DRIVE-TORQUE CAPACITY OF CRANKSHAFT FLANGES

As a recent picture in SPORT AVIATION shows, it is important to ensure that the drive-torque capacity of a crankshaft flange-propeller hub combination is adequate. By definition, only one of the four strokes accomplished by a four-stroke-cycle engine makes a positive contribution to rated engine torque. The torque contribution by the engine during the other three strokes is negative. It follows that the instantaneous peak torque must be far greater than rated engine torque. Also, the ratio of instantaneous peak torque to rated torque will vary with the number and arrangement of engine cylinders, among other things.

Most common flanges designed to drive wood propellers can be idealized into two distinct torque-transmission systems. The flat hub face can be thought of as driven by static-friction or the propeller can be considered as driven by the drive bushings incorporated in the flange. It is not possible to add the drive capacity of one system to that of the other since if the propeller is considered to be driven by static-friction, the drive bushings will not feel an imposed load and when drive bushings are considered the transmission mechanism, some movement of the hub against the flange must occur so that the static-friction mechanism cannot apply and scorching of the propeller hub boss will occur.

It is well known that the maximum friction force parallel to a contact face is defined by the compression force perpendicular to that face (F_c) multiplied by a coefficient of friction (f_o) dependent upon the two materials in contact. This fact can be used to calculate the maximum resisting torque due to propeller hub to engine flange contact if the compression stress in the hub and the applicable friction coefficient are known. By using a value of f_o equals 0.6 for the static-friction coefficient of dry hard-wood against steel (1), and a value of F_c equals 600 psi for the compression stress at the flange face (2), the equation defining the maximum instantaneous torque which can be transmitted from a crankshaft flange to a propeller hub made from Yellow Birch can be derived:

$$Q = \frac{\pi}{144} f_o F_c (D^3 - d^3) - \frac{n\pi}{96} f_o F_c d_{bc} d_{db}^2$$
$$= 7.85\ (D^3 - d^3) - 11.78\ (n d_{bc}\ d^2_{db})$$

where Q = Maximum instantaneous torque (ft.lb.)

D = Outer diameter of flange (inches)

d = Pilot stub diameter (inches)

n = Number of bolts

d_{bc} = Bolt circle diameter (inches)

d_{db} = Drive bushing diameter (inches)

We've used this equation to calculate the maximum allowable torque corresponding to several standard flange designs. See Table #1 for these calculations.

If it is assumed that the drive bushings in the flange must bear the torque load, the equation $Q=F_bAR$ can be applied, where F_b equals the allowable bearing stress for the drive bushings bearing against the sides of the holes provided for them, A equals the total drive bushing bearing area, and R equals the drive bushing radius from the crankshaft axis (equals bolt circle radius).

The total drive bushing bearing area equals drive bushing diameter multiplied by drive bushing contact length in the propeller and multiplied by the number of bushings (number of bolts), and the accepted value for allowable bearing stress in Yellow Birch is 790 psi (3). It follows that the equation $Q=F_bAR$ can be reduced to calculate maximum allowable torque directly from the drawing of an engine flange:

$$Q = F_b\ (n\ L_{db}\ d_{db})\ d_{bc}/24$$

$$= 32.92\ (n\ L_{db}\ d_{db}\ d_{bc})$$

Where:

Q = Maximum allowable torque (ft.lb.)

n = No. of bolts (No. of drive bushings)

L_{db} = Drive bushing length in propeller (inches)

d_{db} = Drive bushing diameter (inches)

d_{bc} = Bolt circle diameter (inches)

We've used this equation to calculate the maximum allowable instantaneous torque load transmitted to a propeller hub by drive bushings for several standard flanges. The results have been arranged in Table #2.

Installations with satisfactory service histories indicate that the drive-torque capacity of that crankshaft flange is adequate. It follows that the instantaneous peak engine torque is less than the static-friction drive-torque capacity of the flange. Several examples of satisfactory installations are shown in Table #3 which also shows the ratios of calculated torque capacity to rated torque for each installation.

After deciding that a flange is suitable for a particular installation, we must install the propeller so that the maximum torque capacity is actually obtained. As can be seen by a comparison of Tables #1 and #2, the maximum torque reaction is available by the static-friction drive mechanism which depends on the compression stress between the wood face and the crankshaft flange. This brings us back to the previously used figure of 600 psi for compression stress perpendicular to the grain in Yellow Birch which approximates the stress Yellow Birch can maintain over a long period of time. The applicable value of Young's Modulus across the grain in Yellow Birch is approximately E_T = 92,500 psi (2). It follows from the stress vs. strain equation that the propeller hub should be compressed about 0.006 inches per inch of hub thickness. A direct method to attain this strain would be to measure the propeller hub thickness (T) before installation and tighten the attaching bolts evenly until the propeller hub would measure (T) – 0.006(T). Another method would be to calculate the number of attaching bolt revolutions required to compress the hub by the desired amount.

The Sensenich W66LM propeller can be used to illustrate the second approach. The W66LM has a 5.375 inch thick hub and is attached to an SAE #2 crankshaft flange by 3/8-24 bolts. Therefore, 5.375 multiplied by 0.006 equals the desired compression equals 0.032 inches. The thread pitch of 24 threads per inch indicates that a 3/8-24 bolt will compress the propeller hub 0.042 inches per revolution. The desired compression (0.032) divided by compression per revolution (0.042) equals the number of revolutions required of each bolt after the propeller hub, flange face, and front face plate are in contact with each other. Therefore, the desired compression stress will be attained when each attaching bolt is tightened 0.76 revolutions after contact.

No further consideration can be given to the drive bushing (back-up) torque transmission mechanism and the propeller installation is now able to transmit its maximum driving capacity.

SENSENICH CORPORATION
Henry Rose
Henry S. Rose
Chief Engineer
Robert E. Bristol
Robert E. Bristol
Engineer

Ref: (1) Mechanical Engineer's Handbook, Lionel S. Marks, 1941
(2) ANC 18, Design of Wood Aircraft Structures, June 1951
(3) CAM 14, Aircraft Propeller Airworthiness, May 1946

Table #1: Torque Capacity Due to Static Friction

$$Q = 7.85(D^3 - d^3) - 11.78(nd_{bc}d_{db}^2)$$

Flange	D(in.)	d(in.)	n	d_{bc}(in.)	d_{db}(in.)	Q(ft.-lb)
SAE #1	5.50	2.63	6	4.375	0.625	1,000
SAE #2	6.00	2.63	6	4.75	0.625	1,400
SAE #3	6.50	2.63	8	5.25	0.625	1,800
SAE #4	7.00	2.63	8	6.00	0.688	2,300
AN 20,30	8.25	4.76	8	7.00	0.453	6,900
W68LY	7.75	2.50	6	5.50	0.75	3,300
McCulloch	5.00	2.62	6	4.00	0.344	800

Table #2: Torque Capacity Due to Drive Bushings

$$Q = 32.9(nl_{db}d_{db}d_{bc})$$

Flange	n	l_{db}(in.)	d_{db}(in.)	d_{bc}(in.)	Q(ft.-lb.)
SAE #1	6	0.438	0.625	4.375	240
SAE #2.	6	0.438	0.625	4.75	260
SAE #3	8	0.438	0.625	5.25	380
SAE #4	8	0.438	0.688	6.00	480
AN 20, 30	8	5.912	0.437	7.00	4,800
W68LY	6	0.75	0.75	5.50	610
McCulloch	6	0	0	4.00	0

Table #3: Satisfactory Installations

Flange	Engine	No. of Cyl	Q-Rating	Table #1 Q-Rating	Table #2 Q-Rating
SAE #1	Cont. C90-8F	4	191 ft-lb	5.2	1.3
SAE #2	Lyc. O-235-C1	4	218 ft-lb	6.4	1.2
SAE #3	Lyc. O-290-C	4	253 ft-lb	7.1	1.6
SAE #4	Cont O-470-11	6	434 ft-lb	5.3	1.1
AN 30 spline	R-985	9	1,028 ft-lb.	6.7	4.7
W68LY	Lyc. O-360-A	4	350 ft-lb	9.4	1.7
McCulloch		4-cyl., 2-cycle	91 ft-lb	8.8	—

NOTE: Q-Rating $= \left(\frac{\text{Rated H.P.}}{\text{Rated RPM}}\right)\left(\frac{16{,}500}{\pi}\right)$

OR

$$\frac{HP \times 5252}{RPM} = Q \text{ (TORQUE)}$$