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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 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. 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 vs. 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 vs. speed curve the basic starting system can be analyzed using the following equation:

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

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 vs. 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

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2. (Continued):

included. This is accomplished by determining the steady-state accessory torque vs. 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.

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: $T = I \alpha$

(2) Windage of rotating parts in air

Equation 2: $T = K \times$ 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 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.

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3.1 (Continued):

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.

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:

(3) $T = D/rev. \times \Delta \text{ Pressure}$
$D/rev. = \text{constant } (K_1)$
$\Delta P = f(\mu, \omega)$
$T = K_1 \times f(\mu, \omega)$

(4) $T = K_2 \times f(\mu, \omega)$ K_2 = Geometrical Constant

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

Equation (5) $T = K_3 \times f(\mu, \omega)$

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

3.2 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.

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3.2 (Continued):

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.

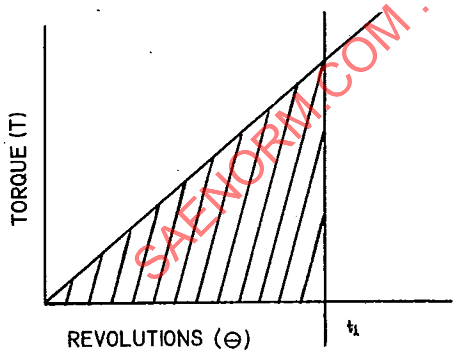
3.3 Effect of Work on Fluid Temperature: It is assumed that any work absorbed by an accessory above its normal steady-state output energy is transferred into heating of its internal fluid.

Therefore:

Dynamic viscosity (μ) is a function of its temperature (Te)
Equation (6): $\mu = f\ (Te)$
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 vs. Revolutions

3.4 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 $W_1 > W_2$.

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3.4 (Continued):

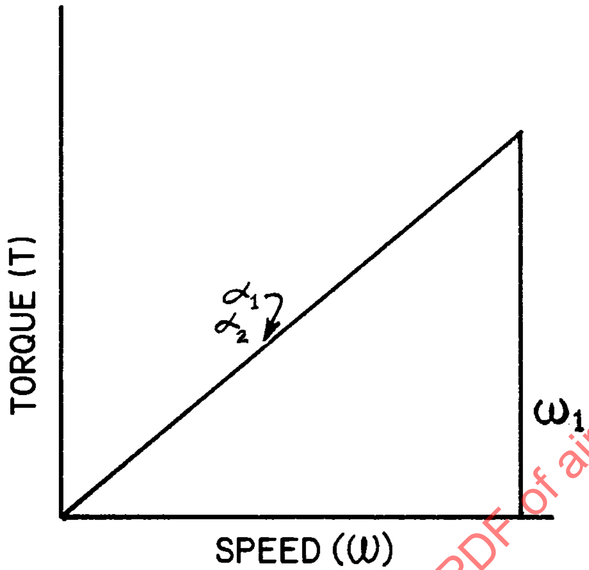

FIGURE 2. Torque vs. Speed

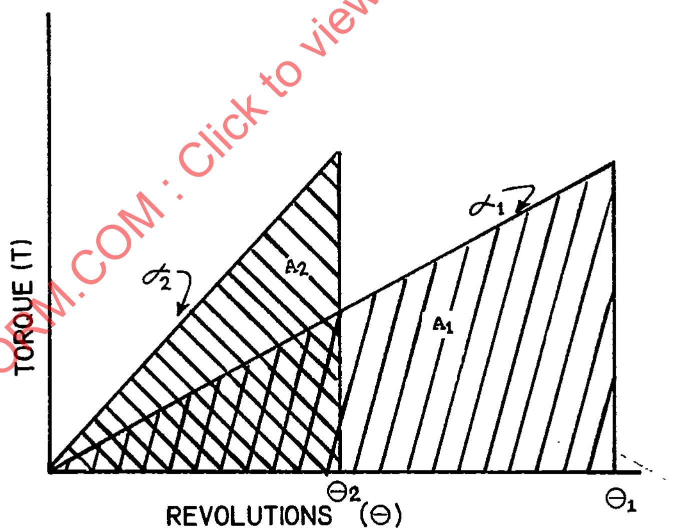

FIGURE 3. Torque vs. Revolutions for Two Acceleration Rates

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: α_1 = 10 radians/sec^2
α_2 = 20 radians/sec^2
ω_1 = 50 radians/sec

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3.4 (Continued):

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

substituting $\theta = 1/2\ \alpha \times t^2$
$\theta_1 = 125$ radians
$\theta_2 = 62.5$ 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 $\mu_1 = .06 \frac{\text{slugs}}{\text{ft. sec.}}$ and $\mu_2 = .12 \frac{\text{slugs}}{\text{ft. sec.}}$

$$W_1 = 2W_2$$
$$\mu_1 = 1/2\ \mu_2$$

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.

$$\text{But, } T_1 < T_2$$

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

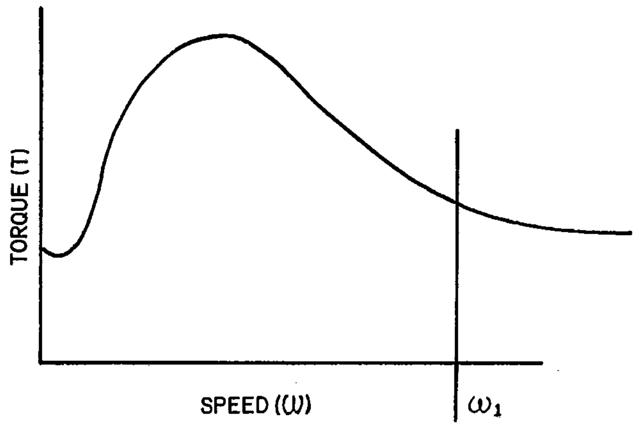

FIGURE 4. Torque vs. Speed

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3.4 (Continued):

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

The change in torque required by bearings and other components that operate with an interference or tight fit at -65°F condition and loosen as temperature increases also affects the cold start in a manner similar to oil viscosity changes.

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.

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

This equation is not very useful because it is very difficult to determine or measure fluid temperature at the critical moving surfaces during the start cycle. Therefore, it is necessary to reflect the viscosity variable as a measurable function. Equations (6), (7) and (8) showed how viscosity is a function of work (W) or (T x θ).

$$\text{Equation (9)} \quad \mu = f\,\Sigma\,(T \times \theta)$$

From this 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 done on the fluid up to the speed in question.

$$\text{Equation (10)} \quad T = K \times f\,[\omega,\ f\,\Sigma\,(T \times \theta)]$$

4. TEST METHODS FOR DETERMINING DRAG TORQUE:

4.1 Steady-State Torque and Polar Moment of Inertia: The component to be tested is conditioned in the required environment and then driven to a predetermined speed. The torque required 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 is determined through testing or calculations. The accessory drag torque is then added to the engine drag curve and the inertia is added to the system inertia.

This method is greatly influenced by the lapse of time between data point measurements. For instance, on a -65°F test the hydraulic and lubricating fluids are very viscous, but warm up rapidly as work is being done on them. Once the constant speed data point is reached (acceleration = 0) the torque continues to decrease with time as the fluids become less viscous. The problem of an accurate reading is influenced by:

1. The torque measurement is made after too much work is put into the component producing too low a value.

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4.1 (Continued):

2. The torque measurement is made before enough work is put into the component which produces too high a value. This is possible when actual engine acceleration rates are lower than used on the test.

This test method is considered acceptable at moderate temperatures where the change in oil viscosity is not large.

4.2 Uniform Acceleration Rate: The component is accelerated at a fixed rate during which time the accessory torque is continually monitored. This test method is satisfactory for applications where the engine drag is large compared to the accessories. However, on a small engine, a more accurate method of determining accessory drag may be desired.

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

4.3 Variable Acceleration Rate: Measure torque absorption of an accessory using a variable acceleration rate to more closely simulate actual engine conditions. This is an improvement over the other two methods as it more closely simulates actual operational start conditions.

It is necessary in this approach to determine the approximate engine acceleration profile. This acceleration profile would then be segmented into constant acceleration rates between predetermined speeds. An example of this is shown in Figure 11, which also dramatizes the difference between variable rate acceleration and constant rate acceleration.

An undesirable factor in this test method is each change in temperature or starting mode requires a new acceleration rate. This becomes even more complicated because the rate of acceleration is dependent upon the drag characteristics of the accessories.

4.4 Analytical Conversion of Constant Acceleration to Variable Acceleration: A fourth method of torque absorption measurement utilizes the data obtained from accelerating the accessory at a constant rate of acceleration. From these data a method can be derived which enables calculation of accessory drag torque at any desired rate of acceleration.

From Section 3, $T = K\, f\,(\omega, W)$

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

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4.4 (Continued):

To simplify this equation 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 written 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.

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

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

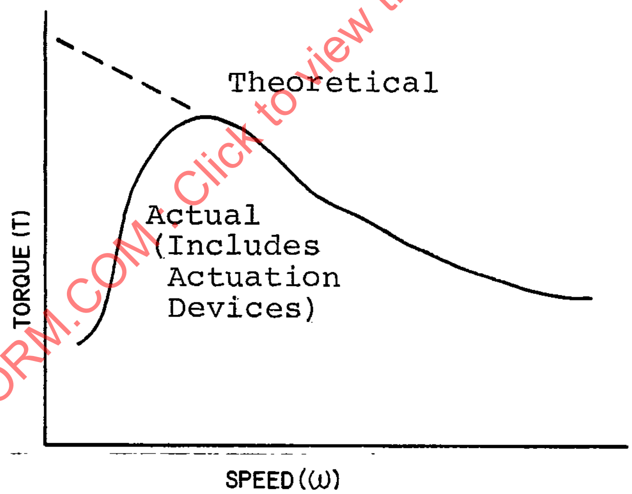

FIGURE 5. Typical Torque vs. Speed for Constant Acceleration

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

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4.4 (Continued):

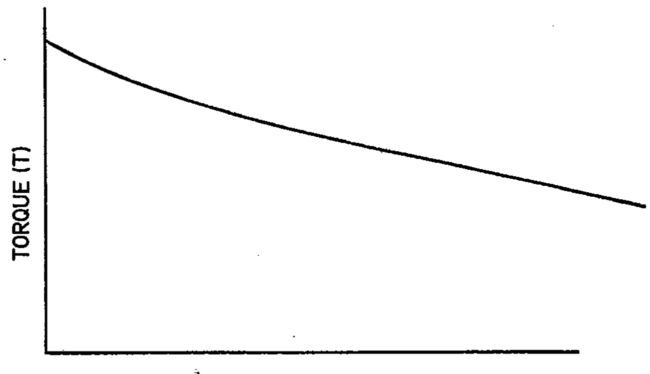

FIGURE 6. Torque vs. Revolutions

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 rpm/sec. 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,

$$\theta = \frac{\omega}{2} \times \frac{\omega}{\alpha}$$

$$= \frac{500}{2} \times \frac{500}{150} \times \frac{1}{60}$$

$$= 13.9 \text{ revolutions}$$

where: θ = revolutions
ω = velocity, rpm
α = acceleration, rpm/sec.
t = time = $\frac{\omega}{\alpha}$, sec.

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:

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

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

$$Y = \frac{400}{500}\ \frac{1}{(1.309)} = 0.611 \text{ units}$$

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 semi-log paper, as in Fig. 7.

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4.4 (Continued):

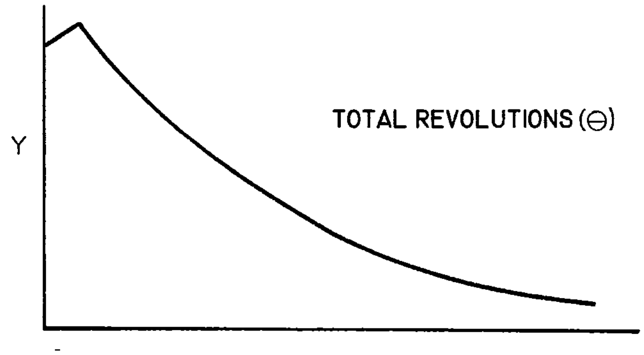

FIGURE 7. Y vs. Total Revolutions

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 fluid 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 it is 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 speed 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.

$$\text{Then, } T = Y \cdot \omega \cdot W$$

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.

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4.4 (Continued):

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.

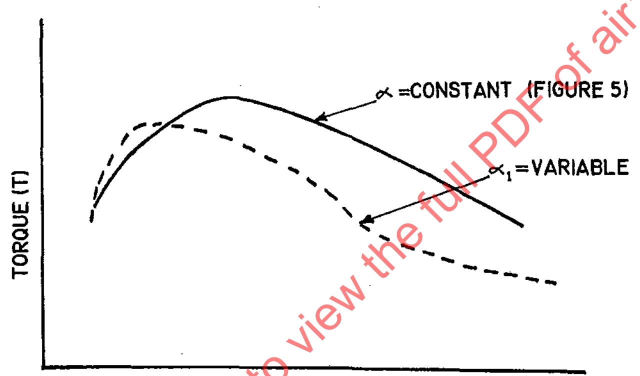

FIGURE 8. Torque vs. Speed for Constant and Variable Acceleration

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.

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