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TEST OF A DUAL-ROTATION AXIAL-FLOW FAN

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ADVANCE RESTRICTED REPORT

TEST OF A DUAL-ROTATION AXIAL-FLOW FAN

By E. Barton Bell and Lucas J. DeKoster

SUMMARY

A dual-rotation axial-flow fan composed of two oppositely rotating 24-blade rotors was built and tested. The outside diameter was 21 inches and the hub diameter was $14\frac{1}{2}$ inches. The fan was tested with various combinations of front and rear blade angles and with two combinations of front and rear solidities. Pressure and torque coefficients and efficiency were determined and are presented as functions of a quantity coefficient.

In order to have the torque coefficients of the front and rear rotors remain nearly equal over the operating range with fixed blade angles, it was necessary to reduce the number of blades in the rear rotor to one-half the number of blades in the front rotor. More than twice as much pressure rise was obtained from the dual-rotation fan as from a previously tested single-stage fan without contravanes. The maximum efficiency of the dual-rotation fan was slightly lower than the efficiency of a single-stage fan with contravanes and higher than the maximum efficiency of a single-stage fan without contravanes.

INTRODUCTION

Previous NACA tests of axial-flow fans reported in references 1 and 2 were restricted to a single-stage fan operated with and without contravanes. Reference 1 for a single-stage fan with 24 blades indicated that higher pressures were obtainable with than without contravanes. Reference 2 indicated that not much increase in pressure could be expected from an increase in the solidity above the value of 0.86 for 24 blades. In order to get pressures appreciably higher than those obtainable with a fan such as was used in reference 1, it is therefore necessary to use more than one stage. A simple multistage arrangement involves the use of two rotors turning in opposite directions. Such a fan has

been built and tested by the propeller-research section. The program included tests with various combinations of front and rear blade angle and with two combinations of front and rear solidity.

DESCRIPTION OF APPARATUS

The dual-rotation fan was composed of two similar single-rotation fans mounted with their rotor ends together. The spacing between the center lines of the blades in the front and the rear hubs was $3\frac{3}{8}$ inches. The outside diameter and the hub diameter were 21 inches and $14\frac{1}{2}$ inches, respectively, in order to make these fans interchangeable with the fan of reference 1. The general arrangement of the fan and test set-up is shown in figures 1 to 3. The blades of R.A.P. 6 section used in each rotor were alike except that they were of opposite hand. The blades in the rear rotor were the blades used in the fan of reference 1. Each hub had spaces for 24 blades. For operation with 12 blades, one-half the blades were removed from a 24-blade rotor and the holes were filled with steel plugs, which gave surfaces flush with the hub.

Power was supplied to each rotor by a 25-horsepower three-phase alternating-current induction motor. These motors were supplied from a common source with alternating current of variable frequency. A water rheostat for each phase of this variable-frequency supply was connected in series with each motor. The voltage drop across the motors could be independently varied by varying the resistance of these rheostats; this arrangement gave a small amount of differential speed control. The front motor was air cooled and the rear motor was water cooled.

The motors were supported inside the steel fan casing on four radial streamline struts. The struts supporting the front motor were drilled for static orifices for measuring the pressure to be used to determine quantity of air flow. All electrical connections to the motors were carried through the struts. The fan casing was machine-finished on the inside to a diameter of 21 inches. At the location of the rotors the diameter was relieved to a diameter of $21\frac{1}{8}$ inches, which gave $1/52$ -inch clearance at the tips of the blades.

The two parts of the multistage fan were bolted and doweled together and mounted on a steel frame. This frame

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was supported on four rollers, which rested on hardened and ground steel tracks. The frame was restrained from rolling fore-and-aft by a thrust wire, one end of which was carried over a pulley to a dial balance. The other end was connected to counterweights through a bell crank. The entrance bell and exit cones were separately supported free of the fan. A cylinder having the diameter of the hub was extended upstream beyond the pressure field of the entrance. The flow through the test section was therefore confined between the cylindrical hub fairing and the cylindrical outer casing.

A diffuser extended downstream of the fan. At the outlet of the diffuser a variable restriction consisting of a number of overlapping movable streamline slats controlled the pressure rise through the fan. The air was exhausted through this restriction into a muffler.

METHOD OF TESTING

Preceding the actual tests of the fan, the torque output of each motor was calibrated with a Prony brake against wattmeter and voltmeter readings. The electric tachometers, one on each motor, were calibrated against commercial 60-cycle alternating current. The tachometer calibrations were checked at various times during the testing. The pressure orifices for determining quantity of air flow immediately ahead of the front rotor were calibrated by running the fan in series with a previously calibrated venturi.

As in references 1 and 2, the pressure rise across the fan was assumed to be the thrust on the effective disk area divided by the effective disk area. The thrust on the disk area was obtained from balance and counterweight readings corrected for the force due to the pressure inside the hub diffuser cone. For conditions in which the torques on the front and the rear rotors are unequal, the pressure coefficients may be somewhat affected by the rear-motor supporting struts. For unequal-torque conditions, there will be twist in the air behind the rear rotor and ahead of these struts. This twist will produce lift and drag on the struts. Whether there is a positive pressure or negative pressure component to the resultant of this lift-and-drag force depends on the magnitude of the twist in the air. For no twist (equal torque) there is only drag on these struts and

the pressures are conservative. Under most test conditions this effect will be small and has therefore been neglected in the calculations of the pressure coefficients.

Three series of tests were made. One series consisted of a number of tests with the blade angles in the front rotor varying in 5° increments from 15° to 50° and the blade angles in the rear rotor so adjusted that the rear rotor would absorb approximately the same torque as the front rotor at maximum pressure. Twenty-four blades were used in each rotor during these tests. Another series of tests was made with the front and rear blade angles equal but with 24 blades in the front rotor and 12 blades in the rear rotor. This rotor arrangement is shown in figure 4. A third series was made with 12 blades in the rear rotor so adjusted that the torque at maximum pressure was the same as that of the front rotor with 24 blades.

The pressure and the quantity of air were varied during each test by adjusting the slats in the variable restriction. The following observations were made at each test point: voltmeter, wattmeter, and tachometer readings for each motor; manometer readings of pressure at static orifices and inside the hub diffuser; balance reading and amount of counterweight; wet and dry bulb temperatures; and barometric pressure. All tests were run at rotational speeds between 3000 and 3600 rpm.

RESULTS AND DISCUSSION

The results of this investigation are presented in terms of the fan coefficients used in references 1 and 2. Torque coefficients are given for the individual components of the fan and for the fan as a whole. These coefficients and the symbols used are defined as follows:

$$C_p = \frac{\Delta p}{\rho n^2 D^2} \quad \text{pressure coefficient}$$

$$C_{T_F} = \frac{T_F}{\rho n^2 D^5} \quad \text{torque coefficient, front rotor}$$

$$C_{T_R} = \frac{T_R}{\rho n^2 D^5} \quad \text{torque coefficient, rear rotor}$$

$$C_T = \frac{T}{\rho n^2 D^5} \quad \text{total torque coefficient}$$

$$\eta = \frac{1}{2\pi} \frac{C_p}{C_T} \frac{Q}{nD^3} \quad \text{efficiency}$$

$$Q/nD^3 \quad \text{quantity coefficient}$$

where

Δp pressure rise across fan, pounds per square foot

ρ mass density of air, slugs per cubic foot

n average rotational speed, revolutions per second
 $[(n_F + n_R)/2]$

n_F rotational speed front rotor, revolutions per second

n_R rotational speed rear rotor, revolutions per second

D fan diameter (1.75 ft)

T total torque ($T_F + T_R$), foot-pounds

T_F torque on front rotor, foot-pounds

T_R torque on rear rotor, foot-pounds

Q quantity rate of flow, cubic feet per second

The following additional symbols are used in the text and figures:

R radius to outside of fan ($D/2$)

r_o radius of hub ($0.69R$)

σ solidity $\left[\frac{Bb}{\pi R(1 + r_o/R)} \right]$

B number of blades

B_F number of blades in front rotor

B_R number of blades in rear rotor

b blade width

β_F blade angle on front rotor at 0.714R, degrees

β_R blade angle on rear rotor at 0.714R, degrees

All coefficients are presented on the basis of an average rotational speed because, with available equipment, it was impossible to synchronize the two motors if the loads were appreciably different. This method of calculating the coefficient leads to a small error in all cases where the torque is appreciably different between the rotors. Using an average rotational speed has the effect of decreasing the high torque coefficient and increasing the low torque coefficient. The individual curves, in terms of Q/nD^3 , are also brought more closely together than when individual speeds were used. The magnitude of this discrepancy is usually very small because it was possible to keep the two components of the fan synchronized over most of the range of operation. The deviations from synchronization for all the conditions tested are shown in figure 5.

The results of the test with 24 blades in both front and rear rotors and with the rear blade angles adjusted to give approximately equal torque at maximum pressure are shown in figure 6. The individual torque curves (fig. 6(b)) show the large differences in torque on the two rotors at any condition other than that for which they were set. In order to obtain equal torque at maximum pressure, it was necessary to set a large difference in blade angle between front and rear rotors. This fact is explainable by the increased velocity over the rear blades, which arose from the rotation imparted to the air by the front rotor. This effect is similar to the effect of contravanes on a single-stage fan. At Q/nD^3 values larger than 0.3 this arrangement of adjusting the blades provided higher pressures than either of the other arrangements tested. The slope of the pressure curves is, however, very steep. If the fan is required to operate at anything other than a fixed Q/nD^3 , this variation of pressure is a disadvantage.

In order to bring the blade angles more closely together, to decrease the slope of the pressure curves, and to keep the torques of the front and rear rotors more nearly equal, several tests were run with blade angles the same on front and rear and with 24 blades ($\sigma = 0.86$) in the front rotor and 12 blades ($\sigma = 0.43$) in the rear rotor. The results of these tests, given in figures 7(a) to 7(d), show that the slope of the pressure curves has been decreased but

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 the rear fan is still drawing more power. Although the rear fan is operating with only one half as many blades as the front fan, the power it absorbs differs from that of the front fan by an amount that is about the same over the operating range. The maximum pressure available is not so great with this arrangement as with the arrangement having 24 blades in each rotor at values of Q/nD^3 larger than 0.3 but is slightly greater at values of Q/nD^3 less than 0.3. The maximum efficiency is about 2 or 3 percent lower.

In order to bring the power requirements of the two fans more closely together and to determine the effect on the efficiency, several tests were run with 24 blades in the front hub, 12 blades in the rear hub, and with the blade angles of the rear rotor adjusted to absorb torque equal to that of the front rotor at maximum pressure. The results of these tests are given in figure 8. Figure 8(a) shows that, at the lower blade angles, the torques of the front and the rear rotors were practically equal for the range of operation. The slopes and the maximum values of the pressure-coefficient curves and the maximum efficiency were about the same as with front and rear blade angles set equal. For the test with $\beta_F = 50^\circ$ and $\beta_R = 41^\circ$ the torque was not equal over the operating range but the maximum efficiency was about 5 percent higher than with equal blade angles. Some other solidity ratio between front and rear might have brought the $\beta_F = 50^\circ$ curve to the desired condition.

A comparison of the pressure coefficients of the dual-rotation fan with 24 blades in each hub with the pressure coefficients of the single-rotation fan of reference 1 shows that the maximum pressure coefficients of the dual-rotation fan are more than twice as great as for the single-stage fan without contravanes and about equal to the sum of the maximum pressures of this fan without contravanes and this fan with contravanes set 70° . The maximum efficiency of the dual-rotation fan was slightly less than the single-rotation fan with contravanes set 70° . It appears that still higher pressures might be obtained from a dual-rotation fan if contravanes were installed ahead of the front rotor. This installation would have the effect of loading the front rotor and unloading the rear rotor. The front and rear blade angles could then be brought more closely together.

SUMMARY OF RESULTS

From tests of an axial-flow fan having two oppositely turning rotors with variable number of blades, and previous

tests on a single-stage fan, the following results were obtained:

1. For quantity coefficients above 0.3 the greatest pressure rise was obtained from 24 blades in each rotor with the blades adjusted to absorb equal torque at maximum pressure. The efficiency of the dual-rotation fan was slightly less than the maximum efficiency of the single-stage fan with contravanes.
2. The slope of the pressure curves was made less steep by reducing the number of blades in the rear rotor.
3. The torque coefficients of the front and the rear rotor were made nearly equal over the operating range by reducing the number of blades in the rear rotor and adjusting the blade angles of the rear rotor.
4. The dual-rotation fan provided more than twice as much pressure rise at the same quantity of air flow as the previously tested single-stage fan without contravanes.

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REFERENCES

1. Bell, E. Barton: Test of a Single-Stage Axial-Flow Fan. Rep. No. 729, NACA, 1942.
2. Bell, E. Barton, and DeKoster, Lucas J.: The Effect of Solidity, Blade Section, and Contravane Angle on the Characteristics of an Axial-Flow Fan. NACA A.R.R., Dec. 1942.

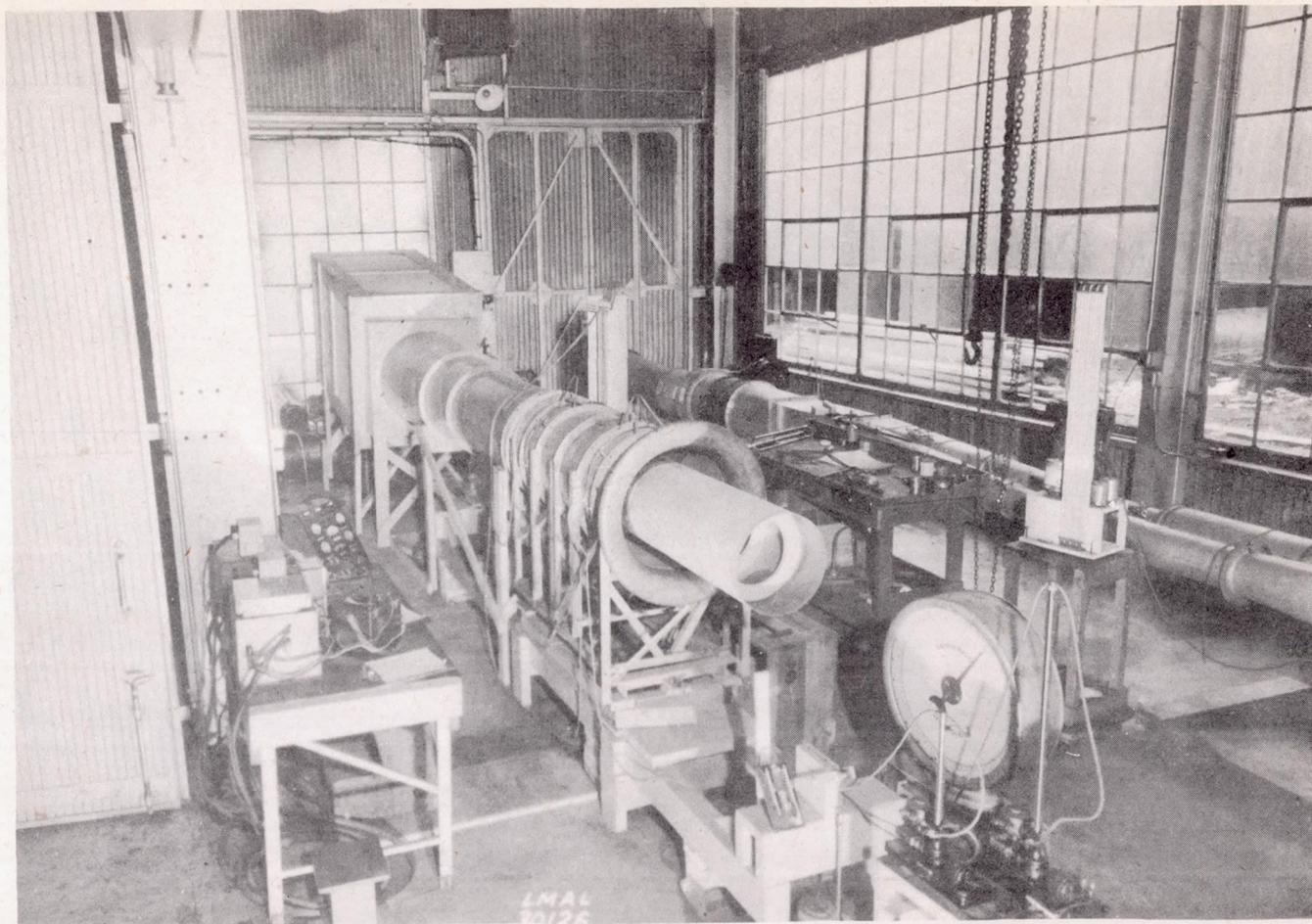


Figure 1.- Three-quarter front view of dual-rotation axial-fan test arrangement.

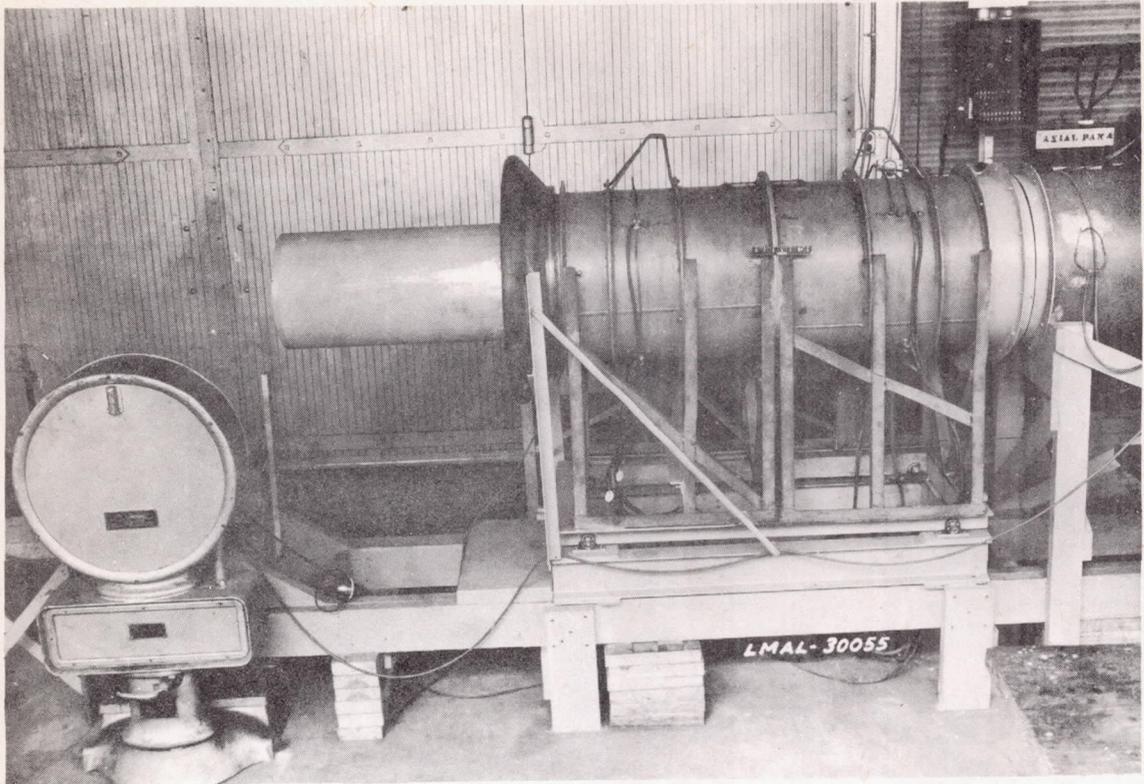


Figure 2.- Side view of dual-rotation axial-flow fan test section.

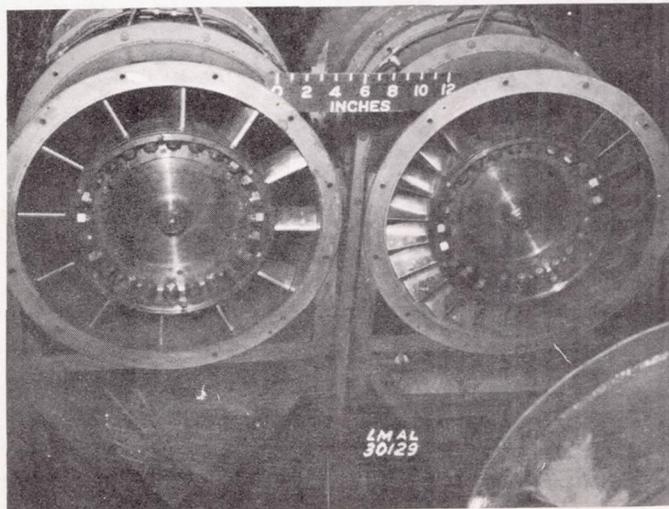


Figure 4.- Rotors of dual-rotation axial-flow fan; 24 blades in front rotor, 12 blades in rear rotor.

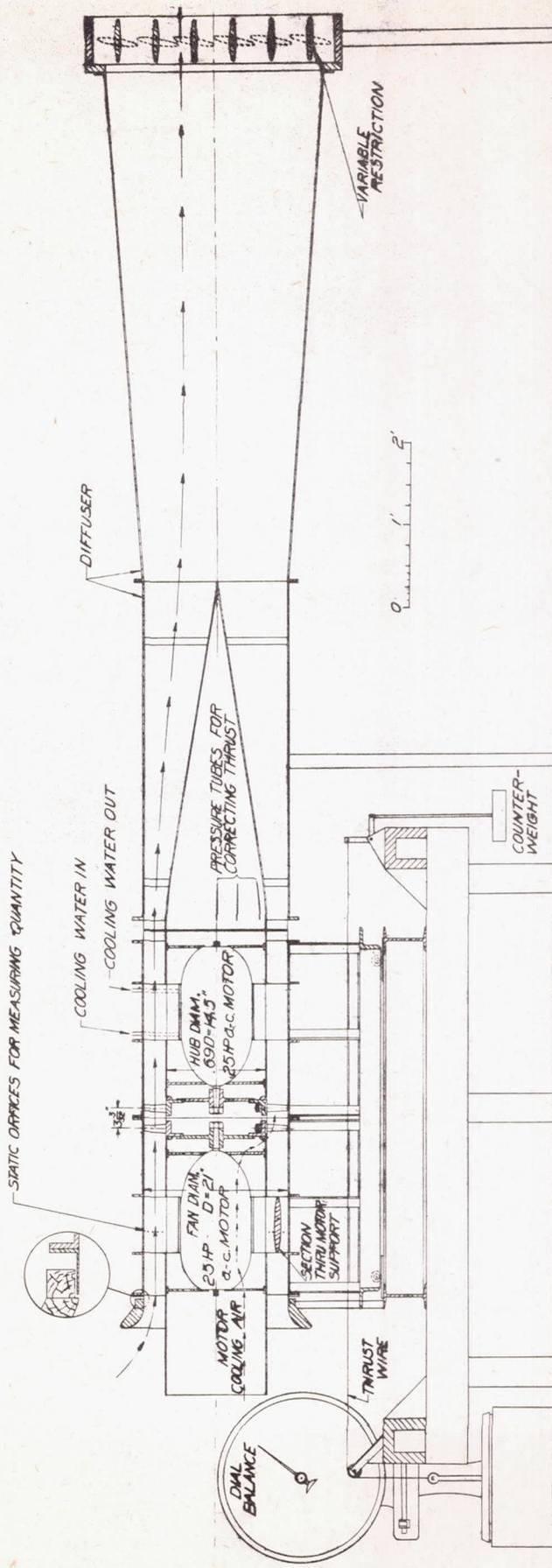
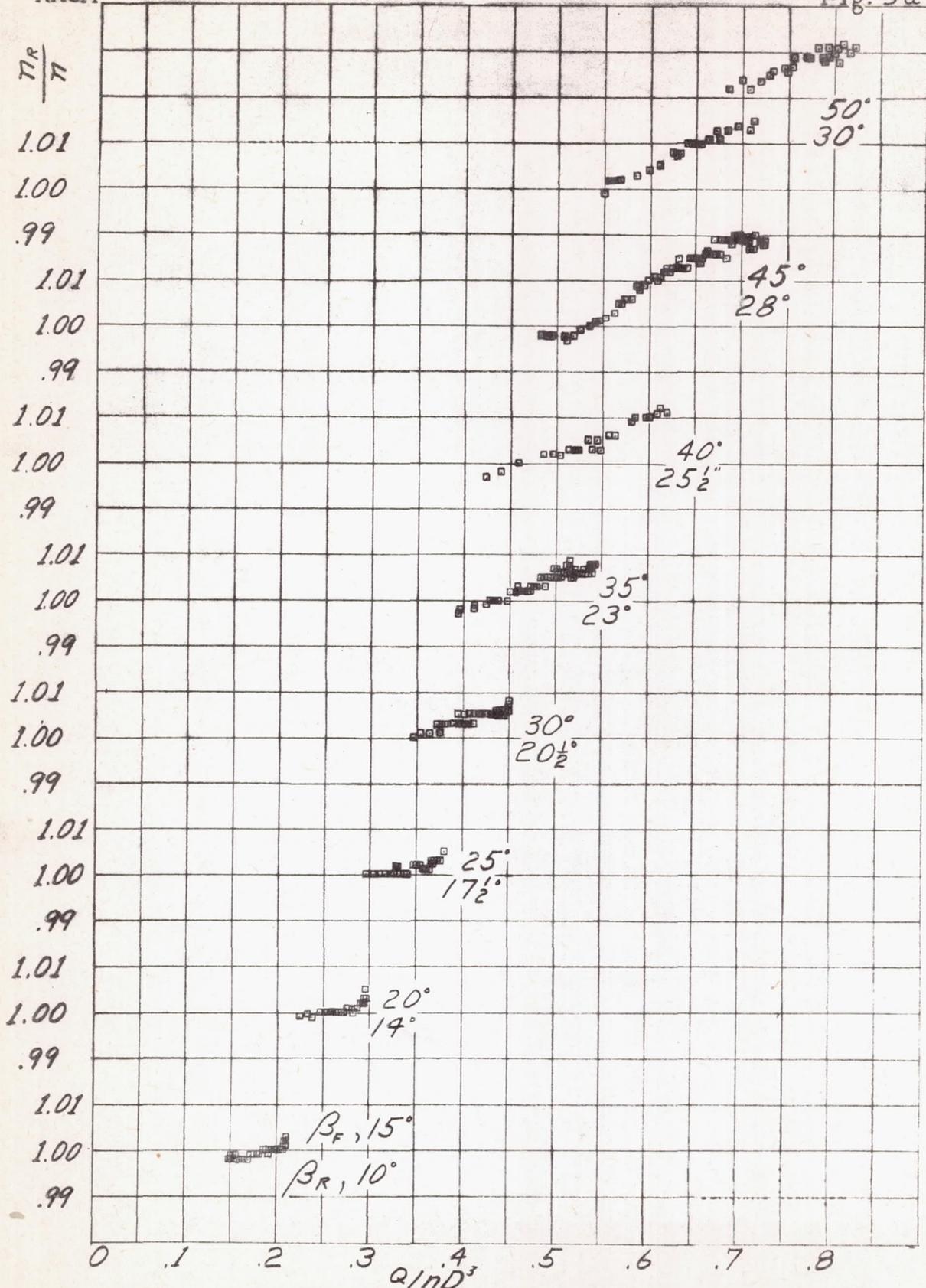


Figure 3.—Dual-rotation axial-fan test arrangement.

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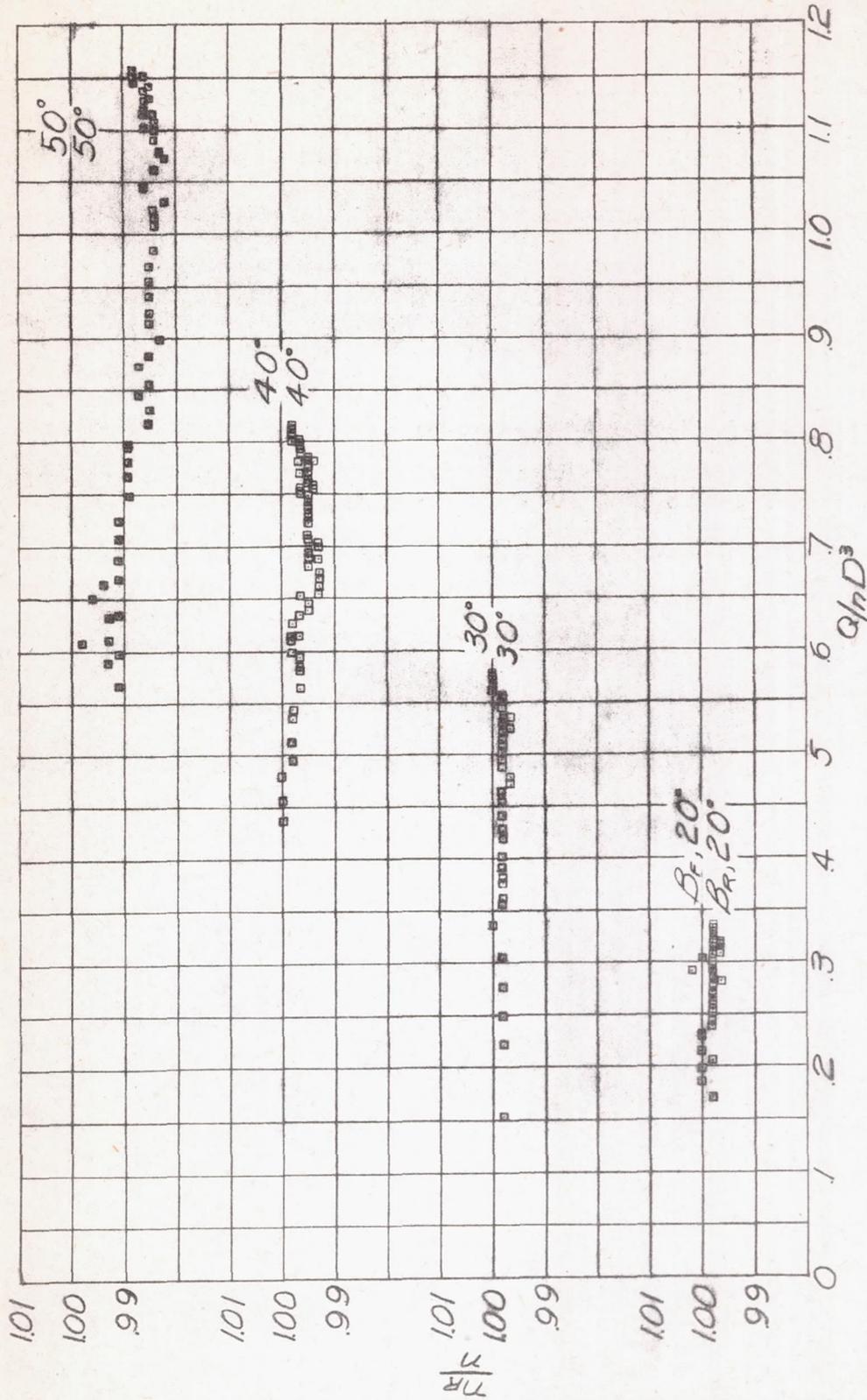
Fig. 5a



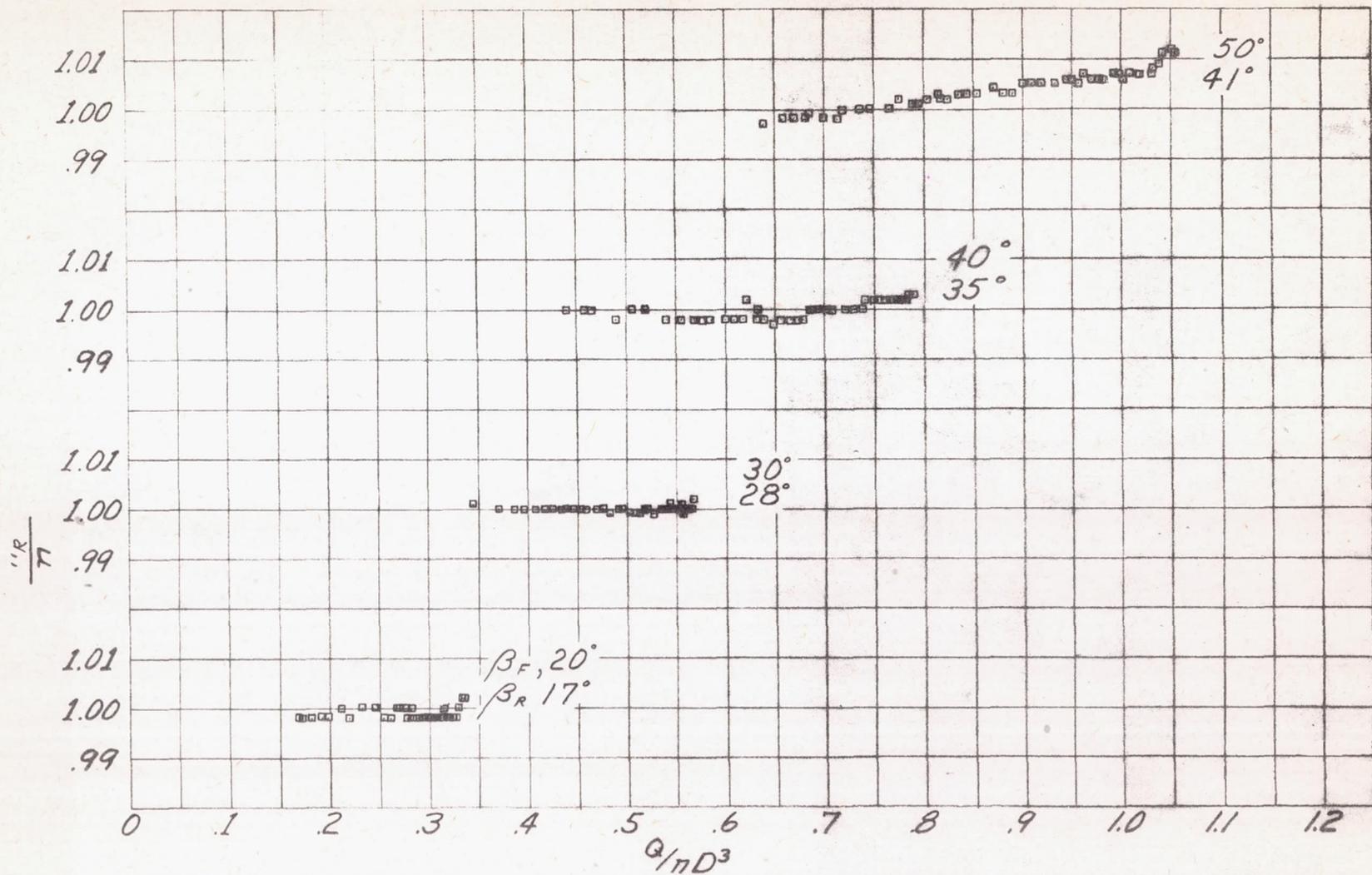
(a) Tests with 24 blades in front and rear rotor; $C_{TF} = C_{TR}$ at $C_{p, \max}$
 Figure 5. — Ratio between rear rotor speed and average rotor speed.

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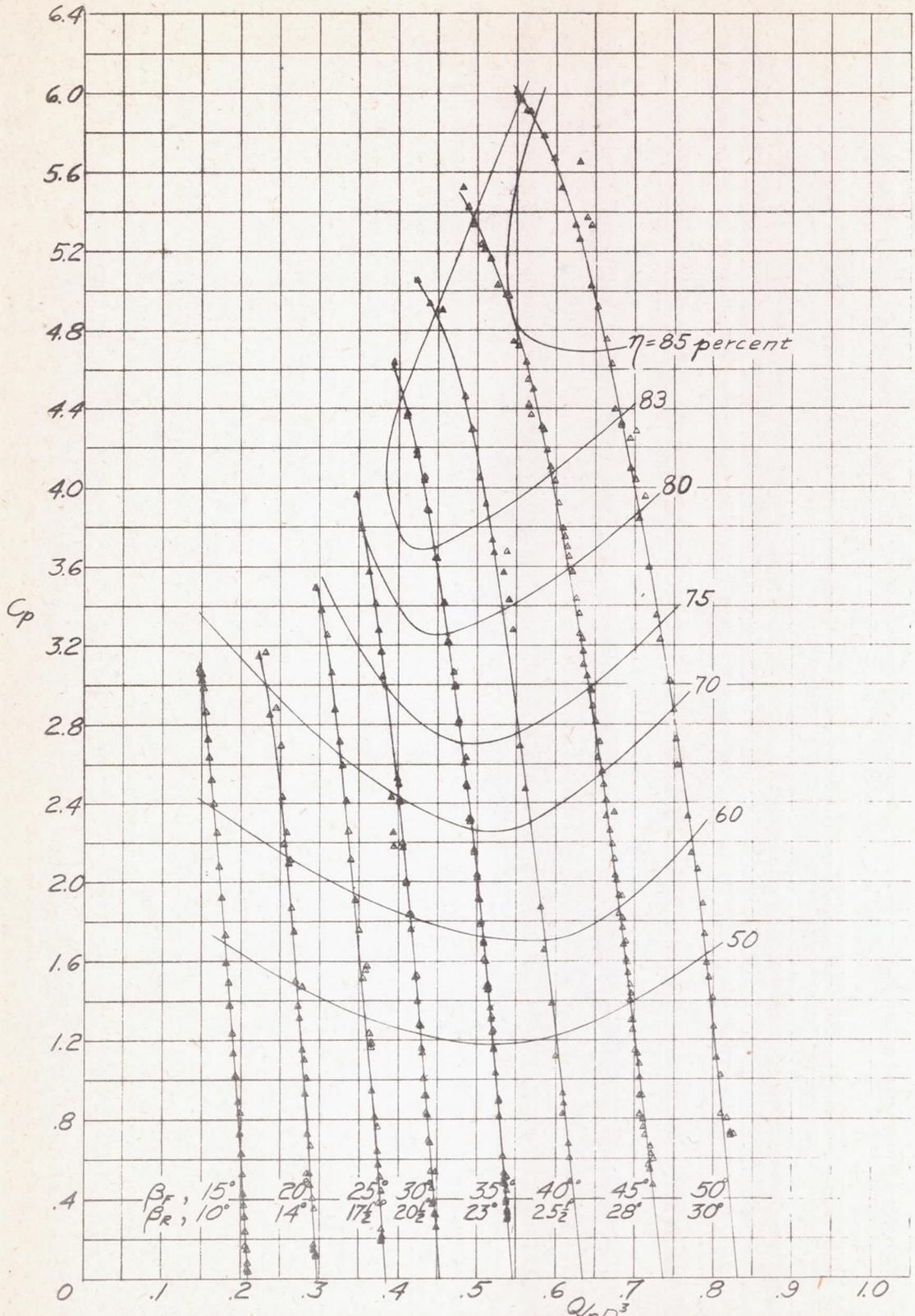
Fig 5b



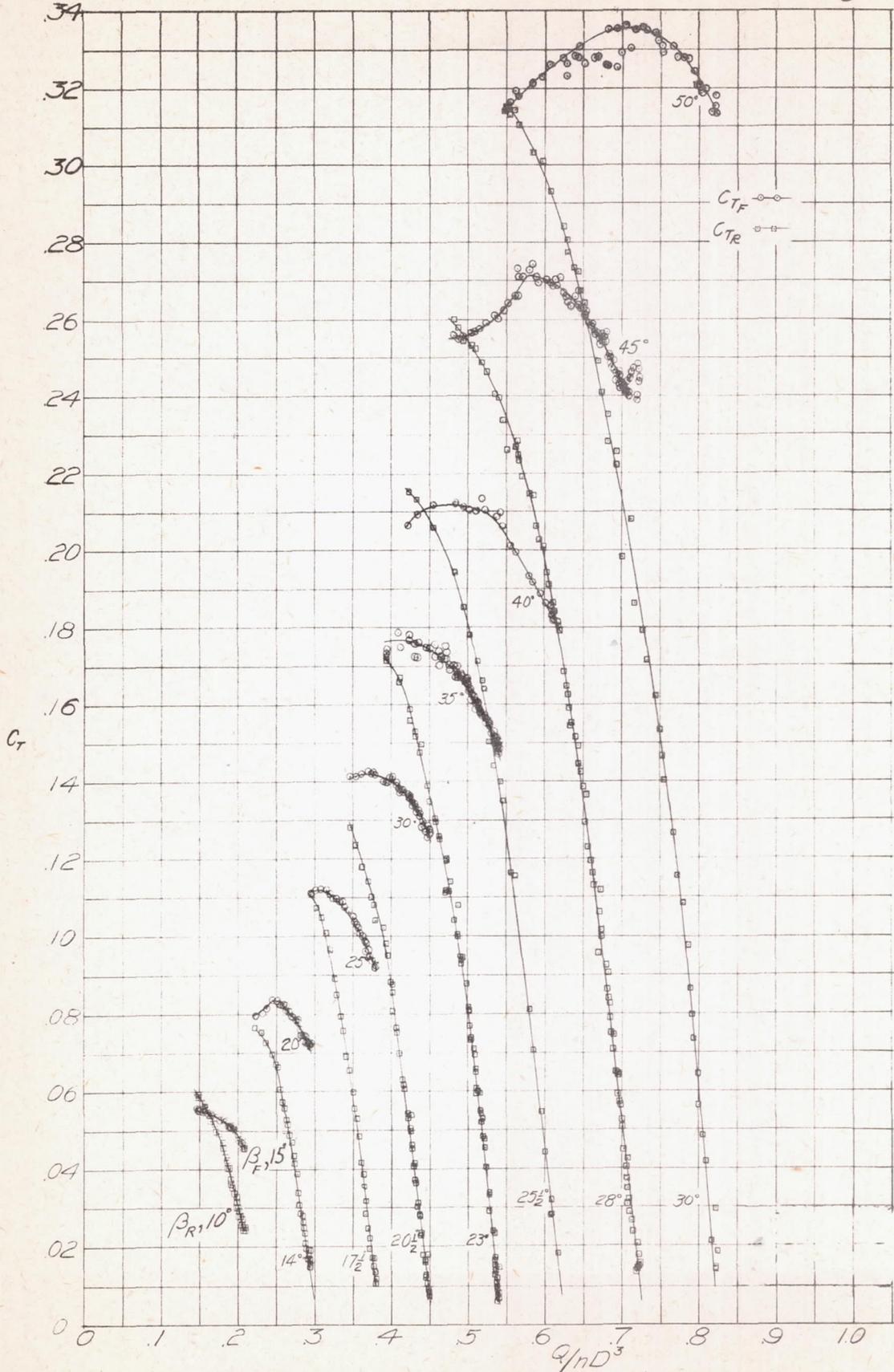
(b) Tests with 24 blades in front rotor, 12 blades in rear rotor; $\beta_F = \beta_R$.
Figure 5. - Continued.



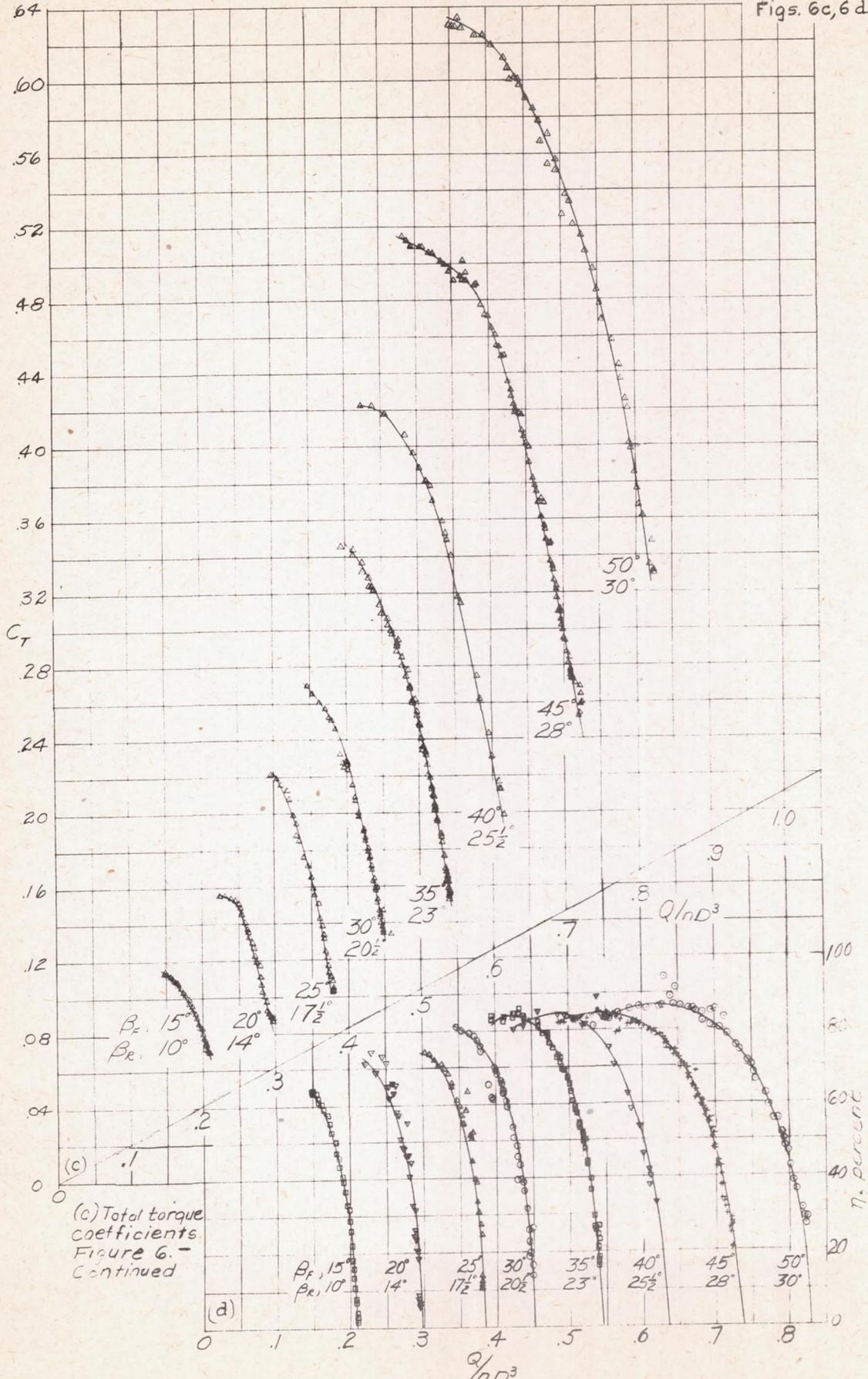
(c) Tests with 24 blades in front rotor, 12 blades in rear rotor; $C_{TF} = C_{TR}$ at C_{Pmax}
 Figure 5.- Concluded.



(a) Pressure coefficients showing lines of constant efficiency.
 Figure 6.- Tests with 24 blades in front and rear rotor; $C_{TF} = C_{TR}$ at C_{Pmax} .

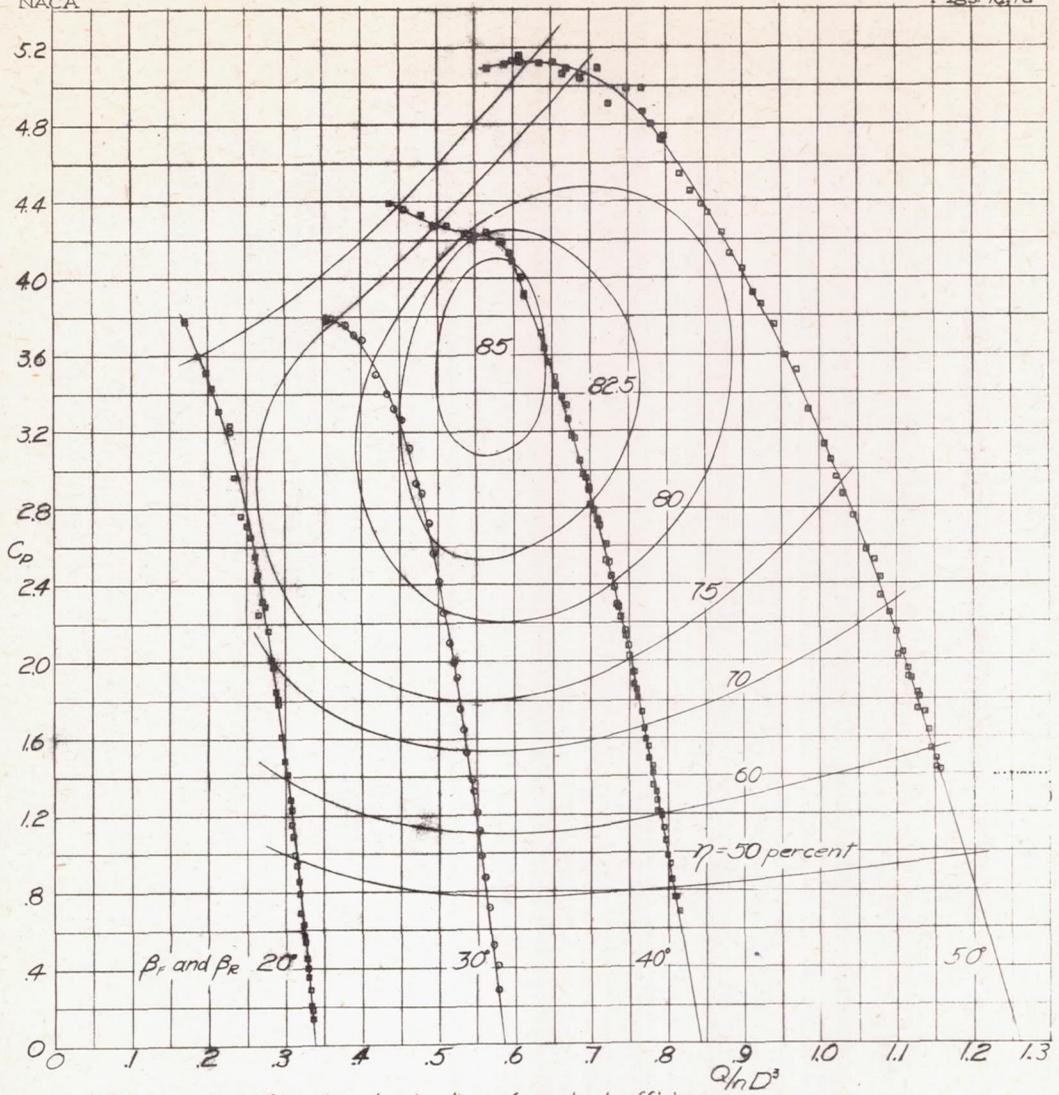


(b) Individual torque coefficients $B_F, 24; B_R, 24$.
Figure 6.- Continued.

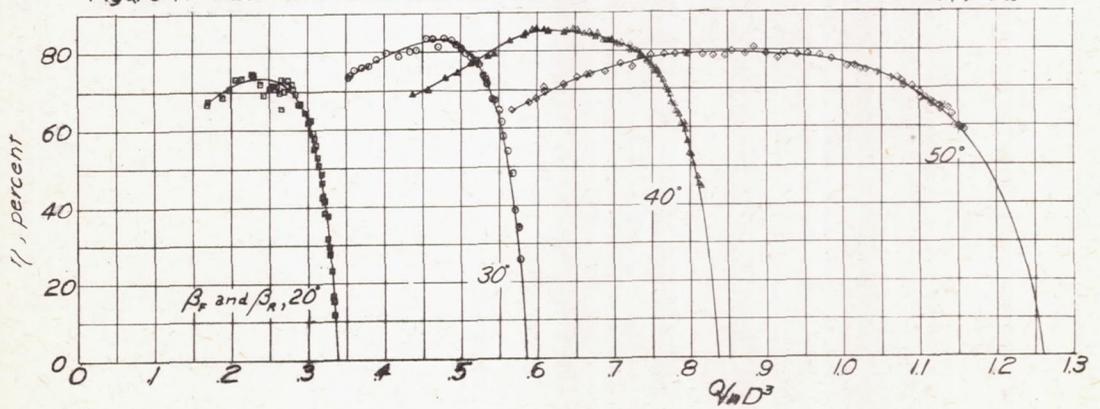


(c) Total torque coefficients. Figure 6. - Continued

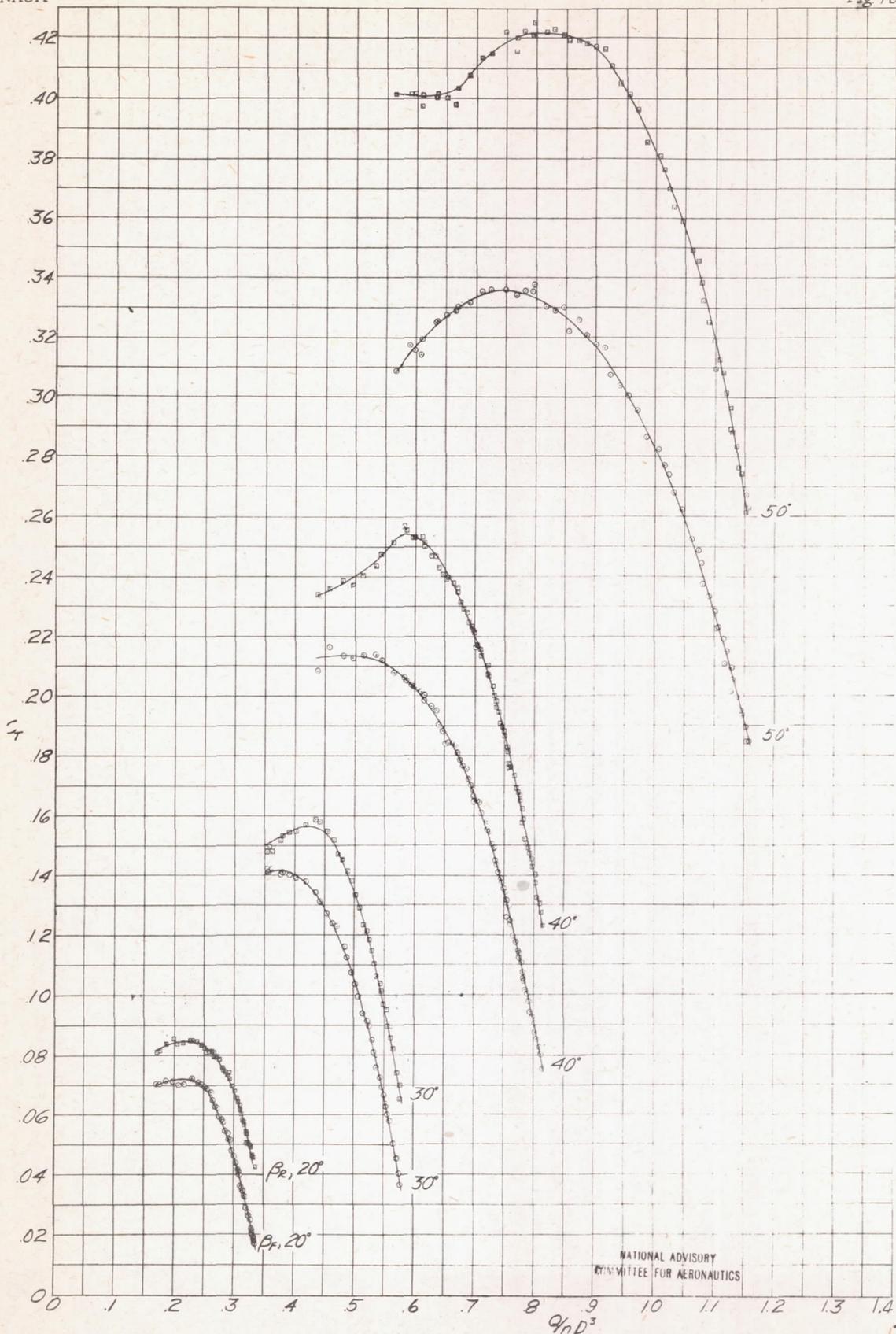
Fig. 6. - Concluded (d) Fan efficiencies $B_F, 24; B_R, 24.$



(a) Pressure coefficients showing lines of constant efficiency.
 Figure 7.- Tests with 24 blades in front rotor and 12 blades in rear rotor; $\beta_F = \beta_R$.

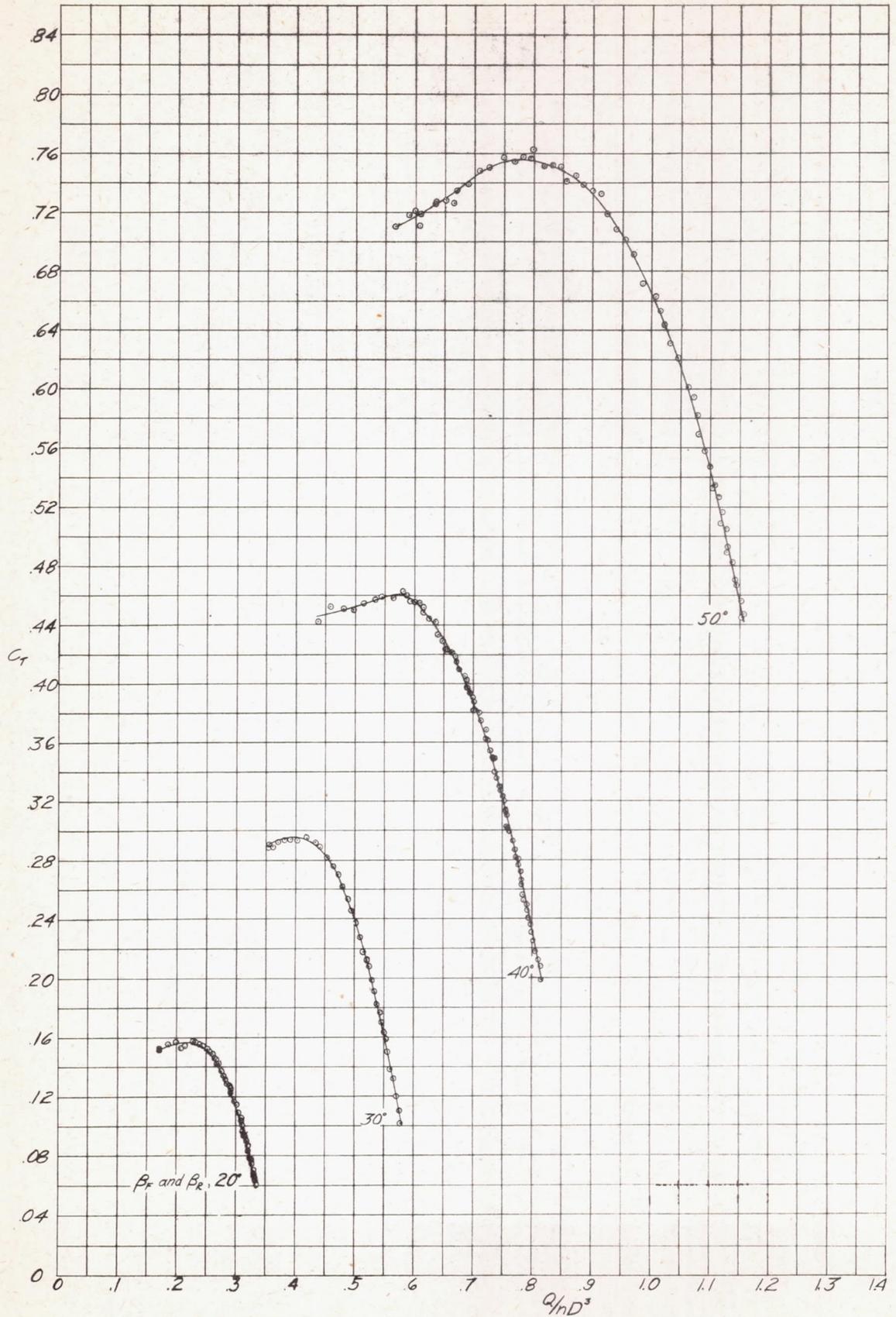


(d) Fan efficiencies.
 Figure 7.- Concluded.

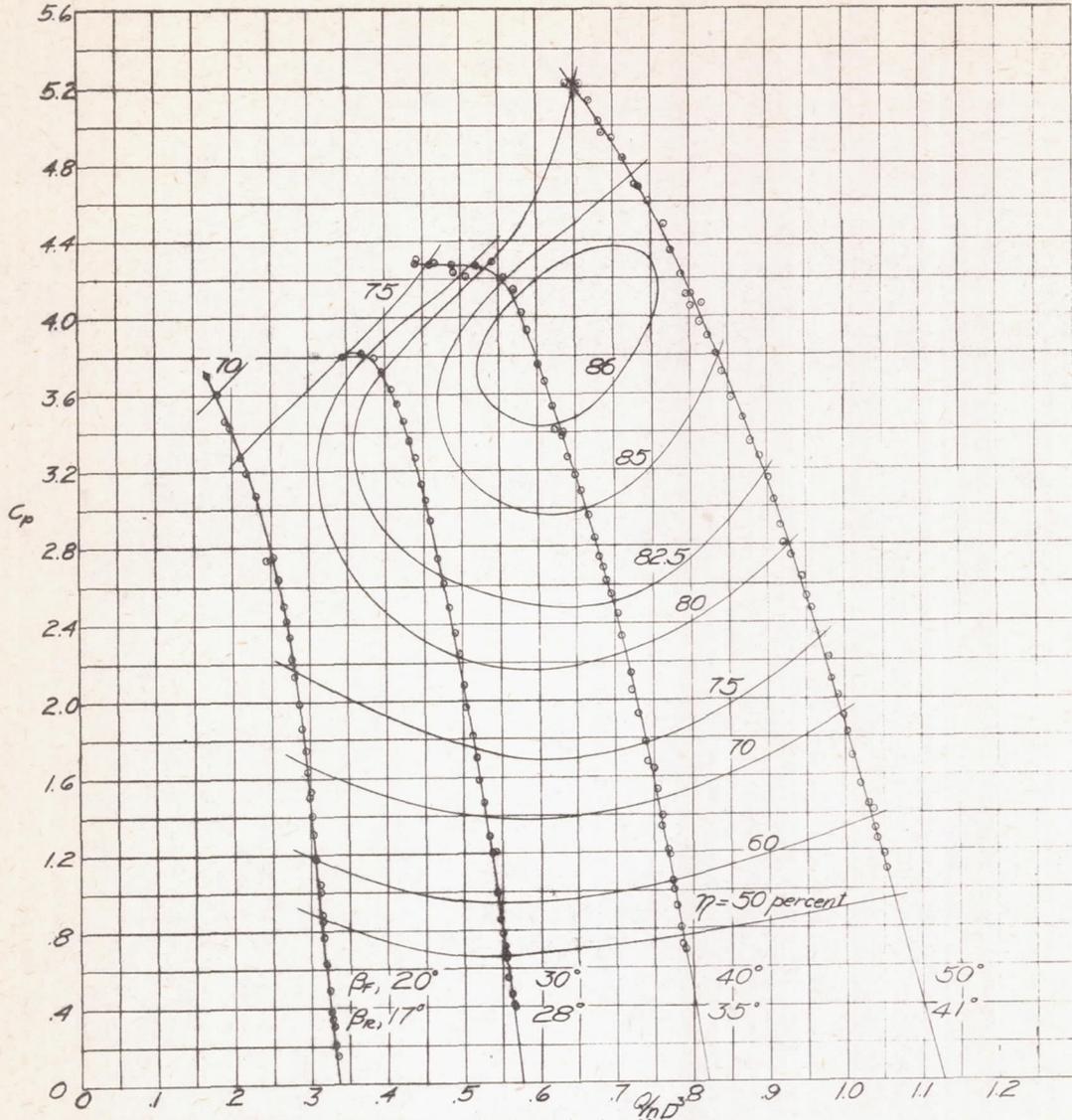


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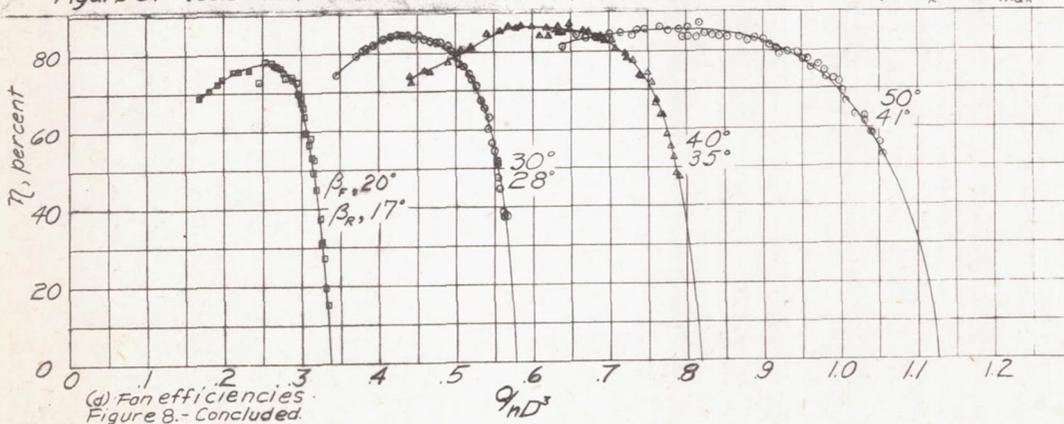
(b) Individual torque coefficients.
Figure 7.-Continued.



(c) Total torque coefficient.
Figure 7.- Continued.



(a) Pressure coefficients showing lines of constant efficiency.
 Figure 8.- Tests with 24 blades in front rotor, 12 blades in rear rotor; $C_{Tf} = C_{Tr}$ at $C_{p,max}$.

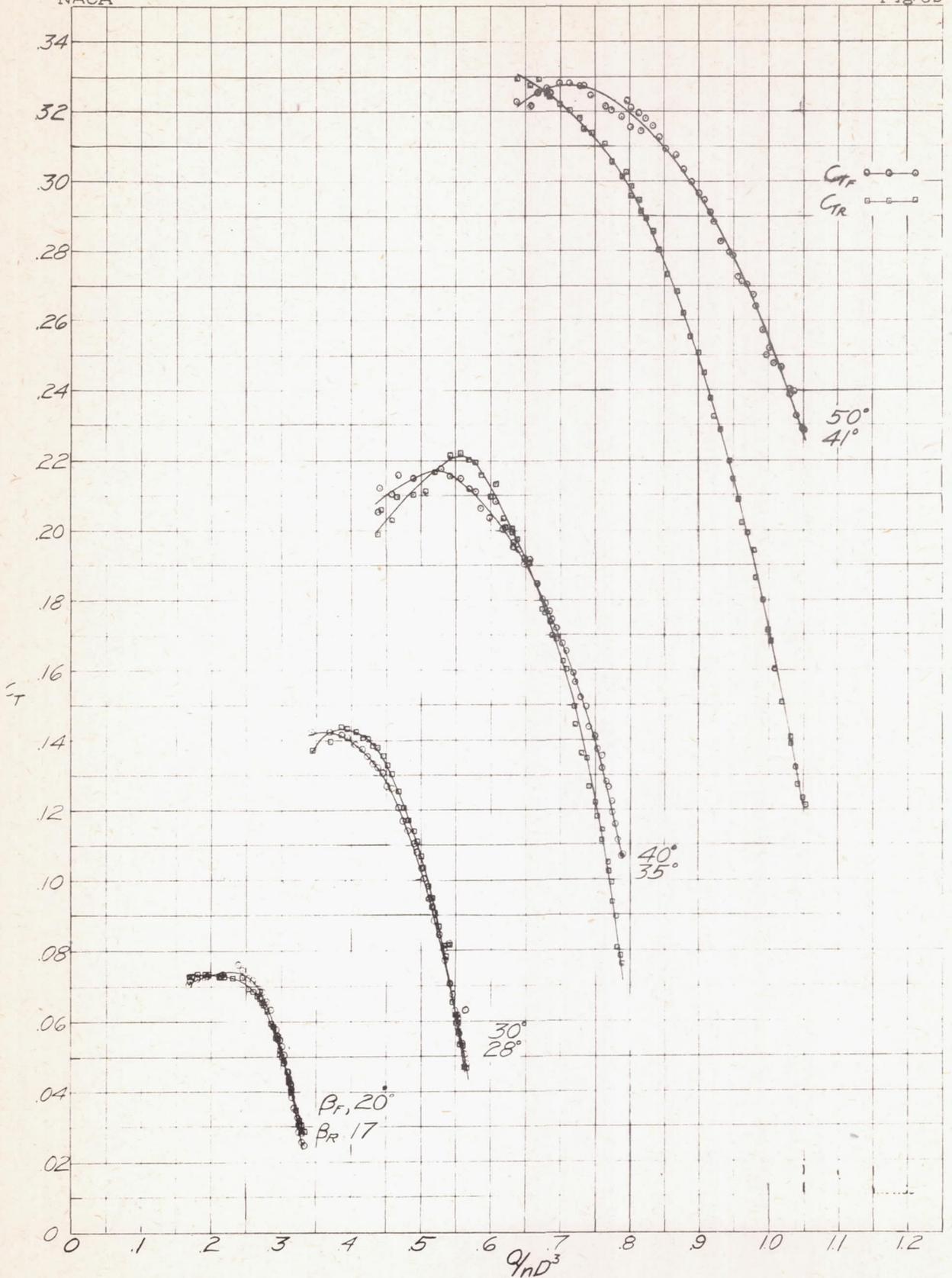


(d) Fan efficiencies.
 Figure 8.- Concluded.

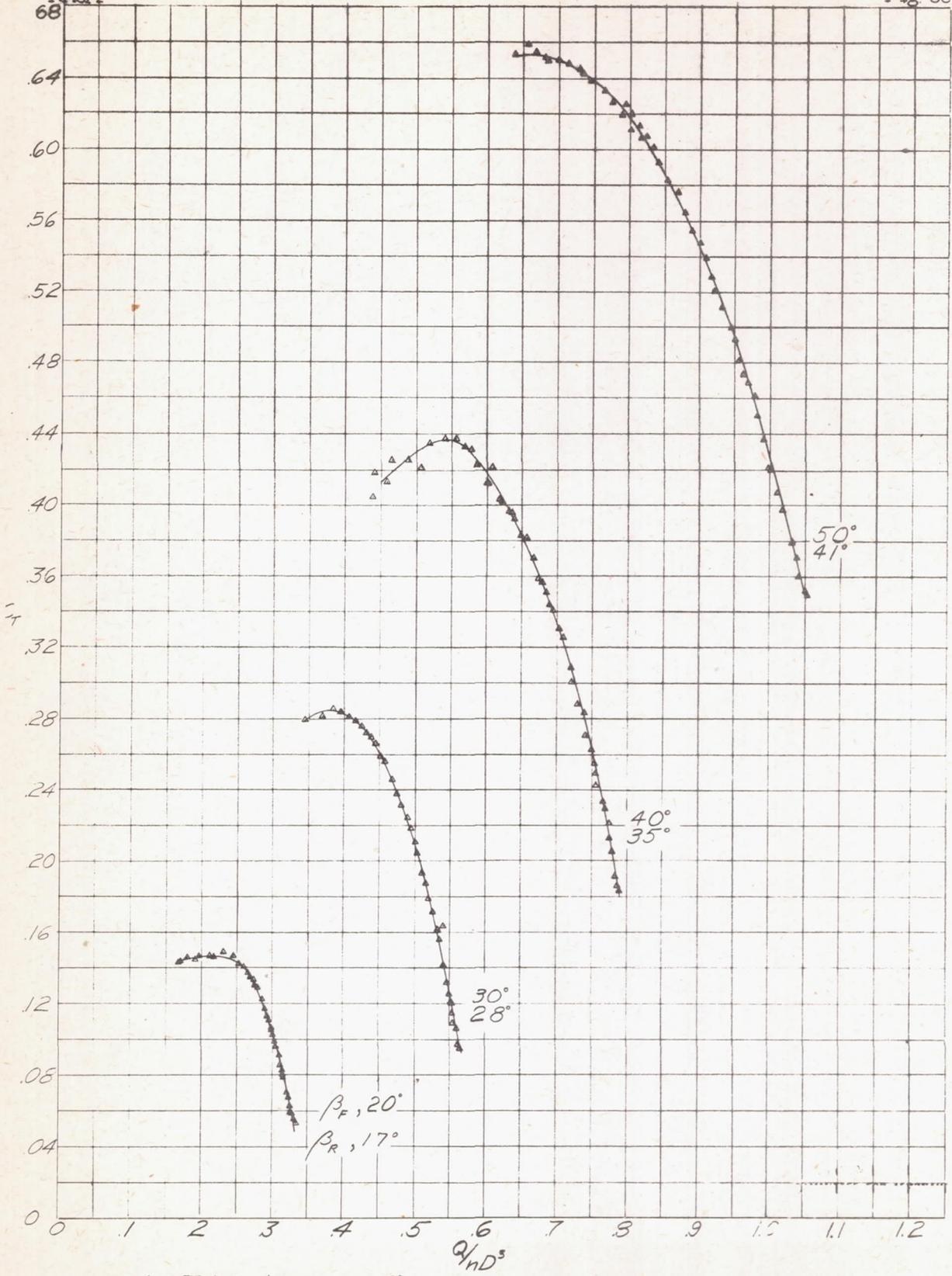
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Fig 8b



(b) Individual torque coefficients.
Figure 8.- Continued



(c) Total torque coefficients.
Figure 8.- Continued.