shaft
Su=95000;                //[lb/in^2] Ultimate strength (Appendix 4)
Sy=60000;                //[lb/in^2] Yield strength (Appendix 4)

Mmax=F*L/4;              //[lb*in] Maximum bending moment
Mmin=-F*L/4;             //[lb/in^2] Minimum bending moment

Csurface=1;              //[] As surface is polished
Csize=0.85;              //[] Assuming 0.5<D<2
```

```
Ctype=1;                        //[] Bending stress

Sn=Csize*Csurface*Ctype*(0.5*Su); //[lb/in^2] Endurance limit

if Mmax==abs(Mmin) then
    Sm=0;                      //[lb/in^2] Mean stress
end

Sa=Sn/SF;                      //[lb/in^2] As (1/SF)=(Sm/Sy)+(Sa/Sn) from soderberg equation

Sa_Z=(Mmax-Mmin)/2;            //[lb*in^2] Product of altenating stress and section modulus

Z=Sa_Z/Sa;                     //[in^4] Section modulus

D=(32*Z/%pi)^(1/3);            //[in] Diameter of shaft

D1=1.375;                      //[in] Next higher available is 1.375 in. so use D1

mprintf('\n The required diameter of rotating shaft is %f in.', D1);
```

Scilab code Exa 5.2 Design of a Cantilever Beam using the Soderberg Method

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-5.2 Page No.95\n');

Su=90000;                       //[lb/in^2] Ultimate strength (Appendix 8)
Sy=37000;                       //[lb/in^2] Yield
```

```
strength (Appendix 8)
Sni=34000;                   //[lb/in^2] Endurance
   limit (Appendix 8)
SF=1.6;                      //[] Safety factor

F=1000;                      //[lb] Load
L=12;                        //[in] Length of
   cantilever beam

Mmax=F*L;                    //[lb*in] Maximum bending
    moment
Mmin=0;                      //[lb*in] Minimum bending
    moment

Csize=0.85                   //[] Assuming 0.5<D<2 in
Ctype=1;                     //[] Bending stress
Csurface=1;                  //[] As surface is
   polished

Malt=(Mmax-Mmin)/2;          //[lb*in] Alternating
   bending moment

Mmean=(Mmax+Mmin)/2;         //[lb*in] Mean bending
   moment

Sn=Csize*Csurface*Ctype*Sni; //[lb/in^2] Modified
   endurance limit

Z=((Mmean/Sy)+(Malt/Sn))*SF; //[in^3] Section
   modulus

D=(32*(Z)/%pi)^(1/3);          //[in] Diameter of bar

mprintf('\n The required diameter of bar using the
   soderberg method is %f in.',D);
```

Scilab code Exa 5.3 Design of a Cantilever Beam using the Modified Goodman Method

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-5.3 Page No.97\n');

Su=90000;                    //[lb/in^2] Ultimate strength (Appendix 8)
Sy=37000;                    //[lb/in^2] Yield strength (Appendix 8)
Sni=34000;                   //[lb/in^2] Endurance limit (Appendix 8)
SF=1.6;                      //[] Safety factor

F=1000;                      //[lb] Load
L=12;                        //[in] Length of cantilever beam

Mmax=F*L;                    //[lb*in] Maximum bending moment
Mmin=0;                      //[lb*in] Minimum bending moment

Csize=0.85                   //[] Assuming 0.5<D<2 in
Ctype=1;                     //[] Bending stress
Csurface=1;                  //[] As surface is polished

Malt=(Mmax-Mmin)/2;          //[lb*in] Alternating bending moment

Mmean=(Mmax+Mmin)/2;         //[lb*in] Mean bending moment
```

```
Sn=Csize*Csurface*Ctype*Sni; //[lb/in^2] Modified endurance limit

Z=((Mmean/Su)+(Malt/Sn))*SF; //[in^3] Section modulus

D=(32*(Z)/%pi)^(1/3);        //[in] Diameter of bar

mprintf('\n The required diameter of bar using the soderberg method is %f in.',D);

//Note that the modified Goodman results in a less conservative size as would be expected from figure 5.10
```

Scilab code Exa 5.4 `Design of Water Pump Connecting Rod`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-5.4 Page No.98\n');

Su=95000;               //[lb/in^2] Ultimate strength
Sy=60000;               //[lb/in^2] Yield strength
SF=1.5;                 //[] Safety factor

Fmax=1000;             //[lb] Maximum load
Fmin=-6000;            //[lb] Minimum load

Fmean=(Fmax+Fmin)/2;    //[lb] Mean load
Fmean=abs(Fmean);       //[lb] Considering absolute value
Falt=(Fmax-Fmin)/2;     //[lb] Alternating load

```

```
Csize=1                      //[] Assuming b<0.5 in
Ctype=0.8                    //[] Axial stress
Csurface=0.86                //[] Machined surface Figure
    5.7b

Sn=Csize*Csurface*Ctype*(0.5*Su); //[lb/in^2]
    Modified endurance limit

A=((Fmean/Sy)+(Falt/Sn))*SF;      //[in^2] Area of
    cross section of rod

b=sqrt(A);                        //[in] Side of
    square cross section

mprintf('\n The required square size in the center
    section is %f in.',b);
```

Scilab code Exa 5.5 Factor of Safety for Design with Stress Concentration Factor

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-5.5 Page No.100\n');

Su=80000;                     //[lb/in^2] Ultimate
    strength
Sy=71000;                     //[lb/in^2] Yield strength

D=0.6;                        //[in] Diameter of shaft
d=0.5;                        //[in] Diameter of shaft
    at notch
r=0.05;                       //[in] Radius of notch
Z=%pi*d^3/16;                 //[in^3] Polar section
    modulus
Tmax=200;                     //[lb*in] Maximum load
```

```
Tmin=0;                     //[lb*in] Minimum load

Smax=Tmax/Z;                //[lb/in^2] Maximum stress
Smin=Tmin/Z;                //[lb/in^2] Minimum stress

Smean=(Smax+Smin)/2;        //[lb/in^2] Mean stress
Salt=(Smax-Smin)/2;         //[lb/in^2] Alternating stress

Csize=0.85;                 //[] Assume 0.5<D<2 in
Csurface=0.88;              //[] Machined surface Figure 5.7b
Ctype=0.6;                  //[] Torsional stress

Sn=Csize*Csurface*Ctype*(0.5*Su);   //[lb/in^2] Modified endurance limit

Kt=1.32;                    //[] (D/d)=1.2, (r/d)=0.1 from Appendix 6c

N=inv(Smean/(0.5*Sy)+Kt*Salt/Sn);   //[] Safety factor

mprintf('\n The factor of safety for this design is %f',N);
```

Scilab code Exa 5.6 Factor of Safety for Design when Desired Life is known

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-5.6 Page No.102\n');

//From Example Problem 5.5
Sy=71000;                   //[lb/in^2] Yield strength
```

```
Smax=8148.7331 ;              //[lb/in^2] Maximum stress
Smin=0;                       //[lb/in^2] Minimum stress
Smean=(Smax+Smin)/2;          //[lb/in^2] Mean stress
Salt=(Smax-Smin)/2;           //[lb/in^2] Alternating stress
Sn=18000;                     //[lb/in^2] Modified endurance strength
Kt=1.32                       //[] Stress concentration factor

Nd=100000;                    //[cycles] Desired life

Snn=Sn*(10^6/Nd)^0.09;              //[lb/in^2]

N=inv(Smean/(0.5*Sy)+Kt*Salt/Snn);  //[] Factor of safety

mprintf('\n The new factor of safety for this condition is %f.',N);
```

Chapter 6

Fasteners and Fastening Methods

Scilab code Exa 6.1 Torquing Method

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-6.1 Page No.120\n');

As=0.334;                //[in^2] Tensile stress area
    (Table 6.1)
Sp=85000;                //[lb/in^2] Proof strength (
    Table 6.3)
D=3/4;                   //[in] Nominal diameter of
    thread

Fi=0.85*As*Sp;           //[lb] Desired intial preload
C=0.2;                   //[] Torque coefficient

T=C*D*Fi;                //[in*lb] Torque

mprintf('\n The required torque is %f lb*in.',T);
```

Scilab code Exa 6.2 Turn of Nut Method

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-6.2 Page No.121\n');

L=5;                    //[in] Length of engagement
E=30*10^6;              //[lb/in^2] Modulus of
   elasticity
As=0.334;               //[in^2] Tensile stress area
   (Table 6.1)
Sp=85000;               //[lb/in^2] Proof strength (
   Table 6.3)
Fi=0.85*As*Sp;          //[lb] Desired intial preload

Delta=Fi*L/(As*E)       //[in] Elongation

pitch=0.1;              //[in] Pitch for 3/4 UNC
TA=Delta*360/pitch;     //[Degree] Torque angle

mprintf('\n The angle of rotation needed is %f
   degree.',TA);
```

Scilab code Exa 6.3 Elastic Analysis of Bolted Connections

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-6.3 Page No.122\n');

```

```
Alpha=6.5*10^-6;          //[in/(in*F)] Thermal
   expansion coefficient (Appendix 8)
L=5;                      //[in] Length of engagement

Delta=0.01204;          //[Degree] Elongation

DT=Delta/(Alpha*L);       //[F] The temperature we
   would need to heat this bolt above the sevice
   temperature

mprintf('\n The temperature we would need to heat
   this bolt above the sevice temperature is %f F.',
   DT);
```

Scilab code Exa 6.4 Elastic Analysis of Bolted Connections

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-6.4 Page No.124\n');

Dp=20;                    //[in] Pressure vessel head
   diameter
Ds=1.25;                  //[in] Stud diameter
Ls=6;                     //[in] Stud length
Af=50;                    //[in^2] Clamped area of
   flanges

E=30*10^6;                //[lb/in^2] Modulus of
   elasticity
C=0.15;                   //[] Torque coefficient
Si=120000;                //[lb/in^2] Proof strength (
   Table 6.3)
A=1.073;                  //[in^2] Tensile stress area
   (Table 6.1)
```

```
Fi=0.9*Si*A;            //[lb] Desired intial load

T=C*Ds*Fi;              //[lb*in] Torque

mprintf('\n1. The required torque is %f lb*in.',T);

Pp=500;                  //[lb/in^2] Pressure inside the pressure vessel
Ap=%pi*Dp^2/4;           //[in^2] Pressure vessel head cross section area

Kb=A*E/Ls;               //[lb/in] Stiffness per stud
Kf=Af*E/Ls;              //[lb/in] Stiffness per flange
Fe=Pp*Ap;                //[lb] Force on pressure vessel head

Ft=10*Fi+(10*Kb/(10*Kb+Kf))*Fe;  //[lb] Total load on the bolt

mprintf('\n2. The total load on the bolt is %f lb.',Ft);
```

Chapter 7

Impact and Energy Analysis

Scilab code Exa 7.1 Impact Energy

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-7.1 Page No.137\n');

D=2;                    //[in] Diameter of bar
W=500;                  //[lb] Weight
h=1;                    //[in] Height from which the
   weight falls
A=%pi*D^2/4;            //[in^2] Area of cross section
    of bar
L=10;                   //[in] Length of bar
E=30*10^6;              //[lb/in^2] Modulus of
   elasticity

S=(W/A)+(W/A)*(1+(2*h*E*A/(L*W)))^(0.5);  //[lb/in
   ^2] Stress in the bar

mprintf('\n Stress in the bar is %f lb/in^2.',S);

Delta=S*L/E;            //[in] Deflection
```

```

mprintf('\n Deflecton in the bar is %f in.',Delta);
```

Scilab code Exa 7.2 `Velocity and Impact`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-7.2 Page No.139\n');

W=2000;                  //[lb] Weight of automobile
L=36;                    //[in] Length of stop
D=2;                     //[in] Diameter of steel bar
V=5*5280*12/3600;        //[in/s] Velocity of automobile

A=%pi*D^2/4;             //[in^2] Area of cross section of bar
E=30*10^6;               //[lb/in^2] Modulus of elasticity

k=A*E/L;                 //[lb/in] Stiffness of the bar
g=386;                   //[in/s^2] Acceleration due to gravity

Delta=sqrt(2/k*W*(V^2/(2*g)+0));  //[in] Deflection

mprintf('\n The deflection in the bar is %f in.', Delta);

S=Delta*E/L;             //[in] Stress in the bar

//Note-In the book Delta=0.124 is used instead of Delta=0.123800

mprintf('\n The stress in the bar is %f lb/in^2.',S)
```

;

Scilab code Exa 7.3 Impact on Beam

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-7.3 Page No.141\n');

W=3000;                  //[lb] Weight of automobile
L=40*12;                 //[in] Length of the beam
I=64.2;                  //[in^4] Moment of inertia of
    the beam
Sy=48000;                //[lb/in^2] Yield strength of
    the beam
c=8/2;                   //[in] Distance from the
   outermost fiber to neutral axis
E=30*10^6;               //[lb/in^2] Modulus of
   elasticity
g=32.2;                  //[ft/s^2] Acceleration due
   to gravity

M=I*Sy/c;                //[lb*in] Moment at which
   beam will yield
F=4*M/L;                 //[lb] Force at which beam
   will yield

Delta=F*L^3/(48*E*I);    //[in] Deflection
KE=F*Delta/2;            //[lb*in] Kinetic energy

V=sqrt(2*g*KE/W);        //[in/s] Velocity
V=V/5280*3600;           //[miles/hr] Velocity

mprintf('\n At %f miles/hr velocity the beam will
   yield.',V);
```

Scilab code Exa 7.4 Designing for Impact

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-7.4 Page No.143\n');

D=3/4;                  //[in] Diameter of the bolt
At=0.334;               //[in^2] Area of thread
As=%pi*D^2/4;           //[in^2] Area of shank

//Note-In the book As=0.442 in^2 is used instead of As=0.4417865 in.

E=30*10^6;              //[lb/in^2] Modulus of elasticity
Lt=2;                   //[in] Length of the thread
Ls=6;                   //[in] Length of the shank
h=0.03;                 //[in] Height from which the weight falls
W=500;                  //[lb] Falling load

Kt=At*E/Lt;             //[lb/in] Stiffness of threaded portion
Ks=As*E/Ls;             //[lb/in] Stiffness of shank

K=Kt*Ks/(Kt+Ks);        //[lb/in] Overall stiffness

Delta=(W/K)+(W/K)*sqrt(1+2*h*K/W); //[in] Deflection

A=[Ls/E, Lt/E; 0.442, -0.334];
b=[Delta; 0];
S=A\b;

```

```
S=max(S);                //[lb/in^2] Maximum stress in
    the bolt

//Note-In the book Delta=0.0048 in is used instead
    of Delta=0.0047619 in.

mprintf('\n The maximum stress in this bolt is %f lb
    /in^2.',S);

Ln=8;                    //[in] Length when shank has
    same area as threads
Kn=At*E/Ln;              //[lb/in] Stiffness
Deltan=(W/Kn)+(W/Kn)*sqrt(1+2*h*Kn/W);   //[in]
    Deflection
S=Deltan*E/Ln;           //[ln/in^2] Stress

mprintf('\n If shank has the same area as threads
    then stress is %f lb/in^2 and deflection is %f in
    .',S,Deltan);
```

Chapter 8

Spring Design

Scilab code Exa 8.1 Design of Helical Compression Spring

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-8.1 Page No.160\n');

Dm=0.625;                //[in] Mean diameter of spring
F=35;                  //[lb] Load


K=1.25;               //[] Wahl factor for Dm/Dw=6.25 (
    figure 8.8)
Q=190000;             //[lb/in^2] Expected ultimate
    strength

LF=0.263;             //[] Loading factor

Dw=(K*8*F*Dm/(LF*%pi*Q))^(1/2.846); //[in] Wire
    diameter

mprintf('\n The wire diameter of spring is %f in.',
    Dw);
```

```
//Use U.S Steel 12-gage wire: Dw=0.105 in.
```

Scilab code Exa 8.2 Determination of number of coils

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-8.2 Page No.163\n');

Dw=0.105;             //[in] Wire diameter
Dm=0.620;             //[in] Mean diameter of spring
F=35;                 //[lb] Load
G=11.85*10^6;         //[lb/in^2] Shear modulus of
    elasticity
Delta=0.5;            //[in] Deflection

Na=Delta*G*Dw^4/(8*F*Dm^3); //[] Number of active
    coils

Nat=Na+2;             //[] Total number of coils

Lf=2;                 //[in] Free length of spring

P=(Lf-2*Dw)/Nat;      //[in] Pitch (Table 8.1)

mprintf('\n Pitch is %f in.',P);

k=G*Dw^4/(8*Dm^3*Na); //[lb/in] Spring rate

mprintf('\n Spring rate is %f lb/in.',k);

mprintf('\n The total number of coils necessary to
    meet design criteria are %f.',Nat);

```

```
//Note: The deviation of answer from the answer
    given in the book is due to round off error.(In
    the book values are rounded while in scilab
    actual values are taken)

//Note: The deviation of answer from the answer
    given in the book is due to round off error.(In
    the book values are rounded while in scilab
    actual values are taken)
```

Scilab code Exa 8.3 `Stability of Spring`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-8.3 Page No.165\n');
Lf=2;               //[in] Free length of spring
Dm=0.620;           //[in] Mean diameter of spring

R=Lf/Dm;            //[] Free lengtth to mean diameter
    ratio

mprintf('\n The ratio of the free length of spring
    to mean diameter of spring is %f.',R);
mprintf(' From Figure 8.9 for squared and ground
    ends, this is a stable spring.');
```

Scilab code Exa 8.4 `Deflection of Spring`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-8.4 Page No.165\n');
```

```

F=35;                  //[lb] Load
k=73.3;                //[lb/in] Spring rate

x=F/k;                 //[in] Deflection

mprintf('\n The deflection in the spring would be %f in.',x);
```

Scilab code Exa 8.5 `Flat Springs`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-8.5 Page No.166\n');

b=12;                       //[in] Width of plate
h=1;                        //[in] Thickness of plate
L=72;                       //[in] Length of plate
I=b*h^3/12;                 //[in^4] Moment of inertia

Delta=4;                    //[in] Deflection
E=10*10^6;                  //[lb/in^2] Modulus of elasticity

F=3*Delta*E*I/L^3;          //[lb] Force

mprintf('\n The force at this point is %f lb.',F);

k=F/Delta;                  //[lb/in] Stiffness

mprintf('\n stiffness is %f lb/in.',k);

//Note: The deviation of answer from the answer given in the book is due to round off error.(In
```

```
the book values are rounded while in scilab
actual values are taken)

//Note: The deviation of answer from the answer
given in the book is due to round off error.(In
the book values are rounded while in scilab
actual values are taken)
```

Scilab code Exa 8.6 Energy from Deflection

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-8.6 Page No.167\n');

F=322;                  //[lb] Force
Delta=4;                //[in] Deflection

U=F*Delta/2;            //[in*lb] Energy

mprintf('\n The energy from the 4-inch deflection
   was %f lb*in.',U);
```

Chapter 10

Pneumatic and Hydraulic Drives

Scilab code Exa 10.1 `Calculation of Hydraulic Cylinder Diameter and Standard Rod S`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-10.1 Page No.195\n');

P=100;                  //[lb/in^2] Hydraulic pressure
F=450;                  //[lb] Extension force
Fr=400;                 //[lb] Retraction force

A=F/P;                  //[in^2] Cross section area
D=sqrt(4*A/%pi);        //[in] Bore of cylinder

mprintf('\n The bore of cylinder is %f in.',D);

//Use 2.5in bore cylinder

Dm=2.5;                 //[in] Bore of cylinder
Dr=1;                   //[in] Diameter of rod
A2=%pi*Dm^2/4-%pi*Dr^2/4; //[in^2]
```

```
F2=P*A2;                    //[lb] Force

if F2>=Fr then
    mprintf('\n The diameter of rod is %f in.',Dr);
else
    mprintf('\n This would not meet requirement');
end

//This would meet requirement

Ab=%pi*Dm^2/4;              //[in^2] Cross section area
//Note-In the book V=180.7 is used instead of V=180.64158
d=20;                       //[in] stroke
V=Ab*d+A2*d;                //[in^3] Volume per cycle
t=2;                        //[s] Cycle time
FR=V/t;                     //[in^3/s] Flowrate

FR=FR*7.48*60/1728;         //[gal/min] Flowrate

mprintf('\n Flow rate required is %f gal/min.',FR);
```

Scilab code Exa 10.2 Pneumatic Pop Rivet Gun

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-10.2 Page No.198\n');

Pa=100;                     //[lb/in^2] Air pressure
Da=4;                       //[in] Diameter
Aa=%pi*Da^2/4;              //[in^2] Cross section area

F1=Pa*Aa;                   //[lb]
Do=1;                       //[in]
```

```
Ao=%pi*Do^2/4;          //[in]
Po=F1/Ao;               //[lb/in^2]

mprintf('\n The oil pressure is %f lb/in^2.',Po);

D2o=3;                  //[in]
A2o=%pi*D2o^2/4;        //[in^2]
F2=Po*A2o;

mprintf('\n Force F on piston rod is %f lb.',F2);

D=1;                    //[in]
d=4;                    //[in]
A=%pi*D^2/4;            //[in^2]

V=A*d;                  //[in^3]

mprintf('\n The volume in 1-inch cylinder for the 4-
   inch travel is %f in^3.',V);

A3=%pi*3^2/4;           //[in^2]
l3=V/A3;                //[in]

mprintf('\n Travel for 3-inch cylinder is %f in.',l3
   );
```

Chapter 11

Gear Design

Scilab code Exa 11.1 Double Reduction Spur Gear Set

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-11.1 Page No.217\n');

N2=60;
N1=20;
N3=20;
N4=60;

Vr=(N2/N1)*(N4/N3);

//Output speed
n1=3600;
n4=n1/Vr;

mprintf('\n The output speed is %f rpm.',n4);

//Output torque
T1=200;
T4=T1*Vr;
```

```
mprintf('\n The output torque is %f lb*in.',T4);

//Input horsepower
hpi=T1*n1/63000;

mprintf('\n The input horsepower is %f hp.',hpi);

//Output horsepower
hpo=T4*n4/63000;

mprintf('\n The output horsepower is %f hp.',hpo);
```

Scilab code Exa 11.2 Double Reduction Spur Gear Set with Idler

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-11.2 Page No.219\n');

Na=20;
Nb=65;
Nc=20;
Nd=22;
Ne=60;

//train value
Vr=(Nb/Na)*(Nd/Nc)*(Ne/Nd);

mprintf('\n Train value = %f ',Vr);

//Output speed
na=3000;
ne=na/Vr;

```

```
mprintf('\n \Output speed = %f rpm.',ne);

//Output torque
Ta=10;
Te=Ta*Vr;

mprintf('\n Output torque = %f lb*in.',Te);

//Direction

mprintf('\n Direction\n   If Gear A is clockwise,\n   Gear B is counterclockwise.\n   Gear C is counterclockwise.\n   Gear D is clockwise. \n   Gear E is counterclockwise.');

//Output power
P=Te*ne;
P=P*%pi/60;
 mprintf('\n Output power = %f W.',P);
```

Scilab code Exa 11.3 Calculation of Pitch Diameter Circular Pitch and Shaft Centre

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-11.3 Page No.231\n');

Np=16;
Ng=32;
Pd=8;

//Pitch diameter
Dp=Np/Pd;

mprintf('\n Pinion pitch diameter is %f in.',Dp);
```

```
Dg=Ng/Pd;

mprintf('\n Gear pitch diameter is %f in.',Dg);

//Circular pitch
Pc=%pi*Dp/Np;

mprintf('\n Circular pitch is %f in.',Pc);

//Centerline distance
CC=(Dp+Dg)/2;

mprintf('\n Centerline distance is %f in.',CC);
```

Scilab code Exa 11.4 Bevel Gear

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-11.4 Page No.236\n');

//Torque in input shaft
hp=1.5;
n=3450;
T=63000*hp/n;

mprintf('\n Torque in input shaft is %f lb*in.',T);

//Note-In the book T=27.4 in-lb is used instead of T
    =27.391304

//Output torque
Ng=24;
Np=10;
```

```
Tout=(Ng/Np)*T;

mprintf('\n Output torque is %f lb*in.',Tout);

//Output speed
nout=(Np/Ng)*n;

mprintf('\n Output speed is %f rpm.',nout);
```

Scilab code Exa 11.5 Calculation of Gear Train Value Input and Output Torque and S

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-11.5 Page No.241\n');

//Gear train value
Na=12;
Nb=36;
Nc=16;
Nd=64;
Vr=(Nb/Na)*(Nd/Nc);

mprintf('\n Gear train value is %f ',Vr);

//Motor torque
hp=1.5;
n=1750;
T=63000*hp/n;

mprintf('\n Motor torque is %f in-lb.',T);

//Output torque
Tout=T*Vr;

```

```
mprintf('\n Output torque is %f in-lb.',Tout);

//Output speed
nout=n/Vr;

mprintf('\n Output speed is %f rpm.',nout);

//Directions
mprintf('\n Directions\n  Gear A is clockwise.\n
   Gear B is counterclockwise.\n  Gear C is
   counterclockwise.\n  Gear D is clockwise.');

//Output power
hp=T*n/63000;

mprintf('\n Output power is %f hp.',hp);
```

Scilab code Exa 11.6 Precision Spur Gears

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-11.6 Page No.243\n');

//Velocity ratio
N2=2400;
N1=20;
Vr=N2/N1;

mprintf('\n Velocity ratio = %f ',Vr);

mprintf('\n Possible Solution: \n Three sets of
   gears \n   -20 tooth and 80 tooth\n   -20 tooth
   and 100 tooth\n   -20 tooth and 120 tooth.');
```

Chapter 12

Spur Gear Design and Selection

Scilab code Exa 12.1 Forces on Spur Gear Teeth

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-12.1 Page No.254\n');

P=5;
n=1725;
T=63000*P/n;

//Pitch circle diameter
Np=20;
Pd=8;
Dp=Np/Pd;

mprintf('\n Pitch circle diameter = %f in.',Dp);

//Transmitted force
Ft=2*T/Dp;

mprintf('\n Transmitted force = %f lb.',Ft);

```

```
//Separating force
theta=20*%pi/180;
Fn=Ft*tan(theta);

mprintf('\n Separating force = %f lb.',Fn);

//Maximum force
Fr=Ft/cos(theta);

mprintf('\n Maximum force = %f lb.',Fr);
```

Scilab code Exa 12.2 Surface Speed

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-12.2 Page No.256\n');

//Surface speed
Dp=2.5;
n=1725;
Vm=%pi*Dp*n/12;

mprintf('\n Surface speed = %f ft/min.',Vm);
```

Scilab code Exa 12.3 Strength of Gear Teeth

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-12.3 Page No.258\n');
//Pinion
Su=95*10^3;
```

```
Sn=0.5*Su;
Y=0.320;
b=1;
Pd=8;

Fsp=Sn*b*Y/Pd;

mprintf('\n Force allowable for pinion = %f lb.',Fsp
   );

//Gear
Sn=0.5*88*10^3;
Y=0.421;
Fsg=Sn*b*Y/Pd;

mprintf('\n Force allowable for gear = %f lb.',Fsg);
```

Scilab code Exa 12.4 Dynamic Load

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-12.4 Page No.262\n');

//Dynamic load
Vm=1129;
Ft=146;
Fd=(600+Vm)*Ft/600;

mprintf('\n Dynamic load = %f lb.',Fd);

Fs=1900;
Nsf=2;

//Comparing to the allowable stress
```

```
if (Fs/Nsf)>Fd then
    mprintf('\n This is an acceptable design.');
end
```

Scilab code Exa 12.5 Calculation of Factor of Safety Used in Catalog

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-12.5 Page No.263\n');

Su=55*10^3;
Sn=0.5*Su;

Np=24;
Pd=12;
Dp=Np/Pd;

mprintf('\n Pitch circle diameter = %f in.',Dp);

n=1800;
Vm=%pi*Dp*n/12;

mprintf('\n Surface speed = %f ft/min.',Vm);

b=3/4;
Y=0.302;
Fs=Sn*b*Y/Pd;

mprintf('\n Allowable stress = %f lb.',Fs);

Fd=Fs;
Ft=600*Fd/(600+Vm);

```

```
mprintf('\n Force transmitted = %f lb.',Ft);

T=Ft*Dp/2;

P=T*n/63000;

mprintf('\n Power transmitted = %f hp.',P);

//Compared to catalog
hp_catalog=4.14;

Nsf=P/hp_catalog;

mprintf('\n Safety factor = %f .',Nsf);
```

Scilab code Exa 12.6 Spur Gear Design

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-12.6 Page No.266\n');

//Miscellaneous properties
Np=48;
Pd=12;
Dp=Np/Pd;
Vr=3;
Ng=Np*Vr;

//Surface speed
n=900;
Vm=%pi*Dp*n/12;

mprintf('\n Surface speed = %f ft/min.',Vm);

```

```
//Force on teeth
hp=2;
Ft=33000*hp/Vm;

mprintf('\n Force on teeth = %f lb.',Ft);

//Dynamic force
Fd=(600+Vm)*Ft/600;

mprintf('\n Dynamic force = %f lb.',Fd);

//Width
Su=30*10^3;
Sn=0.4*Su;
Y=0.344;
Nsf=2;
b=Fd*Nsf*Pd/(Sn*Y);
b=round(b);

mprintf('\n Width = %f in.',b);

if (8/Pd)<b&b<(12.5/Pd) then
    mprintf('\n This is an acceptable design.');
end
```

Scilab code Exa 12.7 Buckingham Method of Gear Design

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-12.7 Page No.270\n');

Su=95*10^3;
Sn=0.5*Su;
Np=24;
```

```
Pd=16;
Dp=Np/Pd;

//Torque
n=3450;
P=3;
T=P*63000/n;

mprintf('\n Torque = %f in-lb.',T);

//Force transmitted
Ft=2*T/Dp;

mprintf('\n Force transmitted = %f lb.',Ft);

//Surface speed
Vm=%pi*Dp*n/12;

mprintf('\n Surface speed = %f ft/min.',Vm);

//Force allowable
Y=0.337;
b=1;
Fs=Sn*b*Y/Pd;

mprintf('\n Force allowable = %f lb.',Fs);

//Dynamic load using Buckingham's equation
C=830;
Fd=Ft+0.05*Vm*(b*C+Ft)/(0.05*Vm+(b*C+Ft)^0.5);

Nsf=1.4;
if (Fs/Nsf)>Fd then
    mprintf('\n This is a suitable design');
end
```

Scilab code Exa 12.8 Wear of Gears

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-12.8 Page No.272\n');

Ng=42;
Np=24;
Q=2*Ng/(Ng+Np);

Kg=270;
Dp=1.5;
b=1;

Fw=Dp*b*Q*Kg;
Fd=699;
Nsf=1.2;

if (Fw/Nsf)<Fd then
    mprintf('\n (Fw/Nsf)<Fd So this would not be
        suitable design');
end

//If the surfaces each had a BHN = 450

Kg=470;
Fw=Dp*b*Q*Kg;

if(Fw/Nsf)>Fd then
    mprintf('\n\n If the surfaces each had a BHN =
        450');
    mprintf('\n (Fw/Nsf)>Fd So this would be
        suitable design.');
```

```
end
```

Chapter 13

Helical Bevel and Worm Gears

Scilab code Exa 13.1 `Helical Gears`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-13.1 Page No.280\n');

//Pitch-line velocity
Nt=24;
Pd=12;
Dp=Nt/Pd;
n=1750;
Vm=%pi*Dp*n/12;

mprintf('\n Pitch-line velocity = %f ft/min.',Vm);

//Transmitted force
hp=5;
Ft=33000*hp/Vm;

mprintf('\n Transmitted force = %f lb.',Ft);

//Axial force
```

```
psi=15*%pi/180;
Fa=Ft*tan(psi);

mprintf('\n Axial force = %f lb.',Fa);

//Separating force
theta=20*%pi/180;
psit=atan(tan(theta)/cos(psi));
Fn=Ft*tan(psit);

mprintf('\n Separating force = %f lb.',Fn);
```

Scilab code Exa 13.2 Helical Gear Stresses

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-13.2 Page No.282\n');

//Normal plane pitch
Pd=16;
psi=45*%pi/180;
Pdn=Pd/cos(psi);

mprintf('\n Normal plane pitch = %f in.',Pdn);

N=24;
S=30000;
b=0.5;
Ne=N/cos(psi)^3;
Y=0.427;
Fs=S*b*Y/Pdn;

mprintf('\n Allowable force = %f lb.',Fs);

```

```
Dp=24/16;
n=600;
Vm=%pi*Dp*n/12;

mprintf('\n Surface speed = %f ft/min.',Vm);

Ft=Fs/((600+Vm)/600);

mprintf('\n Force transmitted = %f lb.',Ft);

P=Ft*Vm/33000;

mprintf('\n Power rating = %f hp.',P);

//Note-There is an error in the answer given in
   textbook
```

Scilab code Exa 13.3 Bevel Gear Forces

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
   \n EXAMPLE-13.3 Page No.286\n');

Np=24;
Ng=36;
Pd=8;
Yp=33.7*%pi/180;
Yg=56.3*%pi/180;
theta=14.5*%pi/180;

//Pitch diameter
Dp=Np/Pd;

mprintf('\n Pitch diameter = %f in.',Dp);
```

```

//Transmitted force
n=2200;
P=8;
T=63000*P/n;

Ft=2*T/Dp;

mprintf('\n Transmitted force = %f lb.',Ft);

//Separating force - Pinion
Fnp=Ft*tan(theta)*cos(Yp);

mprintf('\n Separating force-Pinion = %f lb.',Fnp);

//Separating force-Gear
Fng=Ft*tan(theta)*cos(Yg);

mprintf('\n Separating force = %f lb.',Fng);

//Axial force-Pinion
Fap=Ft*tan(theta)*sin(Yp);

mprintf('\n Axial force-Pinion= %f lb.',Fap);

//Axial force-Gear
Fag=Ft*tan(theta)*sin(Yg);

mprintf('\n Axial force-Gear = %f lb.',Fag);
```

Scilab code Exa 13.4 `Worm Gear Forces`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
```

```
    \n EXAMPLE−13.4 Page No.288\n');

//Pitch diameter
Ng=60;
Pd=6;
Dp=Ng/Pd;

mprintf('\n Pitch diameter = %f in.',Dp);

//Circular pitch
Pc=%pi*Dp/Ng;

mprintf('\n Circular pitch = %f in.',Pc);

L=Pc;

//Lead angle
D=2;
LA=atan(L/(%pi*D));
LA=LA*180/%pi;

mprintf('\n Lead angle = %f deg.',LA);

//Centerline distance
CC=(D+Dp)/2;

mprintf('\n Centerline distance = %f in.',CC);

//Velocity ratio
Ntgear=60;
Nstarts_worm=1;
Vr=Ntgear/Nstarts_worm;

mprintf('\n Velocity ratio = %f',Vr);

//Output speed
nin=1750;
nout=nin/Vr;
```

```
mprintf('\n Output speed = %f rpm.',nout);
```

Scilab code Exa 13.5 Calculation of Normal Circular Pitch Dynamic Load Force Allow

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-13.5 Page No.292\n');

//Normal circular pitch
Pc=0.524;
LA=4.77*%pi/180;
Pcn=Pc*cos(LA);

mprintf('\n Normal circular pitch = %f in.',Pcn);

//Force transmitted
hp=5;
n=29.2;
T=63000*hp/n;
Dp=10;
Ft=2*T/Dp;

mprintf('\n Force transmitted = %f lb.',Ft);

Vm=%pi*Dp*n/12;

//Dynamic load
Fd=(1200+Vm)*Ft/1200;

mprintf('\n Dynamic load = %f lb.',Fd);

//Force allowable
Su=95*10^3;
```

```
Y=0.392;
b=1;
Sn=0.5*Su;
Fs=Sn*Y*b*Pcn/%pi;

mprintf('\n Force allowable = %f lb.',Fs);

//Safty factor
Nsf=Fs/Fd;

mprintf('\n Safty factor = %f .',Nsf);

//Note-There is an error in the answer given in
    textbook
```

Scilab code Exa 13.6 Efficiency of Worm Gear Drive

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-13.6 Page No.294\n');

//Efficiency
LA=4.77*%pi/180;
f=0.03;
e=tan(LA)*(1-f*tan(LA))/(f+tan(LA));

mprintf('\n Efficiency = %f .',e);

//Torque input
hp=5;
n=1750;
T=63000*hp/n;

mprintf('\n Torque input = %f in-lb.',T);
```

```
Vr=60;
Tout=0.73*Vr*T;

mprintf('\n Output torque = %f in-lb.',Tout);
```

Scilab code Exa 13.7 Heat Generated

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-13.7 Page No.296\n');

hpin=5
e=0.73;
Q=(1-e)*hpin*2544;

mprintf('\n Heat generated by system = %f Btu/hr.',Q
    );

//Note-There is an error in the answer given in
    textbook
```

Chapter 14

Belt and Chain Drives

Scilab code Exa 14.1 Calculation of Front Force Net Driving Force Force on Shaft e

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-14.1 Page No.306\n');

//Torque on small pulley
hp=2;
n=2450;
T=63000*hp/n;

mprintf('\n Torque on small pulley = %f in-lb.',T);
r=6/2;
Fd=T/r;

//Front force
Fb=10;
Ff=Fd+Fb;

mprintf('\n Front force = %f lb.',Ff);

//Force pulling the shafts
```

```
Ft=Ff+Fb

mprintf('\n Force pulling the shafts = %f lb.',Ft);

//Surface speed
D=2*r;
Vm=%pi*D*n/12;

mprintf('\n Surface speed = %f ft/min.',Vm);

//Ratio
D2=10;
Mw=D2/D;

mprintf('\n Ratio = %f .',Mw);

//Output speed
no=n/Mw;

mprintf('\n Output speed = %f rpm.',no);

//Note-There is an error in the answer given in textbook
```

Scilab code Exa 14.2 Calculation of Angle of Contact Length of Belt Ratio and Surf

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-14.2 Page No.310\n');

//Length of belt
C=19;
D1=4;
D2=6;
```

```
L1=2*C+1.57*(D1+D2)+(D2-D1)^2/(4*C);

//Assuming a 54-inch belt is available
L=54;

mprintf('\n Length of belt = %f in.',L);

//Centerline distance
B=4*L-6.28*(D2+D1);

C=(B+sqrt(B^2-32*(D2-D1)^2))/16;

mprintf('\n Centerline distance = %f in.',C);

//Ratio
Mw=D2/D1;

mprintf('\n Ratio = %f.',Mw);

//Surface speed
n=1800;
Vm=%pi*D1*n/12;

mprintf('\n Surface speed = %f ft/min.',Vm);

//Angle of contact

theta=180-2*(180/%pi)*asin((D2-D1)/(2*C));

mprintf('\n Angle of contact = %f deg.',theta);
```

Scilab code Exa 14.3 V Belts

```
clc;
```

```
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-14.3 Page No.315\n');

//Power rating of belt
P1=27+2.98;
SF=1.5;
P=P1/SF;
P=round(P);

mprintf('\n Power rating = %f hp.',P);

//Length of belt
C=20;
D1=8;
D2=16;
L1=2*C+1.57*(D1+D2)+(D2-D1)^2/(4*C);

//Use an 80-inch belt
L=80;

mprintf('\n Length of belt = %f in.',L);
```

Scilab code Exa 14.4 Chain Drive

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-14.4 Page No.321\n');

P=5.31;

mprintf('\n Horsepower rating = %f hp.',P);

Nti=12;
```

```
N1=1800;
N2=900;

//Output sprocket
Nto=(N1/N2)*Nti;

mprintf('\n Number of tooth on output sprocket = %f.
    ',Nto);

//Surface speed
Pc=0.5;
D1=Pc*Nti/%pi;
n=1800;
Vm=%pi*D1*n/12;

mprintf('\n Surface speed = %f ft/min.',Vm);

mprintf('\n Type of lubrication - Bath or disc
    lubrication');

//Length of chain
C=10;
D2=Pc*Nto/%pi;

L1=2*C+1.57*(D1+D2)+(D2-D1)^2/(4*C);

//Use 29 or 30 inch chain

L=30;

mprintf('\n Length of chain = %f in.', L);

hp=5.31;

T=63000*hp/n;

F=2*T/D1;

```

```
mprintf('\n Force in chain = %f lb.',F);

//Comparism with ultimate strength 3700 lb - not a
   valid comparison because of speed etc.
```

Chapter 15

Keys and Couplings

Scilab code Exa 15.1 `Design of Keys`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-15.1 Page No.332\n');

//Torque
P=5;
n=1750;
T=63000*P/n;

mprintf('\n Torque = %f in-lb.',T);

//Length of key for shear
Su=61000;
Ss=0.5*Su;
b=0.125;
D=0.5;
Ls1=2*T/(Ss*b*D);
SF=2.5;

Ls=SF*Ls1;
```

```

mprintf('\n Length of key for shear = %f in.',Ls);

//Length of key for compression
Sc=51000;
t=0.125;
Lc1=4*T/(Sc*t*D);

Lc=SF*Lc1;

mprintf('\n Length of key for compression = %f in.',
   Lc);
```

Scilab code Exa 15.2 Splines

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-15.2 Page No.335\n');

//Torque capacity
Ss=30500;
D=1;
L=2;
T1=Ss*%pi*D^2*L/16;

SF=2;

T=T1/SF;

mprintf('\n Torque capacity 1 = %f in-lb.',T);
n=6;
d=0.81;
A=(D-d)*L*n/2;

```

```
S=1000;
rm=(1+0.810)/4;

T2=S*A*rm;

mprintf('\n Torque capacity 2 = %f in-lb.',T2);
```

Chapter 16

Clutches and Brakes

Scilab code Exa 16.1 Calculation of Torque and Power

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-16.1 Page No.358\n');

//Torque capacity
f=0.3;
N=120;
ro=12;
ri=9;
Tf=f*N*(ro+ri)/2;

mprintf('\n Torque capacity = %f in-lb.',Tf);
n=2000;
//Power

Pf=Tf*n/63000;

mprintf('\n Power = %f hp.',Pf);
```

Scilab code Exa 16.2 Determination of Breaking Torque

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-16.2 Page No.359\n');

//Normal force
W=100;
L=20;
a=4;
N=(W*L)/a;

mprintf('\n Normal force = %f lb.',N);

//Torque friction
f=0.4;
D=12;
Tf=f*N*D/2;

mprintf('\n Torque friction = %f in-lb.',Tf);
```

Scilab code Exa 16.3 Torque Transmitting Capacity

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-16.3 Page No.360\n');

//For alpha=20 deg.
alpha=20*(%pi/180);
f=0.35;
```

```
rm=12/2;
Fa=75;
Tf=(f*rm*Fa)/(sin(alpha)+f*cos(alpha));

mprintf('\n Torque capacity (alpha=20 deg.) = %f in-
   lb.',Tf);

//For alpha=10 deg.
alpha=10*(%pi/180);
Tf=(f*rm*Fa)/(sin(alpha)+f*cos(alpha));

mprintf('\n Torque capacity (alpha=10 deg.) = %f in-
   lb.',Tf);
```

Scilab code Exa 16.4 Calculation of Stopping Force Torque per Brake Normal Brake F

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-16.4 Page No.361\n');

//Stopping rate
V=60*5280/3600;
Va=0.5*V;
D=400;
t=D/Va;
a=V/t;

mprintf('\n Stopping rate = %f ft/sec^2.',a);

//Stopping force
W=40000;
g=32.2;
F=W*a/g;

```

```
//Torque
r=36/2;
T=F*r;

mprintf('\n Torque = %f in-lb.',T);

//For each wheel
T=T/10;

//Braking normal force
rm=10;
f=0.4;
N=T/(f*rm);

mprintf('\n Braking normal force = %f lb.',N);

//Note-There is an error in the answer given in
    textbook
```

Scilab code Exa 16.5 Rotational Inertia and Brake Power

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-16.5 Page No.365\n');
W=3500;
V=73;
g=32.2;
V=50*5280/3600;
V=round(V);

//Kinetic energy to be absorbed
KE=W*V^2/(2*g);

mprintf('\n Kinetic energy to be absorbed = %f ft-lb
```

```
.',KE);

//Temperature rise
Uf=KE;
Wb=40;
c=93;
deltaT=Uf/(Wb*c);

mprintf('\n Temperature rise = %f deg.',deltaT);

//Stopping time
a=20;
t=V/a;

mprintf('\n Stopping time = %f sec.',t);

//Frictional power
t=round(t*10)/10;
fhp=Uf/(550*t);

mprintf('\n Frictional power = %f hp.',fhp)
```

Chapter 17

Shaft Design

Scilab code Exa 17.1 Design Stresses in Shaft

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-17.1 Page No.379\n');

hp=5;
n=1750;
T=63000*hp/n;

//Torsional stress in the shaft
D=0.75;
Z1=%pi*D^3/16;

Ss=T/Z1;

mprintf('\n Torsional stress in the shaft = %f lb/in
   ^2.',Ss);

//Load at the gear pitch circle
Nt=40;
Pd=10;
```

```
Dp=Nt/Pd;

Ft=2*T/Dp;

mprintf('\n Load at gear pitch circle = %f lb.',Ft);

//Resultant force on the shaft
theta=20*%pi/180;
Fr=Ft/cos(theta);

mprintf('\n Resultant force = %f lb.',Fr);

//Maximum moment
L=15;
Mm=Fr*L/4;

mprintf('\n Maximum moment = %f in-lb.',Mm);

//Stress
D2=0.75;
Z2=%pi*D2^3/32;
Z2=round(Z2*1000)*10^-3;

S=Mm/Z2;

mprintf('\n Stress = %f lb/in^2.',S);

//Note-There is an error in the answer given in textbook
```

Scilab code Exa 17.2 Combined Stresses in Shaft

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
```

```
\n EXAMPLE−17.2 Page No.383\n');

//Combined stress using the maximum shear stress theorem

Ss=2170;
S=8780;
Sr=sqrt(Ss^2+(S/2)^2);

mprintf('\n Combined stress = %f lb/in^2.',Sr);
```

Scilab code Exa 17.3 `Combined Stress Using Maximum Normal Stress Theory`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE−17.3 Page No.383\n');

//Combined stress using the maximum normal stress theory
Ss=2170;
S=8780;
Sr=S/2+sqrt(Ss^2+(S/2)^2);

mprintf('\n Combined stress = %f lb/in^2.',Sr);

//Note−There is an error in the answer given in textbook
```

Scilab code Exa 17.4 `Comparison of Stresses to Allowable Values and Endurance Limi`

```
clc;
clear;
```

```
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-17.4 Page No.385\n');

//Modifying factors for Sn
Su=88000;
Csize=0.85;
Csurface=0.88;
Ctype=1;

Sn=Csize*Csurface*Ctype*(0.5*Su);
Kt=2.3;
S=9300;

N=Sn/(Kt*S);

if N>2 then
    mprintf('\n It would be an acceptable design.');
else
    mprintf('\n N<2,So this is not a suitable design for long term use.');
end
```

Scilab code Exa 17.5 Critical Speed

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E. \n EXAMPLE-17.5 Page No.388\n');

//Deflection
D=0.75;
E=30*10^6;
L=15;
F=96;
I=%pi*D^4/64;
```

```
delta=F*L^4/(48*E*I);
delta=floor(100*delta)*10^-2;
Nc=188/sqrt(delta);

mprintf('\n Critical speed = %f rpm.',Nc);
```

Chapter 18

Power Screws and Ball Screws

Scilab code Exa 18.1 Torque and Power in Power Screw

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-18.1 Page No.399\n');

//Torque
Dp=(1.5+1.208)/2;
F=5800;
L=1/3;
f=0.15;

Tup=(F*Dp/4)*(L+%pi*f*Dp)/(%pi*Dp-f*L);

mprintf('\n Torque up = %f in-lb.',Tup);
```

Scilab code Exa 18.2 Efficiency of a Power Screw

```
clc;
```

```
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-18.2 Page No.400\n');

//Lead angle
L=1/3;
Dp=1.354;
LA=atan(L/(%pi*Dp));

mprintf('\n Lead angle = %f deg.',LA*180/%pi);

//Efficiency
f=0.15;
e=tan(LA)*(1-f*tan(LA))/(tan(LA)+f);

mprintf('\n Efficiency = %f',e*100);

//Power
n=175;
T=454;
P=T*n/63000;
Pt=P*2;

mprintf('\n Power = %f hp per lead screw.',P);

if f>tan(LA) then
    mprintf('\n It is self-locking');
end
```

Scilab code Exa 18.3 Acme Threads

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-18.3 Page No.402\n');
```

```
L=1/4;

Dp=1.375;
LA=atan(L/(%pi*Dp));

mprintf('\n Lead angle = %f deg.',LA*180/%pi);

//Torque
phi=14.5*%pi/180;
f=0.15;
F=5800;
Tup=(F*Dp/4)*(cos(phi)*tan(LA)+f)/(cos(phi)-f*tan(LA
    ));

mprintf('\n Torque = %f in-lb.',Tup);

//Power
n=175*4/3;
P=Tup*n/63000;

mprintf('\n Power = %f hp per lead screw.',P)
```

Scilab code Exa 18.4 `Torque and Ball Screws`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-18.4 Page No.405\n');

//Torque
L=0.5;
F=5800/2;
T=0.177*F*L;

mprintf('\n Torque = %f in-lb.',T);
```

```

//Power
n=175*2/3;
P=T*n/63000;

mprintf('\n Power = %f hp.',P);
```

Chapter 19

Plain Surface Bearings

Scilab code Exa 19.1 `Shaft Considerations`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-19.1 Page No.417\n');

//Length
F=20;
n=500;
PV=3000;
L1=%pi*F*n/(12*PV);

//Use 7/8-inch or longer bearing

L=7/8;

mprintf('\n Length of bearing = %f in.',L);

//Maximum pressure
A=(3/4)*(7/8);
P=F/A;

```

```
mprintf('\n Maximum pressure = %f lb/in^2.',P);

//Maximum velocity
D=3/4;
V=%pi*D*n/12;

mprintf('\n Maximum velocity = %f ft/min.',V);
```

Scilab code Exa 19.2 Wear

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-19.2 Page No.421\n');

//Life in hours of operation
t=0.01;
K=12*10^-10;
P=30.5;
V=98;
T=t/(K*P*V);

mprintf('\n Life = %f hours.',T);

//Note-There is an error in the answer given in
    textbook
```

Chapter 20

Ball and Roller Bearings

Scilab code Exa 20.1 Life Expectancy of Ball Bearing

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-20.1 Page No.431\n');

//L10 design life
Cd=5050;
Pd=2400;
k=3;
Ld1=(Cd/Pd)^k*10^6;
Ld=Ld1/(1750*60);

mprintf('\n L10 design life = %f hr.',Ld);
```

Scilab code Exa 20.2 Selection of Bearing to Meet Given Criteria

```
clc;
clear;
```

```
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-20.2 Page No.432\n');

//Dynamic load capacity
T=200;
n=1750;
L=T*n*60/10^6;
Pd=2400;
Ld=21;
Lc=1;
k=1/3;

Cd=Pd*(Ld/Lc)^k

mprintf('\n Dynamic load capacity required = %f lb.'
    ,Cd);

mprintf('\n Bearing 6211 meets this criterion.');
```

Scilab code Exa 20.3 Selection of Bearing to Meet Given Criteria

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell, P.E.
    \n EXAMPLE-20.3 Page No.434\n');

R=1200;
Ft=500;
n=1500;
L10=5000;

//Assume thrust factor=1.6

Y=1.6;

```

```
Pd=0.56*R+Y*Ft;

Ld=n*L10*60/10^6;
Lc=1;
k=3;
Cd=Pd*(Ld/Lc)^(1/k);

//For bearing number 6215

Cd1=11400;
Cs1=9700;

//Verify the assumption for Y
Ft_Cs1=Ft/Cs1;

Y=(0.056-Ft_Cs1)*(1.99-1.71)/(0.056-0.028)+1.71;

Pd=0.56*R+Y*Ft;

Cd=Pd*(Ld/Lc)^(1/k);

if Cd>Cd1 then
    mprintf('\n Since Cd of bearing < Cd required, So bearing number 6215 is not acceptable.');
end

//For bearing number 6216
Cd2=12600;
Cs2=10500;

Ft_Cs2=Ft/Cs2;
Y=(0.056-Ft_Cs2)*(1.99-1.71)/(0.056-0.028)+1.71;

Pd=0.56*R+Y*Ft;

Cd=Pd*(Ld/Lc)^(1/k);

if Cd<Cd2 then
```

```
    mprintf('\n Since Cd of bearing > Cd required , So bearing number 6215 meets the design criteria .');
end
```

Scilab code Exa 20.4 `Life of 6200 Series Bearing`

```
clc;
clear;
mprintf('MACHINE DESIGN \n Timothy H. Wentzell , P.E. \n EXAMPLE-20.4 Page No.436\n');

//Thrust factor
Ft=300;
Cs=2320;
Ft_Cs=Ft/Cs;

Y=(0.17-Ft_Cs)*(1.45-1.31)/(0.17-0.11)+1.31;

mprintf('\n Thrust factor = %f ',Y);

V=1.2;
X=0.56;
R=1000;

P=V*X*R+Y*Ft;

Cd=3350;
Pd=1095;
k=3;

Ld=(Cd/Pd)^k*10^6;

mprintf('\n Life = %f revolutions.',Ld);
```