A comprehensive punch-press project for an undergraduate course on mechanical systems design

Petru A. Simionescu*

Texas A&M University Corpus Christi,
Corpus Christi, TX 78412, USA
Email: pa.simionescu@tamucc.edu
*Corresponding author

Eric W. Constans

Rose-Hulman Institute of Technology,
Terre Haute, IN 47803, USA
Email: constans@rose-hulman.edu

Abstract: In this paper, a semester-long project for an undergraduate mechanical-systems-design course is presented. The project requires students to perform kinematic and load torque analysis of a crank-driven punch press, choose a suitable electric motor, and calculate the moment of inertia of the flywheel such that an imposed coefficient of speed fluctuation is ensured. Then students select a pair of sheaves and the V-belt of the first stage of the press transmission, and size for endurance the second stage consisting of a pinion-shaft and the mating gear. In the process, students perform kinematic and energy balance calculations which they compare with Working Model 2D simulations of the crank press. While collaboration is encouraged, students choose their own set of design parameters, and present their work as individual written reports. This comprehensive project was very well received by the undergraduate mechanical engineering students at Texas A&M University Corpus Christi and can be adopted as is or with modifications at other universities.

Keywords: project-based learning; engineering calculations; computer simulation; crank press.

Reference to this paper should be made as follows: Simionescu, P.A. and Constans, E.W. (2022) 'A comprehensive punch-press project for an undergraduate course on mechanical systems design', *Int. J. Student Project Reporting*, Vol. 1, No. 1, pp.83–98.

Biographical notes: Petru A. Simionescu teaches and conducts research at Texas A&M University in Corpus Christi. He received his BSc in Mechanical Engineering from Politehnica University of Bucharest in 1992, Doctorate in Technical Sciences from the same university in 1999, and PhD degree from Auburn University in 2004. His research interests include kinematics, dynamics, design and optimisation of mechanisms and machines, CAD, and information visualisation. He has authored over 80 technical publications including one reference book titled *Computer Aided Graphing and Simulation Tools for AutoCAD Users* and has been granted nine patents.

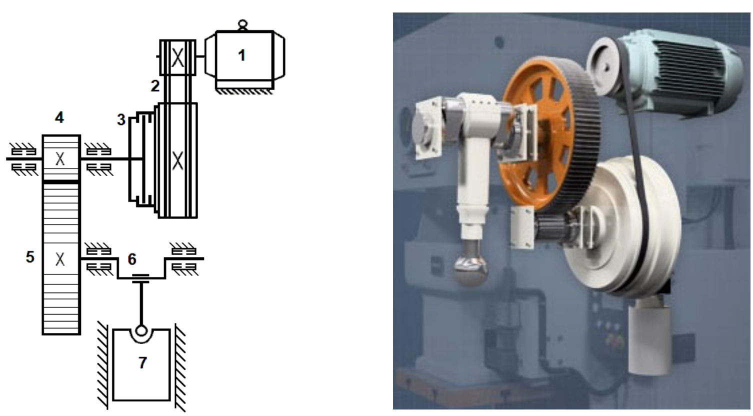
Eric W. Constans is a Professor of Mechanical Engineering at Rose-Hulman Institute of Technology. Before that, he served for six years as the Chair of the Mechanical Engineering program at Rowan University. He received his BS degree from the University of Washington, and his MS and PhD in Mechanical Engineering from the Pennsylvania State University. Prior to joining academia, he worked as a researcher in the Acoustics Center of Continental A.G., in Hannover, Germany. His interests include engineering education, acoustics, and vibration, and have published over 50 works, including the book *Introduction to Mechanism Design: with Computer Applications*.

1 Introduction

The benefits of project-based learning to engineering education are well recognised. These include heightened motivation, enhanced student participation in the learning process (active and self-learning), promoting critical and proactive thinking, etc. It also facilitates the development of soft skills, such as communication, working effectively in teams, project and time management, awareness of societal and ethical issues, etc. (Frank et al. 2003; Gary, 2015; Kokotsaki et al., 2016).

In this paper, a project assignment for a senior level class on mechanical systems design is presented. As part of this semester-long project, students perform kinematic and load torque analysis of a crank-driven punch press. The calculated average torque is then used in selecting an electric motor and in determining the moment of inertia flywheel of the press, such that a desired coefficient of speed fluctuation is ensured. Finally, students choose the V-belt with sheaves, and size the pinion-shaft and its mating gear, and produce a detailed mechanical drawing of the pinion shaft. This is a comprehensive project which combines subjects taught in classes such as engineering graphics, manufacturing processes, kinematics and dynamics of machinery and machine component design.

Figure 1 Main components of a crank press (see online version for colours)



(a) (b)

Notes: 1 – Motor; 2 – Belt transmission; 3 – Clutch; 4 – Pinion; 5 – Large gear/flywheel; 6 – Crankshaft; 7 – Slide-block.

2 Project statement

A mechanical power press is a machine used in manufacturing industry to apply pressure by means of slides or rams upon a pair of dies, for the purpose of shearing, punching, forming or assembling metal or other material (Lange, 1995; Hosford and Caddell, 2011). Of the known mechanical presses (Lange, 1995; Ham and Jang, 2009; Halicioglu, 2015a, 2015b), the student project discussed in this paper refers to a crank-slider press driven from a motor via a belt, in series with a pinion-gear pair, as shown in Figure 1 (Karakoulidis, 2015; Convergence Training, 2021).

Each student chooses their own specification for the press according to the recommendations in Table 1, and perform the design and calculations steps listed in Appendix 1. While collaboration is encouraged, students present their work individually in one midterm and one final written report.

Table 1 Project specifications with recommended value ranges

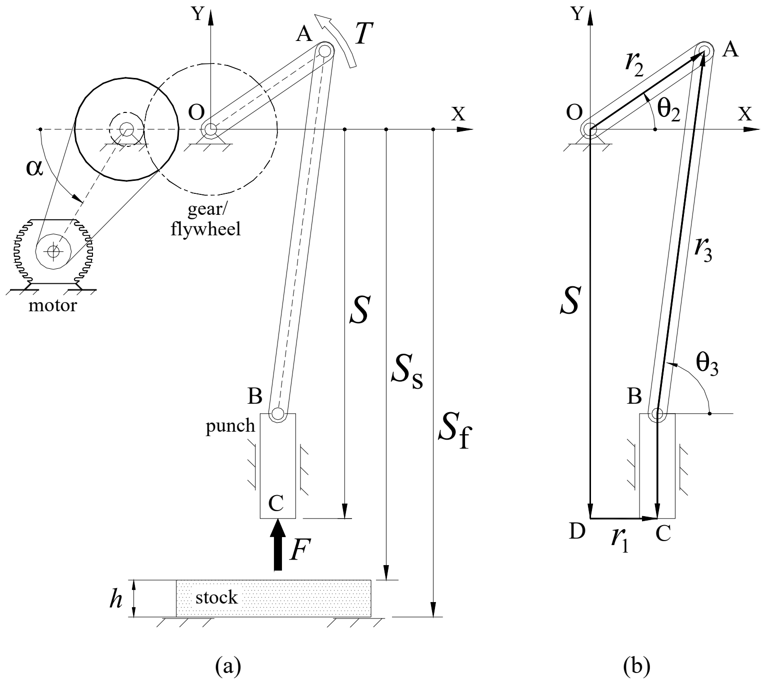
1	The press punches holes in aluminium sheets of one of the following grades (choose one): 1100-H12; 1100-H18; 3003-0; 3003-H12; 3003-H18; 6061-0
2	The hole diameter is between 40 and 160 mm
3	The stock thickness h is between 10 and 25 mm
4	The punch stroke $S_{max} – S_{min}$ is 15 to 25 times the stock thickness (bigger value for thinner material). Note that for negligible slider eccentricity, the crank length is half the punch stroke, i.e., $r_2 = 0.5(S_{max} – S_{min})$.
5	Slider eccentricity r_1 is between 0.5 and 1.25 times the workpiece thickness
6	Connecting-rod length r_3 is 2.5 to 4 times the punch stroke
7	After penetrating the workpiece, the punch continues to travel at least three times the workpiece thickness
8	The number of punches performed (equal to the average crank rotational speed n_p) should be between 80 and 120 holes per minute
9	Belt angle α is between 50 and 80 degrees

Note: See also Figure 1

To simplify the analyses in this project, the following assumptions are made:

- the material to be punched has an ideal plastic behaviour
- the friction in the joints of the crank-slider mechanism and between the punch and the workpiece are neglected
- the motor torque is constant and independent of the rotor speed
- the motor power is transmitted to the punch without losses
- the V-belt does not slip on its sheaves
- all inertias, other than that of the flywheel, are neglected.

Figure 2 Crank-slider punch press, (a) schematics (b) vector loop with main kinematic parameters



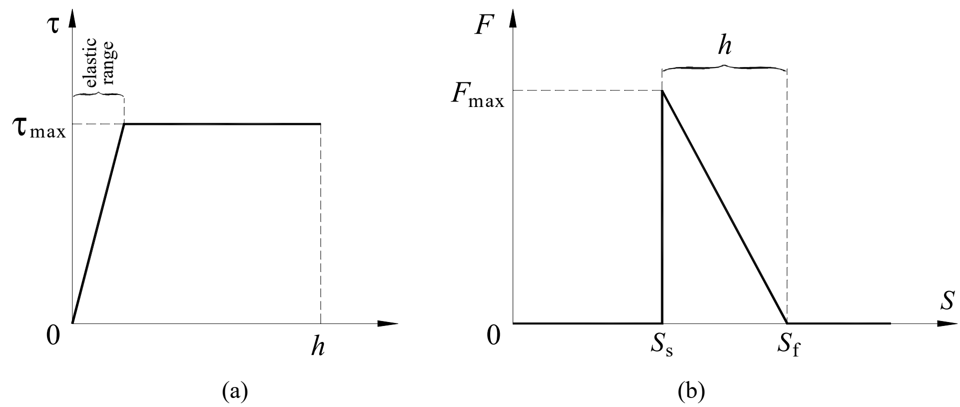
The press (Figure 2) employs a crank-slider mechanism of crank length OA = r_2, connecting-rod length AB = r_3, eccentricity CD = r_1, and punch length BC = L_p. The press punches holes of diameter d into aluminum stock of thickness h and shear strength τ_{max}, at a rate of n_p holes per minute. The actual punch begins when the displacement S equals S_s and ends when S equals S_f. According to the simplifying assumptions listed above, force F acts only when the punch is moving down and $S_s < S < S_f$. Assuming the workpiece shear-stress τ versus punch penetration S as shown in Figure 3(a) with negligible elastic range, the punch force F will peak right when the punch makes contact with the stock (i.e., for $S = S_s$), and will decreases linearly as the penetration progresses, reaching zero when $S = S_f$. The maximum punch force F_{max} depends on the shear strength of the material and on the shear area A_{shear} (equal to the side area of the hole) according to the equation:

$$F_{\max} = \tau_{\max} \cdot A_{shear} = \tau_{\max} \cdot (\pi \cdot d \cdot h) \tag{1}$$

The punch force F versus punch displacement S will therefore be:

$$F(S) = \begin{cases} \dfrac{S_f - S}{S_f - S_s} \cdot F_{\max} & \text{for } S_s \leq S_f \text{ and } \dfrac{dS}{dt} < 0 \\ 0 & \text{otherwise} \end{cases} \tag{2}$$

Figure 3 (a) Shear stress τ vs. punch penetration and (b) Punch force f vs. punch displacement for an ideally plastic material with negligible elastic range



3 Punch press linkage kinematics and inverse dynamics

The loop-closure equation of the crank-slider mechanism in Figure 2(b) is:

$$\overrightarrow{OA} - \overrightarrow{BA} + \overrightarrow{BC} - \overrightarrow{DC} - \overrightarrow{OD} = 0 \tag{3}$$

and projects on the X and Y axes as:

$$\begin{cases} r_2 \cdot \cos\theta_2 - r_3 \cdot \cos\theta_3 + r_1 = 0 \\ r_2 \cdot \sin\theta_2 - r_3 \cdot \sin\theta_3 - L_P - S = 0 \end{cases} \tag{4}$$

The unknowns in scalar equation (4) are the punch displacement S and transmission angle θ_3. To solve for displacement S for any crank angle θ_2, equation (4) are rearranged as

$$\begin{cases} r_2 \cdot \cos\theta_2 - r_1 = r_3 \cdot \cos\theta_3 \\ r_2 \cdot \sin\theta_2 - L_P - S = r_3 \cdot \sin\theta_3 \end{cases} \tag{5}$$

and after squaring and adding them, a quadratic equation in of the form S is obtained:

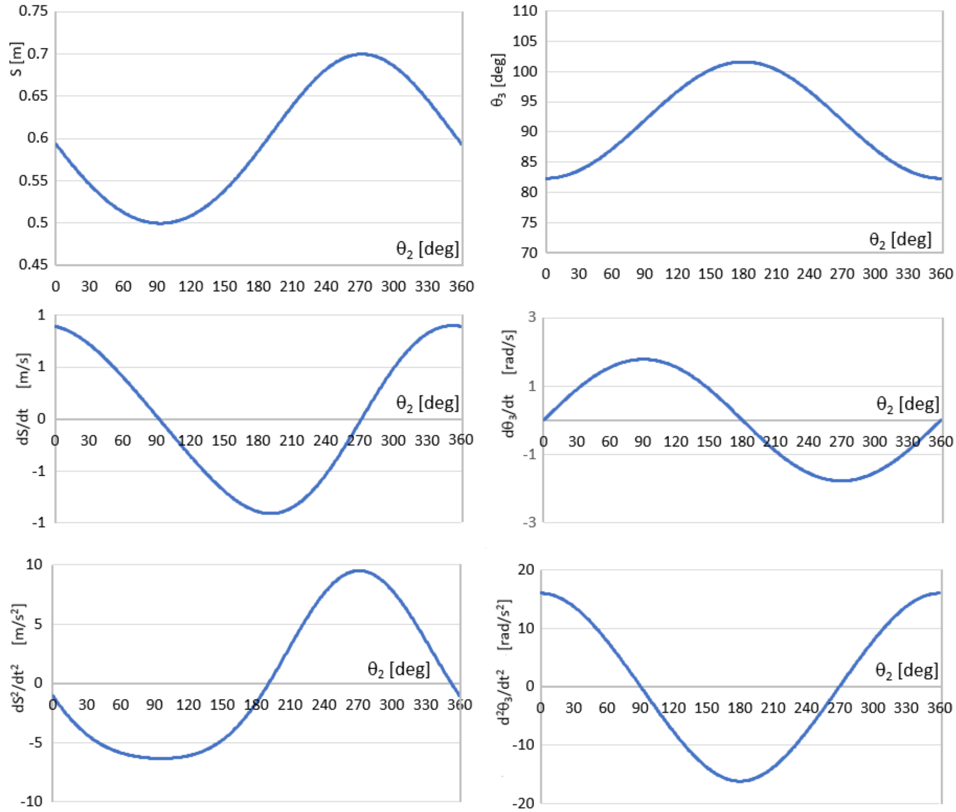
$$S^2 - 2 \cdot b \cdot S + c = 0 \tag{6}$$

For a given crank angle θ_2, the two solutions of this quadratic correspond to the two possible assembly configurations of the mechanism (Uicker et al., 2016; Constans and Dyer, 2018).

Once slider displacement S is determined, the preferred way of calculating the unknown transmission angle θ_3 is by applying the arctangent function of two arguments to the ratio of equation (5) as explained in Simionescu (2014).

The velocity problem is easier to solve by evaluating the first-time derivatives of equation (4), which yields a set of two linear equations in the unknowns $\mathrm{d}S/\mathrm{d}t$ and $\mathrm{d}\theta_3/\mathrm{d}t$. Before solving for velocities, punch displacement S and transmission angle θ_3 must be determined for the same crank angle θ_2, while the angular velocity of the crank (i.e., $\mathrm{d}\theta_2/\mathrm{d}t = \pi n_p/30$) must be specified as input.

Figure 4 Sample plots of punch displacement s and transmission angle θ_3 and their first- and second-time derivatives (see online version for colours)



Accelerations d^2S/dt^2 and $d^2\theta_3/dt^2$ are solutions to the set of two linear equations resulting from evaluating the second-time derivatives of equation (4). A constant angular velocity of the crank is assumed and therefore $d^2\theta_3/dt^2 = 0$. Evidently, dS/dt and $d\theta_3/dt$ must be known before attempting to calculate d^2S/dt^2 and $d^2\theta_3/dt^2$.

Figure 5 Plot of the crank torque and average torque vs. crank angle (see online version for colours)

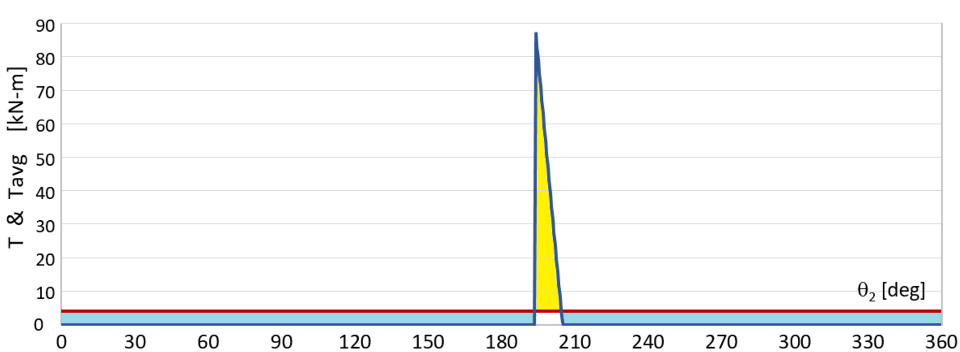


Figure 4 shows sample plots of the position, velocity and acceleration of punch displacement S and transmission angle θ_3 of a mechanism with r_1, r_2, r_3 and n_p chosen according to the specifications summarised Table 1 (see also Appendix 2).

Given the punch force F as function of the punch displacement S [see equation (2)], the crankshaft load torque can be calculated relatively easily by applying the principle of Virtual Work, i.e.,

$$T \cdot d\theta_2 = F \cdot dS \tag{7}$$

which is equivalent to:

$$T(\theta_2) = F \cdot dS/d\theta_2 = F \cdot \frac{dS/dt}{d\theta_2/dt} \tag{8}$$

Once the crank torque T vs. crank angle θ_2 becomes known as discrete successive values, the average torque per punch defined as:

$$T_{avg} = \frac{1}{2\pi}\int_0^{2\pi} T(\theta_2)\, d\theta_2 \tag{9}$$

can be relatively easily calculated using trapezoidal rule.

4 Flywheel moment of inertia calculation

Figure 5 shows the overlapped plots of the instantaneous crank torque T and of the average crank torque T_{avg} as functions of the crank angle θ_2. The area above the T_{avg} line and coloured in yellow, corresponds to the energy released by the flywheel as it slows down from its maximum speed $n_{p\max}$ right before the punch begins, to its minimum speed n_{pmin} when the punch ends. The two areas below the T_{avg} line shown in cyan, when taken together, are equal to the area in yellow, and correspond to the energy supplied by the motor to the flywheel as it accelerates from $n_{p\min}$ to $n_{p\max}$.

Given the energy ΔE exchanged with the flywheel (equal either to the yellow or to the two cyan areas in Figure 5 multiplied by π / 180) and the average (nominal) speed n_p of the flywheel, the required moment of inertia I of the flywheel for which a desired coefficient of speed fluctuation CS is ensured can be now determined. This coefficient of speed fluctuation is defined as:

$$C_S = \left(n_{p\max} - n_{p\min}\right)/n_p = \left(\omega_{p\max} - \omega_{p\min}\right)/\omega_p \tag{10}$$

and is recommended to be around 0.2 for punch presses and rock crushers (Dresig and Holzweißig, 2010; Norton, 2020; Uicker et al., 2016).

For machinery exhibiting only one torque spike such as in Figure 5, the energy exchanged with the flywheel can be calculated as:

$$\Delta E = \frac{1}{2}\int_0^{2\pi} \left|T(\theta_2) - T_{avg}\right| d\theta_2 \tag{11}$$

The integral in equation (11) can be evaluated numerically using the trapezoidal rule.

Figure 6 Free-body diagram of the flywheel of the punch press

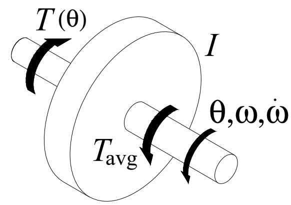


Figure 6 shows a free-body diagram of the flywheel, with T_{avg} the constant torque delivered by the electric motor as it is amplified through the press transmission. The equation of motion of the flywheel can be stated as:

$$T_{avg} - T(\theta) = I \cdot \dot{\omega} = I \frac{d\omega}{d\theta} \cdot \frac{d\theta}{dt} = I \cdot \omega \frac{d\omega}{d\theta} \tag{12}$$

This equation is integrated as

$$\int_{\theta_f}^{\theta_s} \left(T_{avg} - T \right) d\theta = \int_{\omega_{p\min}}^{\omega_{p\max}} I \cdot \omega \, d\omega = \frac{1}{2} I \left(\omega_{p\max}^2 - \omega_{p\min}^2 \right) \tag{13}$$

where θ_s and θ_f are the crank angles for which the maximum and minimum angular velocities of the crankshaft occur. Because the left-hand-side integral has the same value as the one in equation (11) we can further write:

$$\Delta E = \frac{1}{2} I \left(\omega_{p\max} - \omega_{p\min} \right) \left(\omega_{p\min} + \omega_{p\max} \right) \tag{14}$$

or

$$\Delta E = \frac{1}{2} I \left(n_{p\max} - n_{p\min} \right) \left(n_{p\min} + n_{p\max} \right) \left(\frac{\pi}{30} \right)^2 \tag{15}$$

The above two equations relate the change in kinetic energy ΔE and the required mass moment of inertia I of the flywheel, for which its speed fluctuation is limited between ω_{pmin} and ω_{pmax} or n_{pmin} and n_{pmax}.

If we approximate the average angular velocity of the crankshaft as:

$$\omega_p = \left(\omega_{p\max} + \omega_{p\min} \right) / 2 \tag{16}$$

equations (10) and (14) yield

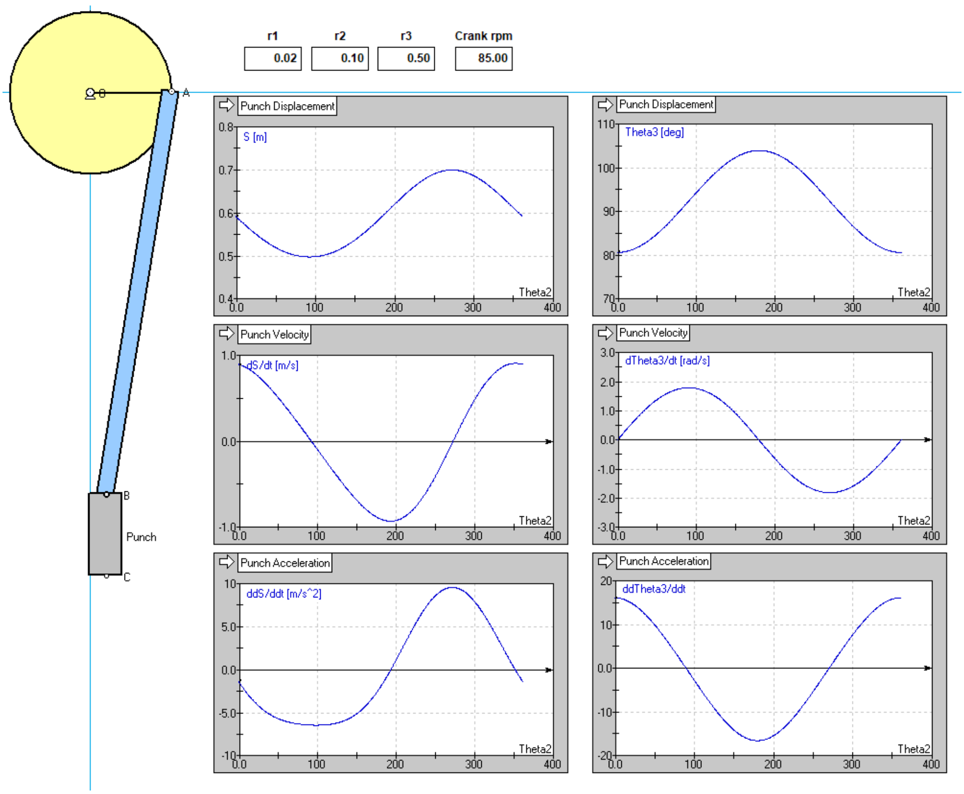
$$I = \frac{\Delta E}{C_s \cdot \omega_p^2} = \frac{\Delta E}{C_s \cdot n_p^2} \left(\frac{30}{\pi} \right)^2 \tag{17}$$

which is the required mass moment of inertia of the flywheel.

5 Working Model 2D simulation

Students are required to validate their Excel calculations using Working Model 2D motion simulation software (WM2D in short) available from Design Simulation (2021). A crank-slider mechanism with constant rotational input is required to be modelled in WM2D part of the midterm project report (Figure 7). Side-by side comparisons between the Excel plots and WM2D meters are expected part of students' reports.

Figure 7 Kinematic simulation done using Working Model 2D (see online version for colours)



For their final report, a second WM2D simulation must be prepared (see Appendix 3), where the driving motor is of the constant-torque type instead, and the resistance to workpiece penetration is modelled using a conditional force applied at the punch head. This conditional force is programed using the WM2D formula language based on equation (2). The maximum punch force $F_{\max}$ calculated with equation (1) is supplied to the WM2D program via a textbox control. In the simulation with the screenshot in Appendix 3, workpiece location S_f and workpiece thickness h are specified using textbox controls, as well as the crankshaft torque T_{avg}, and the moment of inertia I of the flywheel. The value of the crank initial speed which is about equal to np also provided via a textbox control.

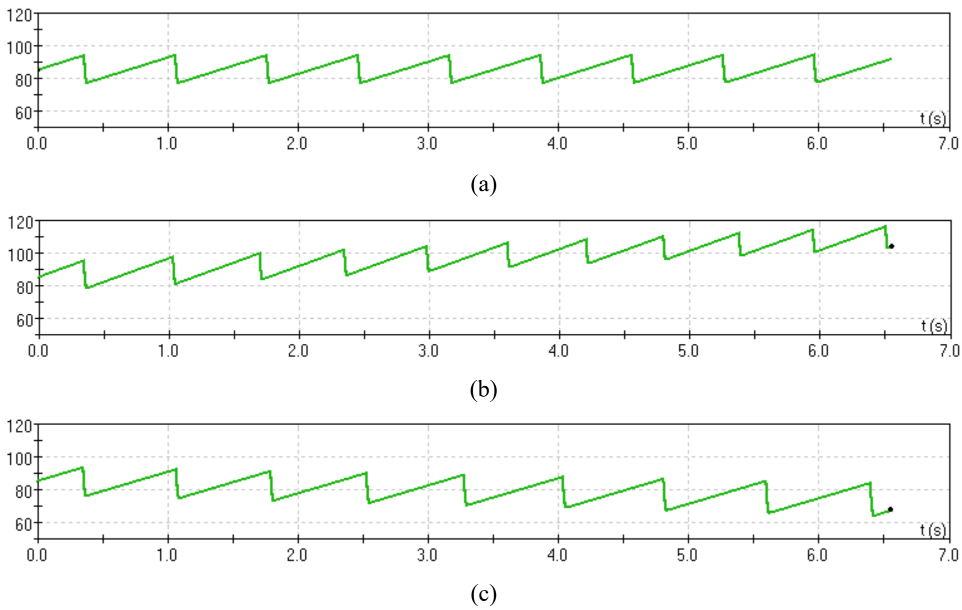
For the motor torque T_{avg} and flywheel moment of inertia properly selected, the rotational speed of the crank should fluctuate between the prescribed limits n_{pmin} and

n_{pmax} over multiple simulation cycles. A tendency of the crank rpm to increase indicates that T_{avg} is bigger than necessary. Conversely, if the crank tends to lose speed, it means that T_{avg} is smaller than required (see Figure 8).

The prescribed coefficient of speed fluctuation C_S can be validated using equation (10) and the n_{pmin} and n_{pmax} values extracted from the crank rpm vs. time plot output by WM2D. For example, the simulation with the screenshot in Appendix 3 yields a CS value close to the one imposed through the project specifications [see equation (18)], while the calculated average crank speed is also close to the one considered in Excel calculations, i.e., n_p = 85.6 rpm.

$$C_S = \frac{n_{p\max} - n_{p\min}}{\left(n_{p\max} + n_{p\min}\right)/2} = \frac{(93.37 - 77.23)}{(93.967 + 77.23)/2} = \frac{16.74}{85.6} = 0.1996 \approx 0.2 \tag{18}$$

Figure 8 Crank *rpm* vs. time for, (a) torque T_{avg} correctly calculated (b) T_{avg} too big (c) T_{avg} too small (see online version for colours)



6 Mechanical design and engineering analyses

After the average crank torque T_{avg} is determined, the corresponding input power can be straightforwardly calculated. This power, corrected with an overload factor of about 1.75, serves to choose a three-phase induction motor for the press.

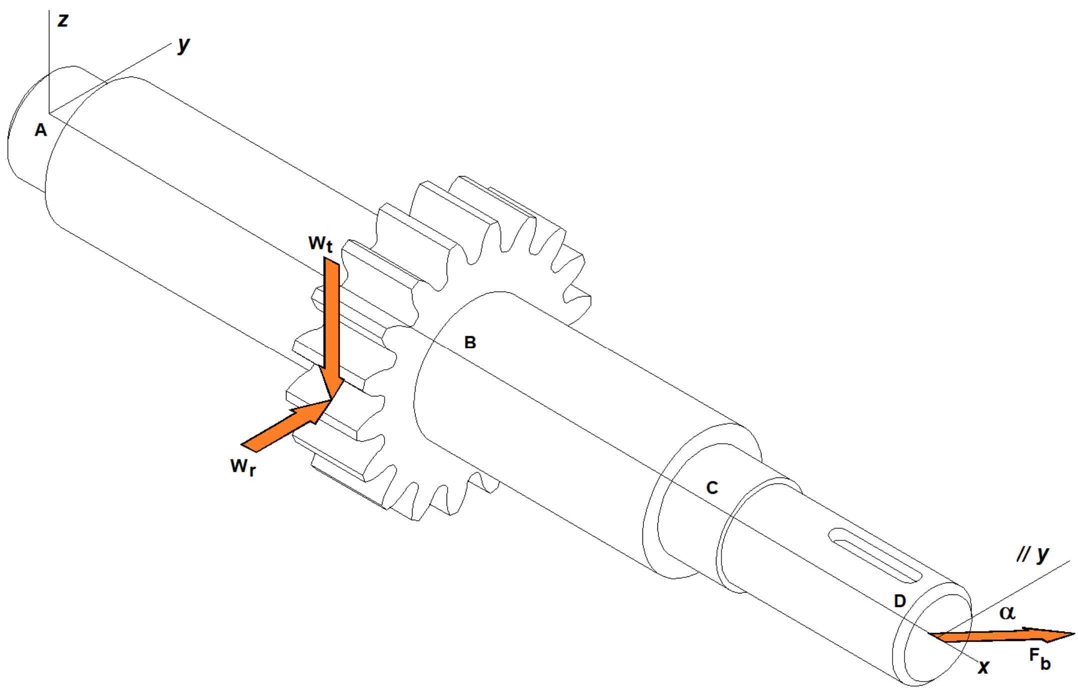
The next step involves dividing the total transmission ratio i_{total} between the belt transmission (with a ratio i_b) and the gear transmission (with a ratio i_g – see Figure 2) according to the equation:

$$i_{total} = \frac{n_{mot}}{n_n} = i_b \cdot i_g = \frac{D_2}{D_1} \cdot \frac{N_{gear}}{N_{pinion}} \tag{19}$$

where D_1 and D_2 are the pitch diameters of the small and of the big belt sheaves respectively. The number of teeth of the pinion N_{pinion} should be at least 17 teeth if standard involute gears are to be used. A larger N_{pinion} value may be however required to allow for a big enough gear and implicitly N_{gear}, such that the desired moment of inertia I is achieved. This is because the gear is also the flywheel of the press as shown in Figure 1.

To select the V-belt and sheeves, students follow the theory and sample problems available in the textbook (Mott et al., 2017). Similarly, students follow Mott et al. (2017) for calculating the gear-pinion pair, to size for fatigue the pinion shaft and to select a pair of rolling-element bearings to fit the journals at points A and C of the shaft as shown in Figure 9. Based on these analyses, students are required to prepare a mechanical drawing of the pinion shaft, where they practice assigning dimensioning and tolerance as they have learned in an earlier class on engineering graphics and machine component design.

Figure 9 Schematic for pinion shaft calculation, where W_r and W_t are the radial and tangential gearing forces, and F_b is the resultant belt force



7 Student perception

The project described in this paper has been assigned in the last three years to the undergraduate students in the Engineering Department at Texas A&M University Corpus Christi. Below are some of the favourable comments formulated by students in their end-of-semester course evaluations.

- I enjoyed doing the punch-press project. I think it was a good taste of what a full-scale mechanical design and analysis may entail.
- First time I have ever gone so in depth in calculations about a machine. Felt good! Also, I had no idea how well Excel could be utilised for different types of problems.
- I liked the project. Never working with the Working Model program before it was interesting. Refresher on AutoCAD was great as well.
- The project though tough, was extremely helpful in testing my engineering skills.
- The project required in depth critical analysis that facilitated a greater understanding of the course material and its significance in the workplace.
- The project was very cool. Real–world applications to theoretical content.
- The project was very interesting and fun to do. Though it had its challenges, it was worth it.

There were fewer negative comments received as well, mainly from students dissatisfied with the workload, and the use of Working Model 2D software.

- The project takes up too much time and is rather difficult. By the time I have completed the project I will have spent approximately 60–70 hours on it.
- The project is hard and takes forever to do it.
- I learned a lot because of the project, but why would we use Working Model and not Solid Works?
- The biggest challenge, in my opinion, was Working Model. I loved the idea behind it. I hated the lack of resources for learning it. Took more time trying to figure out the software than doing the calculations and typing the report.

8 Conclusions

In this paper, a comprehensive project for an undergraduate course on mechanical systems design has been presented. Students perform kinematic and load torque analysis of a crank-driven punch press. Based on the calculated required torque, they choose an electric motor, and size the flywheel of the press such that a suitable coefficient of speed fluctuation is satisfied. The kinematic and load calculations are implemented in Office Excel, and are validated using Working Model 2D simulations. Students then select the V-belt and sheaves for the first stage of the press transmission, size for endurance the second transmission stage consisting of a pinion shaft and its mating gear, and select a pair of rolling-element bearings for the pinion shaft. Finally, students generate detailed mechanical drawings of the pinion shaft with dimensions, tolerances and surface finishes. The project was well received by most mechanical engineering students at Texas A&M University Corpus Christi, and can potentially be adopted in other undergraduate engineering programs.

9 Lessons learned

When the project was first introduced, students were required to submit only one report at the end of the semester. To keep them on track however, a midterm project report was additionally introduced, covering only steps 1 to 11 in Appendix 1. It was also noted that occasionally students produce pinion shafts that appear too slender, and a shaft deflection analysis requirement has been recently added to the final project and is not documented here.

References

Constans, E. and Dyer, K.B. (2018) *Introduction to Mechanism Design: with Computer Applications*, CRC Press, Boca Raton FL.

Convergence Training (2021) Mechanical power press safety [online] http://www.convergencetraining.com/mechanical-power-press-safety.html (accessed 28 October 2021).

Design Simulation (2021) Working model 2D [online] http://www.design-simulation.com/WM2D/.

Dresig, H. and Holzweißig, F. (2010) *Dynamics of Machinery: Theory and Applications*, Springer-Verlag, Berlin.

Frank, M., Lavy, I. and Elata, D. (2003) 'Implementing the project-based learning approach in an academic engineering course', *International Journal of Technology and Design Education*, Vol. 13, pp.273–288, https://doi.org/10.1023/A:1026192113732.

Gary, K. (2015) 'Project-based learning', *Computer*, Vol. 48, No. 9, pp.98–100.

Halicioglu, R., Dülger, L.C. and Bozdana, A.T. (2015a) 'Mechanisms, classifications, and applications of servo presses: a review with comparisons', *Proc. Inst. Mech. Engr. Part B Journal of Engineering Manufacture*, Vol. 230, No. 7, pp.1177–1194.

Halicioglu, R., Dülger, L.C. and Bozdana, A.T. (2015b) 'Structural design and analysis of a servo crank press', *Engineering Science and Technology, An International Journal*, Vol. 19, pp.2060–2072.

Ham, K.C. and Jang, D.H. (2009) 'Kinematical analysis on the several linkage drives for mechanical presses', *Journal of Mechanical Science and Technology*, Vol. 23, pp.512–524.

Hosford, W.F. and Caddell, R.M. (2011) *Metal Forming: Mechanics and Metallurgy*, Cambridge University Press, New York.

Karakoulidis, K. (2015) 'Automated determination of the power required and selection of electric motors for forging fly-press mechanisms', *Journal of Engineering Science and Technology Review*, Vol. 8, No. 3, pp.78–82.

Kokotsaki, D., Menzies, V. and Wiggins, A. (2016) 'Project-based learning: a review of the literature', *Improving Schools*, Vol. 19, No. 3, pp.267–277.

Lange, K. (1995) *Handbook of Metal Forming*, Society of Manufacturing Engineers, Southfield, Michigan, USA.

Mott, R.L., Vavrek, E.M. and Wang, J. (2017) *Machine Elements in Mechanical Design*, Pearson, New York, NY.

Norton, R.L. (2020) *Machine Design: An Integrated Approach*, 6th ed., Pearson, New York.

Simionescu, P.A. (2014) *Computer Aided Graphing and Simulation Tools for AutoCAD Users*, CRC Press, Boca Raton, FL.

Uicker Jr., J.J., Pennock, G.R. and Shigley, J.E. (2016) *Theory of Machines and Mechanisms*, Oxford University Press.

Appendix 1

Project report requirements

1 Research known designs of slider-crank punch presses (consider patents also), and identify their main parts and functions.

2 Using the shear strength of the aluminum material selected and the diameter of the hole to be punched, derive the equations of the punch force vs. punch penetration, assuming a linear reduction of the shear force during punching.

3 Write the vector equations and the corresponding scalar equations for position, velocity and accelerations of the slider displacement and transmission angle.

4 Calculate the minimum and maximum slider displacement values and draw the press mechanism at scale in these two positions.

5 Calculate the minimum and maximum transmission angle values and draw the press mechanism at scale in these two positions.

6 Program in Excel the equations derived at steps (4) and (5) with columns for (A) time, (B) crank angle in radians, (C) crank angle in degrees, (D) punch displacement, (E) punch velocity, (F) punch acceleration, (G) transmission angle, (H) punch force, (K) required crank moment calculated using the principle of virtual work.

7 Use trapezoid integration to calculate the average torque required per one press cycle.

8 Assuming a constant input motor-torque, calculate the required flywheel moment of inertia to be placed on crankshaft that will limit the coefficient of speed fluctuation of the press to acceptable values.

9 Model the press in Working Model 2D. Assign small masses and small moments of inertia to the slider and to the connecting rod of the press, such that their flywheel effect is negligible.

10 Plot in Working Model 2D the crank rpm vs. time and use it to confirm that the press satisfied the desired coefficient of speed fluctuation.

11 Select a three-phase motor for the press.

12 Select a V-belt and pulley for the first stage of the press transmission.

13 Calculate and size the gear pair of the press, for the pinion being solid with its shaft.

14 Calculate for infinite fatigue life and size the intermediate shaft of the press. Assume the pinion to be halfway between the bearings, and the distance between bearings to be 2.5 – 3 times the pitch diameter of the pinion (see Figure 9).

15 Select a pair of bearings for the pinion shaft of the press.

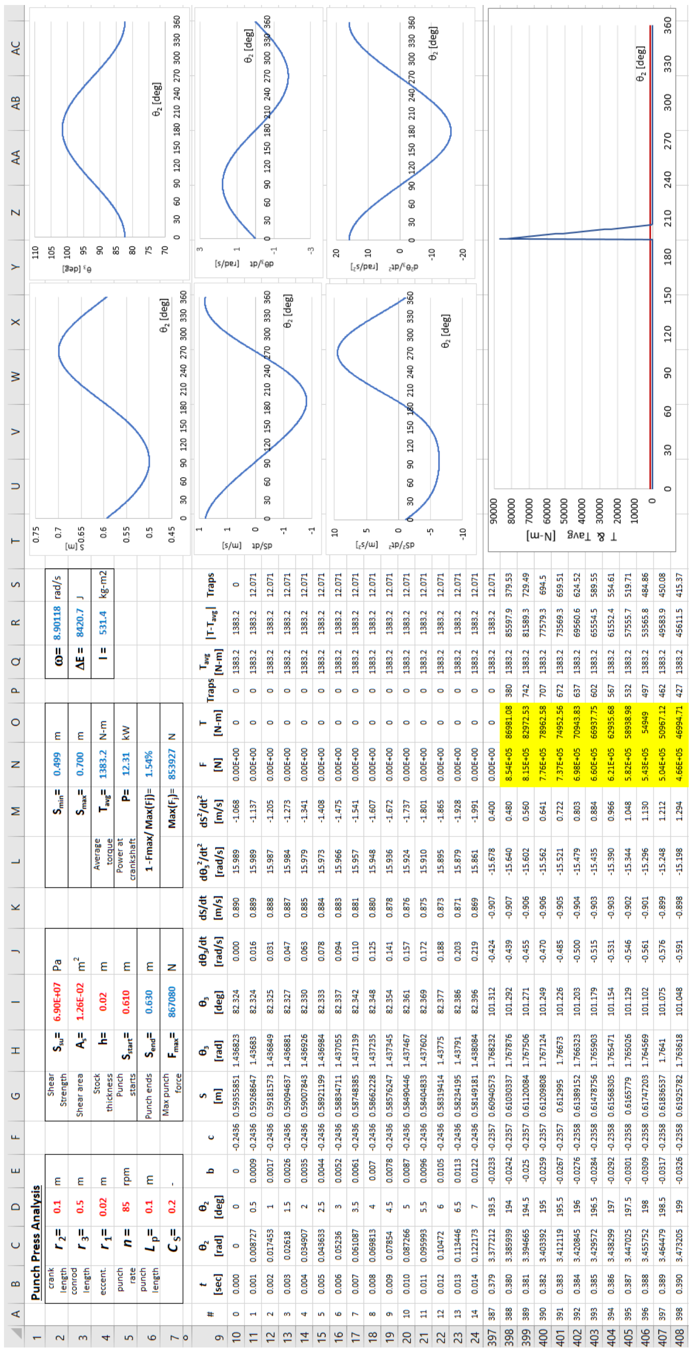
16 Draw full page, landscape the pinion-shaft with dimensions, tolerances and surface finishes.

17 Draw at scale a schematic of the entire mechanism as shown in Figure 2(a).

Appendix 2

Excel spreadsheet for kinematic and torque analysis

Figure A1 Excel spreadsheet for kinematic and torque analysis (see online version for colours)



Appendix 3

Working model 2D dynamic simulation

Figure A2 Working model 2D dynamic simulation (see online version for colours)