

AEROSPACE INFORMATION REPORT

SAE AIR1087

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Issued	1969-01
Reaffirmed	1994-02
Revised	1999-04

Aircraft Accessory Drag Torque During Engine Starts

FOREWORD

Changes in this revision are format/editorial only.

1. SCOPE:

This SAE Aerospace Information Report (AIR) presents a discussion of drag torques of engine accessories operating at cold temperature. Various conditions of temperature, speed, and acceleration rates are reviewed showing the merits of each method and recommending a standard test method.

1.1 Purpose:

The purpose of this Aerospace Information Report is to present a brief discussion of the drag torques of aircraft accessories operating at cold temperatures, specifically -65 °F. The parameters affecting the magnitude of torque at various speeds and acceleration rates are reviewed. In conclusion, this report establishes the difference in accessory torque characteristics that are obtained by various test procedures and recognizes the need for a standard test method.

2. REFERENCES:

There are no referenced publications specified herein.

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3. MAIN ENGINE STARTING SYSTEM ANALYSIS:

The presently accepted method for analyzing the starting system for gas turbine engines involves the generation, by the engine manufacturer, of an engine torque versus speed curve at various temperatures with related light-off, minimum assist and idle speeds. Using this engine torque curve, engine inertia and starter output torque versus speed curve the basic starting system can be analyzed using the following equation:

$$T_{\text{net}} = I_T \times \alpha$$

where,

T_{net} = net accelerating torque by the algebraic addition of engine torque and starter output torque at a common shaft.

I_T = summation of engine and starter inertia reflected at a common shaft.

α = resultant rate of acceleration of a common shaft.

By taking small increments of the net torque versus speed curve the average acceleration rate and time for each increment can be used to determine total time to engine idle.

In many cases this method of calculation provides the information necessary to complete the starting system analysis. If a more accurate analysis is required the effects of engine driven aircraft accessory gearbox loads can be included. This is accomplished by determining the steady-state accessory torque versus speed and algebraically adding this torque magnitude to the engine torque curve and adding the accessory inertia to the total system inertia. This method is satisfactory for normal accessory loads and relatively large engines. At temperatures below -40 °F, and in particular -65 °F, the loads from hydraulic devices and gearboxes can be increased many times because of the large increase in oil viscosity at these temperatures.

The engine and accessory starting system analysis at -65 °F is more involved because of increased and transient torque loading due to large changes in oil viscosity in the engine, gearbox and hydraulic accessories. To make a detailed analysis of the -65 °F system, special attention must be given to these loads before they can be added to the steady-state engine torque curve in the start calculation.

4. COLD START DRAG TORQUE ANALYSIS FOR ACCESSORY LOADS:

4.1 Basic Parameters:

The torques required to rotate hydraulic devices and gearboxes during an engine acceleration are influenced by the following parameters:

- (1) Inertia (I) of the rotating mass

$$\text{Equation 1: } T = I \alpha$$

- (2) Windage of rotating parts in air

$$\text{Equation 2: } T = K \times \text{function of speed } (\omega)$$

- (3) Pumping of fluid

$$\text{Equation 3: } T = \text{Displacement} \times \Delta \text{ pressure}$$

Δ pressure head is also a function of the oil viscosity (μ) and speed (ω).

- (4) Churning and shearing of the oil

$$\text{Equation 4: } T = \text{Geometrical constant} \times \text{a function of speed } (\omega) \text{ and viscosity } (\mu).$$

- (5) Friction

$$\text{Equation 5: } T = \text{a function of contact forces (F) and the coefficient of friction (f)}.$$

Total loads from Items 1 (inertia) and 2 (windage) are easily determined by calculations and steady-state test measurements and impose no particular problems. On large engines the torques from Items 1 and 2 are small compared to total system torque during the engine start cycle and will not be considered in this study. The torque due to friction will not be considered as a separate item in this study.

As noted in equation (3), the torque loads are a function of displacement and pressure head. This displacement per revolution (D/rev.) can be considered a constant for a given fixed displacement unit during the start cycle which results in a flow rate proportional with speed (ω). The total hydraulic head is determined by system operating pressure and flow rate and decreases with increasing fluid viscosity (μ). Pumping pressure is normally a constant designed pressure, but at very cold conditions can be greatly increased. Special considerations would be necessary to determine the pumping torque of variable displacement hydraulic pumps which are pressure compensated and destroked for low flow rates.

4.1 (Continued):

In equation (4), the torque loading is determined by the geometrical size of the unit which can be considered as a constant depicted as an area (A) in square inches, the speed of the individual components shearing the oil (ω) in feet per second, and the dynamic viscosity (μ) in slugs per feet per second.

The following illustrates how equations (3) and (4) can be generalized and combined by separating constants and variables which affect the system starting torque:

$$\begin{aligned} (3) \quad T &= D/\text{revolutions} \times \Delta \text{ pressure} \\ D/\text{revolutions} &= \text{constant } (K_1) \\ \Delta P &= f(\mu, \omega) \\ T &= K_1 \times f(\mu, \omega) \end{aligned}$$

$$(4) \quad T = K_2 \times f(\mu, \omega) \quad K_2 = \text{Geometrical Constant}$$

Equations (3) and (4) can be combined to yield:

$$\text{Equation (5)} \quad T = K_3 \times f(\mu, \omega)$$

Over normal operating temperature ranges, viscosity can be considered a constant and the steady-state torque would vary as a function of speed.

4.2 Factors Affecting Fluid Viscosity:

As seen on the graph of viscosity versus temperature (Figure 10) for a typical aerospace oil, the fluid viscosity changes approximately 50 to 1 between 0 °F and -65 °F. Correspondingly the resistance will vary considerably with small temperature changes in this range. This phenomenon affects overall torque characteristics at low temperatures since local viscosity changes as a function of unit activity.

After the initial revolution of a unit which has been cold soaked to -65 °F, the fluid in contact with the bearings, gears, pistons, and other reciprocating and rotating parts in the unit is subjected to motion and shearing. The work (W) done on the fluid through shearing will in turn heat the oil adjacent to the moving surfaces. This causes a substantial change in local viscosity and therefore a change of work input for the next increment of time.

During a cold start cycle, the oil temperature within a unit at any time after initial start will vary significantly within the unit itself. A rapid temperature rise would be noted adjacent to the moving surfaces which are imparting the most work to the oil, in contrast to a slow temperature rise in a relatively stagnant sump area.

4.3 Effect of Work on Fluid Temperature:

It is assumed that any work absorbed by an accessory above its normal steady-state output energy is transferred into heating of its internal fluid.

Therefore:

Dynamic viscosity (μ) is a function of its temperature (T_e)

Equation (6): $\mu = f(T_e)$

The fluid temperature (T_e) is a function of the work (W) imparted to the fluid by moving parts.

Equation (7): $T_e = f(W)$

Work is equal to a force times distance and in this case

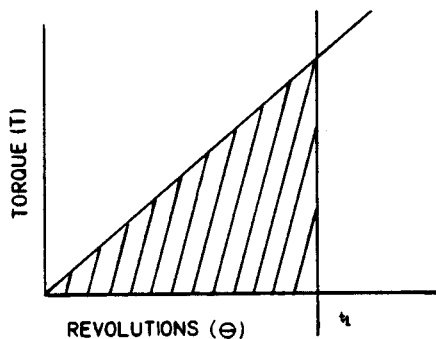
Equation (8) = $W = T \times \theta$

Where, T = torque (to overcome viscous friction)

θ = revolutions or radians

It can be generalized that the fluid temperature or viscosity at any time (t) after start initiation is dependent upon the summation of the torque transmitted to the unit times the number of revolutions to time (t).

In Figure 1 below, the transmitted torque is plotted against revolutions (θ).



(Note: The work accomplished to time (t), is the shaded area under the curve.)

FIGURE 1. Typical Torque Versus Revolutions

4.4 Acceleration Rate Varies the Work Accomplished:

Assume an accessory is accelerated at two different uniform rates α_1 , and α_2 from zero to a speed ω_1 , and there is no change in the fluid temperature, the torque versus speed curve is as shown in Figure 2. For this same accessory, torque versus revolutions is shown in Figure 3 where the areas shown as A_1 and A_2 represent the total work done on the unit where $A_1 > A_2$ and $W_1 > W_2$.

It is noted that with different amounts of work done on the oil, the temperature of the oil at speed ω_1 for the acceleration with α_1 is higher than with α_2 .

As an example, assume:

$$\alpha_1 = 10 \text{ radians/second}^2$$

$$\alpha_2 = 20 \text{ radians/second}^2$$

$$\omega_1 = 50 \text{ radians/second}$$

from,

$$\omega = \alpha \times t$$

$$t_1 = 5 \text{ seconds}$$

$$t_2 = 2.5 \text{ seconds}$$

substituting

$$\theta = 1/2 \alpha \times t^2$$

$$\theta_1 = 125 \text{ radians}$$

$$\theta_2 = 62.5 \text{ radians}$$

As shown in Figure 3, the area under the respective curves would be $A_1 = 2A_2$.

Therefore, the work accomplished with α_1 would be twice the work accomplished with α_2 . If this generalized example is carried a step further, and it is assumed that the accessory temperature is uniform at -65 °F, and W_1 raises the oil temperature 20 °F and W_2 raises it 10 °F it can be seen from Figure 10 that the viscosity $\mu_1 = .06 \frac{\text{slugs}}{\text{feet second}}$ and $\mu_2 = .12 \frac{\text{slugs}}{\text{feet second}}$

$$W_1 = 2W_2$$

$$\mu_1 = 1/2 \mu_2$$

It can now be stated that if all the work is transferred into heating up the oil, the torque at speed μ_1 for run (1) is not equal to the torque for run (2) as originally stated.