

Chapter 6

Engine Design

General Considerations

- The first step in designing a new engine is to choose the desired rated power output and rated speed. The load factor must be considered in choosing the speed.
- Load Factor = $\frac{\text{Average Power Output}}{\text{Maximum Power Output}}$

General Considerations con't

- Engines in tractors, heavy-duty trucks, and other working vehicles have a much higher *load factor* than automotive engines.
- An automotive engine typically uses only a small fraction of its maximum available power while cruising at typical speeds on a highway, but may need to accelerate the engine to very high speed and use maximum power for brief periods.

General Considerations

- For example, an automobile engine might produce its maximum power for a brief period while accelerating to 5000 rev/min when the car is passing another vehicle on a two-lane road, but might have a normal load factor of 0.3 while cruising at 2000 rev/min. Conversely, a tractor engine might run at 2000 rev/min with a load factor close to one for long periods of time while pulling an implement in a field.

General Considerations con't

- After the rated speed and power output are chosen, calculate the *engine displacement* that would be required to produce the power output at an acceptable p_{bme} level. Experience with other engines has shown the p_{bme} levels of 700 to 900 kPa are reasonable.

General Considerations con't

- After the engine displacement is chosen, the designer must decide on the number of engine cylinders needed to achieve the required displacement with reasonable piston size. Then the displacement per cylinder can be calculated.
- Choice of the *bore-stroke ratio* of an engine involves a design compromise. Smaller bore-stroke ratios permit higher compression ratios and the consequent higher combustion efficiencies.

Engine Timing, Firing Orders, and Intervals

- An engine will not run unless the following events are timed to the rotation of the crankshaft:
 1. Opening and closing of the valves
 2. Firing of the spark plugs in SI engines
 3. Start of fuel injection in diesel engines
- Also, in a multi-cylinder engine, the cylinders must fire in the proper order

Engine Timing, Firing Orders, and Intervals con't

- Cylinders are numbered from the belt-pulley (front) end of the engine toward the flywheel (rear) of the engine.
- In-line type engines have the cylinders numbered in the order in which their connecting rods are attached to the crankshaft.

Engine Timing, Firing Orders, and Intervals con't

- V-engines have two numbering conventions. Cylinders can still be numbered in the order in which their connecting rods are connected to the crankshaft, in which case the numbering alternates between cylinder banks. Some manufacturers number all cylinders, front to back, in the right bank and then number all of them in the left bank.

Engine Timing, Firing Orders, and Intervals con't

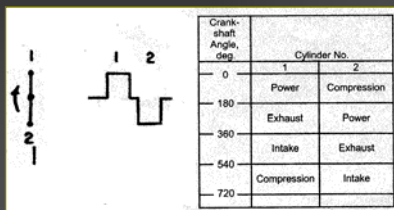
- The *standard direction of rotation* of the engine crankshaft is clockwise, as viewed from the front of the engine.
- Increasing the number of cylinders in an engine produces smoother power flow by reducing the average *firing interval*. The average firing interval can be calculated by:

Engine Timing, Firing Orders, and Intervals con't

- **AFI = $180 \text{ cy} / n$**
 - Where AFI = average firing interval in crankshaft degrees
 - $\text{cy} = 2$ for two-cycle engines or 4 for four-cycle engines
 - n = number of cylinders in the engine
- The firing order (FO) is simply the order in which the cylinders fire. In a two-cylinder, four-cycle engine, the firing order is 1-2, and the average firing interval is 360° . Because of the crankshaft arrangement, the firing intervals are not uniform.

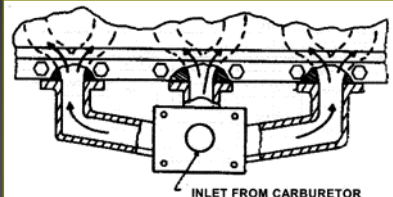
Engine Timing, Firing Orders, and Intervals con't

- There are two power strokes in one crankshaft revolution, followed by another revolution in which there are no power strokes.



Engine Timing, Firing Orders, and Intervals con't

- The need to balance air delivery to the various cylinders influences the choice of which firing order is actually used.



Engine Timing, Firing Orders, and Intervals con't

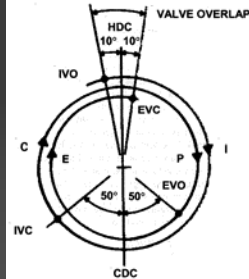
- If either of the FO branches beginning 1-2 were selected, there would be two consecutive air surges to the front branch of the manifold and then oscillation between the center and rear branches for remaining air surges.
- For uniform air delivery, it is better to eliminate the two consecutive surges in a single branch by selecting a FO beginning with 1-5. By far the most common FO used for 6-cylinder engines is: **1-5-3-6-2-4**

Valve Timing in an Engine

- Within the combustible range of the air-fuel ratios in an engine, the required air delivery rate is much larger than the required fuel delivery rate.
- Engine designers have several methods for maximizing the air pumping capacity of engines. Valve diameters are made as large as possible within the constraints of the allowable diameter of the combustion chamber.
- Sometimes 4 valves (2 intake, 2 exhaust) are used within each cylinder to achieve greater flow area.

Valve Timing in an Engine can't

- Remember, in a four-stroke-cycle engine, two revolutions of the crankshaft are required per engine cycle. A *valve-timing spiral* provides a convenient way of displaying the full 2 revolutions of the crankshaft needed to show all valve events.



Valve Timing in an Engine can't

- The optimum valve timing is a function of engine speed. The valve events are closest to the respective ends of the piston strokes at low engine speeds and move farther away from the dead centers as the engine speed increases.

Engine Balance

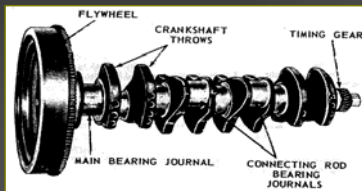
- There are several sources of potential unbalance in an engine:
 1. Rotating masses
 2. Reciprocating masses
 3. Rotational speed fluctuations
 4. Resulting from torque pulses
 5. Crankshaft twist resulting from torque pulses

Rotating Mass Unbalance

- The crank throws on the crankshaft of multi-cylinder engines are a possible source of rotating unbalance. There are two types of crankshaft balancing, *static* and *dynamic*.
- Static balancing is checked by laying the crankshaft on a pair of knife edges. The crankshaft is in static balance if it can be placed in any angular orientation and not roll on the knife edges.

Rotating Mass Unbalance con't

- Dynamic balance is achieved by use of counterweights installed opposite each crank throw. Engine manufacturers spin each crankshaft on a balancing machine and remove mass from the counterweights as needed to bring the crankshaft into dynamic balance.



Piston-Crank Dynamics

- Piston-crank kinematics affect engine balance and torque production. The piston movement as a function of crankshaft rotation is given by:

$$\frac{S}{R} = [1 - \cos\theta] + \frac{L}{R} \left[1 - \sqrt{1 - \left(\frac{R}{L} \sin\theta\right)^2} \right]$$

- Where S = piston displacement from HDC, m
- R = crank throw radius, m
- θ = crankshaft position, radians, measured from HDC
- L = connecting rod length, m

Piston Crank Dynamics con't

- Through the binomial series expansion:
 - $S / R = 1 - \cos \theta + (R / 2L)\sin^2 \theta$
- By differentiating, the approximate piston velocity is:
 - $v = R\omega [\sin \theta + (R / 2L)\sin 2\theta]$
 - Where $v = dS/dt$ = piston velocity (m/s)
 - $\omega = d\theta/dt$ = crankshaft speed, rad/s

Piston Crank Dynamics con't

- Note that ω is used instead of N to avoid the need for conversion factors in the piston-crank dynamics equations.
- By setting the derivative w.r.t θ equal to zero, the maximum piston speed can be shown to occur when the crankshaft is at:

$$\theta = \arccos\left[\frac{L}{4R}(-1 + \sqrt{1 + (8R^2 / L^2)})\right]$$

Piston Crank Dynamics con't

- Typically, the L/R ratio is close to 3 and the maximum piston speed occurs when the crankshaft is 1.27 to 1.31 radians (73° to 75°) from' HDC. Differentiating velocity w.r.t. time gives the equation for piston acceleration:
 - $a = R\omega^2[\cos \theta + (R/L) \cos 2\theta]$
 - Where a = piston acceleration, m/s^2

Piston Crank Dynamics con't

- Note that the maximum acceleration occurs at HDC and at CDC. The piston inertia force is given by:
 - $F = mR\omega^2[\cos \theta + (R/L) \cos 2\theta]$
 - Where F = inertial force, N
 - m = translational mass, kg

Piston Crank Dynamics con't

$$F = mR\omega^2[\cos \theta + (R/L) \cos 2\theta]$$

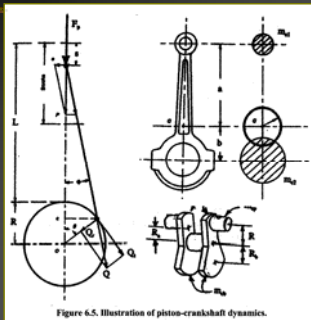


Figure 6.5. Illustration of piston-crankshaft dynamics.

Instantaneous Torque and Flywheels

- Piston-crank kinematics can also be used in calculating the instantaneous torque produced by each cylinder. The instantaneous torque is force, Q_t , multiplied by the crank radius, R:

$$\frac{T}{F_p R} = \left[1 + \frac{\cos \theta}{\sqrt{(L^2 / R^2 - \sin^2 \theta)}} \right] \sin \theta$$

- Where T = instantaneous torque, N*m
- F_p = instantaneous force on piston, N
- R = crank-throw radius, m

Instantaneous Torque and Flywheels con't

- The force on the piston at any time is the piston top area multiplied by instantaneous cylinder pressure. The radial force, Q_r , is also of interest, since it provides loading on the main bearings:

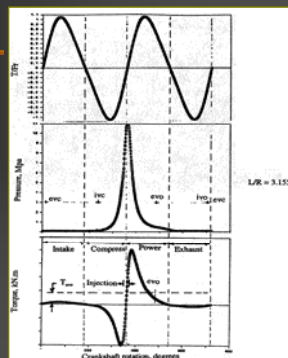
$$\frac{Q_r}{F_p} = \cos\theta - \frac{(R/L)\sin^2\theta}{\sqrt{1-(R^2/L^2)\sin^2\theta}}$$

Instantaneous Torque and Flywheels con't

- Neither the instantaneous torque nor the radial bearing loads can be calculated until the cylinder pressure is known as a function of crank angle.

Instantaneous Torque and Flywheels con't

- This figure shows instantaneous torque for a specific CI engine. Inserting the engine L/R ratio into the previous equation produced the diagram $T/F_p R$. The middle graph shows instantaneous cylinder pressures as measured with a transducer, and the bottom graph shows instantaneous torques.



Flywheel Design

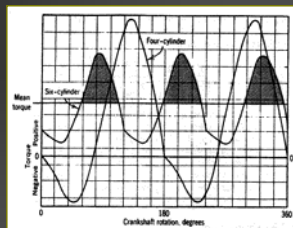
- The instantaneous torque is less than the average torque, T_{ave} , during most of the cycle. The average torque output is equal to the average torque imposed by the load on the engine; when T_{ave} is less than the load torque, the engine will stall unless a flywheel is used to supply part of the torque demand.

Flywheel Design con't

- The flywheel accelerates and stores kinetic energy when the instantaneous torque is greater than T_{ave} ; it decelerates and gives up kinetic energy when T_{ave} is greater than the instantaneous torque.
- The flywheel design consists of providing sufficient inertia such that cyclic speed changes are within desired limits.

Flywheel Design con't

- In a multi-cylinder engine, the contributions of the cylinders combine to produce an instantaneous torque curve that is less peaked than the chart shown previously.
- Combined instantaneous torque curves for a 4-cylinder and 6-cylinder engine are shown.



Flywheel Design con't

- Each of the shaded areas illustrated for the 6-cylinder engine are where the instantaneous torque exceeds the mean load torque; on a torque versus crank-angle plot, the area of each shaded region is equal to the kinetic energy that can be stored by the flywheel for release when the mean torque exceeds the instantaneous engine torque.

Flywheel Design con't

- The mass moment inertia of the flywheel is:
 - $I_f = (900 \Delta E k) / \pi^2 N_e^2$
 - Where I_f = required mass moment of inertia, $\text{kg} \cdot \text{m}^2$
 - N_e = average engine speed, rev/min
 - ΔE = kinetic energy transfer (J), $= \lambda W$
 - Where λ = ratio from Table 6.2
 - W = indicated work per revolution, J/rev
 - $k = 50/p$ = coefficient of speed variation
 - Where p = percent by which the speed is permitted to vary from N

Flywheel Design con't

- The indicated work per cycle can be estimated by:
 - $W = 60,000 P_b / N_e e_m$
 - Where P_b = brake power, kW
 - e_m = mechanical efficiency, decimal (estimated at 0.8 when P_b is full load)
 - The inertia calculated using the equation on the previous slide is the total inertia required to control the speed variation through each engine cycle. The clutch may contribute up to 40% of the required inertia, leaving 60% to be provided by the flywheel itself.
