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R. & M. No.

The Application of Dimensional Relationships to Air Compressors, with special reference to the Variation of Performance with Inlet Conditions.

- By -

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Introductory. (Reasons for Investigation).

It is frequently required to calculate the performance of a compressor at height from tests under ground level conditions of atmospheric pressure and temperature.

Range of Investigation.

It is shown by consideration of the dimensions of the quantities involved that, if heat flow through the casing, distortion of the casing and scale effect are negligible, the compression ratio and the other variables on which the performance of a compressor depends can be expressed as a function of two variables only. If the compression ratios, etc. are plotted against one of these variables with the other as parameter, the resulting diagram is applicable to all inlet conditions.

To check the assumption that heat flow through the casing, distortion of the casing, and scale effect are negligible, experiments were made on a centrifugal type compressor housed in a test chamber in which the pressure and temperature of the air could be varied at will. The effect of heating the compressor casing was also examined, but the inlet conditions were not varied

during /

during these tests.

Conclusions.

Under normal conditions of operation, that is to say, without special arrangements for heating the casing of the compressor, the performance of the compressor can be represented with sufficient accuracy in terms of the two fundamental variables suggested by a consideration of dimensions.

Heating the casing to 100°C. has little effect on the compression ratio, but there is an appreciable increase in the rise of temperature through the compressor, the increase varying from 13°C. at high rates of rotation, to 28°C. at low rates. The adiabatic efficiency appeared to be little affected, but some uncertainty attaches to this conclusion as it is based on certain assumptions as to the mechanical efficiency.

Further Developments.

With the aid of the dimensional relationships derived in the present report, an attempt is being made to reduce the performance of an engine supercharged by a gear driven centrifugal compressor to the product of a power factor analogous to that applicable to a normally aspirated engine, and a function of rotation rate.

Introduction.

A consideration of the dimensions of the physical quantities involved in any problem enables us to reduce the number of variables on which the solution of the problem depends. If there are N physical quantities whose dimensions in length, mass, and time are known, then

by /

by well known methods the problem may usually be expressed in terms of N - 3 groups of the original N quantities; that is to say, the number of quantities we need consider separately is reduced by three. The solution of certain problems is given merely by the reduction in the number of independent quantities obtained in this way; and amongst these is the problem forming the subject of this report, namely, the calculation of the performance of a compressor at one set of intake conditions from tests at another set.

While the information obtained by reference to dimensions is limited in range, it is exact provided that all the quantities which are influencing the results are taken into account. In the application to the compressor, it will be found convenient in practice to neglect scale effect, the effects of distortion of the casing, and the heat flow through the casing, and experiments are required to ascertain the effect of their omission.

In Part I the required dimensional relationships are derived, and in Part II the experiments to test their validity are described.

The discussion in Part I is applicable to centrifugal or displacement compressors. The experiments in Part II were made on a centrifugal compressor only.

PART I.

To simplify the problem in the first instance, it will be supposed that the walls of the compressor are non-conducting. The effect of removing this restriction will be examined subsequently.

<u>The dimensional relationships when the walls of the compressor are non-conducting.</u>

The physical quantities with which we are concerned, associated with the working fluid, may be taken to be density, pressure, the kinematic viscosity, and the ratio of the specific heats. The quantities in which we are interested are therefore:

W, the mass of fluid inspired in unit time, i.e. the mass flow.

p_0 and p_1, the pressures at intake and delivery.

ρ_0 and ρ_1, " densities " " " "

u_0 and u_1, " velocities " " " "

ν, the kinematic viscosity.

γ, the ratio of the specific heats.

n, the rate of rotation of the rotor of the compressor.

D, a linear dimension of the compressor.

P, the power.

We will suppose in the first instance that the compressor is inspiring from a large reservoir in which the pressure is p_0, and delivering to another reservoir

in /

in which the pressure is p_1, so that u_0 and u_1 may be taken to be zero. Of the remaining ten quantities, seven only are independent. It is evident, for example, that if p_0, ρ_0, p_1, n, D, ν, and γ are given, the flow through the compressor is completely defined, and it must therefore be possible to derive the values of the remainder in terms of these. We may choose any seven of the quantities as independent, with the proviso only that they are truly independent; that is to say, that the value of each may be varied without necessarily affecting the values of the others. It is convenient to choose

$$W, p_0, \rho_0, \nu, \gamma, D, n,$$

as independent, leaving

$$p_1, \rho_1, P,$$

as dependent.

By the customary procedure for deriving dimensional relationships,* we find that any non-dimensional variable associated with the performance of the compressor may be expressed as a function of the non-dimensional variables

$$\frac{W}{D^2\sqrt{p_0\rho_0}}, \quad nD\sqrt{\frac{\rho_0}{p_0}}, \quad \frac{nD^2}{\nu}, \gamma$$

and /

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* The rule for the derivation of such relationships may be expressed thus:-

If there are N quantities, of which M are independent, build up the M quantities into M-3 non-dimensional quantities, taking care that each enters at least once. The N-M dependent quantities may then be converted to non-dimensional form and each expressed as a function of the M-3 independent non-dimensional variables.

and therefore

$$\frac{p_1}{p_0}, \frac{\rho_1}{\rho_0}, \frac{T_1}{T_0}, \frac{P}{Wn^2D^2}, \quad \eta = f\left(\frac{W}{D^2\sqrt{p_0\rho_0}}, \frac{nD\sqrt{\rho_0}}{\sqrt{p_0}}, \frac{nD^2}{\nu}, \gamma\right)$$

where f denotes functional relationship.

The temperature ratio T_1/T_0, and η, the adiabatic efficiency, both of which are of course non-dimensional, have been included on the left hand side.

If the working fluid is specified, we do not require to include γ explicitly, and we may replace p_0/ρ_0 by T_0. Further, if the size of the compressor is not varied, D may be omitted. Finally, if scale effect may be supposed small, we shall not require the variable nD^2/ν. The variables on the right hand side of the equations may under these conditions be replaced by

$$\frac{W}{\sqrt{p_0\rho_0}} \text{ or } \frac{W\sqrt{T_0}}{p_0}, \quad \frac{n}{\sqrt{T_0}}$$

From the results of tests of a compressor at any given intake pressure and temperature, we may plot the compression ratio and the other variables on the left hand side of the equations against $W/\sqrt{p_0\rho_0}$ in sets of curves with parameter $n/\sqrt{T_0}$, and the values of the compression ratio and the other dependent variables for any other conditions at the intake may be derived at once. In Figs. 1 and 2 families of curves of this type are shown, derived from tests of a centrifugal compressor. They will be more fully described in Part II.

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If the atmosphere from which the compressor inspires is moving relative to the compressor with velocity V, a new independent variable is, of course, introduced, and we might include it by adding a third non-dimensional variable, $V\sqrt{\rho_0/p_0}$. If we may assume that the air enters the intake of the compressor without appreciable dissipation of energy, then we may avoid including the new variable explicitly by defining p_0 and ρ_0 as the pressure and density of the air when brought to rest adiabatically relative to the compressor; that is to say, p_0 is taken to be the pitot pressure, and to a sufficient approximation for all speeds likely to be encountered, ρ_0 may be considered to be unchanged. It may assist in understanding this if we suppose the air to be brought to rest adiabatically in the reservoir supplying the compressor and use the pressure and density in that reservoir as p_0 and ρ_0 in the dimensional relationships. It is, of course, not necessary that the air should actually be brought to rest before entering the compressor.

Up to the present it has been supposed that there is no transference of heat through the compresser casing. The effect of removing this restriction will now be examined.

<u>The effect of including heat transferences through the walls of the compressor.</u>

In general cooling or heating the compressor casing will give rise to additional independent variables. If, for example, the casing is cooled by a stream of air of constant velocity V, and of pressure and density p_a and ρ_a, it will be necessary to include three new non-

dimensional /

dimensional quantities under the functional sign in the dimensional relationships. The additional quantities may be formed in several ways; they may, for example, be

$$\frac{\rho_a}{\rho_o}, \quad \frac{p_a}{p_o}, \quad \frac{V}{nD}$$

These must therefore be added to our independent variables $W/\sqrt{p_o \rho_o}$ and $n/\sqrt{T_o}$; and it is evident that, except in the artificial conditions, when ρ_a, p_a, V, are respectively proportional to ρ_o, p_o, nD, one or more of these quantities will vary. An increase in the number of independent variables would be inconvenient, for it would lead to more than one parameter in our families of curves representing the performance of the compressor on a non-dimensional basis. A strictly accurate representation of the compressor performance, in terms of two variables, fails, therefore, in this instance; and it may be shown in a similar manner that it will also fail if there is cooling of the compressor casing by convection.

In Part II tests of a centrifugal compressor are described in which the compressor is

(a) in the atmosphere from which the compressor inspires.

(b) in an atmosphere of steam.

In (a) the casing is probably cooled partly by convection and partly by an air stream, while in (b) the casing is heated to a temperature of 100°C. The effect on the compressor performance of the very different conditions of heat flow through the casing which must

exist /

exist in (a) and (b) is found to be small, showing itself mainly in a change in the temperature ratio T_1/T_0. The effect of a large change in the heat flow through the casing appears, therefore, to be small, and since the variables which account for the heat flow probably only vary in practice in narrow limits, it would be expected that only very small errors would result if they were assumed to have constant values.

If, therefore, we carry out the tests on which the diagrams representing the compressor performance are based under heating or cooling conditions similar to those obtaining under the conditions of operation in the aircraft, it seems probable that sufficiently accurate estimates of compressor performance may be obtained from them under all conditions of flight. By this procedure it will be understood that we do not neglect the effect of heat flow entirely. We merely assume that over the conditions of operation it is sufficient to take constant values of the variables which take account of the flow, the constant values lying within their range of variation.

As an alternative to testing the compressor under the appropriate heating or cooling conditions, we may perhaps correct the results obtained under normal compressor test bench conditions to allow for the change of heat flow when the compressor is in the aircraft. If, for example, we may assume that the compressor is heated to approximately 100°C. by proximity to the engine, an adjustment of the T_1/T_0 curves in accordance with the results of the tests described in Part II would be sufficient.

Conclusions of Part I.

It has been shown that the performance of a compressor may be expressed in terms of a number of variables, of which the most important are $W/\sqrt{p_0\rho_0}$ and $n/\sqrt{T_0}$. The variables omitted take account of scale effect and heat flow through the casing. Distortion of the casing and rotor may introduce others. The additional variables are not usually taken into account in discussions of compressor performance, and their effect is probably small over the ranges of rotation rates, pressures, etc., required in practice. It will be assumed, therefore, that the compressor performance may be formally represented by the equations

$$\frac{p_1}{p_0}, \frac{\rho_1}{\rho_0}, \frac{T_1}{T_0}, \frac{P}{Wn^2}, \eta = f\left(\frac{W}{\sqrt{p_0\rho_0}}, \frac{n}{\sqrt{T_0}}\right)$$

and in Part II experiments on a centrifugal compressor designed to show the accuracy obtainable by this representation in a specific instance will be described. In these equations p_0 and p_1 may be either static or pitot pressures measured at any two points in the compressor, or in the atmospheres from which the air is inspired, and to which it is delivered respectively, with the proviso that if the compressor is moving relative to the former atmosphere and the intake is facing into the relative wind, the pitot pressure must be used for p_0.

In the experiments described in Part II, p_0 and p_1 are static pressures.

PART II.

In Part I dimensional relationships applicable to air compressors are derived. On practical considerations it is convenient to assume scale effect heat flow through the casing, and distortion of the casing to be negligible. The validity of this assumption in the case of the centrifugal compressor has been checked by tests on a Lion supercharger housed in a test chamber in which the pressure and temperature of the air could be varied at will. The observed values of the compression ratio and the other dependent variables were plotted against $W/\sqrt{p_0/\rho_0}$ for several selected values of $n/\sqrt{T_0}$, for a range of inlet conditions, and the validity of the adopted relationships is indicated by the closeness with which the plotted points lie on a single curve for each value of $n/\sqrt{T_0}$. The effect of varying the intake pressure at approximately constant temperature was first examined, and subsequently the intake temperature was also varied. At a later stage in the experiments the effect of heating the compressor casing was investigated. In these last experiments it was not possible to arrange for variation of the pressure and temperature of the inspired air, and they do not therefore check the validity of the dimensional relationships.

The supercharger, as initially constructed, was of the exhaust-driven type. For the purposes of these tests the exhaust turbine was detached from the

compressor /

compressor unit, which was fitted to a gearbox designed for experimental work. The impeller was driven by an electric motor through a train of gears providing a velocity ratio of 10 to 1. The chief features of the internal construction of the compressor are shown on Fig. 8.

Tests on the effect of a reduction in intake pressure.

These tests were performed at ground atmosphere intake temperature, which varied from 6° to 18°C. during the course of the experiments. Four series of tests were carried out, at intake gauge pressures of 0, -5, -10 and -15 inches of mercury. The rotation rate of the impeller was adjusted to provide results at four different values of the variable $n/\sqrt{T_0}$ at each of the four conditions of inlet pressure.

The supercharger was mounted within a closed test chamber, from the interior of which its air supply was drawn. Fig. 9 shows the test chamber with the end-cover removed; the general arrangement of the supercharger mounting and part of the delivery system are visible. During the tests air was admitted to the chamber from the surrounding atmosphere through a pipe A fitted with a valve for control of the internal pressure, and the supercharger delivery system was connected to an exhauster pump through a pipe B, enabling the delivery pressure at constant intake conditions to be varied over a wide range.

The pressure and temperature of the air at the supercharger inlet were measured respectively by a

mercury /

mercury gauge connected to an open-ended tube in the side of the test chamber remote from the inlet pipe A and by a nitrogen transmitting thermometer. The spiral bulb of the thermometer is clearly visible in front of the wire gauze covering the air intake in Fig. 9. The delivery pressure and temperature were measured at the junction of the two delivery branches by a "static" tube and mercury gauge and a mercury-invar transmitting thermometer. Both thermometers were calibrated before the commencement of the tests.

The rate of air discharge was determined by means of a calibrated parabolic nozzle fitted in the delivery pipe-line about 10 feet from the compressor. The test conditions were identical with those under which the nozzle had previously been calibrated by orifice plates, using Watson's coefficients for sharp-edged orifices. The nozzle discharge coefficient obtained at the lowest value of the range of Reynold's number covered by the calibration tests was 2%. below that resulting from German experiments on similar nozzles;* and at the higher limit of the range the coefficient was 1%. higher than that of the German tests.

From the measured quantities the values of p_1/p_0, T_1/T_0, $W/\sqrt{p_0/\rho_0}$, $n/\sqrt{T_0}$, have been calculated, and are plotted in Fig. 1. The units adopted are as

follows/

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* Verein Deutscher Ingenieure, No. 27, 1929.

follows:-

n, revolutions of the impeller per minute.

T_o and T_1, degrees C. absolute.

W, lb. of air delivered per minute.

p_o and p_1, lb. per sq. in. absolute.

ρ_o and ρ_1, lb. per cubic foot.

On Fig. 1 the ratios p_1/p_o and T_1/T_o at each of the conditions of intake pressure are plotted for constant values of $n/\sqrt{T_o}$. As $\frac{\rho_1}{\rho_o} = \frac{p_1}{p_o} \div \frac{T_1}{T_o}$, the density ratio has not been included in the diagram.

Slight discrepancies exist in the families of pressure ratio curves, and, as in general the observations obtained during each separate test are consistent, it appears improbable that the discrepancies are entirely accounted for by experimental errors. The maximum difference in the value of the pressure ratios in any of the groups of curves, however, does not exceed 3%. The temperature ratio curves at the various intake pressures are in almost exact agreement, and have been combined in a single curve for each value of $n/\sqrt{T_o}$.

To check the agreement of the curves representing values of the variable $P/(Wn^2)$ and the adiabatic efficiency at constant values of $n/\sqrt{T_o}$, the values of the power, P, are required. They may be obtained either

(i) by deducting the mechanical losses from the measured power

(ii) from the temperature rise through the compressor with the aid of the well-known formula

$$P = W K_p (T_1 - T_0) + \frac{W}{2g}(u_1{}^2 - u_0{}^2) + Q$$

where K_p is the specific heat at constant pressure, and Q is the rate of loss of heat from the compressor casing.

The electric motor driving the supercharger was of the swinging field type, and was provided with means for measuring the power output. The measured power required to be corrected for the mechanical losses. Unfortunately, the losses in the gearbox were probably varying irregularly for the following reason. The gear-box was erected as an integral part of the compressor, and was therefore subjected to the pressure within the test chamber. At low pressures the extractor pump provided to maintain a steady oil level on the gearbox was unable to deliver against the differ nce of pressure from the test chamber to the atmosphere, and the varying oil level led to a variation in the mechanical efficiency which could not therefore be determined with sufficient accuracy. Consequently, the first method was abandoned.

The second method of obtaining the power P evidently only rechecks the consistency of the temperature ratios. For neglecting the small term $(W/2g)(u_1{}^2 - u_0{}^2)$ and assuming the heat flow, Q, to be negligible, the above expression for P may be written

$$\frac{P}{Wn^2} = K_p \frac{T_0}{n^2}\left(\frac{T_1}{T_0} - 1\right)$$

and at constant values of $n/\sqrt{T_0}$, the variation of $P/(Wn^2)$ is

dependent /

dependent on the variation of T_1/T_0.

For a similar reason the agreement of the adiabatic temperature efficiency curves at constant values of the parameter $n/\sqrt{T_0}$ does not convey any information additional to what is contained in the agreement of the curves of pressure ratio and temperature ratio. It was thought, however, that the magnitude of the resulting discrepancies in the adiabatic efficiency might be of interest, and the curves have, therefore, been plotted and are shown in Fig. 2.

Tests on the effect of a reduction in intake temperature.

The test plant was similar to that used during the tests at reduced intake pressure, but the inlet pipe to the test chamber was connected to a refrigerating plant by which a continuous circulation of cold air through the chamber was maintained. The temperature within the test chamber was dependent upon the internal pressure, and, as the tests required the maintenance of definite temperatures throughout a considerable range of air flow through the compressor, it was necessary to have an independent control of the pressure within the chamber, other than that provided by the compressor itself. Both the chamber and the supercharger delivery system were therefore connected to separate exhauster pumps, the former through the pipe C. (Fig. 9).

Tests were carried out at intake temperatures of -15°C. and -25°C., the intake pressure being regulated as required to obtain the desired temperatures. The speed of the impeller was adjusted to provide results at each of the four values of the variable $n/\sqrt{T_0}$

at /

at which the tests at low intake pressures were performed and the same quantities were measured as during those tests.

During all the experiments at low temperatures, but more particularly at an intake temperature of -25°C., sudden fluctuations of the pressure-measuring gauges were noticed, and the formation and breaking away of ice suggested itself as the most probable cause of this unsteadiness. It was thought that any deposition of ice would be most likely to occur at the entrance to the compressor, before the rise of temperature due to compression, and arrangements were made in some of the tests to measure the pressure before and after the guide vanes at the air intake. The ratio of these pressures has been plotted on Fig. 3, and the contrast between the uniformity of the results at normal temperature and the irregularities at the reduced temperatures is significant. The results obtained at -25°C. inlet temperature have not been included in the non-dimensional diagrams as the majority of them were inconsistent, due probably to restriction of the passage through the inlet guide vanes by the deposition of ice.

The pressure ratios, temperature ratios, and adiabatic efficiencies calculated from the tests at normal temperature and pressure and at -15°C., are plotted on Figs. 4 and 5, the units adopted being the same as in Figs. 1 and 2. The pressure ratio curves show slight discrepancies, similar to those resulting from the tests at reduced intake pressures. There appears to be a small disagreement in the temperature ratio curves at the two higher values of the variable $n/\sqrt{T_0}$; the

maximum /

maximum difference in the values of the temperature ratios is equivalent to an error of 4°C. in the delivery temperature in the tests at -15°C. intake temperature. The efficiency curves reflect the discrepancies in the pressure ratio and temperature ratio curves from which they are deduced.

Tests on the effect of heating the compressor casing.

Arrangements were made during these tests for the supercharger to take its air supply directly from the atmosphere by leading a pipe through a pressure-tight gland in the end-cover of the test chamber to the supercharger inlet. The portion of the pipe inside the chamber was lagged with layers of asbestos, felt, aeroplane fabric and rubber, and the intake temperature was determined by means of a transmitting thermometer fixed in the pipe a few inches from the supercharger inlet, the capillary tube passing down the centre of the pipe to the outside atmosphere. The barometric height was regarded as the initial pressure condition. The delivery pressure and temperature and the air quantity were determined as in the previous tests. The chamber was disconnected from the cold air plant, and was fitted with a pipe through which steam could be admitted, an exhauster pump serving to extract the condensate and maintain a continuous flow of steam through the interior of the chamber.

Two series of tests were carried out at each of the four original values of the variable $n/\sqrt{T_0}$. Steam was passed through the chamber during the first series only: in the second, the exhauster pump was allowed to draw a

small /

small quantity of air through the chamber from the surrounding atmosphere, in order to prevent any appreciable rise in the temperature of the air within the chamber.

During the tests with steam-heating, the internal temperature of the test chamber, measured at a point near the top, was maintained at 100°C., and the temperature of the external surface of the supercharger casing in the region of the delivery volute, measured by means of a washer-type thermo-couple, was found to be practically the same as that of the chamber.

The pressure ratios and temperature ratios obtained from the two series of tests are shown on Fig. 6. The temperature ratio at definite values of the variables $W/\sqrt{p_0 \theta_0}$ and $n/\sqrt{T_0}$ is increased from 13°C. at the high rates of rotation to 28°C. at the low rates by the external heating of the compressor, and there appears to be a tendency for the pressure ratio to be decreased, but the latter effect is evidently negligible.

It will be observed that, although the full-line curves on Figs. 1, 3 and 6 represent the results of tests at approximately the same intake temperature and pressure, the curves on Fig. 6 differ appreciably from those on Figs. 1 and 3. This difference is probably due to the alteration in the geometry of the supercharger by the fitting of the long inlet pipe for the tests to which Fig. 6 relates.

To enable the adiabatic efficiency to be obtained, the power P absorbed in compressing the air is required. The second of the methods of obtaining it referred to above, viz: by reference to the temperature

rise /

rise through the compressor, cannot be used in this instance because it cannot be assumed that the rate of heat flow through the casing, Q, is negligible. The difficulty which arose in the first method through the irregular variation of the mechanical efficiency due to variation of the oil level in the gearbox was not present in these tests because the pressure in the test chamber was approximately atmospheric, and it was possible to compare the adiabatic efficiencies with and without heating of the casing on the following basis.

Let η_m be the mechanical efficiency. Then if η is the adiabatic efficiency, and η' is the efficiency based on the total power input,

$$\eta' = \eta_m$$

In Fig. 7 values of η' are plotted against $W/\sqrt{p_0/\rho_0}$ for the four selected values of $n/\sqrt{T_0}$, with and without heating of the casing. It will be noted that the values of η' at the same values of the two variables are not very different. Now at the same values of $W/\sqrt{p_0/\rho_0}$ and $n/\sqrt{T_0}$, W and n will have approximately the same values since the inlet conditions varied little, and the torque and rotation rate will therefore be nearly the same. It would be expected, therefore, that η_m at corresponding points on the curves under the two conditions of operation would be approximately the same, and it is indicated that the values of the adiabatic efficiency η were not very different. Some uncertainty arises, however, because the temperature of the oil in the gearbox would be higher when the casing was heated, and the consequent

change/

change of mechanical efficiency, is unknown. It is thought that it would be small.

Conclusions of Part II.

The results of the tests show that the performance of a centrifugal compressor may be represented with sufficient accuracy as a function of the two variables

$$\frac{W}{\sqrt{p_o/\sigma}} \quad , \quad \frac{n}{\sqrt{T_o}}$$

and the performance of a centrifugal compressor at height may therefore be readily obtained from diagrams based on tests at ground level intake conditions in the manner indicated.

The effect on the compressor performance of heating the casing to 100°C. was not large, showing itself mainly as an increase in the temperature rise through the compressor. The applicability of the dimensional relationships was not tested under these conditions, but for the reasons given in Part I any discrepancies would only be expected to be a small fraction of the total effect on the performance of heating the casing, and would therefore be very small.

It is suggested, therefore, that the performance of a compressor in flight may be obtained with sufficient accuracy from bench tests under ground level conditions if the results are plotted in diagrams of the type shown, and the temperature ratio curves are corrected by some estimated amount based on the temperature of the casing when in proximity to the heated engine parts.

Attached:-

Prints	C.3943.	Fig. 1.
	C.3967.	" 2.
	C.3982.	" 3.
	C.3980	" 4.
	C.3981.	" 5.
	C.3983.	" 6.
	C.3934.	" 7.
Photos.	12003.	" 8.
	12002.	" 9.

TESTS AT VALUES OF $\frac{n}{\sqrt{T_0}}$ EQUAL TO 828, 944, 1066, AND 1188 AT VARIOUS INTAKE PRESSURES. INTAKE TEMPERATURE APPROXIMATELY CONSTANT.

INTAKE GAUGE PRESSURE OF ZERO:
" " " -5 INS. OF MERCURY:
" " " -10 " "
" " " -15 " "

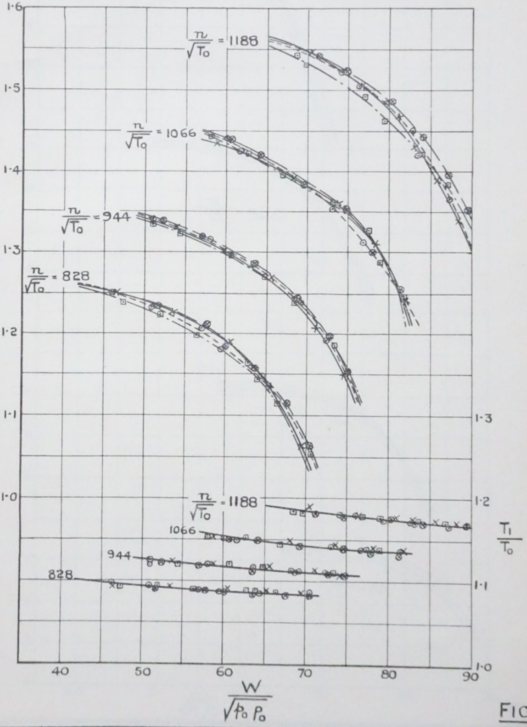

FIG. I.

ISSUED BY DRAWING OFFICE DIRECTORATE OF TECHNICAL DEVELOPMENT AIR MINISTRY	TITLE:- LION SUPERCHARGER. TESTS AT VARIOUS INTAKE PRESSURES.	DRAWN G.V.B.	TRACED N.A.H. 31·1·30	SKETCH Nº C3943
	ISSUE Nº 1	CHECKED G.V.B.	APPROVED [illegible]	
	ALTN Nº			

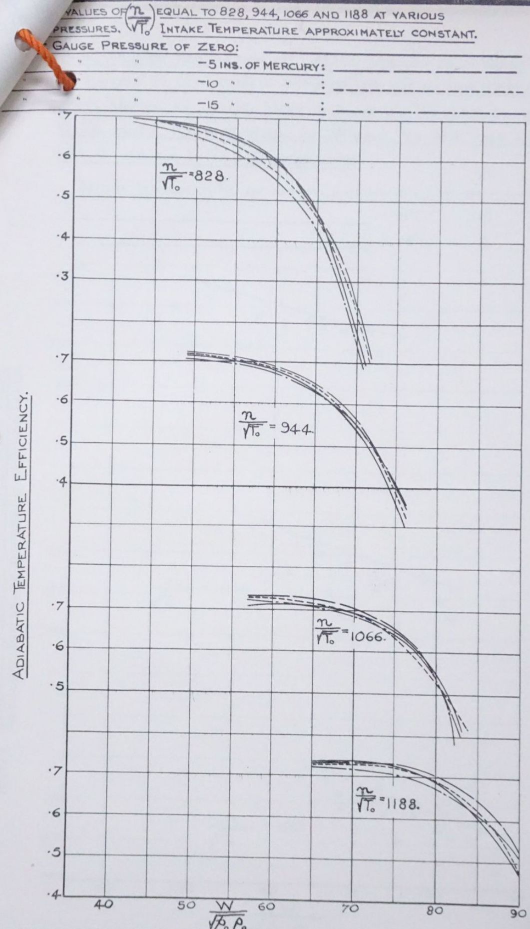

FIG. 2.

ISSUED BY DRAWING OFFICE DIRECTORATE OF TECHNICAL DEVELOPMENT AIR MINISTRY	TITLE:- LION SUPERCHARGER. TESTS AT VARIOUS INTAKE PRESSURES.	DRAWN G.V.B.	TRACED E.B. 1-3-30
	ISSUE Nº 1	CHECKED G.V.B.	APPROVED W.L.M.
	ALTN Nº		SKETCH Nº C.3967.

TESTS AT VALUES OF $\frac{n}{\sqrt{T_0}}$ EQUAL TO 828, 944, 1066 AND 1188 AT DIFFERENT INTAKE TEMPERATURES AND PRESSURES.

INTAKE PRESSURE AND TEMPERATURE 14 LB. PER SQ. IN. ABS. AND +13°C. APPROX. } ——×——×——×——

INTAKE PRESSURE AND TEMPERATURE 10·5 LB. PER SQ. IN. ABS. AND −15°C. APPROX. } ---⊙------⊙----⊙---

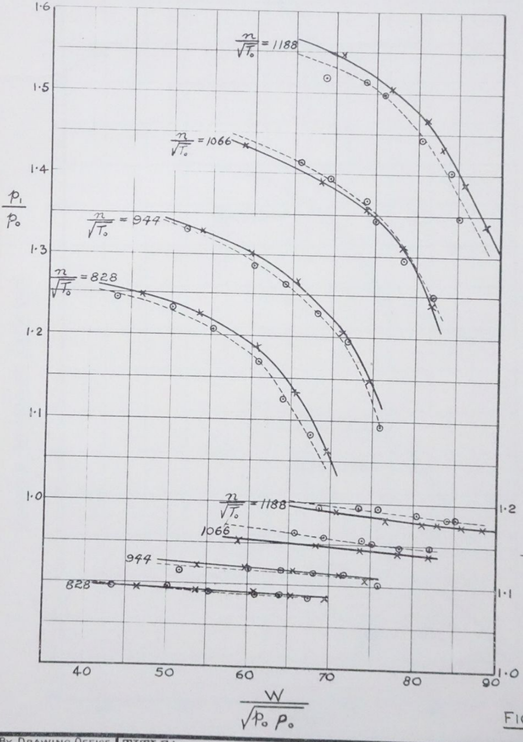

FIG. 4.

ISSUED BY DRAWING OFFICE DIRECTORATE OF TECHNICAL DEVELOPMENT AIR MINISTRY	TITLE:- LION SUPERCHARGER. TESTS AT VARIOUS INTAKE TEMPERATURES				DRAWN G.V.B.	TRACED E. 25-4-30	SKETCH Nº C.3980.
	ISSUE Nº	1			CHECKED G.V.B.	APPROVED W.W.J.	
	ALTN Nº						

ESTS AT VALUES OF $\frac{n}{\sqrt{T_o}}$ EQUAL TO 828, 944, 1066 AND 1188.
AT DIFFERENT INTAKE TEMPERATURES AND PRESSURES.

TAKE PRESSURE AND TEMPERATURE 14 PER SQ. IN. ABS. AND + 13°C. APPROX.	———
INTAKE PRESSURE AND TEMPERATURE 10·5 LB. PER SQ. IN. ABS. AND −15°C. APPROX.	- - - -

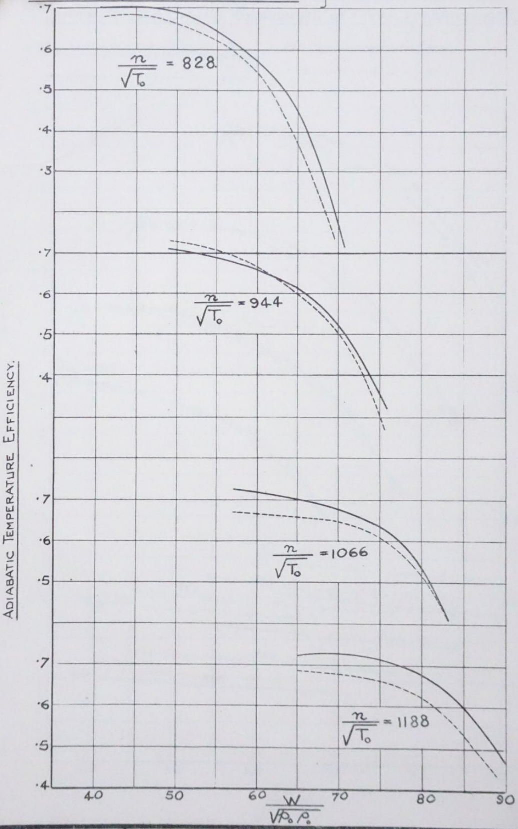

FIG. 5.

ISSUED BY DRAWING OFFICE DIRECTORATE OF TECHNICAL DEVELOPMENT AIR MINISTRY	TITLE:- LION SUPERCHARGER. TESTS AT VARIOUS INTAKE TEMPERATURES.	DRAWN G.V.B.	TRACED E.B. 25-4-30	SKETCH Nº C.3981.
	ISSUE Nº 1	CHECKED G.V.B.	APPROVED WWM	
	ALTN Nº			

Tests at values of $\frac{n}{\sqrt{T_0}}$ equal to 828, 944, 1066 and 1188;

(A) Supercharger Casing surrounded by atmospheric air at a temperature of approximately 15°C.

(B) Supercharger Casing surrounded by steam at approx. 100°C.

Tests under conditions "A": —×—×—×—

" " " "B": --⊙--⊙--⊙--

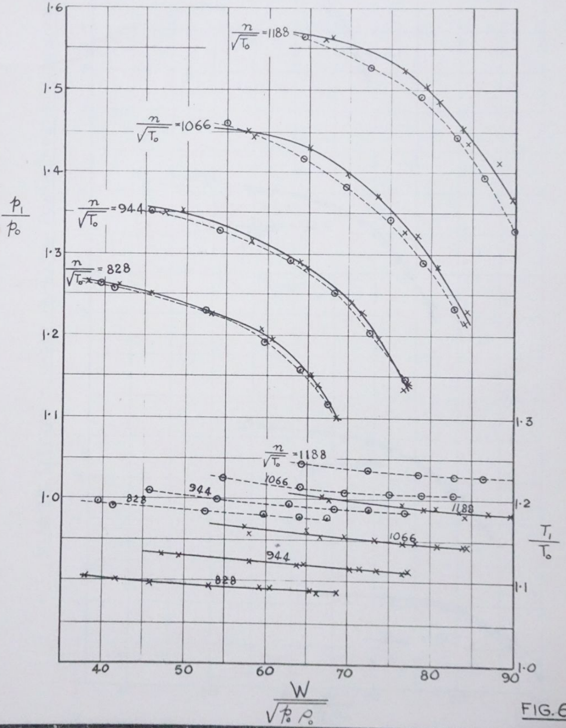

FIG. 6.

Issued by Drawing Office Directorate of Technical Development AIR MINISTRY	TITLE:- Lion Supercharger. Effect of Heating Supercharger Casing.	Drawn G.V.B.	Traced E.B. 28-4-30	SKETCH Nº
	Issue Nº 1	Checked G.V.B.	Approved WWM	C.3983.
	Altn Nº			

[T]ESTS AT VALUES OF $\frac{n}{\sqrt{T_0}}$ EQUAL TO 828, 944, 1066 AND 1188;
(A) SUPERCHARGER CASING SURROUNDED BY ATMOSPHERIC AIR AT A TEMPERATURE OF APPROXIMATELY 15°C.
([B]) SUPERCHARGER CASING SURROUNDED BY STEAM AT APPROX. 100°C.

TESTS UNDER CONDITIONS "A": —×—×—×—
" " " "B": -⊙-----⊙-----⊙--

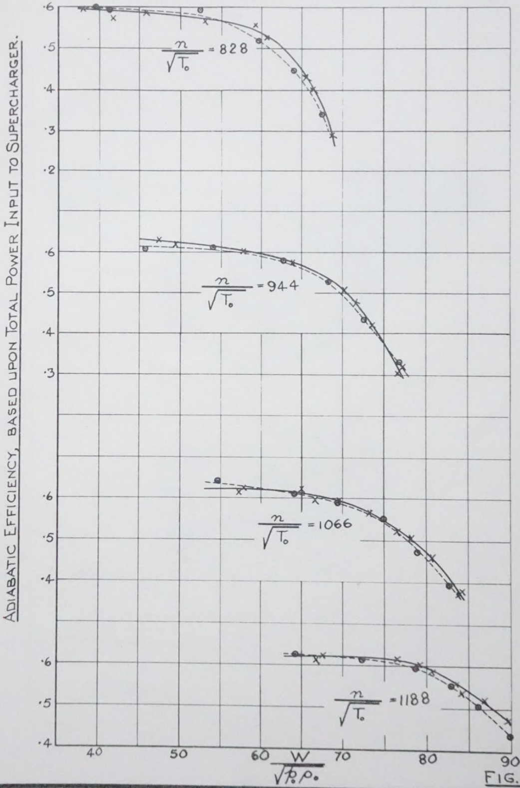

FIG. 7.

ISSUED BY DRAWING OFFICE DIRECTORATE OF TECHNICAL DEVELOPMENT AIR MINISTRY	TITLE:- LION SUPERCHARGER. EFFECT OF HEATING SUPERCHARGER CASING	DRAWN G.V.B.	TRACED E 28-4-30	SKETCH Nº C.3984.
	ISSUE Nº 1	CHECKED G.V.B.	APPROVED W.W.A.	
	ALTN Nº			

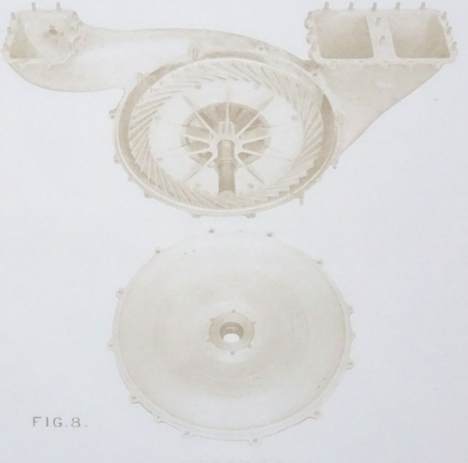

FIG. 8.

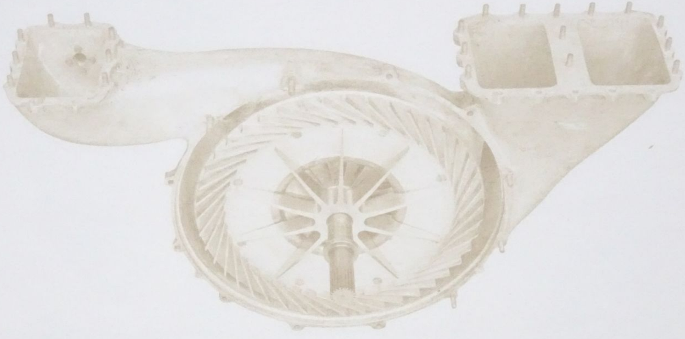

FIG. 8.

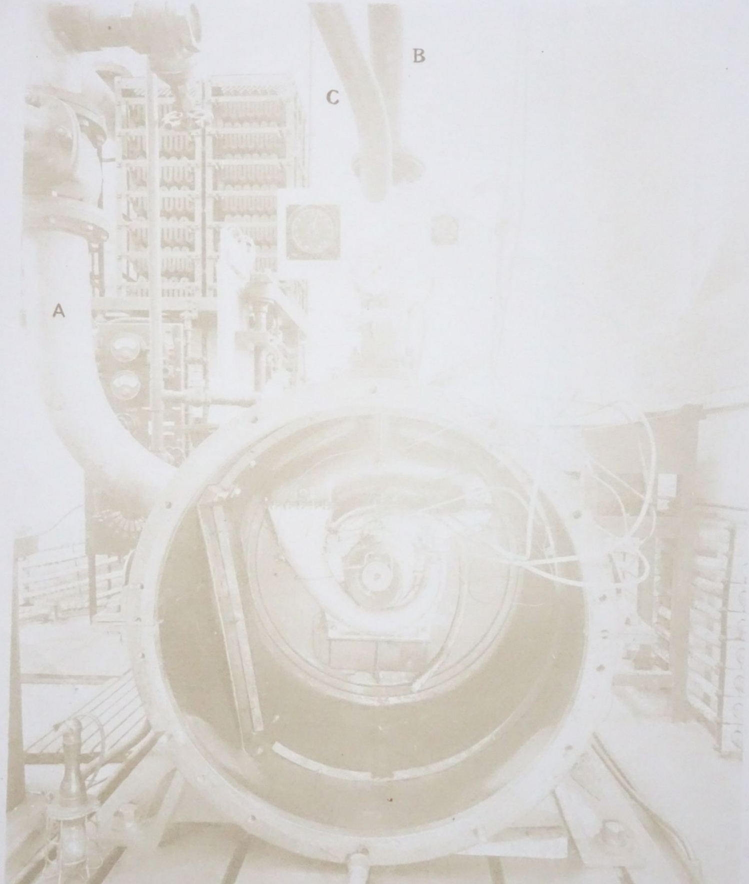

FIG. 9.

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