Society of Automotive Engineers, Inc.
TWO PENNSYLVANIA PLAZA, NEW YORK, N.Y. 10001

AEROSPACE INFORMATION REPORT

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AIRCRAFT ACCESSORY DRAG TORQUE DURING ENGINE STARTS

1. PURPOSE

The purpose of this Aerospace Information Report is to present a brief discussion of the drag torques caused by accessory units while starting under -65 F conditions. The various parameters affecting the magnitude of the drag torque at various speeds and acceleration rates are reviewed. In conclusion, this report establishes the difference in accessory drag torque characteristics that are obtained by various test procedures, and thereby this difference warrants the need for the establishment of a standard test method.

2. TURBINE ENGINE STARTING SYSTEM ANALYSIS

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

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

where, T_{net} = net accelerating torque by the algebraic addition of the engine torque and the starter steady state output torque,

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

α = resultant rate of acceleration of a common shaft.

By taking small increments of the torque vs. speed curve, the average acceleration rate (α) and time (t) for each increment can be calculated. A summation of the time for each increment can be used to determine the resulting time to engine idle.

In many cases, the above basic method of calculation provides the necessary information to complete the starting system analysis. If the system analysis must be more refined, the aircraft engine accessory loads may be analyzed for their effect on the starting systems. Presently this is accomplished by determining the steady-state accessory drag vs. speed and adding this torque magnitude to the engine curve and adding the accessory inertia to the total system inertia. Again this method is satisfactory for normal accessory loads and with relatively large engines. This brings us to the main point of discussion in this paper. At temperatures below -40 F, and in particular -65 F, the normal accessory loads from hydraulic devices and gearboxes can be increased by a factor of many times because of the large change in oil viscosity at these temperatures.

The engine and accessory starting system analysis at -65 F becomes more involved because of the increased and variable torque loading due to high oil viscosity change in the engine, the gearbox, and hydraulic accessory devices. To make a refined analysis of the -65 F starting system, special attention must be given to this drag loading before it can be added to the steady state engine torque curve in the starting calculation.

3. COLD START DRAG TORQUE ANALYSIS FOR ACCESSORY LOADS

3.1 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
   Equation 1: $T = I \times \alpha$
2) Windage of rotating parts in air
   Equation 2: T = K x Function of Speed (ω)
3) Pumping of fluid
   Equation 3: T = Displacement x Δ pressure
   Δ pressure head is also a function of the oil viscosity (μ) and speed (ω)
4) Churning and shearing of the oil
   Equation 4: T = Geometrical constant x a function of speed (ω) and viscosity (μ)
5) Friction
   Equation 5: T = A function of contact forces (F) and the coefficient of friction (f)

Total loads caused by Items 1 (Inertia) and 2 (Windage) are easily determined by calculations and steady state test measurements and therefore pose no particular problems. On large engines the torques observed by these items are relatively very small because the accessory is accelerating at slow rates during the engine starting cycle and therefore need 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 Item 3, the torque loads are a function of displacement and pressure head. The displacement per revolution (D/Rev) can generally be considered a constant for a given fixed displacement unit during the start cycle, and therefore the fluid flow will vary directly with the unit

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speed (ω). The pumping head is determined by the hydraulic system operating pressure plus flow losses due to the fluid viscosity (μ) and the rate of flow. The pumping pressure is normally a constant designed pressure, but at very cold conditions this pressure 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 system flow.

In Item 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 (fps), and the dynamic viscosity of the fluid (slug/fps).

The following section illustrates how the equations for Items 3 and 4 can be generalized and combined by separating constants for a given unit and the variables which affect the start torque loading:

Item 3: $T = D/Rev \times \Delta p$

Equation 3: $D/Rev = \text{Constant } (K)$

$\Delta p = K \times f(\mu, \omega)$

$T = K \times f(\omega, \mu)$

Item 4: $T = K_{Geometry} \times f(\omega, \mu)$

Equation 4: K_{Geo} = A constant which can be likened to an area of a paddle wheel with radius (r) moving through a fluid.

Therefore with some liberties, Equations 3 and 4 can be combined:

Equation 5: $T = K \times f(\omega, \mu)$

NOTE: In normal operating temperature ranges, μ can be considered as a constant and the steady state torque would vary as a function of speed.

3.2 Factors Affecting Fluid Viscosity

As can be seen on the chart of (μ) dynamic viscosity vs. temperature (Fig. 10), the fluid viscosity changes approximately 6 to 1 between -65 F and -40 F. Correspondingly the resistance will vary considerably with a slight change in temperature within this range of temperature.

After the initial revolution of a unit which has been cold soaked at -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 by shearing in turn will heat the oil adjacent to the moving components.

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

3.3 Fluid Temperature Affected by Work

It can be assumed that any work (W) 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 the 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

θ = Revolutions or Radians

It is thereby generalized that the fluid temperature or viscosity at any time (t_1) after start initiation is dependent upon the summation of the torque transmitted to the unit times the number of revolutions to time (t_1).

In Fig. 1 below, the torque (T) transmitted is plotted vs. revolutions (θ).

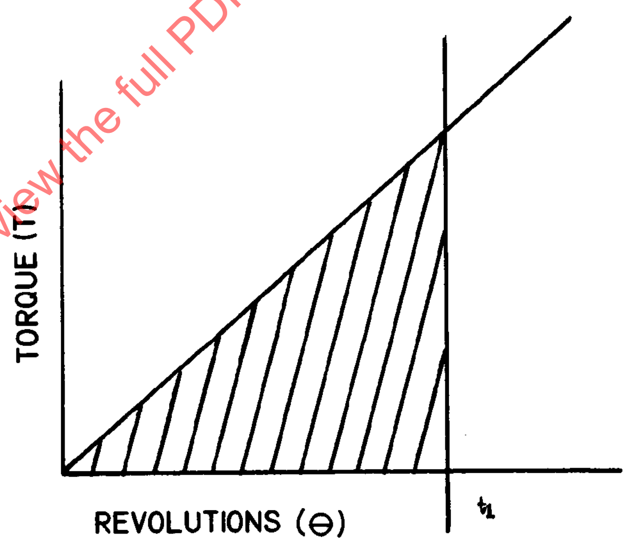

FIGURE 1

The work accomplished to time (t_1) is the shaded area under the curve.

3.4 Starting Acceleration Rate Varies the Amount of Work Accomplished

Assume an accessory is accelerated at two different uniform rates (α_1, α_2) from zero to a speed ω_1 and there is no change in the fluid temperature, the torque vs. speed curve is as depicted in Fig. 2. Then the torque vs. revolutions is shown in Fig. 3, and the area shown as A_1 and A_2 represents the total work (W) accomplished on the unit when $A_1 > A_2$ and $W_1 > W_2$.

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

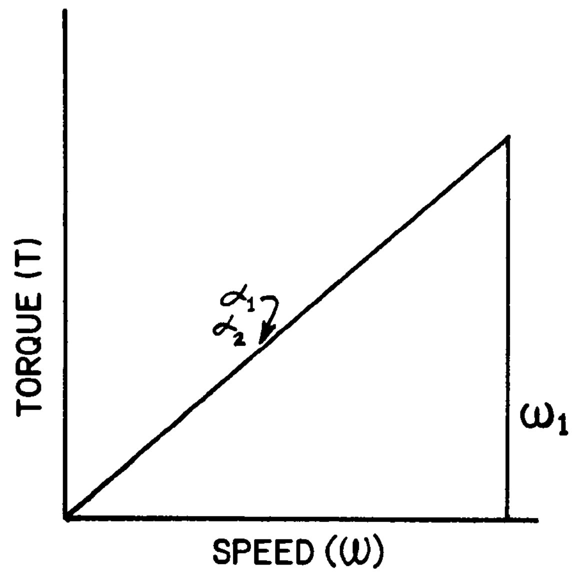

FIGURE 2

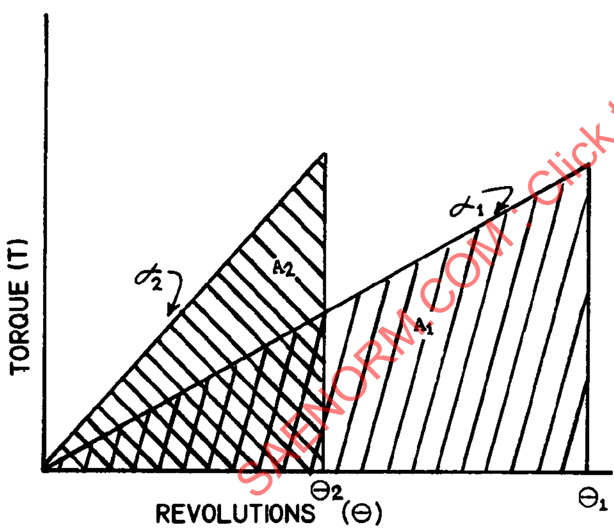

FIGURE 3

Following is an example using actual magnitudes: Assume,

α_1 = 10 radians/sec^2 (95 rps)
α_2 = 20 radians/sec^2 (190 rps)
ω_1 = 50 radians/sec (475 rpm)

From the equation,

$\omega = \alpha \times t$
t_1 = 5 sec
t_2 = 2.5 sec

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

θ_1 = 125 radians (20 rev.)
θ_2 = 62.5 radians (10 rev)

As represented by Fig. 3, the area under the respective curves would be as follows:

$$A_1 = 2\ A_2$$

Therefore, the work accomplished on the run with α_1 would be two times the work accomplished with α_2.

If the generalized example is carried a step further, and it is assumed that the temperature at t_0 = -65 F, and W_1 raises the oil temperature 20 F and W_2 raises the temperature 10 F, it can thus be seen from the viscosity curve (Fig. 10) that the μ_1 is .06 slugs/fps and μ_2 is .12 slugs/fps.

$$W_1 = 2\ W_2$$

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

Thereby it can be stated that if all the work is transferred into heating up of the oil, the torque at speed ω_1 for run (1) will not be equal to the torque for run (2) as stated in the original premise.

But, $T_1 < T_2$

Cold start torque test results of hydraulic or lubricated accessories have shown that the change in fluid viscosity is a dominate factor in determining the shape of the accessory torque vs. speed curve during the accelerated start such that the slope of the curve is not as shown in Fig. 2, but more as shown below in Fig. 4.

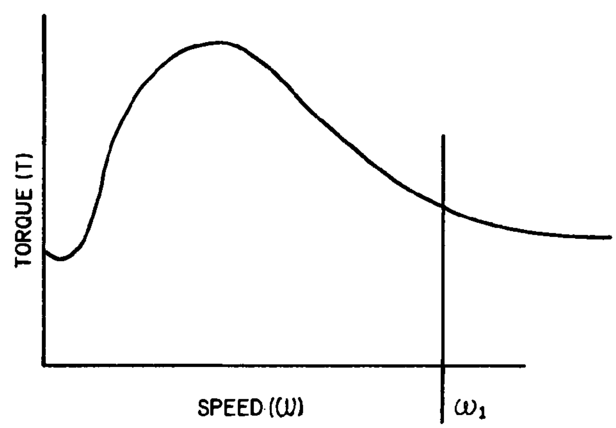

FIGURE 4

This shape curve shows that the decreasing fluid viscosity has a greater affect on the absolute torque than the increasing speed.

Not previously mentioned, but also affecting the cold start torque in a similar manner as the oil viscosity changes, would be the change in torque required by bearings and other components which may be operating with an interference or tight fit at the -65 F condition and which thereafter loosen as the temperature increases.

3.5 Generalized Cold Start Torque Equations for Accessories

Equation 5 noted in section 3.1 combined the pumping and shearing drag torques into one equation and grouped the independent constants.

Equation 5: Torque at any speed (T) = K x function of ω and μ

$$T = K \times f(\omega, \mu)$$

The equation in this form is not very useful because it is very difficult to determine or measure the fluid temperature at the critical moving component areas during the start cycle. Therefore it is necessary to attempt to reflect this viscosity variable as a measurable function.

Equations 6, 7, and 8 show how the viscosity, μ, is a function of work (W) or (T x θ).

Equation 9: $\mu = f\ \Sigma (T \times \theta)$

Therefore it can be stated that the torque at any time or speed during the start cycle is a function of the speed and of the work accomplished on the fluid up to the speed in question.

Equation 10: $T = K \times f\ [\omega, f\ \Sigma (T \times \theta)]$

4. TEST METHODS FOR DETERMINING DRAG TORQUE ON A STARTING SYSTEM

In the past, various accessory manufacturers have utilized different methods in determining the torque required to accelerate their product under certain specific conditions. Some of these methods are described in the following paragraphs.

4.1 Steady State Torque and Polar Moment of Inertia

In this method, the component to be tested is conditioned in the required environment, and then driven to a predetermined speed. The torque to drive the component is then measured while the acceleration rate is zero. From a series of points derived in this manner, a steady state curve of torque versus speed is obtained. The polar moment of inertia of the component can be determined through testing and/or calculations. When this information is obtained, the accessory drag torque is added to the engine drag curve, and the polar moment of inertia is added to that of the system.

This method is greatly influenced by time lapsed from start to when the torque measurement is made. For instance, on a -65 F test the hydraulic and/or lubricating fluids are very viscous, but warm up quite rapidly as work is being done on them. Therefore, once a constant speed is maintained (acceleration = 0), the torque will continue to decrease with time as the fluids become less viscous. The problem of an accurate reading is therefore two-fold:

1) The torque measurement is made after too much work has been put into the component, and is therefore too low a reading (possible at low speeds).
2) The torque measurement is made when not enough work has been put into the component. (This would be possible at speeds where normal engine acceleration rates would be slower than that rate used on the test.)

This test method is considered acceptable at normal temperatures where the change in viscosity of the oil is not as marked. At normal temperatures the steady state torque is less dependent on the time lapse for an accurate torque measurement.

4.2 Uniform Acceleration Rate

In this method the component is accelerated at a fixed rate of acceleration during which time the accessory torque is continually monitored. 150 rps is normally accepted as an average rate to use on the J-57, TF-30, and J-79 engines. As an average rate, it is a little high overall, and would result in a conservative system drag torque.

This test method is satisfactory for applications where the engine drag is very large compared to the accessories. However, on a small engine, a more accurate method of determining the accessory drag may be desired.

The polar moment of inertia of the system can also be determined, and an adjustment made in the torque to account for acceleration of the masses. This would result in a steady state torque curve for that given acceleration. However, a different acceleration rate would mean a different level of energy being put into the fluid and a different rate of change of fluid viscosity, which directly affects the torque absorption of the component.

4.3 Variable Acceleration Rate

Another possible method of measuring the torque absorption of a component is to operate with a variable acceleration rate. The idea here is to simulate the actual engine starting acceleration. This is an improvement over the uniform acceleration rate, as it more closely simulates actual operational start conditions.

It is, of course, necessary in this approach to determine the approximate rate of acceleration of the engine. The acceleration profile would then be segmented into constant rates between certain predetermined speeds. An example of this is shown for an accessory in Fig. 11, which also dramatizes the difference when run at a variable rate of acceleration as compared to a constant rate of acceleration.

The problem in this test method arises in that each change in temperature or starting mode requires a new set of variable acceleration rates. The problem is further complicated as the rate of acceleration of the engine is dependent upon the drag characteristics of the accessories.

4.4 Analytical Interpolation of a Constant Acceleration to Variable Acceleration

A fourth method of torque absorption measurement of interest utilizes the data obtained from accelerating the accessory at a constant rate of acceleration. From this information, a factor can be derived which enables a calculation to be made on the accessory, giving its drag torque at any desired rate of acceleration, constant or variable. The following is a discussion on this method.

As established in Section 3, $T = K\ f(\omega,\ W)$.

Transposing this relationship, $K = \dfrac{T}{f(\omega,\ W)}$

To simplify this equation for use in this study, the constant (K) will be replaced with a dependent variable (Y) which is a function of ω and W such to allow the equation to be rewritten as follows:

$$Y = \frac{T}{\omega \times W}$$

, where Y is a variable unknown in terms of ω and W.

To make this equation useful, the value of (Y) can be determined empirically at specific points during testing as shown in the following text.

From the constant acceleration runs, the torque (T) at a given speed (ω) are known. However, the work done (W) is not known. This must first be established before (Y) can be resolved.

A typical torque versus speed curve for an accessory conditioned to -65 F at a constant acceleration is shown in Fig. 5.

From this data, a plot of torque versus revolutions can be obtained, as shown in Fig. 6. The total area under the curve represents the work (W) accomplished by the accessory.

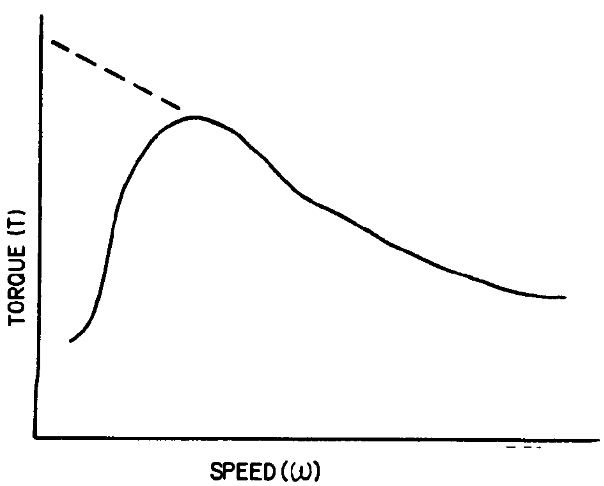

FIGURE 5

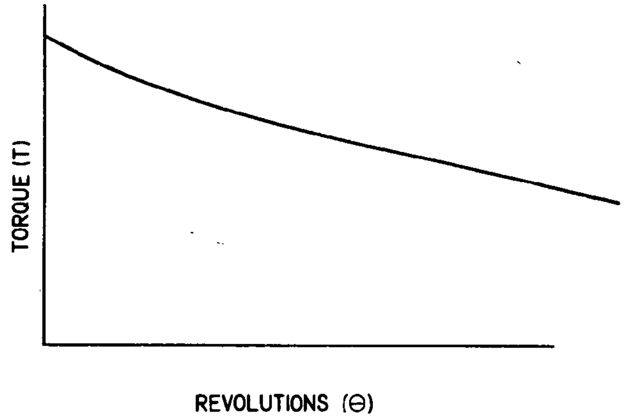

FIGURE 6

A series of values for (Y) can now be obtained in terms of the area under the curve (W), torque (T), and speed (ω).

Example #1: Determine the value of (Y) at 500 rpm when the accessory was accelerated at 150 rps. From Fig. 5, assume that T = 400 in.-lb at 500 rpm. The number of revolutions of the accessory (θ) equals the average speed multiplied by the time element, or:

$$\left(\frac{500}{2}\right)\ \left(\frac{500}{150}\right)\ \left(\frac{1}{60}\right) = 13.9 \text{ revolutions}$$

From Fig. 6, the area under the curve at 13.9 revolutions can now be computed in square inches (or any other convenient terms). In this example, assume area is 1.309 square inches. A value for Y can now be computed in terms of T, ω, and W at a specific point:

$$Y = \frac{T}{\omega W}$$

$$T = 400,\quad \omega = 500,\quad W = 1.309$$

$$Y = \frac{400}{500\ (1.309)} = 0.611 \text{ units}$$

A series of similar calculations, solving for Y, will generate new values for Y at different speeds or total revolutions. (Y) can then be plotted against total revolutions as a curve on log-paper, as in Fig. 7.

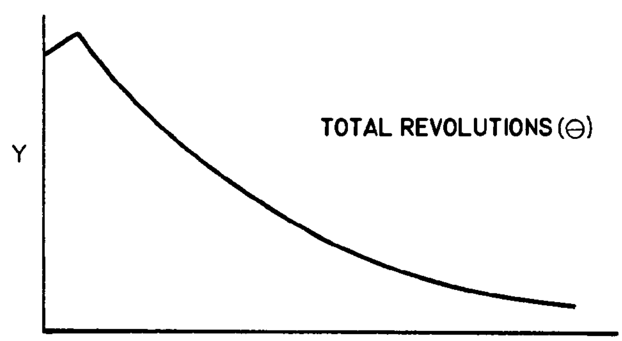

FIGURE 7

It becomes apparent at this point, that although the (Y) factor is a "catch-all" for many factors, variable and constant, its basic function is to describe the fluids viscosity versus temperature curve. Having established the (Y) factor, it is now possible to determine, with a higher degree of accuracy than in methods previously described, the actual component drag curve when installed on an aircraft.

The first step in establishing the desired accessory drag torque curve is to calculate an engine start, utilizing the known drag and start characteristics of the engine and starter. In this start, the accessory drag would be that obtained in the constant acceleration test, i.e., Fig. 5 drag curve.

When this is completed, the acceleration rate in 100 rpm increments and the number of accessory revolutions to that speed are calculated. Then by using the relationship $T = Y \cdot \omega \cdot W$, the torque for the new acceleration rate can be determined.

Example #2: In Example #1, it was determined that it took 13.9 revolutions to get to 500 rpm with a constant acceleration rate. In the engine start calculation above, let us assume that it takes 16 revolutions to get to 500 rpm.

Then, $T = Y \cdot \omega \cdot W$

From the torque vs. θ curve of Fig. 6, we find the work accomplished in 16 revolutions corresponds to an effective area of 1.32 square inches.

From the (Y) versus θ curve of Fig. 7, we find that Y equals 0.50 at 16 revolutions.

Therefore: $T = Y \cdot \omega \cdot W$

Where, $Y = 0.50$, $\omega = 500$, $W = 1.32$

$T = 0.50\ (500)\ (1.32) = 320$ in.-lb

Similar calculations at various speeds will give a torque value for that speed. This will then establish a new drag curve for the accessory at a variable acceleration rate as shown in Fig. 8.

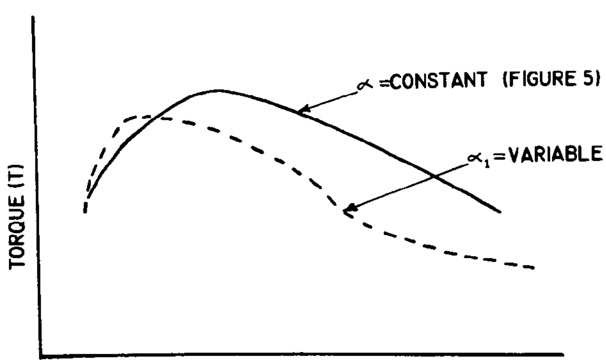

FIGURE 8

A second engine start should now be calculated, using the new accessory drag curve. The same procedure would be followed as above, with the result of a second calculated accessory drag curve as shown in Fig. 9.

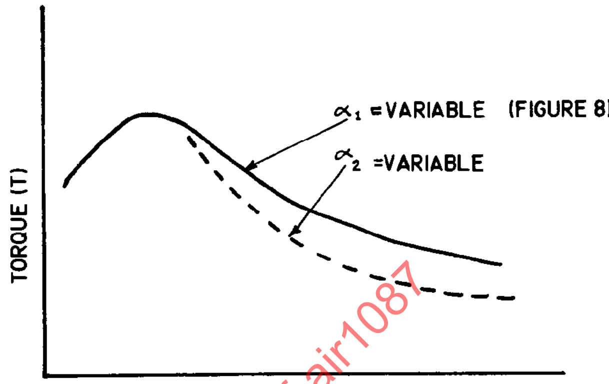

FIGURE 9

For greater accuracy of the drag curve, the procedure can be repeated again. However, a two-step calculation should give drag torque accuracies of about 3%. Fig. 11 shows actual test data (Curves #1, 2, 3) at a constant rate of acceleration vs. a calculated start (Curve #4) using the variable acceleration rate method just described.

When available, this entire series of calculations lends itself to a computer program, increasing the speed and accuracy of the computation while eliminating most of the mechanics.

It should be noted that it may be necessary to perform a constant acceleration test run and a set of calculations for each temperature considered (i.e. -65, -40, and -25 F).

4.5 Pneumatic Powered Acceleration Rate

Testing accomplished utilizing a pneumatic starter as a source for accelerating the accessory and simulated engine inertia at a variable rate has been found to be a convenient method of observing the effects of engine acceleration rates and hydraulic transients on the torque magnitude. In these tests, a pneumatic starter was the power source, and it was coupled to a flywheel simulating the aircraft engine inertia. The flywheel in turn was connected to the accessory. The air supply pressure to the starter was initially programmed such that the starter output torque vs. speed was representative of the net torque available in the starting system to accelerate the engine and accessories.

With this test method, the resultant acceleration rate will instantaneously be modified by changes in the magnitude of accessory loading during the start cycle because of the limited torque supply available from the starter. As shown in Fig. 11, Curve #1 was generated by a constant α of 150 rps, and Curve #4 was generated by calculation and later substantiated by the use of a pneumatic starter as the power source to be a good approximation.

5. EFFECT OF HYDRAULIC TRANSIENTS

At some speed during the start cycle range, the hydraulic accessories may exhibit a change in output characteristic which would be reflected as a transient torque spike on the system drag curve. This transient torque can be considerably higher than the steady state torque when the driving force is essentially an unlimited power supply. However, it is important to remember that the transient condition is an energy condition, and reacts differently with a limited power supply such as an aircraft starter system. Testing has born out that this transient energy can be taken from a rotating energy inertia source (such as the engine) in the form of revolutions per minute, in which case no transient spike would occur at all. In tests conducted of this type, the loss of revolutions by the simulated engine (inertia bed) could not be detected.

The following conclusions were drawn from actual test results.

A. The accessory hydraulic transient is an energy condition only, caused by the high accelerations or pressure change of components; it is not a steady state condition.

B. As long as starter torque in excess of the accessory drag torque is available (not considering the transient), the transient of itself cannot be responsible for an abortive start. The worst result of the transient would be to reverse momentarily acceleration of the engine during its peak spike duration (approximately .25 seconds). This would occur as energy for the transient condition is removed from the energy stored in the engine in the form of inertia for the transient period of time.

In analyzing a start condition, the torque value of the transient can be ignored as a spike on the torque load, and included as an increase in time to idle by the duration of the peak transient condition. (Approximately 0.25 seconds).

PREPARED BY
SUBCOMMITTEE AE-6A, STARTING SYSTEMS INFORMATION, OF
COMMITTEE AE-6, STARTING SYSTEMS