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Tests on a Typical Whittle Compressor

By

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July, 1952

Summary.—Tests have been carried out to determine the performance of a typical Whittle compressor (produced in 1944) as used in the W2/700 series of engines. At 1,500 ft/sec tip speed the overall pressure ratio (total head) was 4·27 and the adiabatic efficiency was 75·6 per cent at a mass flow of 36·3 lb/sec. The overall efficiency falls rapidly from 79·6 per cent at 1,400 ft/sec to 75·6 per cent at 1,500 ft/sec due to choking of the outer half of the impeller eye.

The maximum diffuser efficiency is approximately 80 per cent at a tip speed of 1,500 ft/sec.

1. *Introduction.*—During the development of the Whittle W2/700 engine (rated thrust 2,020 lb) various combinations of impeller and diffuser design were investigated. This report covers a combination which was produced in 1944 towards the end of this development work.

2. *Design Details.*—The impeller was designated No. 8 design and was a 29-vane double-sided impeller. It was designed to remove the danger of impeller failure, due to resonance of the impeller vanes within the engine running speed.

The leading details are shown in Fig. 1.

Impeller tip diameter	20·68 in.
Rotating guide-vane tip diameter	11·81 in.
Rotating guide-vane root diameter	5·8 in.

The non pre-whirl rotating intake guide vanes were bent to the angles shown in Fig. 2 and were designed assuming constant axial inlet velocity.

The diffuser was designated Type 16 and was a corner-vane type, major design details being as follows:

Throat width	2·084 in.
Throat area	33·4 sq in.
Area ratio (first section)	1·56

Inlet hump giving air stream on outside radius an inlet deflection of 4 deg to the diffuser centre-line.

Divergence angle	11 deg
Corner-vane camber angle	90 deg
Overall area ratio	3·2

* N.G.T.E. Report R.121, received 25th August, 1952.

3. *Installation and Instrumentation.*—The compressor was installed in a test cell, the drive being direct from a steam turbine.

Pressure measurements taken included cell pressure (the compressor inlet pressure was assumed equal to this), total-head pressure at outlet, (measured by means of pitot-tubes in the outlet branches after the diffuser), and static pressures in the impeller channel.

To measure the static pressures in the impeller channel the blower casing was drilled at several points as shown in Figs. 3 and 4.

Compressor inlet temperature was measured by two resistance-type thermometers, and the outlet temperatures by four thermocouples mounted in the outlet branch.

The airflow through the compressor was measured by a calibrated venturimeter through which air was drawn into the test cell. The temperature of the air was measured by a resistance element mounted in the venturi inlet.

4. *Testing Procedure.*—The compressor was run at corrected tip speeds of 1,500, 1,400, 1,300, 1,200 and 1,000 ft/sec. At each speed the readings commenced at choking mass flows and the compressor outlet was successively throttled until surging occurred. All static pressure and normal characteristic readings were taken at each throttle setting.

5. *Performance.*—5.1. *Overall Characteristics.*—Pressure ratio and overall adiabatic efficiency were plotted against corrected mass flow and are shown in Figs. 5 and 6 respectively.

Pressure ratio and adiabatic efficiency were calculated from the total head measured in the outlet branches. The values plotted in dotted lines on the same graphs were based on 100 per cent loss of dynamic head after the outlet.

The performance of the compressor at peak pressure ratio assuming no dynamic head loss at outlet was:

Tip speed	ft/sec	1,400	1,500
Peak pressure ratio	—	3·78	4·22
Mass flow	lb/sec	33·0	36·3
Efficiency	per cent	79·0	75·6

There was a rapid fall-off in efficiency from 79 per cent at 1,400 ft/sec to 75·6 per cent at 1,500 ft/sec. Moreover, choking mass flow, which should increase linearly with increasing speed, fails to do so between 1,400 and 1,500 ft/sec. Choking at the impeller eye was responsible for this deterioration in performance.

5.2. *Airflow at the Impeller Eye.*—A series of traverses of the front intake of the compressor, when part of a complete engine, were carried out to determine the radial variation of axial velocity at the impeller eye.

It was found that appreciable pressure losses occur during the passage of the air through the intake chutes and gauzes, but more important, that the axial velocity of the entering air varies considerably from the root to the tip of the rotating guide vanes. Over a range of tip speeds from 1,500 ft/sec to 1,300 ft/sec it was found that the axial velocity at the tip ($11 \cdot 81$ in. diameter) was $1 \cdot 54\ \bar{V}_a$ and at the root $0 \cdot 67\ \bar{V}_a$ with almost linear variation across the eye. $\bar{V}_a$ is the mean axial velocity given by

$$\bar{V}_a = \Sigma \frac{V_a \, . \, 2\pi r \, dr}{A} \, .$$

In Fig. 7 the Mach number (M_n) of the air stream relative to the intake guide vanes has been plotted against distance across the impeller eye for three speeds. The Mach number at which choking occurs has also been deduced and plotted on the same graph. It will be seen that at a tip speed of 1,500 ft/sec the impeller eye is choking from the tip down to the mean diameter of the eye.

A comparison of air inlet angles with blade inlet angles at a tip speed of 1,500 ft/sec shows that the rotating intake guide vanes are working with a large positive incidence (approximately +20 deg) at the root and a small positive incidence (+4 deg) at the mean diameter and the tip.

5.3. *Static Pressures in the Impeller Channel.*—Static-pressure ratio has been plotted against impeller tip speed in Fig. 8. It will be seen that all the pressure rise takes place after the bend in the impeller channel and that considerable depression exists at the eye and bend, although these are partly due to centrifugal action.

The flow in the impeller channel defies accurate analysis by means of these static pressures. The total circumferential variation of static pressure at the impeller tip across half a diffuser pitch was 10 per cent of the mean value at choking mass flow and 15 per cent at surge point. This pressure variation was noticeable throughout the channel and although the amplitude decreased towards the eye, the ratio of amplitude at surge point to that at choking was always about $1 \cdot 5$.

At a given speed the static pressures in the impeller channel showed a slight tendency to rise with reduced mass flow and increased outlet pressure.

5.4. *Diffuser Efficiency.*—Diffuser efficiency and overall compressor efficiency are shown plotted against corrected mass flow for an impeller tip speed of 1,400 ft/sec in Fig. 9.

The curves of diffuser efficiency for other speeds were of a similar shape and the maximum diffuser efficiency in all cases was between 78 per cent and 81 per cent. It will be noticed that the diffuser efficiency, as measured between the throat section of the diffuser and outlet, rapidly attains its maximum value as soon as over-expansion and choking cease in the diffuser throat. Thereafter the diffuser efficiency falls off, rapidly at first, and then more slowly until it remains constant at about 10 per cent below peak value. This fall-off in diffuser efficiency is presumably due to a change in flow conditions through the diffuser. Initially, at peak efficiency the air accelerated from the vortex space into the diffuser throat and at no point attained sonic velocity. However, with further throttling at outlet, the pressure at the throat increases rapidly until the air is being retarded as it passes into the throat. The continued increase in overall compressor efficiency when the diffuser efficiency is falling off is due to this ram effect at the diffuser entry. Eventually this retardation causes a pressure at the throat greater than can be held by the centrifugal force of the air in the vaneless space. The resulting breakdown of radial equilibrium and its consequent disturbances are known as 'surging'.

5.5. *Slip Factor and Temperature Rise.*—In Figs. 10 and 11 are shown slip factor and temperature ratio plotted against corrected mass flow.

Slip factor in this case is the product of the factor allowing for relative eddy and that for friction loss in the vanes.

6. *Conclusions.*—At 1,500 ft/sec the overall pressure ratio (total head) was 4·17 with an adiabatic efficiency of 75·6 per cent at an air mass flow of 36·3 lb/sec. Choking in the outer half of the impeller eye causes the overall efficiency to fall rapidly from 79·6 per cent at 1,400 ft/sec to 75·6 per cent at 1,500 ft/sec.

With a reduction of Mach number at the impeller eye tip and the correct matching of air and blade angles an increase in efficiency of up to 3 per cent should be possible.

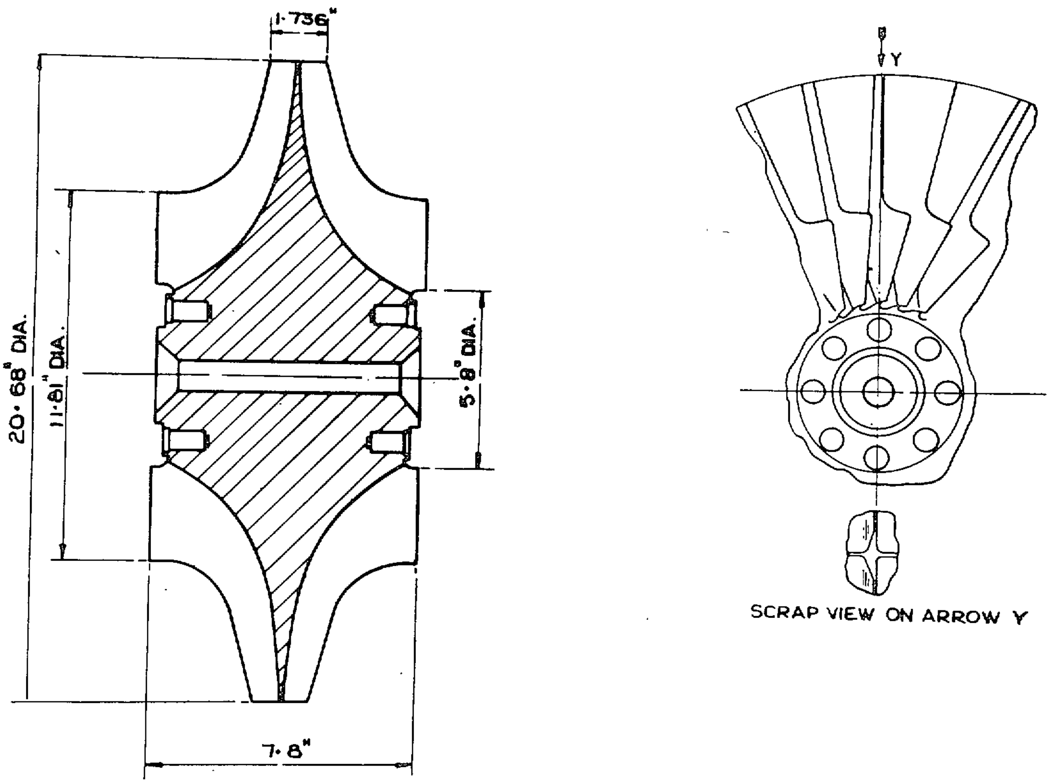

FIG. 1. The No. 8 Design impeller.

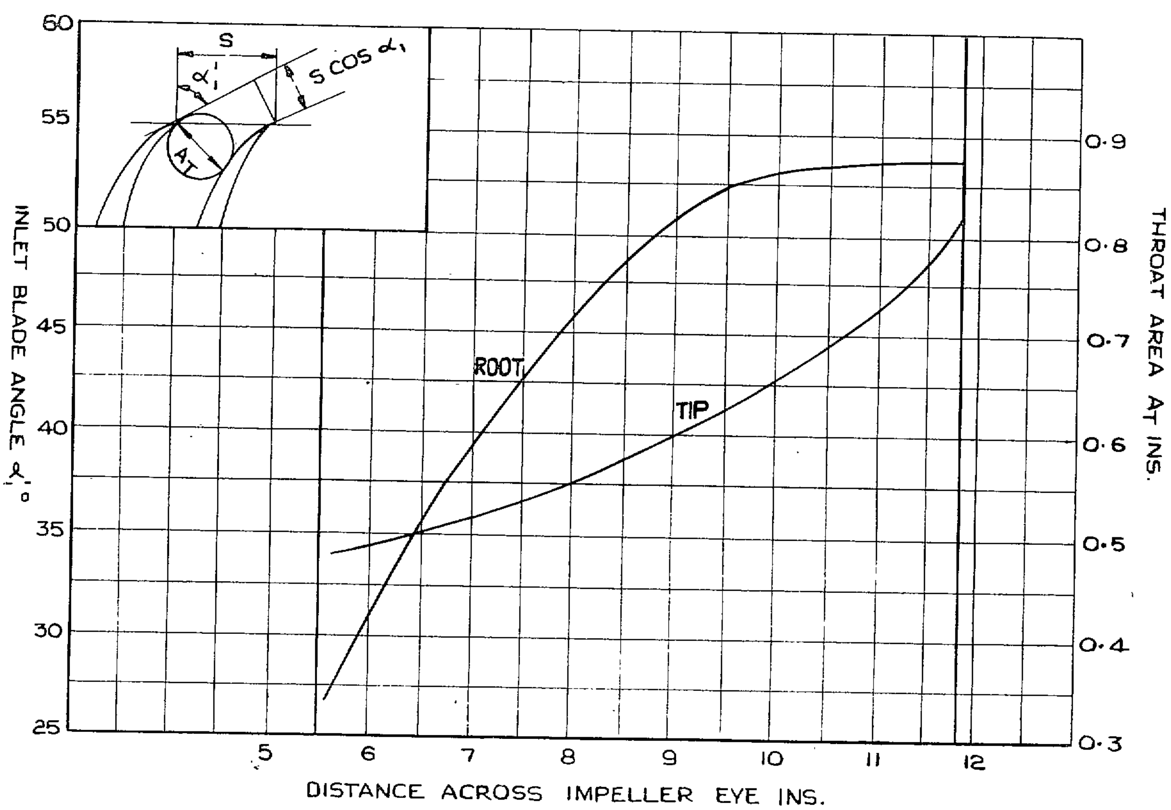

FIG. 2. Blade angles and throat areas.

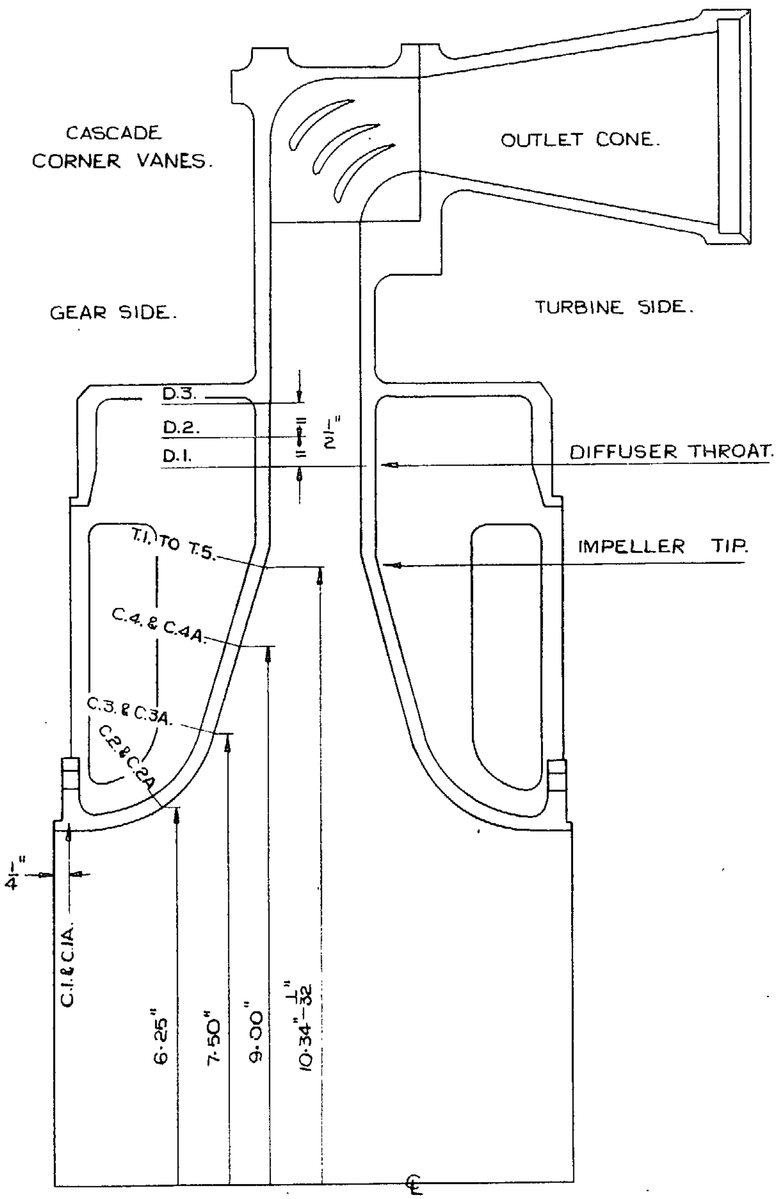

Fig. 3. Location of static points on blower casing.

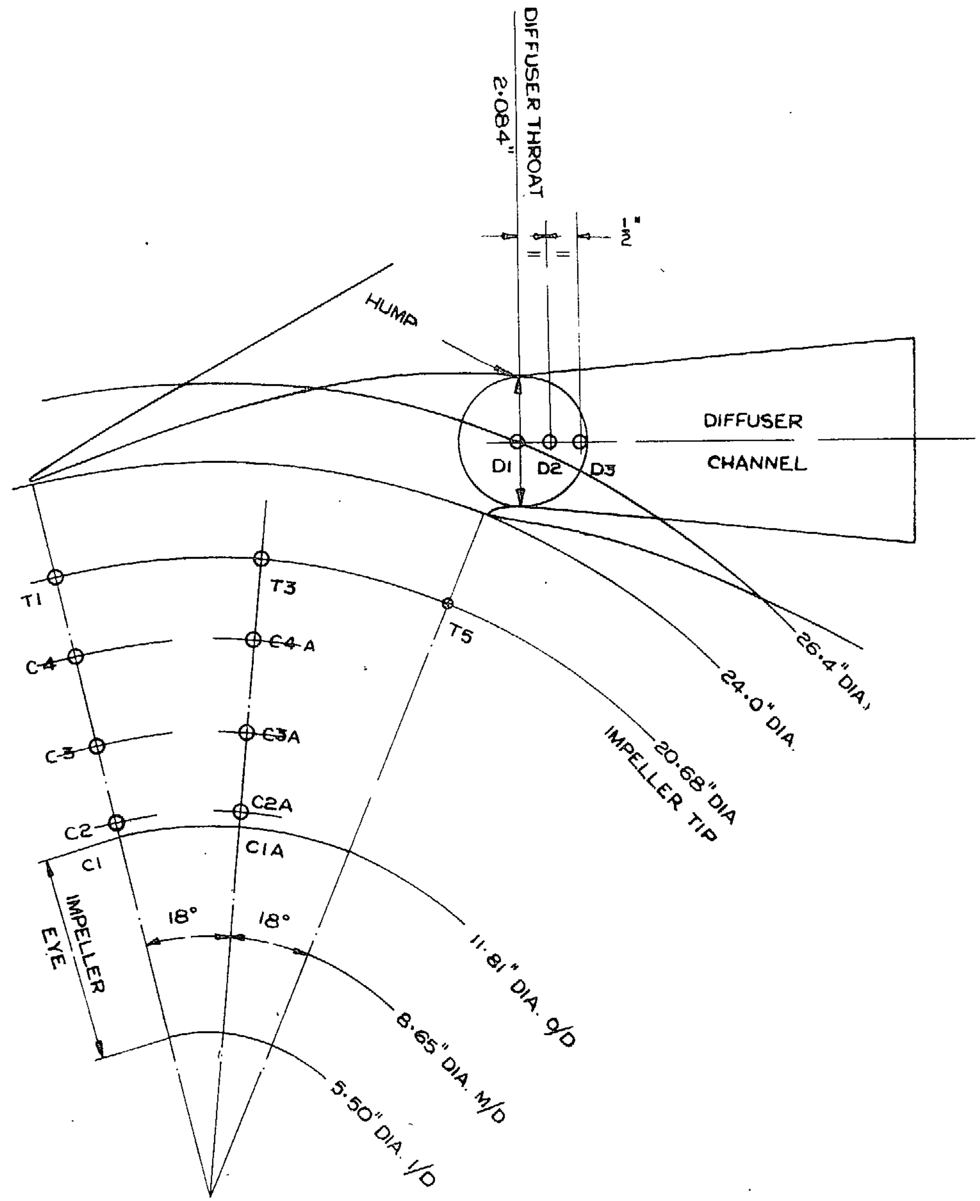

Fig. 4. Location of static points on blower casing.

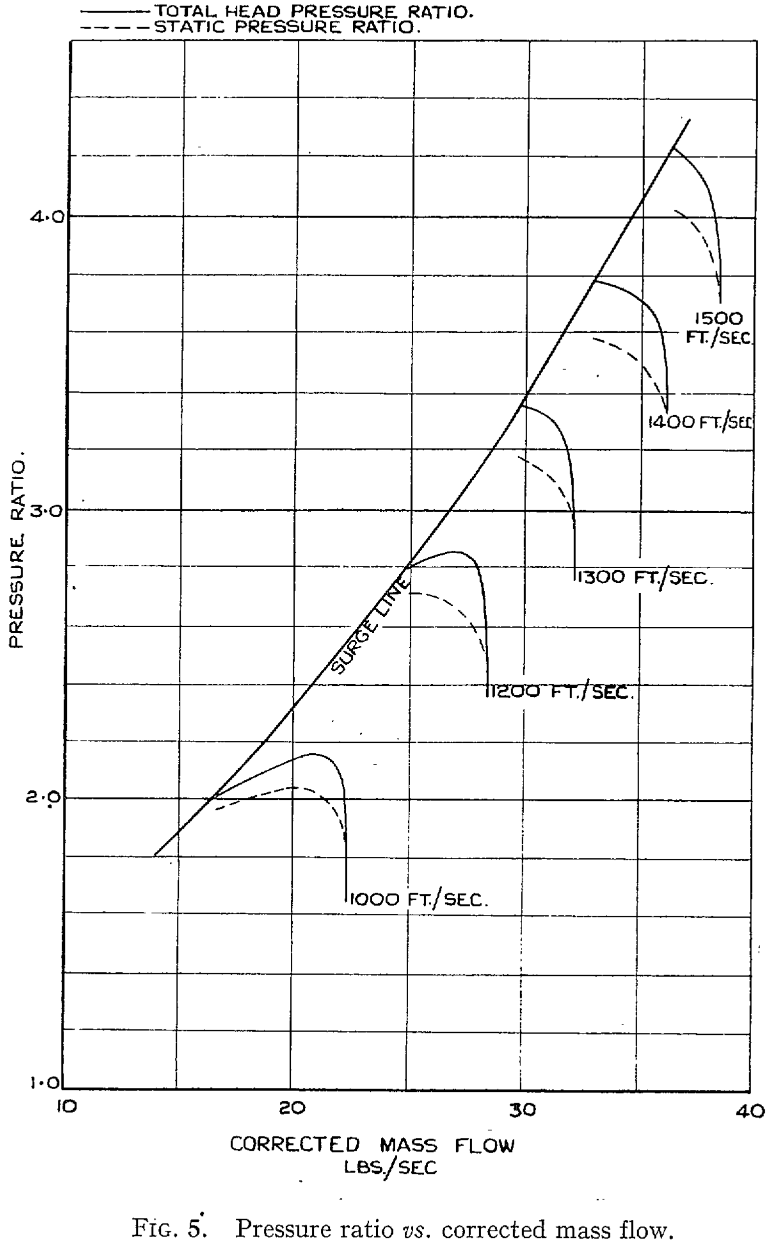

FIG. 5. Pressure ratio *vs.* corrected mass flow.

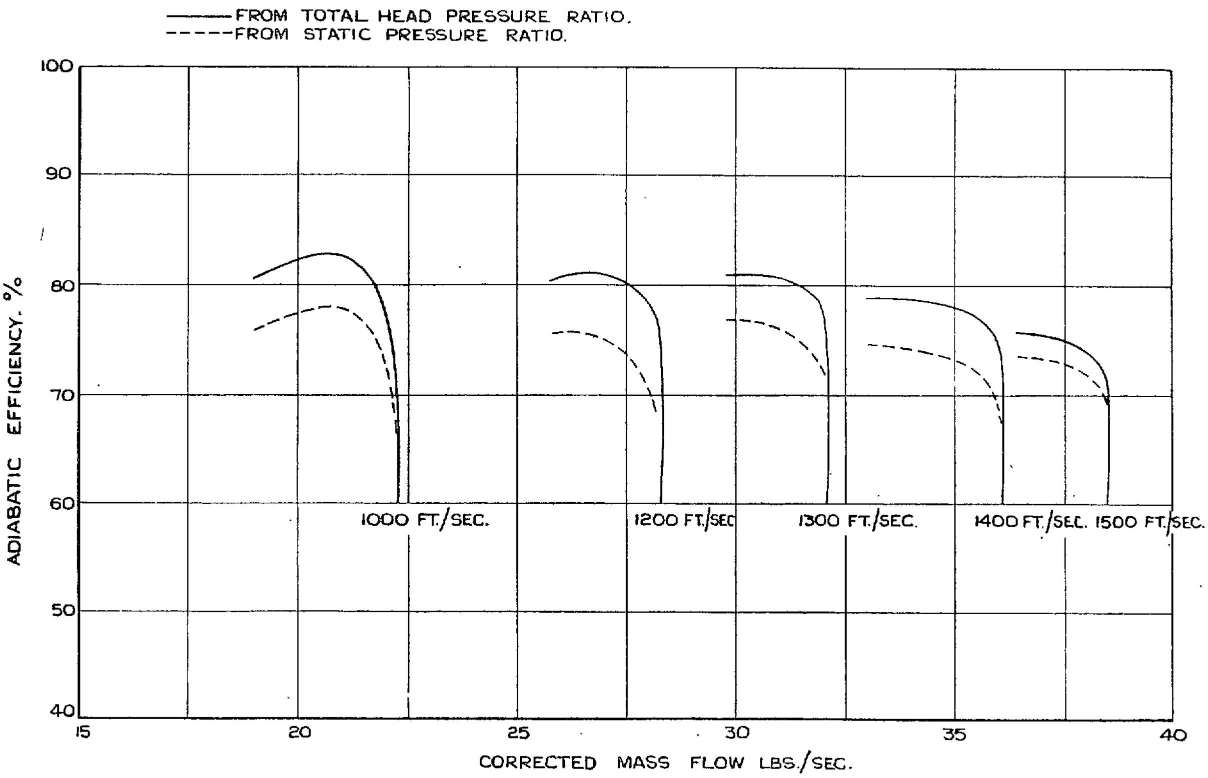

FIG. 6. Overall adiabatic efficiency *vs.* corrected mass flow.

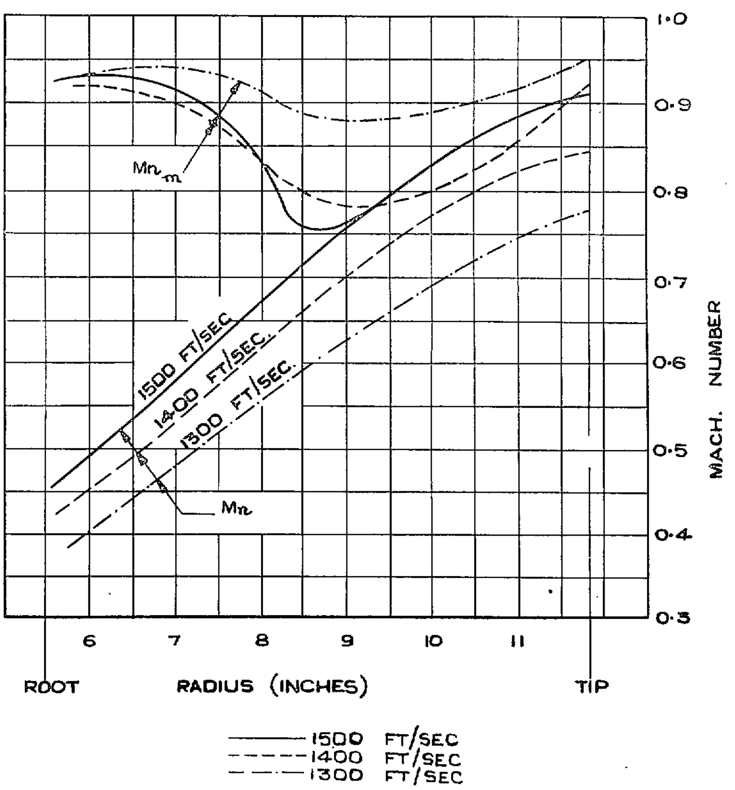

FIG. 7. Variation of Mach number across impeller eye. Choking occurs when $M_n > M_{nm}$.

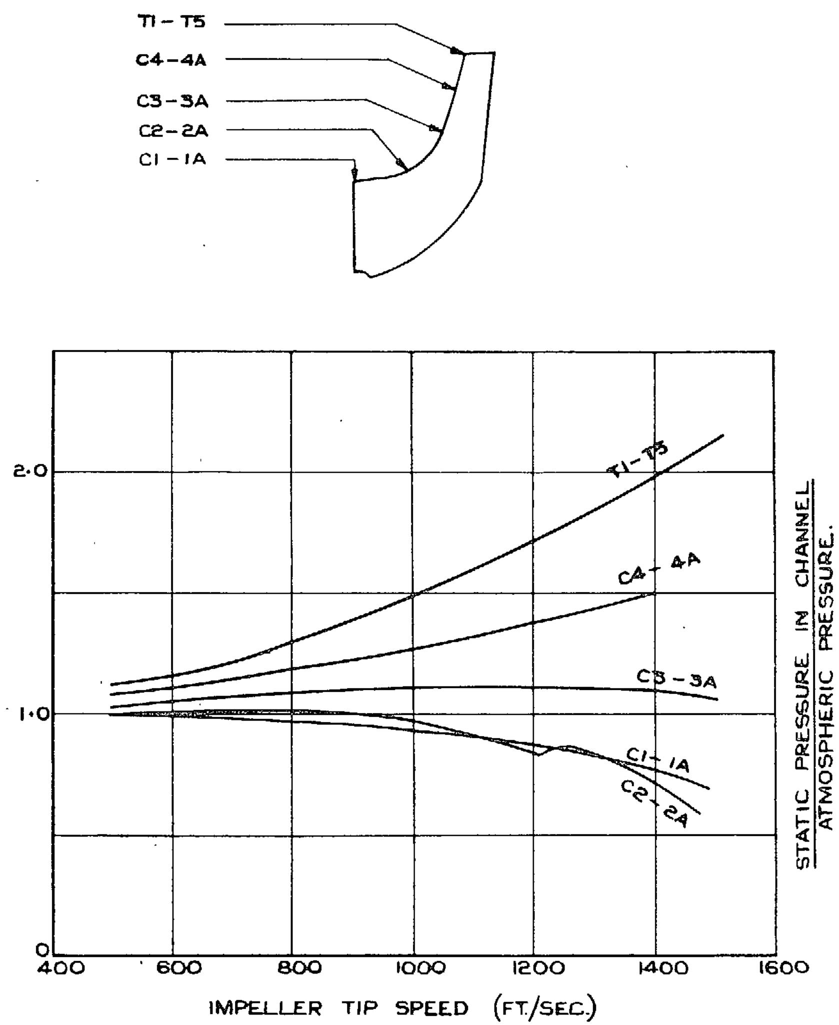

FIG. 8. Static pressures in impeller channels (at peak compression ratio).

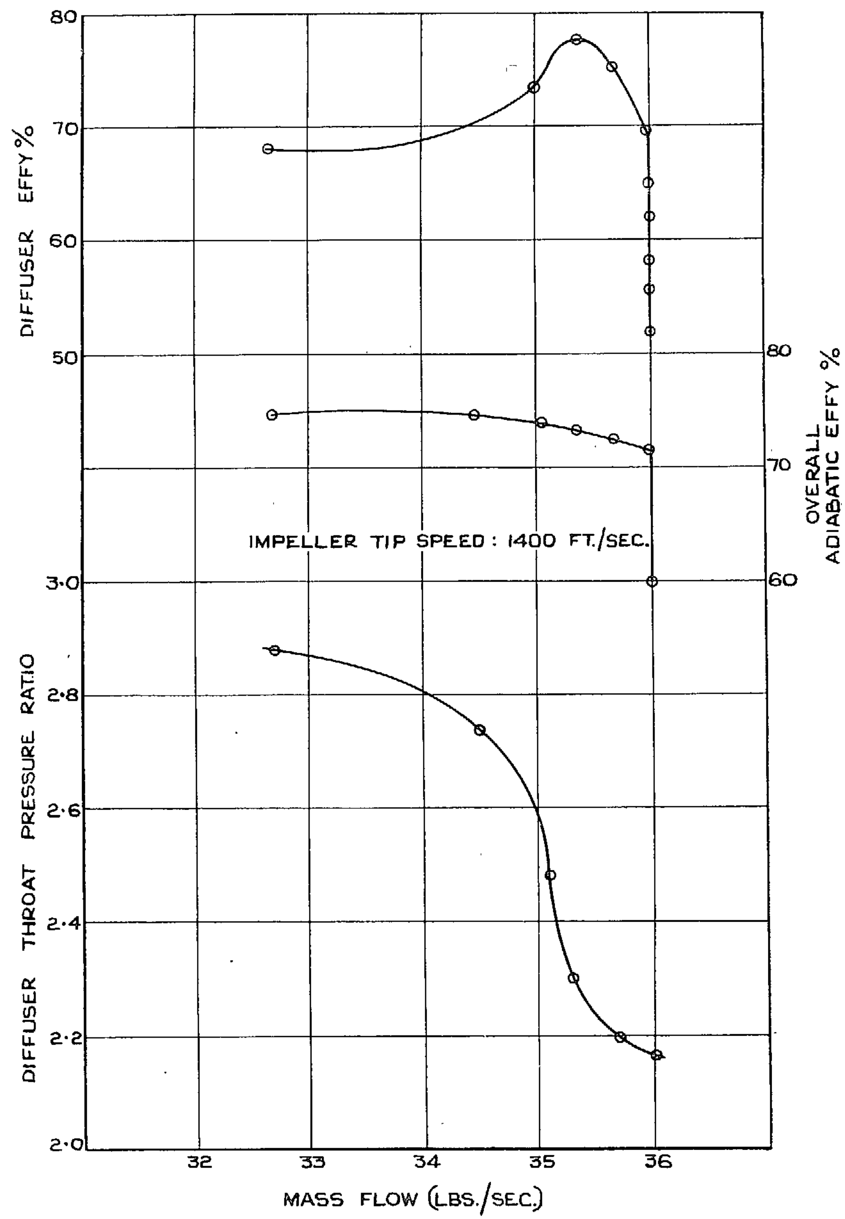

FIG. 9. Diffuser efficiency *vs*. corrected mass flow.

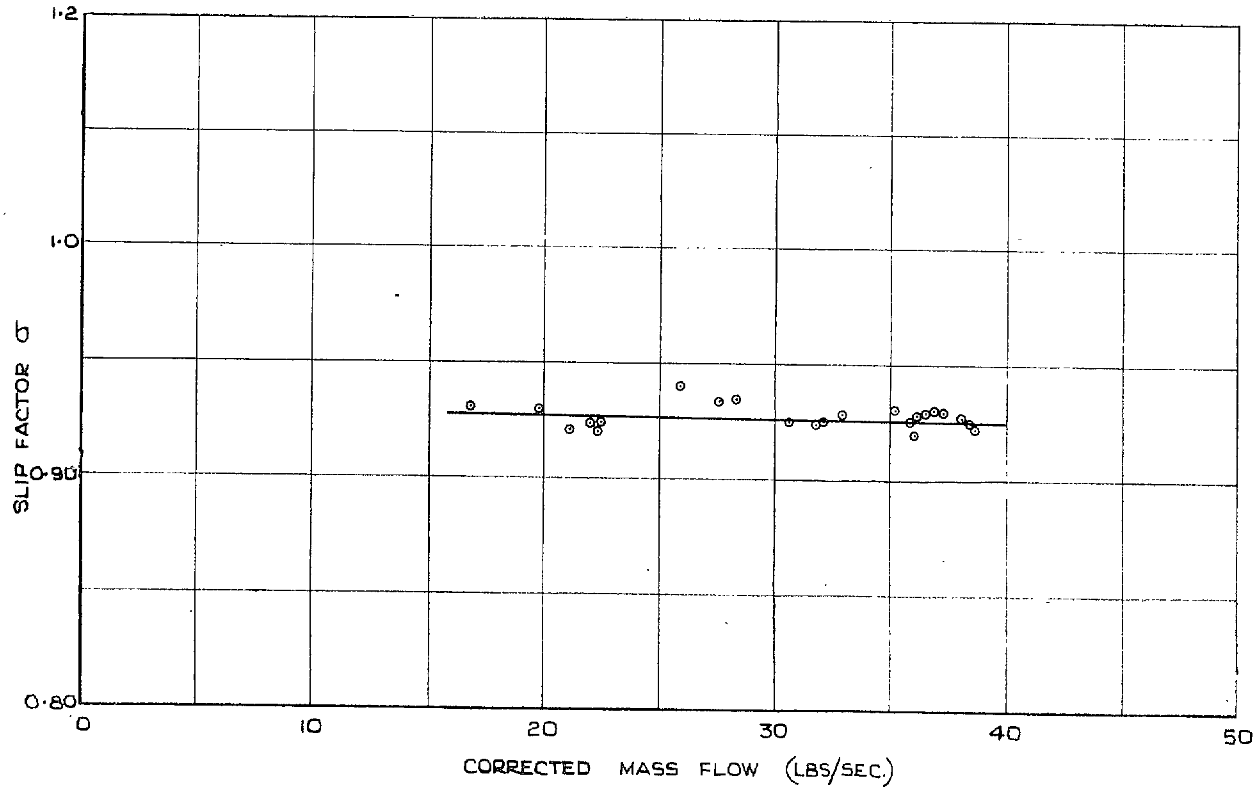

Fig. 10. Slip factor *vs.* corrected mass flow.

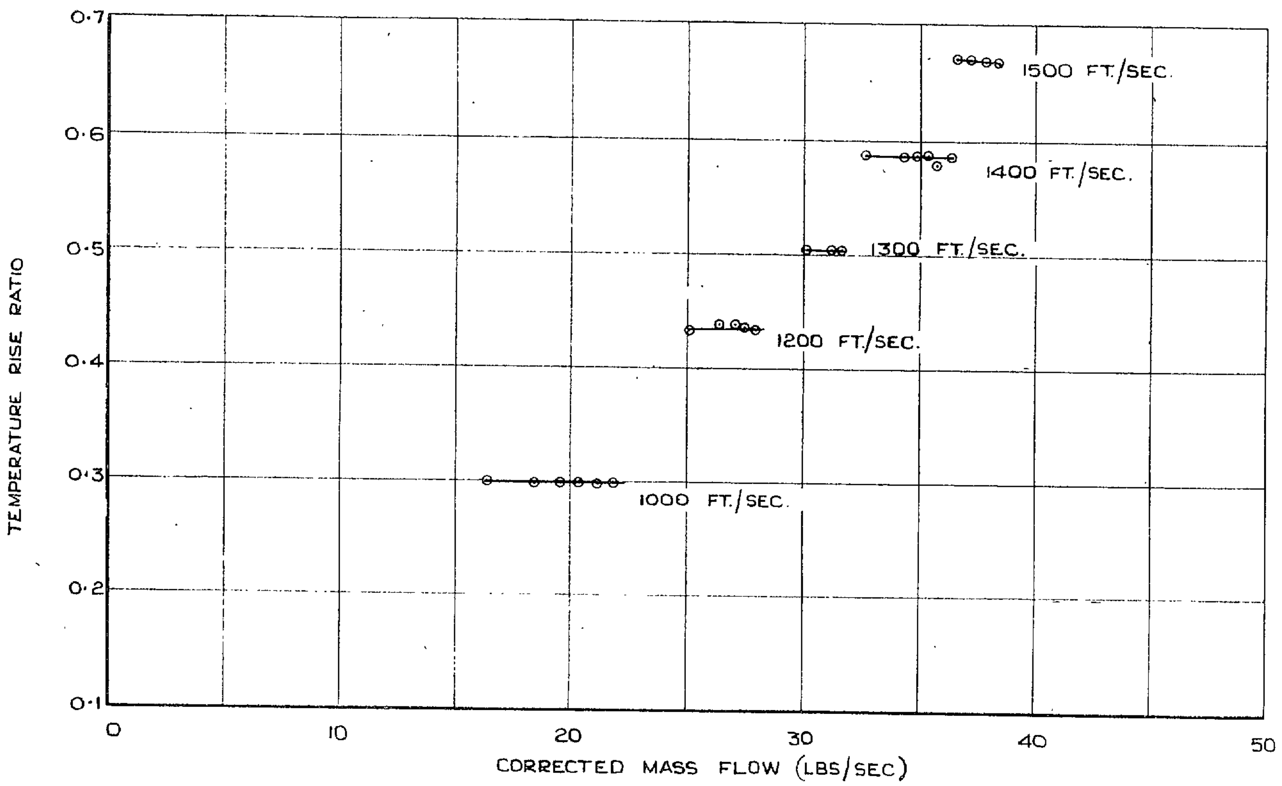

Fig. 11. Temperature rise *vs.* corrected mass flow.

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