

Teaching and Learning Experiences of an Integrated Mechanism and Machine Design Course

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Abstract

The objective of this paper is to discuss some of the issues concerning the teaching and learning experiences of an integrated mechanical assemblies and mechanical engineering design course taught at Kettering University. The integration into one discipline of subjects, which are otherwise commonly taught as separate courses, is also discussed. A course entitled “Analysis and Design of Machines and Mechanical Assemblies” is used as an example. Course objectives and learning outcomes are included along with an example course outline. While the two-course integration into a single one can pose some challenging issues with respect to the pre-requisites needed by the students, it provides a great opportunity to bring out new teaching materials conducive to active learning. The course is designed in such a way that the students are required to complete regular homework, class work and carry out simulation exercises using CAE tools. An example student project will be presented and the learning outcomes discussed.

Introduction

Many universities currently teach kinematics and dynamics of machinery and machine design as two separate courses with some schools still teaching these as three separate courses. However, due to the recent ABET requirements and other curriculum issues, many universities are considering to integrate these courses into a single one. In fact, a common recent trend ^[1] is to teach an integrated course, which includes concepts of statics and basic solid mechanics. Other examples include the integration of technical drawing and solid modeling, dynamics and introductory vibrations, finite element analysis (FEA) and machine component design and system dynamics and controls. Due to a recent curriculum reform at Kettering University (KU), the number of credits for graduation was reduced from 180 to 160 credits. A careful and systematic approach has been taken by the mechanical engineering department at KU to maintain the quality of education of the graduating students while reducing the total number of credits to graduate. This effort took more than two years and the details of such (horizontal and vertical) integration of basic courses were carried out with help from faculty belonging to the Science and Mathematics department. This has been by far the toughest task undertaken by the curriculum

reform task force (CRTF) at KU. Although it may appear that the final curriculum is “diluted” due to the reform, the final outcome has been very beneficial to the students.

In fact, as this paper is going to show, it is possible to combine different subjects in one course without compromising course content. This paper focuses on a course entitled Analysis and Design of Machines and Mechanical Assemblies (MECH – 510). The course outline is included in Appendix A.

The course

MECH-510 is a mezzanine level four credit elective course with four hours of contact (two separate blocks of two hours). It relies on the use of advanced computational and simulation tools, such as SDRC / I-DEAS[®]. The pre-requisites for this course are: MECH-200 (computer aided engineering), MECH-310 (dynamics) and MECH-312 (design of mechanical components). Also, a pre-test is given in the beginning of the course in order to evaluate the level of understanding that students have on these specific subjects.

The course is divided in two major parts: (i) Analysis, (ii) Design with synthesis. The analysis portion consists of kinematic, dynamic and stress analyses. The dynamic analysis portion can also include vibration and durability studies of critical components of a mechanical system. Graphical methods are used to conduct quick velocity analysis and the I-DEAS[®] Mechanism Design module is used to study the acceleration and reaction force characteristics of the mechanical system under scrutiny. Students typically model and conduct fatigue design studies on the Grashof and Non-Grashof linkages and their inversions, single and double slider chains and their inversions, and applications of six-bar mechanisms; namely, the toggle and quick-return mechanisms. The objective is to achieve a good understanding of mechanisms by simply modeling and studying the various inversions. Finally, cams and gears are also modeled in I-DEAS[®].

This subject can be quite challenging both to teach and to learn in the absence of a physical model or a computer model. Parametric studies can be conducted by changing the geometry, function or material of the various links (parts) comprising a mechanism, or by specifying various types of inputs in I-DEAS[®]. Students then model various types of cam follower assemblies and different gear trains. A term project that typically involves a combination of all different types of mechanisms studied (linkage, gear and cam mechanisms) is developed by students. The project includes analysis and design for fatigue life.

Students taking this class should have a senior level background in dynamics, CAE and machine component design. They are expected to use computational tools to conduct parametric studies in order to investigate different mechanism assemblies that are assigned a choice of various engineering materials. For example, a Grashof mechanism modeled in I-DEAS[®] can be easily transformed into a non-Grashof linkage by simply changing the length of one link in its wire-frame stage and updating the assembly to implement the change. All the joint information between each pair of links and the input motions defined earlier for the Grashof condition are preserved after updating for the non-Grashof linkage. This new mechanism when re-solved yields a new set of results thus allowing students to compare different kinematic and dynamic designs quickly. Likewise, a common database with a minimum number of parts in I-DEAS[®] can

be used to assemble a variety of different mechanisms. As for the selection of a topic for the final project, the students can view previous projects that are available in an I-DEAS[®] library. They can also search the Internet and other media, including textbooks, to form an idea of the relative complexity and feasibility of their project.

As it is often the case with integrated courses, currently there is not a textbook available that truly integrates these two subjects. The reference book used in the course is the one by Waldron and Kinzel ^[2] which covers the course material in a sequential manner, namely kinematics and dynamics of machinery followed by machine component design. The subjects are not discussed in a truly integrated manner. Other texts on the subject include the following references: Shigley and Mischke ^[3], Norton ^[4], Chironis ^[5] and Erdman et al. ^[6].

Examples of Student Feedback

In order to evaluate the student response to this new approach to teaching mechanisms and machine component design, they are asked a couple of questions by the end of each term. Below some of the responses to these questions are reproduced. These are samples from the last three terms the course was taught and were edited for clarity and grammar only.

a. Did we achieve integration of Mechanisms and Mechanical Component Design?

"Overall, I think the school has achieved integration of Mechanisms & Design. The class has approximately 70% Mechanisms & 30% Design. I personally liked this distribution. Modeling in I-DEAS[®] was more beneficial than design tips for a course like this. I would like to see more classes such as this one."

"During the first section the course covered kinematic analysis using I-DEAS[®] and hand calculations, which is an essential practice when designing a mechanism. Gears and cams were also addressed and practiced using I-DEAS[®]. The course concluded with force and fatigue analysis. I do think we achieved integration of subjects."

"I believe we achieved a fair amount of integration of mechanisms & design. The class definitely gave me a different look at machines and mechanisms. I think it gave me more of a systematic and logical approach to machines & mechanism assembly."

"The types of mechanisms and proper methods for designing individual components were covered very well."

"Actual designing of individual mechanisms was somewhat limited by time constraints and by the level of I-DEAS[®] knowledge needed to construct the models."

"I suggest that animation should be covered more in-depth during the preceding CAE classes, so that a student can be well prepared for this course. This will also allow more time for actual design work versus learning I-DEAS[®]."

"Yes, I have a better understanding of the material."

“Yes, integration was achieved. More mechanisms than design though.”

“Maybe the course should focus less on I-DEAS[®] mechanism modeling and analysis and more on taking two or three models and conducting design on them.”

“Yes, integration of the two subjects was achieved. The two were not completely balanced, there seemed to be a bias towards the mechanism part of the class. This however is good considering many other classes are aimed specifically towards design.”

“I was satisfied with the integration of the two studies, however, I think design could have been stressed a little more. The mechanism part was covered very well, and most of the homework was directed toward it.”

“Yes, we did achieve integration. The mechanisms we designed in class were easy enough to understand and yet had principles that applied to more complex models.”

“I learned a great deal about different mechanisms and what their function and design requirements were. I also liked how we were able to model these different mechanisms in I-DEAS[®] and see how they function. I wish we had more time to work on our final project. I would have liked to do more analysis of it.”

“There was definitely an integration between mechanisms and design. The design part was a little weak though because we never dealt with different types of materials or actual sizes. It was just designing the mechanism so parts don’t intersect. The mechanisms part was well covered although I would have liked to see more I-DEAS[®] demonstration on how to provide output plots for the gears.”

b. How did you like the selection of the final project that integrates the use of cams/gears and linkage mechanisms? Did we achieve integration of subjects lectured and final project?

“It was difficult to find a project that included cams, gears and four-bar links.”

“The project selection process was very independent. A list of possible topics could be gathered and presented to all students.”

“The project encompasses all the aspects covered in the class. The only issue is the amount of work required in the short period of time.”

“Projects completed in class were beneficial. They were straight to the point and represented the material covered in class. It is a good idea to have a final project mandatory to reflect all components (mechanisms) learned in class. In our case it was a project including gears, cams, and linkages.”

“All of the homework was relevant to the subject matter but there was too much work to do in each assignment. Some of the work was redundant.”

“A possible idea would be giving prescribed attributes for an IC engine and each group would model it. Each group could do analysis on a different part. Then results could be combined and evaluated in class. Maybe each group could provide a presentation of their findings.”

“Other possible ideas:

- Transmission with shift linkage
- Complete differential assembly
- V-configuration engine
- Rotary engine

“The final project was a good way of bringing together everything learned in class”

“The project selection was adequate. I was happy to be able to pick the engine project and to model it in I-DEAS[®]. It would be beneficial though to have a few more options, because I doubt all students would have liked the selections.”

“The only thing I suggest is giving the students more ideas for the project. I had a really hard time finding a mechanism that incorporated cams, links, sliders and gears.”

“Project selection was difficult because it was hard to find something that incorporated all those different mechanisms.”

Lessons learned

By evaluating the course work and projects developed by students during the last three terms the course was taught, the following observations can be made:

1. Overall, the students did an excellent job in terms of understanding graphical velocity analysis in addition to the computer modeling part. They also understood kinematic design of cams and gears reasonably well.
2. The students were more comfortable developing long and separate databases of individual parts for each homework rather than having a minimal number of parts and creating copies of those (as instances). Copies of each individual part from the databases they developed were made in order to create different assemblies in I-DEAS[®]. Some of the reasons why this approach was taken can be traced to limited knowledge of I-DEAS[®], lack of practice and confidence in some of the software features and lack of proper communication between group members.
3. Mainly due to limited knowledge of I-DEAS[®], time was not enough to conduct more useful and interesting parametric studies on the same mechanism. That led to designs that were not “optimized” to satisfy given functional requirements.
4. While selecting the final project most of the students did not refer to previous student projects that are available in a specific I-DEAS[®] library, nor did they search properly the Internet for options and ideas. System operational reasons such as disk quota and difficulty of uploading projects from older versions of I-DEAS[®] somewhat contributed to that.

5. In general it was noticed that the background required from the students in Mechanical Engineering Design was not adequate to satisfactorily complete the final project. This aspect is certainly the weakest link of this course and does pose a challenge that must be overcome by teacher and student during the course term.

Conclusions

In this paper, the status of an integrated course MECH-510: Analysis and Design of Machines and Mechanical Systems offered at Kettering University, Flint, MI is presented. The course integrates the conventional Mechanisms and the Advanced Machine Design courses into a single one offered at the mezzanine level. The integration (first of its kind at KU) was necessary due to reduction of the total number of credits needed for graduation, as well as to present course material in a more meaningful way to students taking this class. A CAE tool (I-DEAS[®]) has been integrated into the course. It was observed that there is a need for a more strong background on the software to fully take advantage of its capabilities in the course. This issue is being addressed in the pre-requisite courses (for instance MECH-200). Feedback from students has been very positive and rewarding as they continue to use the material learned in their senior thesis and wherever appropriate in other courses. There are a few other courses at KU that are the result of integration of one or more courses into a single course. Thus, there is a lot of scope to bring out good textbooks based on this concept.

References

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- [2] Waldron and Kinzel, *Kinematics, dynamics and design of machinery*: John Wiley and Sons, 1999.
- [3] Shigley and Mischke, *Mechanical engineering design*, 6th ed: McGraw Hill, 2001.
- [4] R. L. Norton, *Machine design: an integrated approach*, 2nd ed: Prentice Hall, 2000.
- [5] Chironis, *Mechanisms and mechanical devices source book*: McGraw Hill, 1991.
- [6] Erdman, Sandor, and Kota, *Mechanism design*, vol. 1, 4th ed: Prentice Hall, 2000.
- [7] B. Burges, E. Kiell, and S. Vogt, "Mech - 510: Project of a triple action press," Kettering University, Flint 2001.

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ARNALDO MAZZEI is an Assistant Professor of Mechanical Engineering at Kettering University. He received his Ph.D. in Mechanical Engineering from the University of Michigan in 1998. He specializes in dynamics and vibrations of mechanical systems and stability of drivetrains with universal joints. His current work relates to modal analysis, stability of drivetrains, finite element analysis and CAE. He is a member of ASME, ASEE and SEM.

Appendix A

MECH-510: Analysis and Design of Mechanical Assemblies (4-0-4)

2001 Catalog Data: Credit: 4 (4-0-4)

Prerequisites: Design of Mechanical Components I; Introduction to CAE; Mechanics III

The main aim of this course is to integrate the concepts of kinematic and dynamic analyses to the design of machines and mechanical assemblies used in automotive, medical equipment and other applications. These include (but are not limited to) the analysis and design of reciprocating engine sub-systems such as piston cylinder mechanism, steering linkages, window and door-lock mechanisms, over-head valve linkage system, flywheel, gears and gearboxes, universal couplings and automotive differentials. Synthesis of mechanisms used in the medical equipment area will also be covered. Kinematic and dynamic characteristics such as displacement, velocity, acceleration and forces are analyzed by graphical and analytical methods. CAE tools will be used to perform kinematic, dynamic and stress analyses and fatigue design of these systems. Temperature effects will also be included wherever appropriate. Several practical design projects will be assigned during the term of this course.

Textbook: R. L. Norton, *Machine design: an integrated approach*, 2nd ed: Prentice Hall, 2000.

References: (same as above)

Coordinator: Raghu Echempati

Course learning objectives and outcomes:

Upon completion of this course, the students will be able to

Objective 1: apply the integration of the fundamental concepts of rigid body kinematics in relative motion, solid mechanics, computer aided engineering, by using computational and design tools (ME PEOs¹ 1,2,3,4,5,6,7).

Objective 2: apply fundamental mechanics principles to the kinematic, dynamic and fatigue stress analyses of mechanical components, subsystems and systems (ME PEOs 1,2,3,4).

Objective 3: use state-of-the-art CAE software tools to formulate, conceptualize, design, analyze, and synthesize open-ended problems pertaining to mechanical systems (ME PEOs 1,2,3,4,5).

¹ Mechanical Engineering Primary Educational Objective.

- Objective 4:** *understand and incorporate design standards (for example, ASME and AGMA) in open-ended projects (ME PEOs 1,2,3,4,5,6).*
- Objective 5:** *integrate temperature effects, material selection, manufacturing considerations and other related aspects while designing mechanical systems, subsystems and components (ME PEOs 1,2,3,4).*
- Objective 6:** *develop strategies to improve product and process design based on results obtained (ME PEOs 1,2,3,4,5,6,7).*

Prerequisites by Topics:

1. Basic manufacturing processes.
2. Engineering materials.
3. Solid mechanics and introductory finite element analysis (linear analysis).
4. Dynamics (Kinematics & Kinetics, Energy Methods, Basic Vibration Theory).
5. Computer-aided engineering (solid modeling and design communication).

Topics (*CAE Tools will be used wherever appropriate throughout the course*):

Week	Topic
1.	Introduction to analysis and design of mechanical systems.
2.	Kinematic and dynamic analysis of machines and mechanism systems, including real-world industrial applications.
3.	Analysis and fatigue design of engine mechanism system with applications.
4.	Analysis and fatigue design of overhead valve systems.
5.	Analysis and fatigue design of compound and epicyclic gear trains involving helical gears; AGMA standards.
6.	Analysis and fatigue design of an automotive differential system using bevel and hypoid gears; AGMA standards.
7.	Study and design of worm gears; AGMA standards.
8.	Analysis and design of flywheels and couplings with applications.
9.	Introductory kinematic synthesis and applications to medical devices.
10.	Temperature effects, materials and manufacturing considerations in design; incorporation of ASME standards.
11.	Review.

Schedule: *Two sessions per week of 120 minutes*

Computer usage: PC or Unix-based software will be used.

Laboratory projects: Several laboratory exercises that are open-ended involving computer simulation and parametric studies on the modeling and analysis of machines and mechanical systems will be assigned.

Relationship to professional component: This course is 50 % engineering design.

Appendix B – Sample student project ^[7]

MECH-510: Analysis and Design of Machines and Mechanical Assemblies - Final Project of a Triple Action Press

1. Overview

The four main items of interest were the bending and fatigue analysis of the gears, a finite element analysis of one of the major links, and vibration analysis of one of the major links.

2. Analysis

Bending and Fatigue Analysis of Gear System:

Step 1: *Layout of Gear Train*

The first stage in the analysis of the gear system was to record information about the existing gear train. Figure 1.1 was a representation of the gear system found in the triple-action press located in Kettering Universities Manufacturing Laboratory. Table 1.1 gives the breakdown of the gear sizes for the drive train.

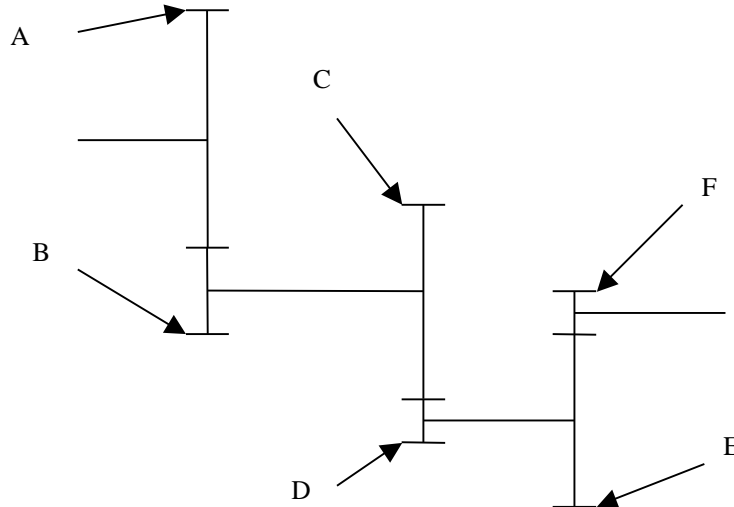


Figure 1.1- Schematic of Gear Train for Three-Punch Press

Gear Designation	Gear Diameter	Gear Radius
Gear "A" (<i>gear</i>)	310 (mm)	155 (mm)
Gear "B" (<i>pinion</i>)	50 (mm)	25 (mm)
Gear "C"	100 (mm)	50 (mm)
Gear "D"	32.5 (mm)	16.25 (mm)
Gear "E"	75 (mm)	37.5 (mm)
Gear "F"	30 (mm)	15 (mm)
Gear "G"	115 (mm)	57.5 (mm)
Gear "H" (<i>motor</i>)	30 (mm)	15 (mm)

TABLE 1.1- Gear information (Pressure angle =25°)

Step 2: *Motor Input Speed and Torque*

The next stage was to determine the engine speed (rpm) and torque. From the faceplate mounted on the electric motor, the operating speed of the engine was 1750rpm with an output of ¼ of a horsepower. Using this information, and the gear diameters from Table 1.1, the following equation was used to find the rpm at gear "A":

$$\text{Speed at Gear "A"} = N_s * \frac{d_H}{d_G} * \frac{d_F}{d_E} * \frac{d_D}{d_C} * \frac{d_A}{d_B}$$

$$\text{Speed at Gear "A"} = (1750\text{rpm}) * \frac{30}{115} * \frac{30}{75} * \frac{32.5}{100} * \frac{50}{310}$$

$$\text{Speed at Gear "A"} = 9.57 \text{ rpm}$$

The same procedure was used to find the torque exerted by gear "B" onto gear "A". This value will be used later to calculate the bending stresses in the gears.

$$\text{Torque at Motor} = \frac{P}{\omega_p} = \frac{(0.25\text{hp}) * (6600\text{in} - \text{lb} / \text{sec}) / \text{hp}}{1750\text{rpm} * (2\pi / 60)\text{rad} / \text{sec} / \text{rpm}}$$

$$\text{Torque at Motor} = 9.00 \text{ in} - \text{lb}$$

$$\text{Torque at Gear "B"} = T_p * \frac{d_G}{d_H} * \frac{d_E}{d_F} * \frac{d_C}{d_D}$$

$$\text{Torque at Gear "B"} = (9.00 \text{ in} - \text{lb}) * \frac{115}{30} * \frac{75}{30} * \frac{100}{32.5}$$

$$\text{Torque at Gear "B"} = 265.4 \text{ in} - \text{lb}$$

Step 3: *Finding Loads on Gears "A" & "B"*

In the following step, the magnitude of the loads on gears “A” and “B” must be found from the torque value found from step 2. The loads were calculated using the following equations:

$$\text{Load on Gears "A" \& "B"} = \frac{T_P}{r_p} = \frac{265.4 \text{ in} - \text{lb}}{1.00 \text{ in}} = 265.4 \text{ lbs}$$

Step 4: *Establishing correction coefficients*

The next stage involves the process of determining the correction coefficient values that are used in the bending stress equation. These coefficients are P_d (*Pitch Diameter*), F (*Face width of gear*), J (*Geometry factor*), K_a (*Application factor*), K_m (*Load distribution factor*), K_v (*Dynamic factor*), K_s (*Size factor*), K_B (*Rim thickness factor*) and K_I (*Idler factor*).

- P_d (*Pitch Diameter*): Industry uses anywhere from 6-10 inches as a standard. Therefore a value of 8 inches was used in the following calculations.
- F (*Face width of gear*): The face-to-face width of Gear’s “A” and “B” were measured to be approximately 2 inches.
- J (*Geometry factor*): The following values were obtained from *Machine Design, An Integrated Approach, 2nd Edition, NORTON, page 717, Table 11-13*.

Diameter of Pinion (Gear B) \cong 2.00 inches \therefore # of Teeth = $8 * 2.00 = 16$ Teeth

Diameter of Gear (Gear A) \cong 12.25 inches \therefore # of Teeth = $8 * 12.25 = 98$ Teeth

* Because there were no J - factors for a gear mesh of 16 to 98, Table 11-13 was interpolated to achieve the following results :

J - Factor of Pinion = 0.37, J - Factor of Gear = 0.498

- K_a (*Application factor*): The application factor is based on the type of service that the drive train will experience during its normal use. Applications in which gears are subjected to severe shock loading have a high application factor. For the case of a triple-action-press, the application factor can be assumed to be around 1.25 because the drive train does not experience heavy shock loading, but does experience medium shock loading.
- K_m (*Load distribution factor*): Because the face-to-face distance is 2 inches, according to Table 11-16, the load distribution factor is 1.6.
- K_v (*Dynamic factor*): This factor takes into account the vibrations and loads created by the gear teeth. Precision gears provide a smooth running gear train. Because the gearbox on the triple-action-press turns at a very low rpm, a gear quality (Q_v) of 6 was chosen. By establishing this variable, the following calculations were made to find the Dynamic Factor:

$$Q_v = 6, \quad B = \frac{(12 - Q_v)^{2/3}}{4} = \frac{(12 - 6)^{2/3}}{4} = 0.8255$$

$$A = 50 + 56(1 - B) = 50 + 56(1 - 0.8255) = 59.772$$

$$V_{t,\max} = [A + (Q_v - 3)]^2 = [59.772 + (0.8255 - 3)]^2 = 3,940.3 \text{ ft/min}$$

$$K_v = \left(\frac{A}{A + \sqrt{V_t}} \right)^B = \left(\frac{59.772}{59.772 + \sqrt{3940.3}} \right)^{0.8255} = 0.553$$

Coefficient	Pinion	Gear
W_t	265.4 lb	265.4 lb
P_d	8 inches	8 inches
F	2 inches	2 inches
J	0.498	0.370
K_a	1.25	1.25
K_m	1.6	1.6
K_v	0.553	0.553
K_s	1.25	1.25
K_B	1	1
K_I	1	1

TABLE 1.2- Coefficient Values for Pinion and Gear

- K_s (*Size factor*): This factor takes into account the tooth size of the gear. For gear “A”, it was found that the average tooth width was 0.300-inches. For the sake of safety, a size factor of 1.25 was chosen.
- K_B (*Rim thickness factor*): This factor takes into account situations in which large gears are made from a rim and spokes, rather than a solid disk. The rim thickness and tooth height are examined using a ratio to determine if there is a chance a failure due to the rim through the gear tooth. Gear “B” was constructed from a solid disk, therefore its thickness factor is 1. For the triple-action-press, gear “A” was constructed using the rim-spoke principle. Therefore, the following calculations were performed to find its corresponding thickness factor:

$$m_B = \frac{t_R}{h_t}, \text{ where } t_R = \text{tooth height and } h_t = \text{the rim height} \therefore$$

$$m_B = \frac{0.375 \text{ in}}{0.250 \text{ in}} = 1.5 \therefore \text{Since } m_B > 1.2, \text{ the thickness factor is 1.}$$

- K_I (*Idler factor*): This variable takes into account the cyclic stresses that an idler gear undergoes. Because neither gear in the drive train being examined is an idler, the idler factor for both gears is 1.

Step 5: Calculate Bending Stress in Pinion and Gear

$$\sigma_b = \frac{W_t * P_d}{F * J} * \frac{K_a * K_m}{K_v} * K_s * K_B * K_I, \text{ Standard Stress Equation}$$

∴ to find the bending stress in the gear ⇒

$$\sigma_{b,gear} = \frac{265.4 * 8}{2 * 0.498} * \frac{1.25 * 1.6}{0.553} * 1.25 * 1 * 1 = 9,637.1 \text{ psi}$$

∴ to find the bending stress in the pinion ⇒

$$\sigma_{b,pinion} = \frac{265.4 * 8}{2 * 0.370} * \frac{1.25 * 1.6}{0.553} * 1.25 * 1 * 1 = 12,971.0 \text{ psi}$$

These values will be later used to determine the factor of safety for the pinion and gear with respect to bending stress.

Step 6: Calculating Gear Surface Fatigue, Calculating C_p

All of the correction coefficients, except C_p , come from Table 1.2. The gears are assumed to be manufactured from steel, which has a $\nu=0.28$ and an $E=30*10^6$. Using these values, and the values from Table 1.2, C_p was calculated using the following equation:

$$C_p = \sqrt{\frac{1}{\pi * [(\frac{1-\nu_p^2}{E_p}) + (\frac{1-\nu_g^2}{E_g})]}} = \sqrt{\frac{1}{\pi * [(\frac{1-0.28^2}{30*10^6}) + (\frac{1-0.28^2}{30*10^6})]}} = 2,276 \text{ psi}$$

The next step is to calculate the “I” value of the gear set:

Pinion/Gear Pair, $I_p = I$, $d_1 = d_p$, $r_1 = r_p$ and $r_2 = r_g$

$$P_1 = \sqrt{\left(r_1 + \frac{1}{P_d}\right)^2 - (r_1 \cos \Phi)^2} - \frac{\pi}{P_d} \cos \Phi = \sqrt{\left(1.00 + \frac{1}{8}\right)^2 - (1 \cos 25^\circ)^2} - \frac{\pi}{8} \cos 25^\circ$$

$$P_1 = 0.3106 \text{ inches}$$

$$P_2 = (r_1 + r_2) \sin \Phi - P_1 = (1.00 + 6.125) \sin 25^\circ - 0.3106 = 2.700 \text{ inches}$$

$$I_{pinion} = \frac{\cos \Phi}{\left(\frac{1}{P_1} \pm \frac{1}{P_2}\right) * d_1} = \frac{\cos 25^\circ}{\left(\frac{1}{0.3106} + \frac{1}{2.7}\right) * 2} = 0.1262$$

Gear/Pinion Pair, $I_g = I$, $d_1 = d_g$, $r_1 = r_g$ and $r_2 = r_p$

$$P_1 = \sqrt{\left(r_1 + \frac{1}{P_d}\right)^2 - (r_1 \cos \Phi)^2} - \frac{\pi}{P_d} \cos \Phi = \sqrt{\left(6.125 + \frac{1}{8}\right)^2 - (6.125 \cos 25^\circ)^2} - \frac{\pi}{8} \cos 25^\circ$$

$$P_1 = 2.516 \text{ inches}$$

$$P_2 = (r_1 + r_2) \sin \Phi - P_1 = (6.125 + 1.00) \sin 25^\circ - 2.516 = 0.495 \text{ inches}$$

$$I_{gear} = \frac{\cos \Phi}{\left(\frac{1}{P_1} \pm \frac{1}{P_2}\right) * d_1} = \frac{\cos 25^\circ}{\left(\frac{1}{2.516} + \frac{1}{0.495}\right) * 12.25} = 0.0306$$

From these “I” values, the surface fatigue stress can now be calculated for the pinion and gear:

$$\sigma_{c, pinion} = C_p * \sqrt{\frac{W_t}{F * I_p * d_p} * \frac{C_a * C_m}{C_v} * C_s * C_f}$$

$$\sigma_{c, pinion} = 2276 * \sqrt{\frac{265.4}{2 * 0.1262 * 2} * \frac{1 * 1.6}{0.553} * 1.25 * 1} = 99,246.6 \text{ psi}$$

$$\sigma_{c, gear} = C_p * \sqrt{\frac{W_t}{F * I_p * d_p} * \frac{C_a * C_m}{C_v} * C_s * C_f}$$

$$\sigma_{c, gear} = 2276 * \sqrt{\frac{265.4}{2 * 0.0306 * 12.25} * \frac{1 * 1.6}{0.553} * 1.25 * 1} = 81,438.8 \text{ psi}$$

Step 7: Calculating Gear Fatigue

The first assumption made was that the gears were constructed from AGMA Grade 2-Steel, through hardened to 250HB. The service life required from the drive train was 10 years at an

operating temperature of 200°F. The first step is to find the allowable bending fatigue strength for the above criteria:

From Figure 11 - 25, pg733 :

$$S_{fb'} = 6235 + (174 * HB) - (0.126 * HB^2) = 6235 + (174 * 250) - (0.126 * 250^2)$$

$$S_{fb'} = 41,860 \text{ psi}$$

The next step is to calculate the life factor based on the above assumptions:

$$N = 1750 \text{ rpm} * \left(\frac{60 \text{ min}}{\text{hr}}\right) * \left(\frac{2080 \text{ hr}}{\text{shift} - \text{yr}}\right) * (10 \text{ yr}) * (1 \text{ shift}) = 2.184 * 10^9 \text{ cycles}$$

$$K_L = 1.3558 * N^{-0.0178} = 1.3558 * (2.184 * 10^9)^{-0.0178} = 0.9246$$

Since the temperature is assumed to be 200°F, K_T can be assumed to equal 1. It is also assumed that the material being used is 99% reliable, making $K_R=1$.

Taking these values into consideration, the corrected bending fatigue strength can now be calculated:

$$S_{fb} = \frac{K_L}{K_T * K_R} * S_{fb'} = \frac{0.9246}{1 * (1)} * 41,860 \text{ psi} = 38,703.8 \text{ psi}$$

The surface fatigue strength is now calculated for the gear set:

$$S_{fc'} = 27000 + (364 * HB) = 27000 + (364 * 250) = 118,000 \text{ psi}$$

The next step is to find the coefficients to calculate the correct surface fatigue strength of the drive train:

$$C_L = 1.4488 * N^{-0.023} = 1.4488 * (2.184 * 10^9)^{-0.023} = 0.8835$$

$$C_T = K_T = 1, \quad C_R = K_R = 1$$

The corrected surface fatigue strength can now be calculated :

$$S_{fc} = \frac{0.8835 * 1}{1 * 1} * 118,000 = 104,253 \text{ psi}$$

Step 8: Calculating and comparing safety factors

The final step is to calculate safety factors for the gear and pinion for fatigue and bending:

$$N_{b, \text{pinion}} = \frac{S_{fb}}{\sigma_{b, \text{pinion}}} = \frac{38703.8}{12971.0} = 2.98 \quad N_{b, \text{gear}} = \frac{S_{fb}}{\sigma_{b, \text{gear}}} = \frac{38703.8}{9637.1} = 4.02$$
$$N_{c, \text{pinion-gear}} = \left(\frac{S_{fc}}{\sigma_{cp}}\right)^2 = \left(\frac{104253}{99246.6}\right)^2 = 1.10 \quad N_{c, \text{gear-pinion}} = \left(\frac{S_{fc}}{\sigma_{cg}}\right)^2 = \left(\frac{104253}{81438.8}\right)^2 = 1.64$$

As shown above, the material and gear selection chosen will be sufficient. To increase the above safety factors the following measures can be taken: change material type, change gear types, change face width, change heat treatment or change gear quality.

3. Finite Element Analysis of Link Pivot Arm

After examining the loads on each component of the triple-action-press assembly, it was concluded that the pivot-link arm for the external punch was the most susceptible component for failure. Therefore, using SDRC / I-DEAS[®] software, a Finite Element Analysis was performed on the component.

The first step in developing the finite model was to determine the loading and constraints on the link member. Therefore, the torque at the main gear "A" was determined by using the following equation:

$$\text{Torque on gear "A"} = 9.00 * \frac{115}{30} * \frac{75}{30} * \frac{100}{32.5} * \frac{310}{50} = 1,645.5 \text{ lb} - \text{inch}$$

∴ the load at the gear can found by the following equation ⇒

$$\text{Load on Gear} = \frac{\text{Torque on Gear}}{\text{Radius of Gear}} = \frac{1645.5}{6.125} = 268.7 \text{ lbs}$$

The next step was to determine the load transferred from the gear to the pivot-link. To accomplish this task, proportional triangles were constructed to find the load transferred to the link at a distance of 1.772 inches from the main gears centerline (see Figure 1.2).

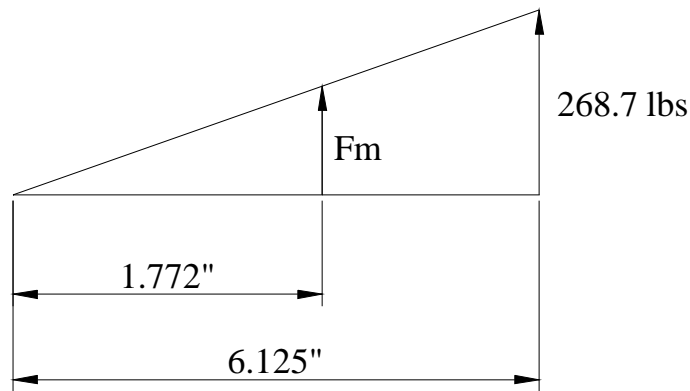


Figure 1.2 –Proportion Triangles

In Figure 1.2, the 6.125” dimension is the radius of Gear “A”, the 1.772” dimension is the distance to the pivot point for the link, the 268.7lbs is the maximum load transmitted through the gear to x=6.125” and Fm is the load needed for constraining the finite element model.

To calculate Fm, setup the following equality :

$$\frac{268.7}{6.125} = \frac{Fm}{1.772} \Rightarrow Fm * 6.125 = 268.7 * 1.772 \therefore Fm \cong 80 \text{ lbs}$$

Now that the load is known, the next step is to determine the proper constraining for the model.

For this particular model, the center pivot point was constrained in all directions except the z-axis. This will allow the part to flex about its pivot point in the assembly. The next point constrained was the far right pivot point. All directions were constrained at this point, as if the part was attached to the ground. The load was then applied to the far left pivot point in the negative y-direction. A mesh of 3-D solid tetrahedral elements was applied to the part, and the results were printed out using SDRC / I-DEAS® Visualizer.

From these results, the maximum principal stress was 805 lbf/in², the minimum principal stress was 197 lbf/in² and the maximum deflection was 9.72e⁻⁵ inches. Overall, the part was more then adequately designed to handle the load requirements. A sketch of the part with loading and constraining conditions is shown in Figure 1.3.

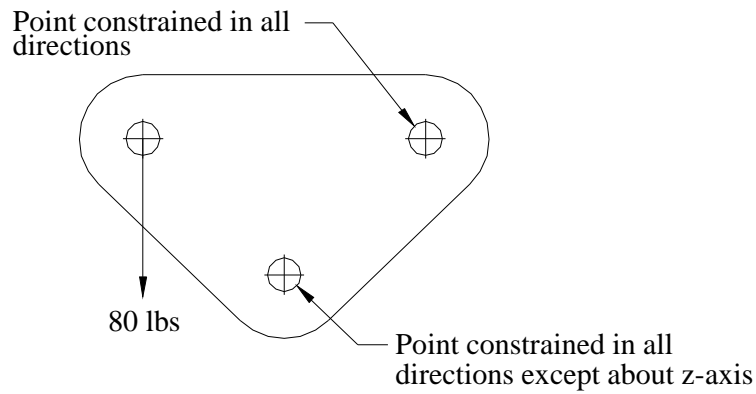


Figure 1.3-Part Loading and Constraining

One of the results for the FEM given by SDRC / I-DEAS[®] is shown in Figure 1.4.

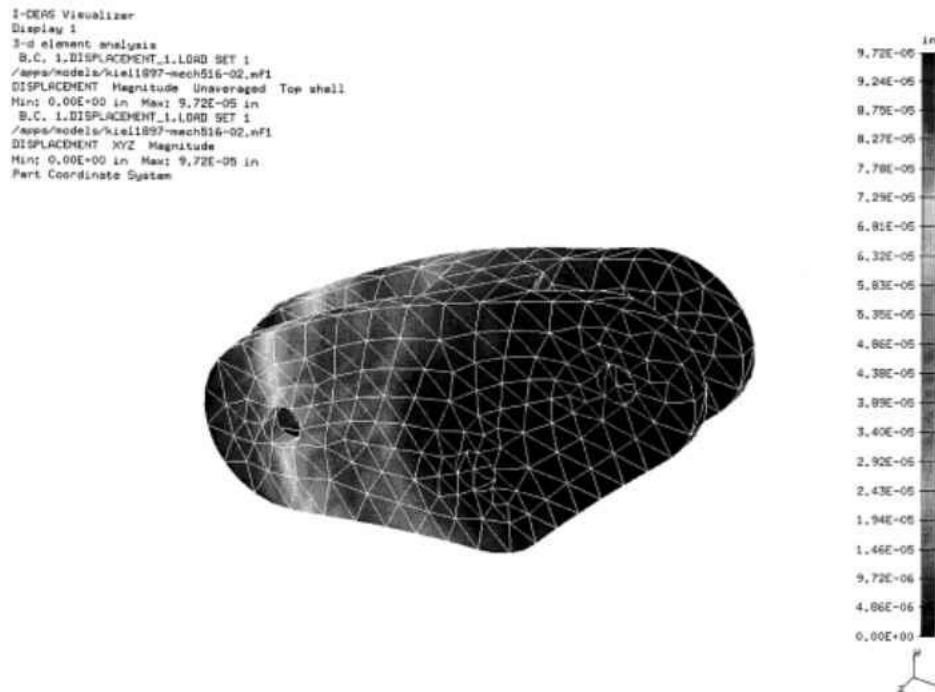


Figure 1.4-Displacement results

4. Vibration Analysis

SDRC / I-DEAS[®] was also used for the vibration analysis. Results are shown in Figure 1.5.

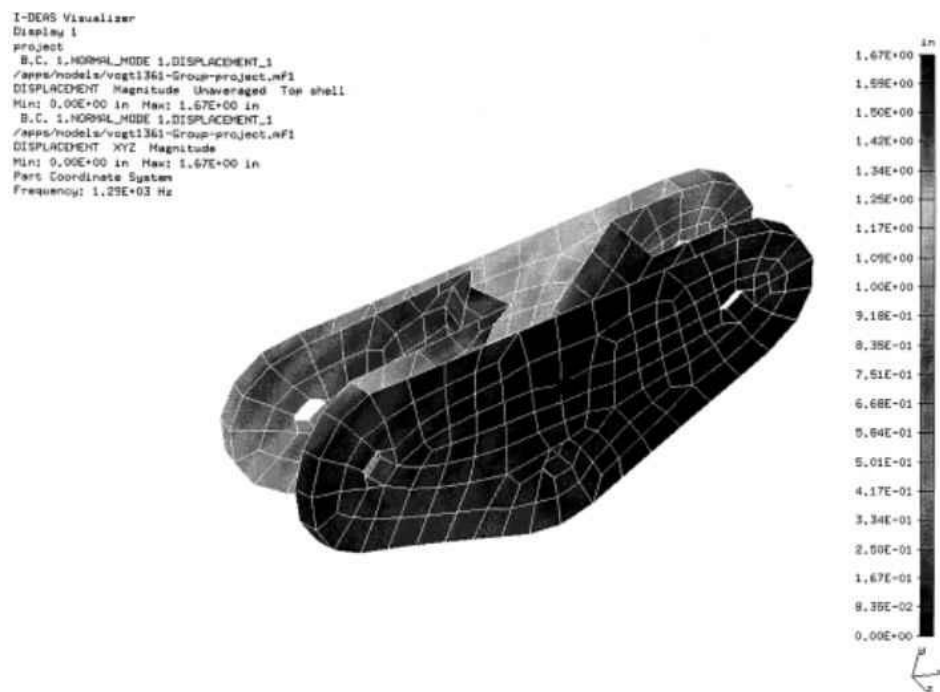


Figure 1.5-Vibration analysis results

The complete triple action press assembly is shown in Figure 1.6.

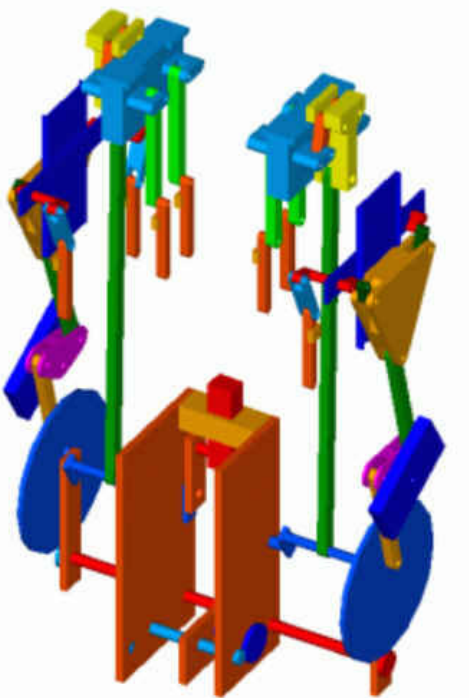


Figure 1.6-Triple action press