ID Number: ______________________________

George W. Woodruff School of Mechanical Engineering
Georgia Institute of Technology

DESIGN QUALIFIER

Spring 2006

WRITTEN EXAMINATION

We are interested in learning what you know and your ability to reason in the formulation and solution of design problems.

If you find any question or part of this exam confusing, please state your assumptions and rephrase the question and proceed.

Please read the entire exam first.

Questions 1 and 2 carry equal points. Both have multiple parts.

Allocate your time carefully so that you cover all three parts that you are being examined on in these two questions, namely, Methods, Realizability and Analysis.

A document containing some formulae is available for you to use in answering Question 2

ORAL EXAMINATION

Please arrive half an hour before the scheduled time for the oral exam. During this period we will give you a question to think about. The scope of the oral exam is as follows:

* provide an opportunity for you to state how design fits into your research activities;
* probe your understanding of the question that we posed to you in the preceding half hour.

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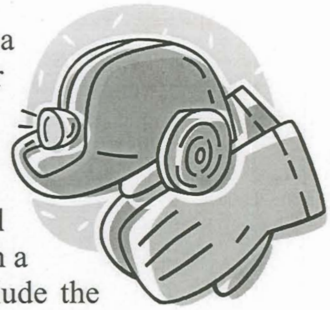
QUESTION 1 – METHOD & REALIZABILITY

THE PROBLEM

Due to the recent mining tragedies, there has been an increased interest in the safety of mine workers. Corporations in the mining industry are searching for ways to eliminate mining accidents altogether; however, because we can never eliminate mining disasters completely, each mining company must develop the tools necessary to protect and rescue miners in the event of a cave-in or explosion. Your task is to design mine safety and rescue equipment. The equipment that you design will have two purposes—to protect miners in hazardous situations and to guide rescue teams to trapped or wounded miners.

MINER'S SAFETY EQUIPMENT

Miner's survival vest



Each time a miner descends into the earth, he must be prepared to face a mining disaster. Since mining accidents are unpredictable, miners must wear their safety gear on the job at all times. To make it easier for miners to perform their mining tasks while carrying personal safety equipment, a particular mining company has commissioned you to design a "Miner's Survival Vest." This safety tool will be worn like a vest and will contain all the necessary tools to allow a miner to survive in the earth for several hours in a disaster situation. In designing your miner's survival vest, be sure to include the means for a miner to survive in the presence of contaminated air, and include tools for basic first aid in the mine. Be sure that the miner's survival vest does not significantly interfere with the miner as he/she works.

Task

Assume that you are in charge of the design team responsible for developing a survival vest for miners. Speculate about the events faced by a miner once there is a disaster and include the following considerations in your design.

- What is the function structure that provides the most flexibility for designing derivative products used in hazardous situations?
- What are the components of your miner's survival vest?
- How can the miner's survival vest make use of the natural resources in the mine?
- How can the miner's survival vest guide rescue workers to trapped / wounded miners?
- How can the miner's survival vest alert rescue teams as to the medical condition of the miner?

Deliverables

Method

1. *Clarify the Task:* State the overall function of your system in solution neutral terms. What are the most important drivers/design criteria? Define a design requirements list.

2. *Conceptual Design:* State and implement the steps (including functional diagrams/decomposition) for transforming the overall function that you have identified into at least three alternative design solutions. Ensure that you have identified the important sub functions. Sketch and describe the workings of these alternatives.

3. *Selection:* Suggest a structured approach to select one of the alternatives for further development.

Realizability

4. *Embodiment:* Further develop the alternative that you have selected.

5. *Costing:* How would you estimate the cost of your design? You may critically evaluate the design in terms of manufacturability, initial cost, maintenance cost, reliability, manipulation performance, and other criteria that you feel are important to consider in this phase of design.

6. *Pricing:* Based on the preceding analysis, how would you estimate the market size for such a system and set the price for selling such a system? Be brief.

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QUESTION 2A – COMPONENTS DESIGN ANALYSIS

2A. Short answer. Please write a complete descriptive answer in the space provided.

a. When the torsion spring is loaded to close the coils, the coil ____________________ decreases and its ____________________ increases as the coil is "wound up"? (0.5 pt.)

b. For a helical extension spring, what do we mean when we say that the end is a standard end? (0.5 pt.)

c. Explain the meaning of Basic Load Rating (Dynamic Load Rating). (0.5 pt.)

d. Compare and contrast ball bearings and roller bearings in terms of speed and load carrying capabilities. (0.5 pt.)

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e. What is a Pressure Line? (0.5 pt.)

f. What is the effect of undercutting on gear tooth? (0.5 pt.)

g. Why is it desirable to have Contact Ratio, $m_p > 1$? (0.5 pt.)

h. Why is Backlash undesirable? (0.5 pt.)

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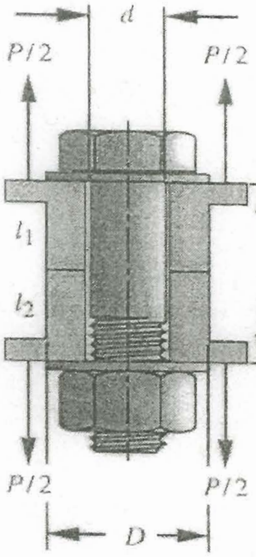
2B. The joint shown below is subjected to a fluctuating load that varies between P = 0 and P = 2000 lb. The bolt is a 5/16 – 18 UNC-2A SAE class 5.2 steel bolt with rolled threads. The joint dimensions are D = 1 in. and l = 2 in. The clamped length is 2 in., and the bolt length is 2.5 in. Both of the clamped parts are steel. The effects of the flanges on the joint stiffness will be ignored. A preload of 90% of proof strength is applied. The following fatigue modifying factors should be assumed: $C_L = 0.90$, $C_S = 1$, $C_F = 0.76$, $C_T = 1$, $C_R = 0.81$, $K_{fm} = 1.1$ and K_f = ? **(0.5 pt)** The bolt stiffness is K_b = 1.059E6 lb/in, K_m = 1.063E7 lb/in.



Determine the following:

a. Resultant bolt load (0.75 pt)
b. Resultant member load (0.5 pt)
c. Alternating stress (0.25 pts)
d. Mean stress (0.25 pt.)
e. Stress due to preload (0.5 pt.)
f. Endurance strength (limit) (1.0 pt)
g. Load factor guarding against bolt failure (0.75 pt)
h. Load factor guarding against joint separation (0.5 pt)
i. Safety factor against static yielding (0.5 pt.)
j. What tightening torque should be used (0.5 pt)

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Table 14-1 Principal Dimensions of Unified National Standard Screw Threads

Data Calculated from Equations 14.1—See Reference 3 for More Information

Size	Major Diameter d (in)	Coarse Threads—UNC: Threads per inch	Coarse Threads—UNC: Minor Diameter d_r (in)	Coarse Threads—UNC: Tensile Stress Area A_t (in^2)	Fine Threads—UNF: Threads per inch	Fine Threads—UNF: Minor Diameter d_r (in)	Fine Threads—UNF: Tensile Stress Area A_t (in^2)
0	0.0600	–	–	–	80	0.0438	0.0018
1	0.0730	64	0.0527	0.0026	72	0.0550	0.0028
2	0.0860	56	0.0628	0.0037	64	0.0657	0.0039
3	0.0990	48	0.0719	0.0049	56	0.0758	0.0052
4	0.1120	40	0.0795	0.0060	48	0.0849	0.0066
5	0.1250	40	0.0925	0.0080	44	0.0955	0.0083
6	0.1380	32	0.0974	0.0091	40	0.1055	0.0101
8	0.1640	32	0.1234	0.0140	36	0.1279	0.0147
10	0.1900	24	0.1359	0.0175	32	0.1494	0.0200
12	0.2160	24	0.1619	0.0242	28	0.1696	0.0258
1/4	0.2500	20	0.1850	0.0318	28	0.2036	0.0364
5/16	0.3125	18	0.2403	0.0524	24	0.2584	0.0581
3/8	0.3750	16	0.2938	0.0775	24	0.3209	0.0878
7/16	0.4375	14	0.3447	0.1063	20	0.3725	0.1187
1/2	0.5000	13	0.4001	0.1419	20	0.4350	0.1600
9/16	0.5625	12	0.4542	0.1819	18	0.4903	0.2030
5/8	0.6250	11	0.5069	0.2260	18	0.5528	0.2560
3/4	0.7500	10	0.6201	0.3345	16	0.6688	0.3730
7/8	0.8750	9	0.7307	0.4617	14	0.7822	0.5095
1	1.0000	8	0.8376	0.6057	12	0.8917	0.6630
1 1/8	1.1250	7	0.9394	0.7633	12	1.0167	0.8557
1 1/4	1.2500	7	1.0644	0.9691	12	1.1417	1.0729
1 3/8	1.3750	6	1.1585	1.1549	12	1.2667	1.3147
1 1/2	1.5000	6	1.2835	1.4053	12	1.3917	1.5810
1 3/4	1.7500	5	1.4902	1.8995			
2	2.0000	4.5	1.7113	2.4982			
2 1/4	2.2500	4.5	1.9613	3.2477			
2 1/2	2.5000	4	2.1752	3.9988			
2 3/4	2.7500	4	2.4252	4.9340			
3	3.0000	4	2.6752	5.9674			
3 1/4	3.2500	4	2.9252	7.0989			
3 1/2	3.5000	4	3.1752	8.3286			
3 3/4	3.7500	4	3.4252	9.6565			
4	4.0000	4	3.6752	11.0826			

Table 14-2 Principal Dimensions of ISO Metric Standard Screw Threads

Data Calculated from Equations 14.1—See Reference 4 for More Information

Major Diameter d (mm)	Coarse Threads: Pitch p mm	Coarse Threads: Minor Diameter d_r (mm)	Coarse Threads: Tensile Stress Area A_t (mm^2)	Fine Threads: Pitch p mm	Fine Threads: Minor Diameter d_r (mm)	Fine Threads: Tensile Stress Area A_t (mm^2)
3.0	0.50	2.39	5.03			
3.5	0.60	2.76	6.78			
4.0	0.70	3.14	8.78			
5.0	0.80	4.02	14.18			
6.0	1.00	4.77	20.12			
7.0	1.00	5.77	28.86			
8.0	1.25	6.47	36.61	1.00	6.77	39.17
10.0	1.50	8.16	57.99	1.25	8.47	61.20
12.0	1.75	9.85	84.27	1.25	10.47	92.07
14.0	2.00	11.55	115.44	1.50	12.16	124.55
16.0	2.00	13.55	156.67	1.50	14.16	167.25
18.0	2.50	14.93	192.47	1.50	16.16	216.23
20.0	2.50	16.93	244.79	1.50	18.16	271.50
22.0	2.50	18.93	303.40	1.50	20.16	333.06
24.0	3.00	20.32	352.50	2.00	21.55	384.42
27.0	3.00	23.32	459.41	2.00	24.55	495.74
30.0	3.50	25.71	560.59	2.00	27.55	621.20
33.0	3.50	28.71	693.55	2.00	30.55	760.80
36.0	4.00	31.09	816.72	3.00	32.32	864.94
39.0	4.00	34.09	975.75	3.00	35.32	1028.39

Table 14-6 SAE Specifications and Strengths for Steel Bolts

SAE Grade Number	Size Range Outside Diameter (in)	Minimum Proof Strength (kpsi)	Minimum Yield Strength (kpsi)	Minimum Tensile Strength (kpsi)	Material
1	0.25–1.5	33	36	60	low or medium carbon
2	0.25–0.75	55	57	74	low or medium carbon
2	0.875–1.5	33	36	60	low or medium carbon
4	0.25–1.5	65	100	115	medium carbon, cold drawn
5	0.25–1.0	85	92	120	medium carbon, Q&T*
5	1.125–1.5	74	81	105	medium carbon, Q&T
5.2	0.25–1.0	85	92	120	low-carbon martensite, Q&T
7	0.25–1.5	105	115	133	medium-carbon alloy, Q&T
8	0.25–1.5	120	130	150	medium-carbon alloy, Q&T
8.2	0.25–1.0	120	130	150	low-carbon martensite, Q&T

* Quenched and Tempered.

Table 14-7 Metric Specifications and Strengths for Steel Bolts

Class Number	Size Range Outside Diameter (mm)	Minimum Proof Strength (MPa)	Minimum Yield Strength (MPa)	Minimum Tensile Strength (MPa)	Material
4.6	M5–M36	225	240	400	low or medium carbon
4.8	M1.6–M16	310	340	420	low or medium carbon
5.8	M5–M24	380	420	520	low or medium carbon
8.8	M3–M36	600	660	830	medium carbon, Q&T
9.8	M1.6–M16	650	720	900	medium carbon, Q&T
10.9	M5–M36	830	940	1 040	low-carbon martensite, Q&T
12.9	M1.6–M36	970	1 100	1 220	alloy, quenched & tempered

$$\text{steels:}\quad \begin{cases} S_{e'} \cong 0.5\,S_{ut} & \text{for } S_{ut} < 200 \text{ ksi } (1\,400 \text{ MPa}) \\ S_{e'} \cong 100 \text{ ksi } (700 \text{ MPa}) & \text{for } S_{ut} \ge 200 \text{ ksi } (1\,400 \text{ MPa}) \end{cases}$$

$$\text{irons:}\quad \begin{cases} S_{e'} \cong 0.4\,S_{ut} & \text{for } S_{ut} < 60 \text{ ksi } (400 \text{ MPa}) \\ S_{e'} \cong 24 \text{ ksi } (160 \text{ MPa}) & \text{for } S_{ut} \ge 60 \text{ ksi } (400 \text{ MPa}) \end{cases}$$

$$\text{aluminums:}\quad \begin{cases} S_{f'_{@5E8}} \cong 0.4\,S_{ut} & \text{for } S_{ut} < 48 \text{ ksi } (330 \text{ MPa}) \\ S_{f'_{@5E8}} \cong 19 \text{ ksi } (130 \text{ MPa}) & \text{for } S_{ut} \ge 48 \text{ ksi } (330 \text{ MPa}) \end{cases}$$

$$\text{copper alloys:}\quad \begin{cases} S_{f'_{@5E8}} \cong 0.4\,S_{ut} & \text{for } S_{ut} < 40 \text{ ksi } (280 \text{ MPa}) \\ S_{f'_{@5E8}} \cong 14 \text{ ksi } (100 \text{ MPa}) & \text{for } S_{ut} \ge 40 \text{ ksi } (280 \text{ MPa}) \end{cases}$$

Steel: $S'_e = 0.45 S_{ut}$

The following are standard thread lengths:

$$l_t = \begin{cases} 2d + \tfrac{1}{4} \text{ in} & L \leqslant 6 \text{ in} \\ 2d + \tfrac{1}{2} \text{ in} & L > 6 \text{ in} \end{cases}$$

In these equations $l_t = l_{thd}$

$$l_t = \begin{cases} 2d + 6 \text{ mm} & L \leqslant 125 \text{ mm} \\ 2d + 12 \text{ mm} & 125 < L \leqslant 200 \text{ mm} \\ 2d + 25 \text{ mm} & L > 200 \text{ mm} \end{cases} \qquad d \leqslant 48 \text{ mm}$$

For a round bolt of diameter d and axially loaded thread length l_t within its clamped zone of length l as shown in Figure 14-21, the spring constant is

$$\frac{1}{k_b} = \frac{l_t}{A_t E_b} + \frac{l - l_t}{A_b E_b} = \frac{l_t}{A_t E_b} + \frac{l_s}{A_b E_b} \tag{14.11a}$$

where A_b is the total cross-sectional area and A_t is the tensile-stress area of the bolt, and $l_s = (l - l_t)$ is the length of the unthreaded shank. The length of the threaded portion is standardized as twice the bolt diameter plus 1/4 in for U.S. bolts (plus 6 mm for metric bolts) up to 6 in (150 mm) long. An additional 1/4 in of thread is provided on longer bolts. Bolts shorter than the standard thread length are threaded as close to the head as possible.[2]

For the cylindrical material geometry in Figure 14-23 (ignoring the flanges), the material spring constant becomes

$$\frac{1}{k_m} = \frac{l_1}{A_{m_1} E_1} + \frac{l_2}{A_{m_2} E_2} = \frac{4 l_1}{\pi D_{eff_1}^2 E_1} + \frac{4 l_2}{\pi D_{eff_2}^2 E_2} \tag{14.11b}$$

where the A_m are the effective areas of the clamped materials and the D_{eff} are the effective diameters of those areas.

If both clamped materials are the same

$$k_m = \frac{A_m E_m}{l} \tag{14.11c}$$

where A_m is the effective area of the clamped material (also see Section 14.8). If A_m can be defined as a solid cylinder with an effective diameter D_{eff}, equation 14.11*c* becomes

$$k_m = \frac{\pi D_{eff}^2}{4} \frac{E_m}{l} \tag{14.11d}$$

$$P = P_m + P_b$$

The compressive load F_m in the material is now

$$F_m = F_i - P_m : \qquad F_m \ge 0^*$$

and the tensile load F_b in the bolt becomes

$$F_b = F_i + P_b$$

We can summarize the information in Figure 14-24 in the following way. The common change in deflection $\Delta\delta$ due to the applied load P is

$$\Delta\delta = \frac{P_b}{k_b} = \frac{P_m}{k_m} \tag{14.13a}$$

or :

$$P_b = \frac{k_b}{k_m} P_m \tag{14.13b}$$

Substitute equation 14.12*a* to get

$$P_b = \frac{k_b}{k_m + k_b} P$$

or

$$P_b = CP \qquad \text{where} \quad C = \frac{k_b}{k_m + k_b} \tag{14.13c}$$

The term C is called the joint's *stiffness constant* or just the **joint constant**. Note that C is typically < 1, and if k_b is small compared to k_m, C will be a small fraction. This confirms that the bolt will see only a portion of the applied load P.

In like fashion,

$$P_m = \frac{k_m}{k_b + k_m} P = (1 - C)P \tag{14.13d}$$

These expressions for P_b and P_m can be substituted into equations 14.12*b* and 14.12*c* to get expressions for the bolt and material loads in terms of the applied load P:

$$F_m = F_i - (1 - C)P \tag{14.14a}$$

$$F_b = F_i + CP \tag{14.14b}$$

Equation 14.14*b* can be solved for the preload F_i needed for any given combination of applied load P and maximum allowable bolt (proof) load F_b, provided that the joint constant C is known.

The load P_0 required to separate the joint can be found from equation 14.14*a* by setting F_m to zero.

$$P_0 = \frac{F_i}{(1 - C)} \tag{14.14c}$$

A safety factor against joint separation can be found from

$$N_{separation} = \frac{P_0}{P} = \frac{F_i}{P(1 - C)}$$

will be demonstrated in an example. The fatigue safety factor can be calculated without drawing the Goodman diagram by employing equation 13.34*b* (p. 851), using notation consistent with this section.

✓

$$N_f = \frac{S_e(S_{ut} - \sigma_i)}{S_e(\sigma_m - \sigma_i) + S_{ut}\sigma_a} \tag{14.16}$$

$$n = \frac{S_y}{\sigma_{max}} = \frac{S_y}{\sigma_m + \sigma_a}$$

✓

$$F_{alt} = \frac{F_b - F_i}{2}, \qquad F_{mean} = \frac{F_b + F_i}{2}$$

where F_b is found from equation 14.14*b* (p. 905) with $P = P_{max}$.

The mean and alternating stresses in the bolt are

✓

$$\sigma_a = K_f \frac{F_{alt}}{A_t}, \qquad \sigma_m = K_{fm} \frac{F_{mean}}{A_t}$$

Table 14-8 Fatigue Stress-Concentration Factors for Bolts

Brinell Hardness	SAE Grade (UNS)	Metric Class (ISO)	K_f Rolled Threads	K_f Cut Threads	K_f Fillet
< 200 (annealed)	≤ 2	≤ 5.8	2.2	2.8	2.1
> 200 (hardened)	≥ 4	≥ 6.6	3.0	3.8	2.3

The stress due to the preload force F_i is

✓

$$\sigma_i = K_{fm} \frac{F_i}{A_t} \tag{14.15c}$$

$$A_m \cong \frac{\pi}{4}\left[\left(\frac{d_2 + d_3}{2}\right)^2 - d^2\right] \qquad (14.17a)$$

The material stiffness k_m is then found from equation 14.11 (p. 903) using diameters d, d_2, d_3 and length $l = l_m$ as defined in Figure 14-31 with $\phi = 30°$.

A more extensive study of joint stiffness using FEA was done by Wileman et al.[13] who fitted an empirical equation to their extensive finite-element-modeled data that defines an approximate material stiffness parameter k_m as a function only of bolt diameter d, clamped length l, and the clamped material's modulus of elasticity E. It works for any units system because the exponential expression is dimensionless.

$$k_m = d\,E\,A\,e^{b(d/l_m)} \qquad (14.17b)$$

The value of k_m is used in equation 14.13c (p. 905) to find the *joint stiffness constant* C. The coefficient A and exponent b in equation 14.17b will vary slightly with the Poisson ratio ν of the material. Table 14-9 shows these equation parameters for several common metals. For materials not shown, use the coefficients from the table for a Poisson ratio that is closest to that of the material used, or use the general expression. Wileman

Table 14-9 Stiffness Parameters for Equation 14.17 [13]

Material	ν	E (GPa)	A	b
Steel	0.291	206.8	0.78715	0.62873
Aluminum	0.334	71.0	0.79670	0.63816
Copper	0.326	118.6	0.79568	0.63553
Gray Cast Iron	0.211	100.0	0.77871	0.61616
General Expression (averaging all four tested materials)			0.78952	0.62914

$$S_e = \frac{C_R C_f C_s C_{IL} C_T}{K_f} S_e' \quad \text{Where:}$$

S_e = endurance limit of mechanical element

C_R = reliability factor

C_s = size factor

C_T = temperature factor

S_e' = endurance limit of test specimen

C_f = finish factor

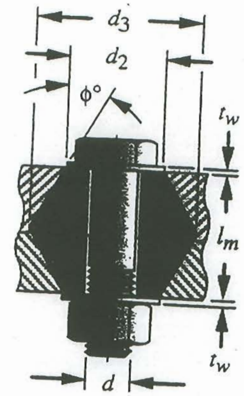
C_{IL} = impact (load) factor

K_f = fatigue notch factor

$T_i \cong K_i F_i d$ $\qquad$ $T_i \cong 0.21 F_i d$

Table 14-11
Torque Coefficient K_i for UNS Standard Threads with Coefficients of Friction $\mu = \mu_c = 0.15$

Bolt Size	K_i UNC	K_i UNF
0		0.22
1	0.22	0.22
2	0.22	0.22
3	0.22	0.22
4	0.22	0.22
5	0.22	0.22
6	0.22	0.22
8	0.22	0.22
10	0.22	0.21
12	0.22	0.22
1/4	0.22	0.21
5/16	0.22	0.21
3/8	0.22	0.21
7/16	0.21	0.21
1/2	0.21	0.21
9/16	0.21	0.21
5/8	0.21	0.21
3/4	0.21	0.21
7/8	0.21	0.21
1	0.21	0.21
1 1/8	0.21	0.21
1 1/4	0.21	0.21
1 3/8	0.21	0.21
1 1/2	0.21	0.20
1 3/4	0.21	
2	0.21	
2 1/4	0.21	
2 1/2	0.21	
2 3/4	0.21	
3	0.21	
3 1/4	0.21	
3 1/2	0.21	
3 3/4	0.21	
4	0.21	

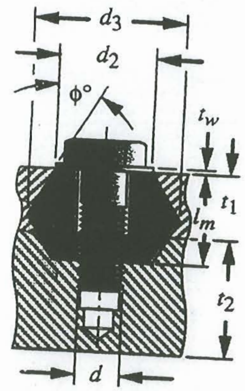


$l_b = l_m + 2t_w$

$d_2 = 1.5d$

$d_3 = 1.5d + l_m \tan\phi$

(a) Bolt-frusta model



$l_m = t_1 + t_2/2$ if $t_2 < d$

$l_m = t_1 + d/2$ if $t_2 \geq d$

$l_b = l_m + t_w$

$d_2 = 1.5d$

$d_3 = 1.5d + l_m \tan\phi$

(b) Cap screw frusta

FIGURE 14-31
Estimating the Material Compressed by a Bolt or Cap Screw Using the