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EIGHTH PARTIAL REPORT ON THE PULSE-JET ENGINE

WHIRLING ARM

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ABSTRACT

A whirling arm, 26 feet in diameter, has been constructed and operated at the Naval Research Laboratory, Chesapeake Bay Annex. Two engines are mounted on the arm, one on each end of the single blade. The engines are started and operated by remote control. The whirling arm permits operation of the pulse-jets under free-flight conditions and provides a means for obtaining data of engine characteristics. The drag of the arm is determined from empirical data and a formula is derived for the drag value at any speed.

PROBLEM STATUS

This is an interim report on this problem; work is continuing.

AUTHORIZATION

NRL Problem P04-01R
NR 484-010

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EIGHTH PARTIAL REPORT ON THE PULSE-JET ENGINE WHIRLING ARM

INTRODUCTION

Prior to June 1949 the operating characteristics of pulse-jet engines designed and developed by the Thermodynamics Branch of the Naval Research Laboratory were evaluated by tests on static reaction stands. These tests proved satisfactory for the determination of such characteristics as maximum and minimum fuel range, static thrust, frequency of the pulse-jet cycle, and starting characteristics.

Most of the practical applications of the pulse-jet engine require that it be under free-flight conditions, and a true evaluation of the engine requires that it be tested while in free flight. For obvious reasons it is impractical to conduct studies over extended time intervals of the available thrust, fuel rate to the engine, vane life, and tube life for a pulse-jet engine while in free flight. The whirling arm presents itself as a practical solution to the problem. Mounted on a whirling arm the engine may be operated under velocity conditions; and measurements of the fuel rate, speed, and frequency of the engine can be easily obtained. By the use of a dynamometer, prony brake, or similar torque-measuring device, the surplus thrust[1] may be determined at any speed within range of the engine. Comparisons of the life of various engine bodies and vanes may be made and the effects of air scoops, cowls, orifice plates, and ducts may be evaluated.

DESIGN AND CONSTRUCTION

The Problem of Centrifugal Force

The whirling arm constructed at Chesapeake Bay Annex was designed for an engine speed (at the tip of the whirling-arm rotor) of 250 miles per hour. An ideal design of a whirling arm would have a rotor diameter of infinity; in which case there would be no centrifugal force to cause an increase in the fuel rate to the engine or to distort the engine. Therefore, it is advisable to design for the largest diameter that is economically practical. The whirling arm at CBA was designed to carry engines 6 inches in diameter which weigh 12 pounds each. Even with the 26-foot-diameter rotor used for the whirling arm, the centrifugal force is 317 G at 250 miles per hour tip speed. Therefore, the 12-pound engine exerts a centrifugal force of ca. 3800 pounds on the rotor in addition to the centrifugal force exerted by the rotor's own weight. Figure 1 indicates the magnitude of the centrifugal force at various tip speeds when a 26-foot rotor is used.

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[1] The term surplus thrust is construed to mean the gross thrust of the engine minus the drag and mechanical friction of the arm and the internal and external drag of the engine.

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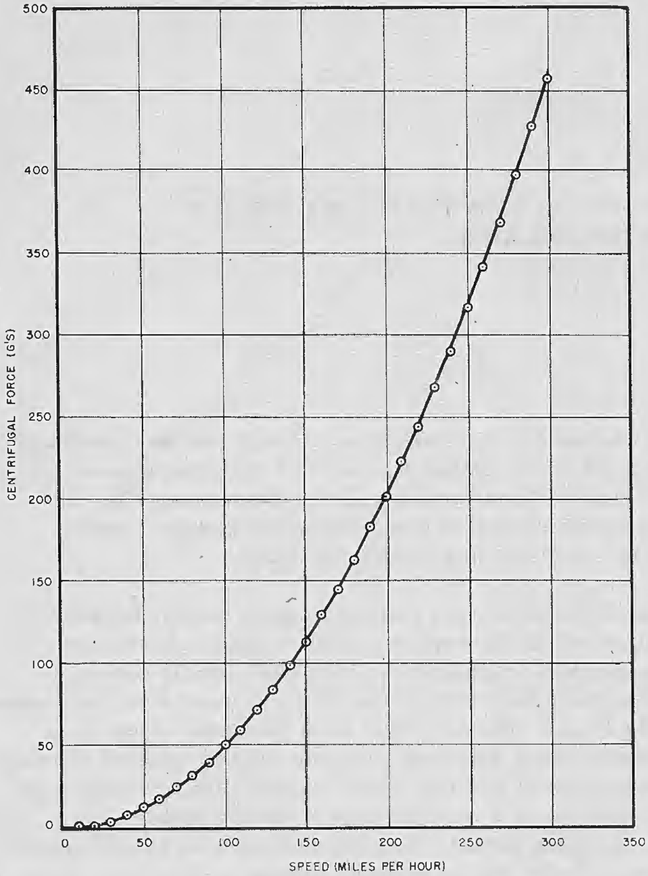

Figure 1 - Effect of tip speed on centrifugal force

Fuel-Rate Control

The centrifugal force acting on the column of fuel in the fuel line through the rotor increases the fuel pressure and fuel rate as the engine accelerates. When the fuel rate exceeds the operating range of the engine it will cause the engine to "rich out." One method of controlling the fuel rate is to use a needle valve in the fuel line near the engine. The valve is mounted in a horizontal position and is free to move radially, parallel to the longitudinal axis of the rotor, but constrained from moving in any other direction by the cylindrical guide sleeve in which it is mounted (Figure 2). With the weight of the needle equal to one-half the weight of the column of fuel, the centrifugal force on each is equal. As the fuel rate tends to increase, the needle moves closer to the valve seat causing a constriction which maintains the fuel rate at the preset value.

Theoretically, this method of control is ideal, but in practice several difficulties arise. At a tip speed of 250 miles per hour, for instance, any error in the weight of the needle is multiplied by 317. It is also difficult to keep the needle from binding in its cylindrical guide.

Another method of controlling the fuel rate is to block off the fuel line at any point along the rotor and drill a small orifice through the block. The size of the orifice should be such that it allows the engine to operate statically but will constrict the fuel rate to the lower part of its range. As the engine speed and centrifugal force increases, the fuel flow increases; but the orifice provides enough constriction to keep the fuel rate within the range of the engine. There are no moving parts required for this type of fuel control and once the orifice size is properly set, there are no further adjustment or maintenance problems for any engine which will operate statically below the preset fuel rate.

This second method has one serious disadvantage; during the acceleration period, the fuel rate indicated by the flow meter is not an accurate indication of the fuel rate at the engine fuel jets. After the engine has reached its operating speed and equilibrium has been established, the flow meter will indicate the true fuel rate to the engine. The fuel range for the 6-inch convergent-divergent engines and the 6-inch straight engines, used on the whirling arm at CBA, is approximately 90 to 150 pounds per hour. An orifice of 0.032 inch in diameter is installed in each fuel line near the hub of the whirling arm. A fuel pressure of 125 pounds per square inch is used.

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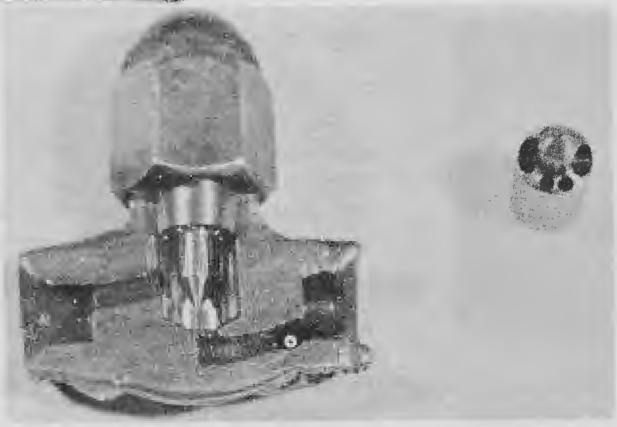

Figure 2 - Fuel flow regulator valve

Engine Distortion

Several of the problems of engine distortion under centrifugal force can be obviated by developing a high-thrust engine of the shortest length and smallest diameter possible, and constructing the engine of strong, light-weight materials. Proper design of the bracket which holds the engine to the tip of the rotor also helps to prevent engine distortion. Distortion of the circular cross section of the engine into an ellipse, with the major axis coincident with the axis of the rotor, is the major problem with this engine design. Figure 3 shows a 6-inch convergent-divergent engine mounted on the rotor tip with an antideformation ring to help prevent ovalization of the engine cross section. The use of antideformation rings has improved engine life over 200 percent. The drag added by the antideformation rings decreases the maximum speed attainable by about two percent. Figure 4 shows an engine failure due to cross-section distortion.

Figure 3 - Engine mounted on rotor tip

Independent Fuel and Air Control

It is possible to run a single fuel line through the shaft of the whirling-arm hub, using a single rotating joint, and then divide the fuel line to operate both engines as shown in Figure 5. However, with this arrangement a single fuel flowmeter must be used; and it is impossible to measure the fuel rate of each engine when both engines are operated simultaneously. An attempt to start the second engine may decrease the fuel rate to the first engine so that it will "lean out." For these reasons the whirling-arm installation at Chesapeake Bay Annex has two individual fuel lines with a fuel flowmeter in each. In order to use individual fuel lines the rotating joints at the rotor hub must be coaxial. This may be accomplished by running one line within the other or by running one line up through the pedestal shaft to one engine and running the other line from above to the second engine. The latter method is used in the Chesapeake Bay Annex whirling arm because of the limited space within the pedestal shaft. Individual control of the starting air to each engine is accomplished by running a single air line and using a solenoid valve in each branch running to the engine. One fuel line is run within the air line so that its rotating joint may be coaxial with the air-line rotating joint. Solenoid valves are also used in each fuel line. Figure 5 shows the arrangement of the solenoid valves on the top of the pedestal.

[redacted]

Figure 4 - Engine failure

Electrical Controls

The use of solenoid valves for on-off switching of the fuel and starting air provides a means for remote control of the engines. The power for the solenoid valves and the spark coils is run through slip rings mounted on the pedestal shaft as shown in Figure 6. All the electrical units are operated from the control cab which is located next to the whirling-arm pit. Figure 7 shows the arrangement of the vibrometer for measuring engine frequency, fuel flowmeters, starting-air pressure gauge, tachometer, prony brake control and the switches for operating the fuel and air solenoid valves. The whirling arm (Figure 8) is located in a pit, seven feet deep and forty feet in diameter, for the safety of the personnel who work with it. Remote control of the whirling arm and engines is desirable for convenience as well as for safety.

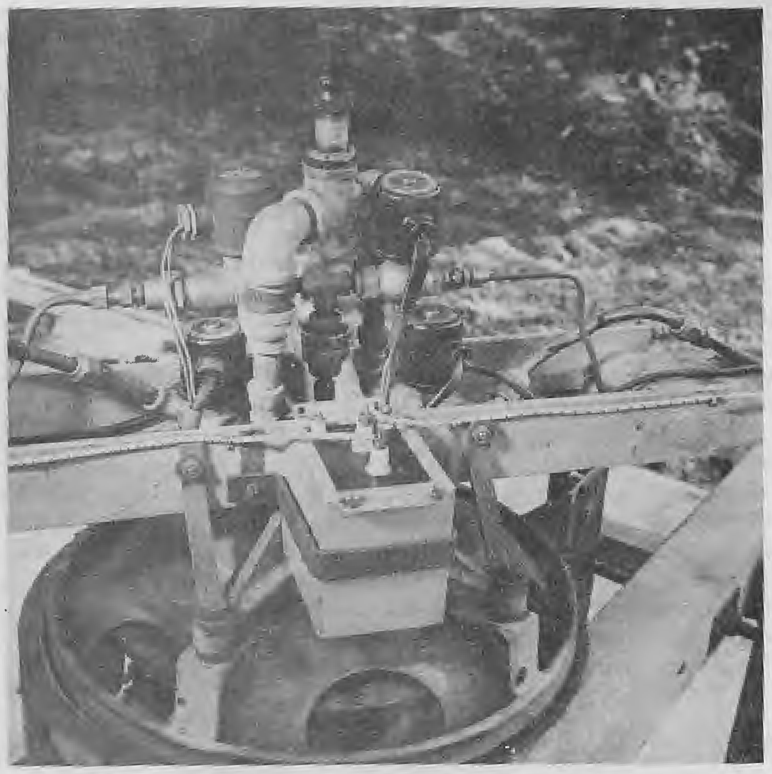

Figure 5 - Arrangement of controls on whirling arm pedestal

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Figure 6 - Slip rings on pedestal shaft

DETERMINATION OF DRAG

As previously stated, the prony brake measures the surplus thrust of the engine at any tip speed within its range. The gross thrust is equal to the sum of the surplus thrust, plus the internal and external drag of the engine, plus the drag and mechanical friction of the arm.

The mechanical friction of the whirling arm is determined by wrapping a light cable around the brake drum, running the cable over a pulley of known friction and moment of inertia, and adding weights to the end of the cable. Weights are added until the whirling arm begins to rotate very slowly with virtually no acceleration. Then, the weights added, minus the frictional force of the pulley, multiplied by the radius of the brake drum, is the frictional torque of the whirling arm.

The moment of inertia of the arm plus engines was determined experimentally by the weight-cable-pulley system. A weight was added to the friction-force weight with the arm held by a restraining electrical cable. The cable was cut and the arm was allowed to accelerate under the influence of the torque generated by the weight acting upon the drum. The time of the weight drop and the time of the pickup sweep immediately after the drop were obtained photographically on a Consolidated Recording Oscillograph.

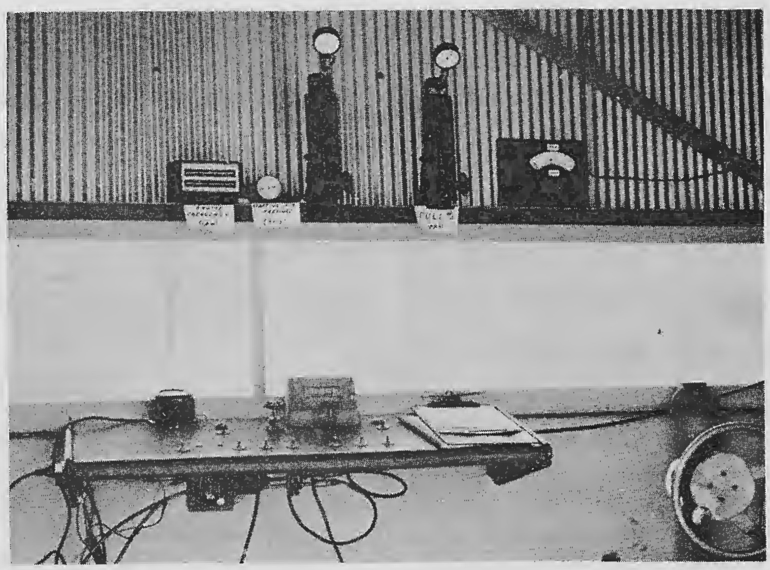

Figure 7 - Inside of control cab

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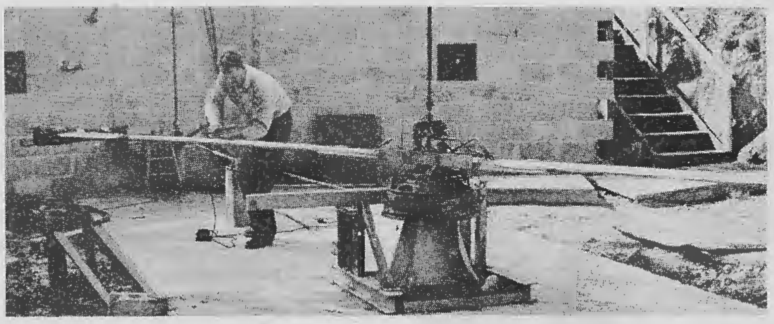

Figure 8 - Whirling arm installation

One recording galvanometer was hooked electrically to the restraining wire and the insulated plate upon which the weight dropped. Another galvanometer was connected in series with two magnetic-sweep coils placed 0.3345 radian apart under the magnetized arms.

In the subsequent discussion, the following notation will be used:

a = empirical constant (0.896 ft-lb-sec$^{\alpha}$)

α = numerical constant (2.35)

C = effective moment of inertia of external weights and pulleys referred to R (lb-ft^2)

g = acceleration of gravity (32.2 ft/sec^2)

I = moment of inertia of arm and engines (lb-ft^2)

I_1 = first approximation to moment of inertia of arm neglecting air resistance term (lb-ft^2)

I_2 = second approximation to moment of inertia including air resistance term (lb-ft^2)

L = torque (ft-lb)

L_d = total drag torque acting upon rotor (ft-lb)

L_ω = air drag as a function of ω

R = radius of drum (ft)

t = time in general (sec)

t_o = time of arm sweep over pickup points at end of drop (sec)

t_m = time of arm sweep between magnetic pickup points (sec)

τ = time of weight drop (sec)

[redacted]

W = weight in excess of friction (lb)

ω_o = angular velocity of rotor at end of drop = $0.3345/t_0$

$\omega(t)$ = angular velocity of rotor as function of time

ω_m = angular velocity of rotor at t_m

$\dot{\omega}$ = rate of change of angular velocity of rotor

Equation (1a) is a fundamental equation of angular mechanics, where L is expressed in ft-lb

$$I\dot{\omega} = gL \tag{1a}$$
$$Id\omega = gLdt \tag{1b}$$
$$I\omega + K = gLt. \tag{1c}$$

Applying (1c) to the arm immediately after drop, and neglecting air friction, a first approximation is

$$C + I_1 \ \omega_0 = gWR\tau \tag{1d}$$

or in functional form

$$\omega\ (t) \cong \frac{gWR}{(C + I_1)}\ t. \tag{1e}$$

The effect of neglected air resistance upon Equation (1) may be investigated by expressing the air resistance torque as a function of the angular velocity. Experimentally it was found that the air resistance torque could be expressed with fair accuracy by

$$L\ (\omega) \doteqdot a\omega^{\alpha}. \tag{2}$$

Air resistance produces a torque which subtracts from WR. If I is the true moment of inertia of the rotor assembly, Equation (1) must be amended to include the air resistance torque. From the fundamental definition of torque in differential form, from which (1) was obtained, when air resistance torque is added the results are

$$(C + I)\ \dot{\omega} = g\ [WR - a\omega^{\alpha}(t)] \tag{3a}$$

$$(C + I)\ d\omega = g\ [WR - a\omega^{\alpha}(t)]dt. \tag{3b}$$

Integrating (3b) gives

$$(C + I)\ \omega_o = gWR\tau - ga\int_0^{\tau} \omega^{\alpha}\ (t)\ dt. \tag{4}$$

To evaluate (4) the technique employed in perturbation theory is used. Since a and ω are experimentally small compared to W and R, the first approximation neglects the small variation in W caused by air resistance as in (1). Substituting (1e) in (4) gives the second approximation,

$$(C + I)\ \omega_o \cong g\left[WR\tau - a\int_0^{\tau}\left(\frac{gWR}{(C + I_1)}\right)^{\alpha} t^{\alpha}\ dt\right]. \tag{5a}$$

[redacted]

 [redacted]

By successive approximation I_2 can be found such that

$$(C + I_2)\ \omega_o = g\left[WR\tau - a\left(\frac{gWR}{(C + I_2)}\right)^{\alpha}\frac{\tau^{\alpha+1}}{\alpha + 1}\right] \tag{5b}$$

or

$$I \cong I_2 = \frac{gWR\tau - ag\left(\frac{gWR}{C + I_2}\right)^{\alpha}\frac{\tau^{\alpha+1}}{\alpha + 1}}{\omega_o} - C\ . \tag{5c}$$

From one weight drop record and other measurements the following values were obtained:

$g = 32.2$ ft/sec^2
$W = 52.94$ lb
$R = 0.812$ ft
$C = 39.7$ ft^2-lb
$\tau = 10.83$ sec

$t_0 = 0.2635$ sec
$\omega_o = \dfrac{0.3345}{0.2635} = 1.269$ rad/sec
$a = 0.869$ ft-lb-sec
$\alpha = 2.35$

Substituting values in (5c) and solving gives:

$$I \cong I_2 = \left[\left(\frac{14991-162}{1.269}\right) - 39.7\right]\ ft^2\text{-}lb = 11{,}685.6 - 39.7 = 11{,}646\ ft^2\text{-}lb.$$

After determining the moment of inertia I, the total drag of the whirling-arm assembly may be determined by using the magnetic pickup and recording setup to obtain a sweep time record during deceleration. The arm is allowed to reach full speed (225 rpm), the recording unit is started, and the fuel to the engines is cut. After a microphonic trace indicates engine stoppage, a series of angular velocities and half-sweep times are determined from the magnetic pickup record, the angular velocities being designated $\omega_1, \omega_2, \omega_3 \ldots \omega_m$ and the times from the middle of one half-sweep to the middle of the next being designated $T_1, T_2, T_3, T_4 \ldots T_m$. The experimental total drag L_d, at the average ω — equal to $(\omega_1 - \omega_2)/2$ — is given by

$$L_d = \frac{I\ \dot{\omega}}{g} = \frac{I\ (\omega_1 - \omega_2)}{gT_1}\ . \tag{6}$$

A sample computation is included to give an idea of the actual values. From one of the sweep time records during deceleration the following values were obtained:

$t_1 = 0.01520$ sec;

$t_2 = 0.01555$ sec;

$T_1 = 0.143$ sec.

Then by (6) from these values ω_1 is 22.006 radians per second and ω_2 is 21.511 radians per second giving an average ω of 21.78 radians per second

$$L_{(21.78)} = \frac{11647\ ft^2\text{-}lb\ (22.006 - 21.511)\ /\text{sec}}{32.2\ ft/\text{sec}^2 \qquad 0.143\ \text{sec}} = 1252\ ft\text{-}lb$$

At this point it is well to point out one of the greatest sources of error. The significant part of t_1 and t_2 is a fraction of a millisecond which must be estimated between rather fuzzy millisecond timing marks. Moreover the broadness of field gives rise to well-rounded swings on which it is difficult to select a peak point with a high degree of time accuracy. It is also possible that the considerable vibration of the arm that has

[redacted]

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been observed statically might well contribute to inaccuracy in measuring the decelerating torque, especially just after a sudden engine cutoff. Several improvements will be made when greater accuracy is demanded. First more expensive film will be run at a higher speed. With the more sensitive film it will be possible to use 10,000-cycle timing traces. Vibration-free contactors are under consideration for they would generate a sharp square wave where points would be determined to a greater accuracy.

An effort was made to determine an empirical drag function of the standard power-series form $L(\omega) = a_o + a_1\omega + a_2\omega^2 + a_3\omega^3 + \ldots\ldots$
Several attempts to evaluate the unknown coefficients by standard determinant methods failed to yield consistent results. A function $L(\omega) = a_o + a_1\omega^k$ was tried and when the three constants were evaluated the empirical curve fell upon most of the experimentally determined points. The relation $\omega = \frac{2\pi(rpm)}{60}$ was used where data was taken in revolutions per minute. The empirical equation as a function of rpm is
$L_d = (0.869(2\pi(\text{rpm})/60)^{2.35} + 2.6)\,ft\text{-}lb.$
The following table compares values of L_d from the empirical computation $(0.869\,\omega^{\alpha} + 2.6)$ *ft-lb* with values observed from the deceleration record.

rpm	ω radians/sec	Empirical L_d (ft-lb)	Observed L_d (ft-lb)
1.16	0.1215	2.61	2.61
11.85	1.24	4.1	6.3
43.2	4.53	32.6	26.5
78.	8.05	120.	138.
99.	10.38	214.	215.
111.	11.69	281.	303.
155.	16.23	606.	572.
177.	18.53	834.	---
196.5	20.55	1060.	1091.
208.	21.78	1211.	1252.
229.	24.	1523.	---

The drag torque and horsepower as a function of rpm are plotted in Figure 9.

Because of compressibility corrections one might have expected a power of α only slightly greater than two. The fact that α is 2.35 may be due to the known results of a diminished drag coefficient as the vanes close up under the decreasing head pressure of falling velocity. Thus the value of α may depend upon the type and stiffness of the vanes used.

After the fuel to the engine is cut off, the internal drag is affected by the decrease in pressure and temperature and the change in the flapper-valve action. These factors may have considerable effect upon the magnitude of the drag torque, but for the present purposes detailed investigation of these factors is not warranted.

WHIRLING-ARM TESTS

The Thermodynamics Branch at Chesapeake Bay Annex has developed a pulse-jet grid for a six-inch diameter engine which has run for forty hours without damage on a static reaction stand.[2] A grid of the same type has been run on the whirling arm for

[2] Schechter, R.M., "Sixth Partial Report on the Pulse-Jet Engine — Grid Development," NRL Report 3537 (Restricted), September 20, 1949.

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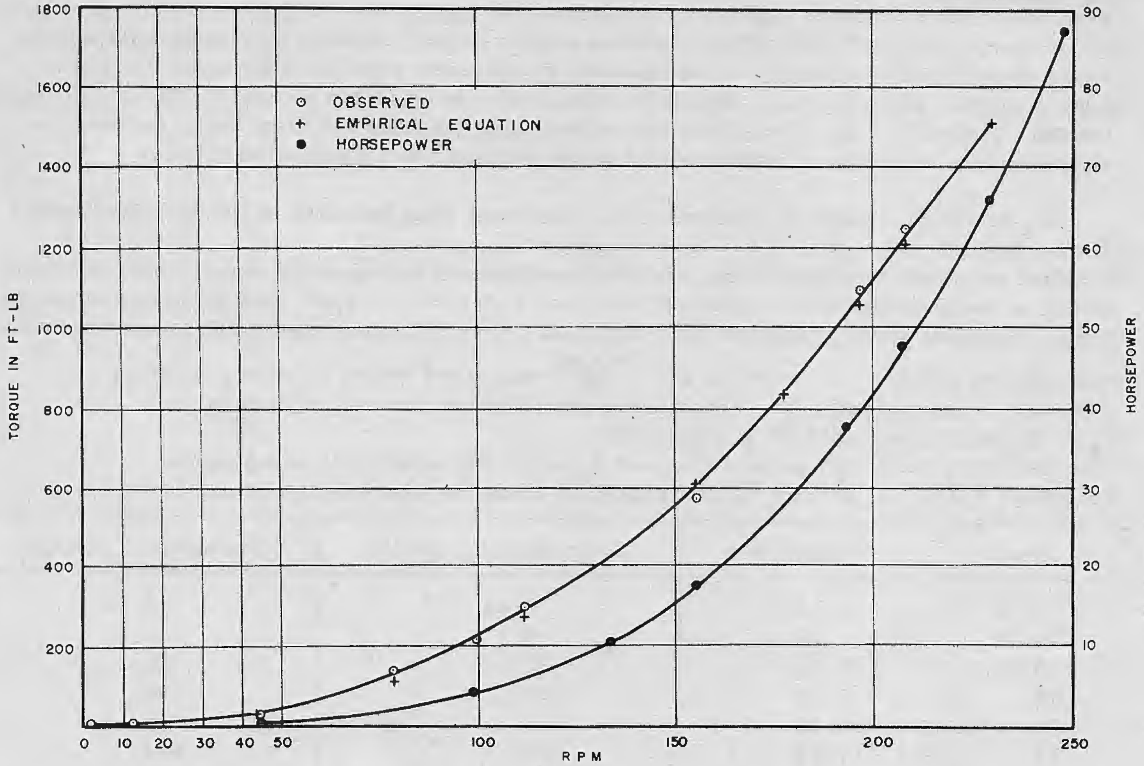

Figure 9 - Drag of whirling arm

two hundred hours without damage. The grid was on a six-inch diameter, convergent-divergent engine with an overall length of thirty-two inches. The fuel system used was of the four-jet, conical-spray type with a Venturi-type entrance to the combustion chamber. The test was run at an average fuel rate of 130 pounds per hour and a tip speed of 152 miles per hour.

The whirling arm has been used to evaluate the effect on engine performance of several types of entrance cowls, air scoops, and tail ducts. Tests of various types of fuel manifolds, Venturi throats, orifice plates, and starting-air manifolds have been made. Engine bodies of various weights and materials have been compared for durability.

CONCLUSIONS

The whirling arm has proved itself to be a valuable apparatus for the dynamic testing of pulse-jet engines and various designs of the component parts. The drag of the arm has been experimentally determined and an empirical equation derived for the drag value at any speed.

RECOMMENDATIONS

Whirling-arm work should be started on larger engines. The arm now in use at the Chesapeake Bay Annex was not constructed to carry engines which weigh more than twelve pounds; therefore, a new arm is in construction which will be capable of carrying thirty-pound engines. It is anticipated that short, high-thrust, eight- and ten-inch diameter pulse-jet engines can be developed which will weigh less than thirty pounds. Engines of this size should have immediate application as pulse-jet helicopter engines or as guided missile prime movers. The new whirling arm will have a thirty-six foot rotor and will be capable of carrying the large engines at a top speed of five hundred feet per second.

ACKNOWLEDGMENTS

T. O. Meyer and R. M. Schechter accomplished the instrumentation described.

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