R. & M. No. 3337

MINISTRY OF AVIATION

AERONAUTICAL RESEARCH COUNCIL
REPORTS AND MEMORANDA

A Compressor Routine Test Code

By N. A. Dimmock

LONDON: HER MAJESTY'S STATIONERY OFFICE
1963
PRICE 18s. 6d. NET

A Compressor Routine Test Code

By N. A. DIMMOCK

COMMUNICATED BY THE DEPUTY CONTROLLER AIRCRAFT (RESEARCH AND DEVELOPMENT), MINISTRY OF AVIATION

*Reports and Memoranda No. 3337**

January, 1961

Summary.

The routine testing of aircraft-type compressors—in the main, axial-flow, multi-stage compressors—requires a compromise between research accuracy and the practical considerations. This test code is the outcome of a survey of compressor testing techniques and instrumentation, initiated and subsequently discussed and endorsed by the Aerodynamics Sub-Committee of the Gas Turbine Collaboration Committee.

The code aims at defining methods of measurement and weighting whereby compressor performance can be obtained sufficiently accurately for a realistic and direct comparison to be made between one compressor and another. The measurement of a quantity at a point in the fluid flow, and the averaging and weighting of such measurements have been treated separately as far as is possible.

The recommendations are given in the main text, whilst additional discussion on these is put into the Appendices.

LIST OF CONTENTS

* Replaces N.G.T.E. Report No. R.246—A.R.C. 22,791.

LIST OF CONTENTS—*continued*

LIST OF APPENDICES

TABLE

LIST OF ILLUSTRATIONS

---

1. *Introduction.*

At the request of the Aerodynamics Sub-Committee of the Gas Turbine Collaboration Committee, a survey was made of the present techniques of routine compressor testing with a view to establishing a recommended code of practice and to go some way towards standardising the measuring instruments. The aim was to increase the accuracy of quoted test results sufficiently for direct comparisons to be made with confidence.

The field of survey was broadly limited to the *routine* testing of aircraft-type compressors—in the main, axial-flow compressors—although brief consideration was to be given to allied problems, for example, that of traversing instrumentation or the effects, on analysis and instrument development, of using fast digital computers for data reduction.

After discussing the preliminary findings of the survey[30, 41], the G.T.C.C. Aerodynamics Sub-Committee endorsed this report as an agreed code of practice for the routine testing of compressors. The recommendations are concentrated into the main text with a minimum of comment, leaving the detailed discussion to a series of Appendices each one covering a particular aspect of the code.

2. *The Problem.*

The problem of accuracy in the measurement of overall compressor performance may be divided into two, the first being that of correctly measuring the required quantity at a point in the fluid flow and the second is how to ensure that a true mean value results from the measurements made. It has been argued that absolute accuracy is not as important as relative accuracy, and this might be true for development work on one (or one type of) test rig. The argument fails when an absolute comparison is required. It is a truism to state that, for accurate comparison in a wide field, completely accurate performance data are essential: nevertheless this is the crux of the matter and any compromise or relaxation introduces a degree of uncertainty which cannot afterwards easily be determined.

Over the years, most of the significant errors in measurement have been considerably reduced[31 to 36], so leaving the last small gains in accuracy the most difficult to achieve.

3. *Systematic Errors.*

Systematic errors can be eliminated, although many of them are frequently ignored because the corrections are often small and laborious to apply. Following is a list of mean practical constants chosen such that corrections are unnecessary for an accuracy to within $\pm 0\cdot1$ per cent for the stated range of conditions, which in turn have been chosen to cover the more usual working conditions. A more detailed discussion of these constants will be found in Appendix I.

(i) Gravitational constant[1] $g = 981\cdot183$ cm/sec^2 (at N.P.L. Teddington) $= 32\cdot191$ ft/sec^2. (*See* also Appendix I.)

(ii) Mean density of mercury[1] between 15° and 25°C corrected for local value of g {as at (i)} $= 13\cdot553$ gm/ml (error at 10° or 30°C $\pm 0\cdot181$ per cent).

(iii) Conversion, in. Hg to lb/sq. in., multiply by $0\cdot4896_3$ conditions as for (ii). (*See* also Figure 15 and Appendix I.)

(iv) Absolute pressures (barometer corrections[2]). What is normally required is an absolute pressure in lb/sq. in. and for this the barometer reading may be added directly to the manometer reading and factored by $0\cdot4896_3$ {(iii) above} if barometer (Fortin) and manometers are all at temperatures between 16° and 28°C. If the barometer is at a different level from the point of required atmospheric pressure the reading can be corrected by $\pm 0\cdot055$ in. Hg per ∓ 50 ft difference in vertical height for a reading of $30\cdot0$ in. Hg, and proportionally adjusted for other barometer readings.

(v) *Standard atmospheric pressure*[1, 2] is 1,013,250 dyn/cm^2 which is equivalent to 760 mm or $29\cdot921$ in. of Hg at 0°C and with $g = 980\cdot665$ cm/sec^2 or to $14\cdot69596$ lb/sq. in. or to a barometer *reading* of $30\cdot000$ in. Hg at local gravity, sea level and at $19\cdot25$°C (Fortin barometer).

(vi) *Capillary depression* is a systematic error to be avoided rather than corrected. The error tends to cancel in a 'U'-tube and, for clean single-limb manometers with a drop or two of dilute phosphoric acid (50/50) on the mercury surface and a tube-bore of not less than 6 mm diameter, the effect of capillarity can safely be ignored unless very low absolute (e.g. severely throttled inlet) pressures are to be measured. Computed corrections for a range of tube-bores and meniscus heights are tabulated in Reference 1.

(vii) *Density of water* and conversion of in. H_2O to lb/sq. in. The density at 20°C and at local gravity (51°N latitude) is $0\cdot99875$ gm/ml. To convert from in. H_2O to lb/sq. in., multiply by $0\cdot036082$ (between 15° and 24°C for an error not exceeding $0\cdot1$ per cent and between 0° and 27°C for an error not exceeding $0\cdot2$ per cent). (*See* also Figure 15 and Appendix I.)

(viii) *Temperature measurement—Weston Cell.*—The potentiometric measurement of thermocouple e.m.f. relies upon a Weston Cell for a voltage standard. Errors due to temperature coefficient (range 0° to 30°C), annual drift, etc., are all very much less than $0\cdot1$ per cent and can be ignored. Thermocouple-wire errors are discussed in Appendix IV.

(ix) *Mechanical equivalent of heat*[3].

$$J = 1400\cdot70 \text{ ft. lb/C.h.u.}$$

(x) *Gas constant for air*[3, 4].

$$R \text{ (or } G) = 96\cdot020 \text{ ft. lb/lb°C.}$$

(xi) *Isentropic index*[3].

$$\frac{\gamma}{\gamma - 1} = 14\cdot588\ K_p \text{ for dry air}$$

where K_p is the true specific heat at the mean temperature. (*See* also Appendix I and Table 1.)

(xii) *Absolute temperature.*

$$T°\text{K} = t°\text{C} + 273\cdot15.$$

(xiii) *Humidity.*—It is usual to make no allowance for the moisture in the air supplied to the compressor, although the error in efficiency is nearly $+0\cdot3$ per cent for $m = 0\cdot01$ (m = weight ratio, water vapour/dry air). It is suggested that a correction for moisture is applied if $m \geqslant 0\cdot01$. (*See* Figure 14 and Appendix I.)

(xiv) *Analysis.*—Considerable discrepancies can arise through different methods of analysis. Some of these can be attributed to a variety of values taken for the factors and constants given in the preceding paragraphs, while others are dependent only on definition. A consistent method of analysis, such as that given in Sections 5.1 and 5.2, should eliminate this type of variation.

4. *Measurement of the Various Quantities.*

As far as is possible, the problem of sampling and weighting has been separated from that of physical measurement at a point in the fluid flow, the latter being the subject of this section.

Adequate time must be allowed for the conditions to become steady. Almost invariably it is the outlet temperature which takes the longest time to stabilise and this measurement can be used to monitor the settling time.

4.1. *Airflow Measurement.*

A general review of airflow measurement is outside the terms of reference of this code and it is a large task by itself. However, there are some features of airflow measurement particular to compressor testing.

An error in mass flow of up to ± 2 or even 3 per cent can be anticipated for an uncalibrated flowmeter, whether of standard design[7] or not. For greater accuracy a calibration is required[28] and, if this is performed carefully, the minimum error is likely to be about $\pm 0\cdot5$ per cent. Whatever type of flowmeter is used, the most frequent troubles stem from poor inlet flow conditions. A particular cause is often an upstream throttle insufficiently far removed from the flowmeter, when the errors vary with the degree of throttling. Palliatives such as perforated plates, or a honeycomb and gauze(s) can be used to improve the entry conditions for the flowmeter; also, the compressor intake can be calibrated against the flowmeter with the throttle fully open, and the intake pressure used subsequently for airflow measurement during throttled conditions. None of these methods can be recommended for accurate airflow measurement, and the flow frequently requires further treatment after the flowmeter to ensure uniform conditions at the compressor intake.

It is recommended that a ducted inlet be used whenever possible, the sequence of components being such that the airflow meter draws air from the filter house (or from a settling room if the air is not filtered) and is followed by the inlet throttle and a settling length, with additional aids if necessary, to give a uniform flow into the compressor inlet (Figure 1). However, with the components in this order, leakage at or after the inlet throttle must be eliminated, particularly for testing with a large amount of inlet throttling. Also, with inlet throttling and reduced mass flow, it is important that the flowmeter manometers are sufficiently sensitive and that the flowmeter itself remains free from the possible effects of a low Reynolds number.

When a test cell is used, care is needed to avoid temperature stratification, pressure variations and swirl at the compressor intake.

4.2. *Rotational Speed.*

The required parameter is corrected speed, i.e.

$$N_c = N \frac{\sqrt{288 \cdot 15}}{\sqrt{(T_{\text{tot}})_{\text{inlet}}}} \{\text{or a given value of } N/\sqrt{(T_{\text{tot}})_{\text{inlet}}}\}.$$

Although the accurate measurement of rotational speed is not particularly difficult, it must be remembered that speed errors become doubled in the pressure ratio and more than trebled in power input. (For the measurement of inlet temperature see Section 4.3.1.) Major sources of error are as follows. With pulse-counting types of tachometer having a long time base, quite large speed fluctuations ($\sim$ 50 rev/min) can pass unnoticed within the counting cycle. This can lead to efficiency errors if the temperature measurement lags or leads the pressure measurement. Driver fatigue is an important factor and is aggravated by unsuitable indicators which lead to eyestrain or which require active concentration. Most pointer-and-scale indicators come into this category, since the set speed often requires the pointer to be kept steady between scale divisions where the onset of a speed change is not easily detected. Digital display counters and stroboscopic or frequency difference methods relieve the driver of a considerable amount of strain and are to be recommended. Some examples of these techniques are given in Appendix II.

4.3. *Temperature Measurement.*

The choice of instruments for temperature measurement narrows down to two types—thermocouples and resistance elements. The general aspect of temperature measurement is discussed in Appendix III, thermocouples in Appendix IV and thermocouple calibration in Appendix V.

Thermocouple wire pairs.—All thermocouple materials should be batch tested (although *see* iron-constantan, Appendix IV) and the commonly used combinations are chromel-constantan, chromel-alumel, copper-constantan and iron-constantan in order of preference. (*See* Appendix IV.)

Thermocouple shields.—The choice of shield depends upon the conditions under which the temperature is to be measured, and the types of shield sketched in Figures 2 and 3 cover most applications, whilst sonic suction pyrometers[26] may be used in particularly difficult circumstances. (*See* Appendices IV and V.)

Probe positions.—*See* Sections 4.3.1 and 4.3.2 and Appendix III.

Other precautions with thermocouples.—There is a general preference for ice-point cold junctions (Figure 5), which should be contained in Dewar flasks; they should be kept dry in sheaths (thin plastic tubing sealed at the lower end is very suitable) and spacers should be used if there are more than four cold junctions per flask. The junctions should be positioned near, but not touching, the bottom of the flask which must first be filled with finely crushed ice and just enough cooled water then added to make the mass sufficiently fluid for the careful insertion of the cold junctions. Ideally, distilled water should be used for both the ice and the water for topping up the flask, but clean tap water introduces a negligible error. It is desirable to insert a calibrated thermometer in the flask as a check on the cold-junction temperature and this is, of course, essential if a well lagged box at nominally ambient temperature is used for the cold-junction assembly, instead of the ice-point method. The wires of the hot junction should be taken all the way to the cold-junction connector (or extension leads of the same batch of wire used) and 'compensating' leads should be avoided. For high accuracy, screw-down connectors and soldered joints should not be used and the copper leads should not have other metals interposed at joints or switches unless positive precautions are taken to avoid thermal gradients; potentiometers should have copper connecting terminals. Thermocouple-wire joints may be cleaned, twisted together and clamped between insulating surfaces and Figure 5 shows a cold-junction assembly using this principle.

Resistance thermometers.—The open-grid or spiral type of resistance element is suitable for inlet-temperature measurement, but in the confined space of the compressor outlet annulus, an element such as the de Gussa resistance thermometer is most satisfactory when mounted in a stagnation pocket of the type sketched in Figure 4. The disadvantages of resistance thermometers are:

(*a*) The heating effect of the bridge current necessitates the use of refined equipment if large errors are to be avoided, particularly in a low-velocity airflow.

(*b*) They are somewhat fragile and not very cheap.

(*c*) The external resistance features in the accuracy of the measurement.

4.3.1. *Inlet temperature.*—Inlet total temperatures need be measured very accurately only if the separate quantity is required directly for determining the temperature rise. Otherwise for speed and mass-flow correction, and for fixing the end point on a thermocouple calibration curve, measurement to within $\pm 0 \cdot 5°C$ is adequate. With an atmospheric intake and a reasonable ducting layout (Section 4.1 and Figure 1) a significant change in total temperature is seldom found between the filter house (or other intake) and the compressor entry. Therefore, the temperature is best measured where the air speed is low upstream from the compressor, unless air bled from the compressor is recirculated, when a measurement must be made within the compressor inlet. Thermocouples or resistance thermometers are both satisfactory. If interstage bleed air is vented to the inlet duct, then care must be taken to ensure that it is well distributed throughout the inlet flow, without local hot-spots.

With a ducted inlet, the temperature variation at the compressor intake should not exceed $\pm 1°C$, but if the compressor is mounted in a test cell, it is often impossible to avoid variations of as much as $\pm 5°C$. The only remedy here is to fit a large number of instruments some of which can be connected to yield a mean temperature reading[10]. (*See* Appendix IV.) The mean inlet temperature is taken to be the arithmetic mean of the individual measurements. (*See* Appendix VI for sampling and weighting.)

4.3.2. *Temperature rise or outlet temperature.*—The mean temperature rise may be measured directly, when thermocouples are used, by connecting inlet and outlet thermocouples in opposition. (*See* Appendix IV.) If this method is used, additional independent inlet and outlet thermocouples should always be fitted, the former being essential to fix the end point on the calibration curve for temperature-rise conversion, as well as for speed and mass-flow correction. Also, the separate thermocouples indicate the variation over the measuring plane and, taken together, afford a check on the temperature-rise measurement.

It is recommended that the elements for outlet-temperature measurement be positioned a short distance downstream from the outlet pitot combs. (*See* also Appendix III for comments on other positions for the outlet temperature instruments.) They should never be less than six in number and eight or more are recommended, the arrangement being such as to give a reasonable radial and peripheral sample. The mean temperature, whether inlet, outlet or temperature rise, is the arithmetic mean of the individual measurements. (*See* Appendix VI for sampling and weighting.)

4.4. *Total-Pressure Measurement.*

4.4.1. *Inlet pressure.*—The quoted mean inlet pressure should be based upon an area-mean total pressure obtained from pitot tubes or combs, there being not less than four combs, each of not less than five tubes, or the equivalent number of single pitot tubes. Figure 6 shows a typical inlet pitot comb.

With only a small variation of pressure over the inlet, and with the radial pitot spacing suitably arranged, an arithmetic averager[38], such as is sketched in Figure 12, may be used to reduce the number of readings to be recorded. (It is preferable to display all the pressures, even though for normal operation only the averaged reading is recorded.)

The limit of inlet flow maldistribution for routine testing should be 5 per cent for the quantity

$$F_P = \frac{(P_{\text{tot}})_{\text{max}} - (P_{\text{tot}})_{\text{mean}}}{(P_{\text{tot}})_{\text{mean}} - P_{\text{stat}}}$$

where $(P_{\text{tot}})_{\text{max}}$ is the highest single value of P_{tot} from any pitot tube or from an exploratory pitot traverse, $(P_{\text{tot}})_{\text{mean}}$ is as defined above and P_{stat} is derived from the value of $(P_{\text{tot}})_{\text{mean}}$, the mass-flow measurement, the inlet temperature and the flow area.

This pressure factor is a better indicator of inlet flow non-uniformity than is the velocity-profile factor since, apart from being arithmetically larger, it appears that the compressor influences the flow upstream, such that the approach velocity is evened out and a corresponding static-pressure variation is left[17]. An area contraction into the compressor intake is the most usual and powerful tool for improving the velocity profile and Prandtl has shown[18] that the percentage variation in kinetic energy is reduced to $1/m^2$ times its original value in passing through a contraction of area ratio m. An almost identical result is obtained more simply by taking the reduction of the fractional perturbation in the velocity to be proportional to the area ratio squared*.

---

* e.g. taking $\dfrac{\bar{v} + \delta v}{\bar{v}} = 1 + x$,

corresponding K.E. $= (1+x)^2 \doteqdot 1 + 2x$ if x is small.

Perturbation is reduced to $\dfrac{2x}{m^2}$ through contraction

$\therefore$ Downstream local K.E. $= 1 + \dfrac{2x}{m^2}$ relative to mean value

$$\text{and corresponding velocity} = \left(1 + \frac{2x}{m^2}\right)^{1/2}$$

$$\doteqdot 1 + \frac{x}{m^2} \text{ if } \frac{2x}{m^2} \text{ is small.}$$

Swirl should be less than $\pm 2 \cdot 5°$ at the compressor inlet, unless the compressor is without inlet guide vanes when the swirl limit should be reduced, the acceptable value depending upon the design of the first-stage rotor blades.

With inlet throttling, the inlet pressure must be kept constant for all the points on a single speed characteristic. An altimeter connected to one of the inlet reference-pressure lines is a suitable indicator for this purpose although some vibration is usually necessary to avoid 'stickiness'. Alternatively a device such as that sketched in Figure 13 may be used so that, when the set test conditions are first reached, a volume of air at inlet pressure is isolated in a coil immersed in an ice bath; a water-filled capillary U-tube is then used to balance subsequent inlet pressures against this reference sample.

4.4.2. *Outlet pressure.*—The measurement of outlet pressure cannot be separated completely from its sampling or weighting. (*See* Appendix VI.) The agreed recommendation on outlet-pressure definition is that it should be derived from a measured mean static pressure, using the mean outlet temperature, the measured mass flow and the flow area in the compressible-flow equation, allowing for swirl angle where this is a design feature. (For static-pressure measurement see Section 4.5 and Appendix VII, and for further comment on sampling and weighting see Appendix VI.) Not less than four equally spaced static tappings should be used at the o.d., while additional tappings both at o.d. and i.d. are desirable. Whenever possible, swirl should be eliminated and a length of parallel annulus should be fitted at the compressor outlet for this measurement in particular, and it is useful also if pitot combs are used.

When pitot combs are used to obtain an outlet mean total pressure, there should be a considerable number fitted (about ten to twelve) with not less than five tubes per comb; the actual number should be prime to the number of O.G.V.'s and the peripheral spacing even. The combs should be of 'Kiel' tubes (*see* Figures 7, 8 and 9) and they should be positioned two to three blade chords or one annulus height downstream from the last blade row. The radial spacing of the tubes should follow a log-linear rule[12] or the tangential rule (centres of equal areas). Pitot combs are a useful means of obtaining the general shape of the outlet velocity profile—*see* Section 5.3.

The method used to obtain the mean outlet pressure should be stated when the compressor performance is quoted.

Cylindrical pitots should be avoided, even when yaw-sensing holes are added, since this type of yawmeter in particular indicates a false flow angle when in a pressure gradient, such as a blade wake, and there is a consequent error in total-pressure reading with the pitot hole displaced from the stagnation point. Also, if this type of probe is cantilevered from one wall, flutter may cause further inaccuracies[16].

4.5. *Interstage Static Pressures.*

Stage characteristics from routine tests on multi-stage compressors can hardly be expected to yield high absolute accuracy, but the static tappings should nevertheless conform to good practice as far as practical considerations will permit. Whenever possible, static tappings should be made to the dimensions of Figure 10a, but where for some reason this cannot be done, the compromise tapping of Figure 10b may be used.

Static tappings, two or more per stage, should be axially positioned in a plane midway between the stator-blade trailing edge and the rotor-blade leading edge. There is some doubt as to the best position for the tapping relative to the stator-blade passage and it is recommended that the position be as close to the passage centreline as is practicable and, of much greater importance, that this relative position is maintained at every stage in the compressor. The mean outlet static pressure can be taken as the arithmetic mean of the individual pressures.

More information on static-pressure measurement will be found in Appendix VII.

4.6. *Torque.*

It is desirable to measure torque when its value can be obtained reasonably directly and simply. Torque measurements become increasingly useful at the lower pressure ratios and hub ratios now being used for the fans of by-pass and ducted-fan engines and in which a correctly weighted mean outlet-temperature measurement is difficult to obtain. The most reliable torque measurements are obtained with electrical drive, by swinging the motor casing, although this method is usually limited to motors of medium or small power. The most direct method for large powers is that employed in the Van Milligan torque meter, where the torsional strain of the torque shaft is measured by an optical system using rotating prisms or mirrors. This equipment is reliable and accurate once the installation teething troubles have been cured (optical line-up, collimator vibration etc.). A lack of associated electronic equipment is considered by many to be a distinct advantage. A gear-box intermediate between the compressor and torque meter greatly increases the difficulty of accurate measurement and should be avoided.

Assuming an accurate torque measurement, then for good correlation of shaft efficiency with blade efficiency (i.e. from the temperature measurement), it is necessary to allow for friction and part of the windage either by estimation or experiment and to use a variable specific heat, or total heat, in the computation of results.

5. *Definition and Interpretation.*

A standardised interpretation of the recorded data is necessary and the following conventions have been agreed and are recommended.

5.1. *Overall Compressor Characteristics.*

(i) *Mass-flow parameter.*—Overall characteristics are plotted with an air mass-flow parameter as the abscissa.

The standard form for this parameter is the corrected mass flow $\underset{\sim}{Q}_c$

$$\underset{\sim}{Q}_c = \frac{14 \cdot 696}{\sqrt{288 \cdot 15}} \underset{\sim}{Q} \frac{\sqrt{(\bar{T}_{\text{tot}})_{\text{inlet}}}}{(\bar{P}_{\text{tot}})_{\text{inlet}}}$$

$$= 0 \cdot 86574 \, \underset{\sim}{Q} \frac{\sqrt{(\bar{T}_{\text{tot}})_{\text{inlet}}}}{(\bar{P}_{\text{tot}})_{\text{inlet}}}$$

where

$$\underset{\sim}{Q} = \text{measured mass flow in lb/sec}$$

$$(\bar{P}_{\text{tot}})_{\text{inlet}} = \text{area-mean inlet total pressure in lb/sq. in.}$$

$$(\bar{T}_{\text{tot}})_{\text{inlet}} = \text{arithmetic-mean inlet total temperature in °K.}$$

Humidity correction should be applied to $\underset{\sim}{Q}$ if significant (Figure 14).

The non-standard alternative forms of the mass-flow parameters are

$$\frac{Q\sqrt{(\bar{T}_{\text{tot}})_{\text{inlet}}}}{(\bar{P}_{\text{tot}})_{\text{inlet}}} \text{ and } \frac{Q_c}{A}$$

where

A = (specified) gross frontal area of the compressor in sq. ft.

(ii) *Pressure ratio.*—The standard form for this parameter is the ratio:

$$\frac{\text{derived mean outlet total pressure}}{\text{area-mean inlet total pressure}}$$

(*See* Sections 4.4.1 and 4.4.2.) Alternative non-standard forms may use mean pressures obtained from weighting methods other than those recommended; if these forms are used, the instrumentation and weighting methods should be specified.

(iii) *Temperature-rise ratio.*—This ratio should always be quoted and is given by

$$T.R. = \frac{\Delta T_{\text{tot}}}{(\bar{T}_{\text{tot}})_{\text{inlet}}}, \text{ where } \Delta T_{\text{tot}} = (\bar{T}_{\text{tot}})_{\text{outlet}} - (\bar{T}_{\text{tot}})_{\text{inlet}}$$

and both inlet and outlet temperatures are arithmetic-mean temperatures over each plane of measurement. Humidity correction should be applied to ΔT_{tot} if significant. If torque is measured, iteration will be necessary to obtain K_p as defined below in (iv).

(iv) *Isentropic efficiency.*—The standard form for efficiency is

$$\eta = \frac{\Delta T'_{\text{tot}}}{\Delta T_{\text{tot}}}$$

where ΔT_{tot} is as in (iii), corrected for moisture if this correction is significant, and

$\Delta T'_{\text{tot}}$ = isentropic temperature rise corresponding to the mean total-pressure rise from

$$\Delta T'_{\text{tot}} = (\bar{T}_{\text{tot}})_{\text{inlet}} \left[\left\{ \frac{(\bar{P}_{\text{tot}})_{\text{outlet}}}{(\bar{P}_{\text{tot}})_{\text{inlet}}} \right\}^{(\gamma-1)/\gamma} - 1 \right]$$

and

$$\frac{\gamma - 1}{\gamma} = \frac{1}{14 \cdot 588 K_p},$$

where K_p is the true specific heat for dry air at constant pressure at the temperature

$$(T_{\text{tot}})_{\text{inlet}} + \frac{\Delta T_{\text{tot}}}{2}.$$

If ΔT_{tot} has been corrected for humidity, then γ must not additionally be corrected since this merely cancels the correction on ΔT_{tot}. If γ is required to be corrected, then for all practical purposes, $\gamma_{\text{moist air}} = \gamma_{\text{dry air}} - m/8$ where m = mass ratio, water vapour/dry air. K_p may be found from Reference 3 or from Table 1 which is abbreviated from Reference 3. Again K_p may be found from

$$K_p = 0 \cdot 27798 + 0 \cdot 037079x - 0 \cdot 021413x^2 - \\ - 0 \cdot 007016x^3 + 0 \cdot 012773x^4$$

where

$$x = (T - 1125)/875$$

and this form is particularly suitable for use with high-speed digital computers although not an exact fit of the curve of Fielding and Topps.

Tables[3] or charts[5] of total heat and entropy functions may be used also for efficiency determination although the random error for $\Delta T_{tot} < 200°C$ may well exceed the systematic error inherent in the recommended method[3].

If torque is used to obtain a shaft efficiency, a humidity correction, if used, must be applied to the isentropic enthalpy rise; this correction is not included on Figure 14, but it may be obtained from References 3, 5 or 6.

Polytropic efficiency, although not the recommended standard form for efficiency, is in many ways superior to isentropic efficiency and may be quoted as a useful additional parameter.

(v) *Rotational speed.*—The preferred form of this parameter is the corrected speed N_c rev/min where

$$N_c = N\sqrt{\frac{288\cdot 15}{(\overline{T}_{tot})_{inlet}}}$$

N being the actual speed in rev/min.

A non-standard form which may be used is the quantity $N/\sqrt{(\overline{T}_{tot})_{inlet}}$.

(vi) *General.*—The conditions of the test should be stated, in particular, the inlet conditions and the basic compressor geometry.

5.2. *Stage Characteristics.*

Although the measured quantities for obtaining stage characteristics are much the same for all test rigs, the use of them in the computation can vary considerably. For instance, the temperature distribution through the compressor may be assumed from the design distribution applied to the measured inlet and outlet temperatures, or the polytropic index of compression may be computed from the overall measured pressure and temperature ratios and used with the interstage static pressures to compute the interstage temperatures. Neither method gives an accurate result as can be deduced from the scatter found in the characteristics for the middle stages of a compressor. This scatter can be reduced by measuring temperatures at planes intermediate between inlet and outlet and using these temperatures in computing the distribution through the compressor. The distribution obtained by dividing the overall temperature rise by the number of stages should not be used.

The method of computation of stage characteristics is of necessity somewhat arbitrary, but with more use now being made of fast digital computers some refinement is possible and the recommended method is as follows:

The measured quantities are:

N_a	Actual speed, rev/min (or N_c from which N_a may be found)
$\underset{\sim}{Q}_a$	Actual mass flow, lb/sec
T_{t1}	Mean inlet total temperature, °K
$T_{t(n+1)}$	Mean outlet total temperature, °K
P_1, P_2, P_n, $P_{(n+1)}$	Mean static pressure at compressor outside diameter. (Suffices denote the stage number entry plane which may be taken as the plane of the rotor leading edge at the mean diameter. $(n+1)$ therefore designates the compressor outlet plane.)

The geometric constants for the compressor are:

A_1 to $A_{(n+1)}$	Annulus area (*see* note above for suffices)
D_{m1} to D_{mn}	Rotor mean diameter at the blade leading edge
α_{01} to $\alpha_{0(n+1)}$	The design air outlet angles from the fixed blade rows (assumed constant).

At any plane where a total temperature and a static pressure are measured the temperature ratio, total/static, may be found from:

$$\frac{T_t}{T} = \frac{1 + \sqrt{\left\{1 + \left(\dfrac{Q_a\sqrt{T_t}}{AP\cos\alpha_0}\right)^2 \dfrac{2R}{g}\,\dfrac{\gamma-1}{\gamma}\right\}}}{2}$$

and from this, the Mach number may be found and used in any of the compressible-flow relationships where, for simplicity, γ is given the same value as for the overall characteristics.

The temperature distribution through the compressor (or each section if the overall temperature is sub-divided) can be determined either by dividing the overall measurement proportionally to the design distribution or by calculating the polytropic index.

If the latter method is used, difficulty arises when there is choking in the last blade row, since the pressure drops considerably, but the total temperature remains constant after the last rotor. This leads to an error in the polytropic index which can have a large effect on the scatter in all the stage characteristics. An improvement results from using the maximum static pressure (usually P_n), rather than the outlet static pressure $P_{(n+1)}$, to calculate the polytropic index; alternatively the index could be calculated over $(n-1)$ stages by subtracting an estimated ΔT_{tn} from the outlet temperature in order to obtain a value for

$$\left(\frac{Q_a\sqrt{T_t}}{AP\cos\alpha_0}\right)_n.$$

With the temperature distribution decided, all the following quantities can be found from the one-dimensional compressible-flow equations and the compressor geometry:

V_a/U	the flow coefficient (with U based on D_m as above)
$\Delta P/\frac{1}{2}\rho U^2$	the static-pressure-rise coefficient
$\Delta P_t/\frac{1}{2}\rho_t U^2$	the total-pressure-rise coefficient
M_0	the actual Mach number defined as $M_0 = V_a/\{\sqrt{(\gamma gRT)}\cos\alpha_0\}$ although it is in fact obtained as an intermediate result in the computation.
M_1	$V_a/\{\sqrt{(\gamma gRT)}\cos\alpha_1\}$ where $\tan\alpha_1 = U/V_a - \tan\alpha_0$

5.3. *Outlet Velocity Profiles and the Total-Pressure Profile Factor.*

Traversing is the only satisfactory method for obtaining detailed information on the compressor outlet flow. {*See* Appendix VIII(A) and Figure 19.} For general purposes however, outlet pitot combs may be used and the total-pressure profile factor, F_p, obtained as defined in Section 4.4.1.

The velocity profile, plotted as $V_a/\bar{V}_a$ against annulus height (or radius) should always be drawn to a standard scale such that, with the annulus height as unity, the ordinate, $\bar{V}_a$, should be two units. The same scale should be used for the total-pressure profile where possible.

6. *Conclusions.*

Individual measurements of pressure, temperature, rotational speed, air mass flow and torque can all be made sufficiently accurately if care is taken and bad practices avoided. The overall accuracy in the determination of compressor performance then hinges on representative sampling and the type of weighting adopted for obtaining a mean pressure and temperature. The sampling must be such as to indicate the distribution of the measured quantity over the whole flow area. Thus, if the measuring plane is chosen where it can be shown that the flow is uniform or has a known constant distribution, the number of measuring points can be reduced accordingly.

Thereafter the type of weighting employed is frequently dominated by the practical limitations of the instrumentation and it is with these practical considerations in mind that the recommended code has been drawn up and summarised as follows:

Airflow meters, whether of standard or non-standard type, should be calibrated in their normal working installation. Throttles and other disturbing influences should not be situated upstream from the airflow meter. (*See* Section 4.1 and Figure 1.)

Rotational speed corrected for inlet temperature may be set accurately in a number of ways, but pulse-counting tachometers should have a high rate of counting and a short time base and rig drivers should be provided with an easily read, sensitive, direct-indicating tachometer. (*See* Section 4.2 and Appendix II.)

Inlet temperature is best measured, in general, in the low-velocity air between the rig entry (filter house, etc.) and the compressor entry. Thermocouples and resistance grids are both satisfactory provided that they are properly applied. (*See* Section 4.3.1, Figures 2, 3 and 4, and Appendices III, IV and V.)

Temperature rise can conveniently be measured directly by thermocouples connected in opposition. Additional individual thermocouples (or resistance elements) are essential both as a check on temperature rise and distribution and, at inlet, for locating the end point for the conversion of the e.m.f. difference to a temperature rise. Outlet temperature instruments should be positioned a short distance downstream from the outlet pitot-comb position. (*See* Section 4.3.2, Figures 1, 2, 3 and 4, and Appendices III, IV and V.)

Inlet total pressure should be an area-mean total pressure obtained from pitot combs; four combs of five or more tubes should be adequate for a reasonably good velocity profile. It is useful to check the area-mean total pressure against the mean total pressure derived from a static-pressure and mass-flow measurement. (*See* Section 4.4.1, Figures 6 to 10, and Appendix VI.)

Outlet mean total pressure should be derived from the outlet static-pressure measurements, the static tappings (four or more) being situated in a parallel annulus whenever possible (Figure 10). When pitot combs are used a large number should be fitted, (about 10 to 12), the number being prime to the number of O.G.V.'s and the peripheral spacing even. Each comb should have five or more tubes of the 'Kiel' type. The pitot combs should be fitted two to three blade chords or about one annulus height downstream from the trailing edges of the O.G.V.'s (*See* Section 4.4.2, Figure 9 and Appendix VI.)

Static pressure should be measured by carefully made tappings situated away from local surface irregularities; interstage tappings should be in the same position relative to the centreline of the stator blade passage for every stage, and if possible, sited axially midway between the stator trailing-edge and rotor leading-edge planes. (*See* Section 4.5 and Figure 10.)

Torque is most reliably measured by the direct measurement of the torsional strain of the torque shaft such as is afforded by the Van Milligan type of torque meter. (Small rigs normally have a swing-mounted driving motor.)

It is usual to measure the temperature rise as well as the torque, while a gear-box intermediate between compressor and torque meter is to be avoided. (*See* Section 4.6.)

A standard form of definition, interpretation and computation should be used to arrive at the final result. (*See* Sections 3.0, 5.1 and 5.2.)

REFERENCES

No.	*Author(s)*	*Title, etc.*
1	G. W. C. Kaye and T. H. Laby ..	*Physical and Chemical Constants.* Longmans, Green & Co., London. 11th Edition, New Impression. June, 1957. (Also 10th Edition. March, 1948.)
2	British Standards Institution ..	Barometer Conventions and Tables. B.S.2520. 1954.
3	D. Fielding and J. E. C. Topps ..	Thermodynamic data for the calculation of gas turbine performance. A.R.C. R. & M. 3099. June, 1954.
4	J. H. Keenan and J. Kaye	*Thermodynamic properties of air including polytropic functions.* John Wiley and Sons, Inc., New York. Second edition. May, 1947.
5	Research and Standards Branch, Bureau of Ships, Navy Department, Washington, D.C.	Gas turbine gas charts. Research Memo. No. 6–44. December, 1944.
6	N.A.C.A. Sub-Committee on compressors	Standard procedures for rating and testing axial-flow compressors. N.A.C.A. Tech. Note 1138. September, 1946.
7	British Standards Institution ..	*Flow Measurement.* B.S. Code 1042.
8	British Standards Institution ..	*Temperature measurement.* B.S. Code 1041. 1943.
9	American Institute of Physics ..	*Temperature, its measurement and control in science and industry.* Reinhold Publishing Corporation. 1941.
10	P. W. Kilpatrick	Accuracy of thermocouples in parallel. *Instruments and Automation.* pp. 1706 to 1709. 30th September, 1957.
11	Meteorological Office, Air Ministry ..	Climatological atlas of the British Isles. Publication M.O.488.
12	F. A. L. Winternitz and C. F. Fischl	A simplified integration technique for pipe flow measurement. *Water Power*, Vol. 9, No. 6, pp. 225 to 234. June, 1957.
13	Massachusetts Institute of Technology	Aerodynamic measurements. (Gas Turbine Laboratory, 1953.)
14	R. E. Rayle	An investigation of the influence of orifice geometry on static pressure measurements. S. M. Thesis, Massachusetts Inst. of Tech., Dept. of Mech. Eng. 1949.
15	D. V. Foster	The performance of the 108 compressor fitted with low stagger free vortex blading. A.R.C. C.P.144. June, 1952.

No.	*Author(s)*	*Title, etc.*
16	F. A. L. Winternitz	Effects of vibration on pitot probe readings. *The Engineer*. 30th March and 6th April, 1956.
17	H. Pearson and A. B. McKenzie ..	Wakes in axial compressors. *J. R.Ae.Soc.*, Vol. 63, p. 415. July, 1959.
18	L. Prandtl	Attaining a steady air stream in wind tunnels. N.A.C.A. Tech. Memo. No. 726. October, 1933. (Translated from : *Handbuch der Experimental Physik*, Vol. IV, Part 2, pp. 65 to 106.)
19	Lars Malenquist	Temperature measurements in high-velocity gas stream. Trans. of the Royal Institute of Technology, Stockholm, Sweden. No. 15. 1948.
20	W. H. P. Leslie	Precision control of shaft speed. *Electrical Energy*, Vol. 1, No. 1, pp. 2 to 5. September, 1956.
21	National Physical Laboratory ..	Calibration of temperature measuring instruments. *Notes on Applied Science* No. 12. 1957.
22	William H. McAdams	*Heat transmission.* McGraw Hill Publishing Company. 2nd Edition, 1942. (Published in London, 1951.)
23	T. M. Stickney	Recovery and time response characteristics of six thermocouple probes in subsonic and supersonic flow. N.A.C.A. Tech. Note 3455. July, 1955.
24	F. A. L. Winternitz and D. Hopkins	Simple total-pressure probes with spherical shields. D.S.I.R. Fluids Report No. 63. January, 1958.
25	Joseph P. Doyle, Jnr.	Instrumentation and data handling methods for determining load distribution on wind tunnel models by pressure distribution measurements. Agard Report No. 114. April/May, 1957.
26	L. Fuller and B. Marlow	An improved design of sonic suction pyrometer. Unpublished M.o.A. Report.
27	R. Shaw	The influence of hole dimensions on static pressure measurements. *J. Fluid Mech.*, Vol. 7, Part 4, pp. 550 to 564. April, 1960.
28	D.S.I.R., National Engineering Laboratory, East Kilbride	Proceedings of a Symposium on flow measurement in closed conduits. 27th to 30th September, 1960. H.M.S.O., London. 1962.
29	M. D. Scadron, C. C. Gettelman and G. J. Pack	Performance of three high-recovery-factor thermocouple probes for room temperature operation. N.A.C.A. Research Memo. E50129. TIB.2444. December, 1950.

No.	*Author(s)*	*Title, etc.*
30	N. A. Dimmock	Unpublished work at N.G.T.E. March, 1959.
31	Rolls-Royce Limited	Test methods for compressors, their accuracies and presentation of results. Ref. GLW/GST.3/KJR. 20th March, 1952.
32	N.G.T.E. Aerodynamics Department	Methods of testing compressors and turbines, their accuracies and presentation of results. ADSC/VG. 24th March, 1952.
33	Armstrong Siddeley Motors Limited	Some notes on the testing of compressors. GWB. 26th March, 1952.
34	D. Napier and Sons Limited ..	Testing methods for axial compressor and turbine units. LAN/BC/811. 26th March, 1952.
35	Bristol Aero Engines Limited ..	Gas turbine engine instruments and measuring techniques. M. Ginniff. May, 1957.
36	D. Napier and Sons Limited ..	Pressure instruments for aerodynamic measurements in gas turbine research. Airflow Laboratory Report No. 147. 1957.
37	D. Napier and Sons Limited ..	Manufacture of $\frac{1}{4}$ in. wedge-type instrument S.5408/1. Combined yaw and total head instrument. REM/MW/1040. 14th March, 1958.
38	C. E. Moss and G. G. Annear ..	Unpublished work at N.G.T.E.
39	B. D. Blackwell, Bristol Aeroplane Co. Limited	Experimental anomalies due to the concept of mean total head. G.N.513. 4th March, 1952.
40	D. Napier and Sons Limited ..	Temperature measurement for aerodynamic research. Airflow Laboratory Report No. 160. December, 1959.
41	N. A. Dimmock	A compressor routine test code. Draft Report for the G.T.C.C. Aerodynamics Sub-Committee.

TABLE 1

Properties of dry air[3]

H = Total heat above 0°K C.H.U./lb

T°K	K_p	γ	$\frac{\gamma}{\gamma - 1}$	$\frac{\gamma - 1}{\gamma}$	H
270	0·2395	1·4010	3·494	0·2862	64·52
280	0·2396	1·4009	3·495	0·2862	66·91
288·16	0·2396	1·4008	3·496	0·2861	68·87
290	0·2397	1·4006	3·497	0·2860	69·31
300	0·2398	1·4003	3·499	0·2858	71·70
310	0·2400	1·3998	3·501	0·2856	74·10
320	0·2401	1·3996	3·503	0·2855	76·50
330	0·2403	1·3991	3·505	0·2853	78·90
340	0·2405	1·3987	3·508	0·2850	81·31
350	0·2407	1·3982	3·511	0·2848	83·71
360	0·2409	1·3977	3·514	0·2846	86·12
370	0·2411	1·3973	3·517	0·2843	88·53
380	0·2414	1·3966	3·521	0·2840	90·94
390	0·2416	1·3961	3·524	0·2837	93·36
400	0·2419	1·3955	3·528	0·2834	95·77
410	0·2422	1·3948	3·533	0·2830	98·19
420	0·2426	1·3940	3·539	0·2826	100·62
430	0·2429	1·3932	3·544	0·2822	103·04
440	0·2432	1·3925	3·549	0·2818	105·48
450	0·2436	1·3916	3·554	0·2814	107·92
460	0·2440	1·3907	3·559	0·2809	110·36
470	0·2444	1·3898	3·565	0·2805	112·80
480	0·2448	1·3889	3·571	0·2800	115·24
490	0·2453	1·3878	3·578	0·2795	117·69
500	0·2458	1·3868	3·586	0·2789	120·14
510	0·2463	1·3857	3·593	0·2783	122·60
520	0·2468	1·3846	3·600	0·2778	125·07
530	0·2473	1·3835	3·608	0·2772	127·54
540	0·2478	1·3824	3·615	0·2766	130·02
550	0·2483	1·3814	3·622	0·2761	132·50
560	0·2488	1·3803	3·629	0·2755	134·99
570	0·2494	1·3791	3·637	0·2749	137·47
580	0·2499	1·3780	3·645	0·2743	139·97
590	0·2505	1·3768	3·653	0·2737	142·48
600	0·2510	1·3757	3·661	0·2731	144·98
610	0·2515	1·3746	3·669	0·2725	147·50
620	0·2521	1·3735	3·678	0·2719	150·01
630	0·2527	1·3723	3·686	0·2713	152·53
640	0·2532	1·3712	3·694	0·2707	155·06
650	0·2538	1·3700	3·702	0·2701	157·60

APPENDIX I

Systematic Errors

A brief discussion follows on some of the values given in Section 3.0. The paragraph numbers correspond to those in the main section.

(i) *Gravitational constant, g.*

Acceleration due to gravity varies by only $0 \cdot 056$ per cent from Plymouth to Aberdeen[1] and, since test plant is usually sited on reasonably level, low-altitude land masses, the variation of the gravitational constant may safely be ignored.

(iii) (iv) and (vii) *Absolute pressures—conversion.*

When a control room is unheated or poorly ventilated, the temperature of the barometer and of the manometers may vary considerably more than the limits quoted in the main section. Under these conditions the barometric pressure should be corrected from (for example) Reference 2 before adding the corrected manometer readings. (*See* Figure 15.)

(xi) *Isentropic index.*

When determining the compressor efficiency, the use of the isentropic index as defined and used in Section 5.1 (iv) introduces a systematic error into the calculated value of the isentropic temperature rise, since the specific heat at constant pressure K_p does not vary linearly with temperature. This systematic error exceeds the random error of enthalpy charts[5] or tables[3,4] and entropy functions only when the temperature rise exceeds about 200°C. The following example indicates the magnitude of the error:

$$T_1 = 290°\text{K},\ T_2 = 525°\text{K},\ P_2/P_1 = 6 \cdot 00.$$

Efficiency from total-heat and entropy-function tables, η_H, is

$$\eta_H = 81 \cdot 49_1 \text{ per cent.}$$

Efficiency from the recommended method of Section 5.1 (ii) η_T, is

$$\eta_T = 81 \cdot 53_2 \text{ per cent.}$$

(xiii) *Humidity.*

In the British Isles the relative humidity is seldom less than 20 per cent and its hourly variation is a maximum in Summer. The largest single factor determining relative humidity at any place is its distance from the sea, although low relative humidity inland is offset, as far as the operative quantity m is concerned, by the higher inland temperatures. (m is ratio of water vapour/dry air by weight.) Similarly, the gradient of relative humidity from South-West to North-East is more than compensated for by the temperature gradient. General information may be obtained from Reference 11, although local measurements at the time of testing are recommended in doubtful conditions and the following types of weather are obviously suspect; warm with early morning mist (fog under any circumstances); fine warm days with cumulus cloud particularly after showers and with a South or South-West wind.

The curves of Figure 14 may be used to evaluate m sufficiently accurately from readings of relative humidity and ambient temperature, and for correcting temperature rise, density and mass flow. If in the example given above, $m = 0 \cdot 01$, then the corrected efficiency becomes $81 \cdot 22_8$ instead of $81 \cdot 49_1$.

APPENDIX II

Corrected Speed Setting

Accurate devices for measuring and indicating rotational speed are numerous as are the principles of operation, while for the automatic control of shaft speed see for example, Reference 20. Here attention will be given to two systems which are based upon the same broad principle and which, whilst no more complicated than many of the other methods, are considered superior from the point of view of display and manual control.

The common principle is the indication of a frequency difference and the common advantage is that of a null-point setting. Both systems use a standard engine-speed indicator (E.S.I.) for rough setting and the specialised equipment for setting precise values of corrected speed.

(i) The first system is described in the appendix of Reference 34 and, as this is not generally available, it will be described briefly here.

A two-phase output signal is obtained from a small cylindrical magnet fitted to the end of the compressor shaft and rotating within a coil assembly attached to the frame of the machine. Thus the size and mass of the additional rotating part is practically insignificant and a generous running clearance can be used. The output is about 0·3 volt at 10,000 rev/min and the phase difference is 90 electrical degrees. The amplified signal is fed to the 'X' and 'Y' plates of a cathode-ray oscilloscope to give a circular trace with the spot rotating at shaft speed. A second signal from an accurate and stable decade oscillator is applied to the suppressor grids of the amplifying valves to modulate the circular trace to a Lissajous figure. The appropriate selection of oscillator frequency in c/sec gives a six-lobed figure such that when the pattern is stationary the shaft speed in rev/min is ten times the frequency reading.

For setting a given corrected speed, the required frequency is first scaled by the ratio $\sqrt{288 \cdot 15}/\sqrt{(T_t)_{\text{inlet}}}$ and set up on the oscillator and the compressor speed finally adjusted to bring the Lissajous pattern stationary. Although the decade oscillator in use is guaranteed to an accuracy within 0·1 per cent and is extremely stable, a check calibration is frequently applied by comparison with the signal from a 1,000 c/sec valve-maintained tuning fork.

(ii) In the other system an accurate high-stability oscillator is again required, but the frequency range is 50 c/sec $\pm$ 2 c/sec corresponding to a range of inlet temperature from about -8°C to $+39$°C, and the frequency-control dial is conveniently calibrated in °C inlet temperature. The armature of an engine-speed indicator runs synchronously with the generator (and usually at one-quarter shaft speed) above a certain critical speed which is about 600 rev/min (2,400 rev/min shaft speed) for one generator driving one indicator and rather higher if two indicators are driven from one generator. If this limitation is an embarrassment, it can be removed as will be described later. Below this critical speed the indicator runs as an induction motor with slip.

In a modification to the E.S.I. the eddy-current drag coupling and pointer mechanism is removed and, in its place, a light disc is fitted to the shaft. A stroboscopic pattern is stuck to this disc and a housing made for a neon strobe-tube driven by a simple power pack and controlled by the oscillator. Nominally this is a corrected-speed-setting device and only rounded values of corrected rev/min can be set, although for normal use, this is no disadvantage since it is what is required. However, if an odd number of corrected rev/min is wanted, the increment of speed $\pm \Delta N$ may be converted into an equivalent temperature which is added to the inlet temperature before setting the oscillator, the resulting corrected speed being sufficiently accurate if the stroboscopic pattern is well chosen. For an E.S.I. driven at one-quarter shaft speed, a suitable pattern has 120 divisions at the outer radius for 100 rev/min steps, 48 divisions inside these for 250 rev/min intervals, then 24 and 12 divisions for 500 and 1,000 rev/min setting. Frequently there is only one take-off shaft for a tachometer drive and it may be necessary to drive four E.S.I. heads from one generator —one pair in the Control Room and the other pair for the driver of the compressor motor or turbine, one of each pair being a standard E.S.I. for rough speed setting. A straightforward three-phase amplifier is suitable for this and has the advantage that its output characteristic may be adjusted to increase the power at low frequencies and so drive the E.S.I. rotors synchronously down to speeds of about half those at which they normally pull into step, i.e. 1,200 rev/min shaft speed.

Yet another method of speed control is given in Reference 20.

APPENDIX III

Temperature Measurement—General

The more usual instruments for temperature measurement are as follows:

Mercury-in-glass thermometers have, until fairly recently been used extensively in compressor testing, but they have now been superseded by other instruments almost entirely. However, they remain as important second-line tools for the calibration and checking of other instruments.

The mercury-in-glass thermometer is inherently accurate to any chosen degree which depends upon calibration[21]; it is rather fragile and is not well suited to grouping in restricted areas of high-speed airflow since the stagnation shield must be comparatively large. (*See* Figure 4.) Direct reading is usually necessary and this may bring the observer into an environment which is sufficiently dangerous or unpleasant to affect the accuracy of reading.

Resistance thermometers can also be very accurate and give a remote reading but, when in the form of a cylindrical probe having a shielded element, the heating effect of the bridge current can cause large errors in a low-velocity air stream. This disadvantage can be overcome by using refined equipment and small currents and by switching on the bridge circuit for only the short period necessary to obtain a reading. The elements are not cheap and are rather fragile although the open-grid or spiral type of element often used for intake-temperature measurement is better in this respect. In a high-velocity flow the problem of recovering the full dynamic temperature is no less than with a thermocouple, and a stagnation pocket such as that illustrated in Figure 4 is necessary. The external resistance features in the accuracy of measurement and is a disadvantage.

Mercury-in-steel remote-reading thermometers are sluggish in response, bulky and not particularly accurate.

Thermistors have a non-linear response and a narrow range and, although very sensitive to small temperature changes, they are not suitable for temperature measurement in routine compressor testing.

Thermocouples appear to be an obvious choice and their main disadvantage lies in their inability to provide an accurate, direct scale reading of temperature. However, it is considered by many people that the advantages far outweigh the disadvantages. Thermocouple probes are fairly robust, cheap and they can be manufactured on site; an accurate null-point method of measurement of the e.m.f. and an accurate calibration of the wire is straightforward. (*See* Appendices IV and V.)

The external resistance does not affect the accuracy of measurement with a null-point method, although it does affect the sensitivity. Appendix IV discusses thermocouple measurements in greater detail, while thermocouple (and shield) calibration is dealt with in Appendix V.

Measuring planes.—While there is little difficulty in positioning the instruments for inlet-temperature measurement (unless interstage bleed air is recirculated) the best position for outlet-temperature measurement is open to question. If the measuring plane is close to the O.G.V.'s, the velocity is usually fairly high and there are both velocity and temperature gradients. If the measuring plane is moved downstream to gain the advantage from mixing and a lower velocity, intermediate heat transfer to the surroundings might result in a lower indicated temperature. It is for these reasons that the recommendation made in Section 4.3.2 is for the outlet-temperature measuring plane to be a short distance downstream from the outlet pitot-comb positions.

If the temperature is measured further downstream than is recommended then an estimate may be made of the heat loss and hence the mean temperature drop. Some simplified equations[22] have been used to compile the following table of combined heat-transfer coefficients for a range of pipe diameters and temperature difference. The assumptions are that the surrounding air and walls remain at about 15°C, that the surface emissivity for fabricated steel ducting is 0·3 and (pessimistically) that the ducting temperature is the same as the outlet-air total temperature.

TABLE 2

Combined Heat-Transfer Coefficients for Circular Ducting, C.H.U./(h) (sq. ft) (°C)

D ft \ Δt°C	50	100	150	200	250	300	350	400
1·0	1·21	1·47	1·70	1·94	2·18	2·45	2·73	3·05
1·5	1·13	1·38	1·60	1·82	2·06	2·32	2·60	2·92
2·0	1·08	1·31	1·53	1·75	1·98	2·24	2·52	2·83
2·5	1·04	1·27	1·48	1·70	1·93	2·18	2·46	2·76
3·0	1·01	1·23	1·44	1·65	1·88	2·13	2·41	2·71
3·5	0·99	1·21	1·41	1·62	1·85	2·10	2·37	2·67
4·0	0·97	1·18	1·38	1·59	1·82	2·06	2·34	2·64
4·5	0·95	1·16	1·36	1·57	1·79	2·04	2·31	2·61
5·0	0·93	1·14	1·34	1·55	1·77	2·01	2·29	2·59
6·0	0·91	1·11	1·31	1·51	1·73	1·98	2·25	2·55
8·0	0·87	1·07	1·26	1·46	1·68	1·92	2·19	2·48
9·0	0·86	1·05	1·24	1·44	1·66	1·90	2·16	2·46

Using these values in an example[30] where the air temperature was 215°C and estimating an equivalent length of cylindrical ducting to replace the awkwardly shaped collector box from a compressor test rig at N.G.T.E. (Figure 17) the calculated temperature drop was 0·66°C. A calculation which included the heat-flow balance to and from the ducting gave an answer of 0·40°C. This may still be pessimistic since it was assumed that the heat loss was distributed evenly throughout the fluid, whereas it is more likely for the low-temperature air to be concentrated near the duct walls and so not to affect the sensing elements if these project reasonably far into the duct.

APPENDIX IV

Thermocouples

A thermocouple probe is fairly robust, cheap to make and it can be manufactured on site; an accurate null-point measurement of the e.m.f. and an accurate calibration of the wire is straightforward. (*See* Appendix V.) Thermocouples can be connected in parallel to measure an arithmetic-mean e.m.f. provided that all the thermocouple loops have the same resistance[10]. They can also be connected in opposition to measure accurately a temperature difference. The external resistance does not affect the accuracy (assuming that a null-point measurement is made—and this is essential for high accuracy) although it does affect the sensitivity.

Thermocouple wire pairs.—The order of preference given in Section 4.3 for the choice of thermocouple pairs follows from a consideration of the properties of each pair.

Chromel-constantan has the highest e.m.f. of all the commonly used base-metal thermocouples (about 0·06 mV/°C) and has a useful upper temperature limit of about 700°C (Reference 8). Both alloys are poor conductors and this eases the problem of heat conduction along the wires. In the normal range of temperature found in compressor testing, the data of Reference 9 suggest that the time-temperature instability is small and unlikely to lead to errors exceeding 0·1°C, while both alloys are reasonably resistant to oxidation.

Chromel-alumel thermocouples give the lowest e.m.f. of the commonly used materials (about 0·041 mV/°C). It has the advantage that both materials are fairly poor conductors of heat. With an upper temperature limit of 1,100°C (Reference 8) the wire (but not the same instruments) can be used also for turbine testing.

A disadvantage lies in its time-temperature instability at the higher temperatures[9] and there is some suspicion that significant instability also exists at lower temperatures (0 to 200°C) although no evidence for this has been found so far; the data of Reference 9 suggest that (for compressor testing) the error is unlikely to exceed 0·2°C.

Copper-constantan is a combination which is frequently used, although the thermal conductivity of the copper wire is an embarrassment and often the reason for discarding this type of element, rather than adopting special construction methods—usually rather complicated—to counteract the effect. The junction gives an e.m.f. of about 0·043 mV/°C and it is subject to oxidation, the products of which diffuse in the junction to give a changing calibration with time and conditions of use. There is an advantage in that the copper wire can be connected to a copper lead without an intermediate cold junction and without generating a significant secondary e.m.f., since commercial copper wire is sufficiently pure for this purpose. The maximum temperature for a reasonable life is about 400°C (Reference 8).

Iron-constantan is a combination giving an e.m.f. of about 0·05 mV/°C and a maximum temperature of operation of about 850°C, but it has some major disadvantages. The chief of these is that a variation of output is often found from one thermocouple to another, when all the junctions are made from the same batches of wire. This necessitates the individual calibration of each element. Again the thermocouple is subject to corrosion and this suggests that the output must change with time as the corrosion products diffuse through the junction while, at temperatures above 400 to 500°C, the time-temperature instability increases rapidly[9]. For these reasons this combination is not recommended.

Thermocouple shields.—For measuring the total temperature in the airflow the thermocouple element must be mounted in a shield, the purpose of which is fivefold:

(i) It must adequately support the element.

(ii) It must bring the airflow almost to rest at the thermo-junction and yet

(iii) pass a flow sufficient to ensure good heat transfer to the junction.

(iv) This bleed flow should bathe the wires for a short distance away from the junction to counteract, as far as possible, the conduction of heat along the wires to or from an external environment.

(v) The shield must protect the junction from radiation whenever this is likely to be a significant source of error.

Figures 2 and 3 show a selection of thermocouple probes to cover a range of application. For example, the probe at the top of Figure 2 is simple to make and suitable for inlet-temperature measurement. The probe (Type B) of Figure 2 could be used for high outlet temperatures with fairly low velocity, while Type C [40] is very useful where a short immersion length is required.

The thermocouple-shield combination should be calibrated in reasonably representative conditions, either to check that the recovery factor is sufficiently near unity for conditions where T_v is small, or to determine its characteristics over a range of Mach number when corrections are to be made for large values of T_v. {T_v is the temperature equivalent of velocity $T_v = V^2/2gJK_p$. The recovery factor is given by $r = (T_i - T)/(T_{\text{tot}} - T)$ where T_i is the indicated total temperature and T is the static temperature.} (*See* the second part of Appendix V, for shield calibration.)

A necessary compromise is that between the strength of the shield and the avoidance of conduction errors. In general therefore, the shield should be made of tubing having the least wall thickness that is considered safe for the particular application. (*See* also Reference 16.)

Error can arise from probes penetrating only a short distance into a duct carrying a low-speed flow at high temperature because the heat-transfer coefficient of the junction and adjacent wires is reduced and the conduction path short.

If this trouble cannot be avoided by re-siting the thermocouple probes and for other difficult conditions, such as hot casings caused by conduction, or narrow passages such as are found in centrifugal-compressor casings, then a sonic suction pyrometer may be used[26]. These instruments are difficult to manufacture in the smaller sizes and require a suction pump adequate to maintain an overall pressure ratio of about 2·5, but they possess the advantage of a constant recovery or correction factor of the order $T_{\text{tot}} = 1{\cdot}027\,T_{\text{indicated}}$. The factor varies slightly from one probe to another but the calibration is simple. (*See* last paragraph of Appendix V.)

APPENDIX V

Thermocouple Calibration

The calibration of a thermocouple probe can be divided conveniently into two sections: (i) finding the relationship of e.m.f. to temperature for the junction and (ii) measuring the dynamic performance of the complete probe.

The following notes aim at a standard rather higher, perhaps, than is essential for the overall temperature measurement in a multi-stage compressor, and might prove useful therefore when small temperature differences are to be measured accurately. It is a good fault to be scrupulously careful when calibrating or using thermocouples.

(i) *Wire calibration.*

The temperatures encountered in compressor testing seldom lie outside the range -10 to $+300$°C while $+350$°C is most unlikely to be exceeded in normal practice. For this reason, a stirred liquid bath is suitable for calibrating thermocouple junctions. Complete probes can be, and often are, calibrated in a liquid bath but this should be avoided whenever possible because of the very real danger of conduction errors. The individual calibration of iron-constantan junctions may necessitate calibrating the complete probe (*see* notes on iron-constantan wire, Appendix IV), and care should be taken that the immersion and the heat transfer to the junction are sufficient to avoid conduction errors. Apart from the combination iron-constantan, it is usually sufficient to calibrate a batch of wire by testing, say, three thermocouples made from lengths of the wires taken from each end and the middle of the batch. This check is worth doing, although inhomogeneity is not often found in base-metal thermocouple wire supplied specifically as such, and annealed or stabilised by the manufacturer. Thermocouple wires should be treated with respect and should not be manipulated in a way which could alter their characteristic, e.g. work-hardening by stretching in order to remove kinks.

The type of calibrating bath is immaterial provided that it satisfies the requirements of the test which are:

(*a*) that the temperature can be controlled within fine limits,

(*b*) that the positioning of the heater, baffles and stirrer is such that the thermocouple element(s) and standard thermometer are in a region of uniform temperature.

(*c*) that the thermocouple junction(s) and the standard thermometer can be given the same adequate immersion and be close together.

The apparatus need not be elaborate and Figure 16 shows simple equipment which has been found quite adequate.

The temperature of the bath may be measured either by N.P.L. calibrated mercury-in-glass thermometers or by a platinum resistance thermometer. The former standard is probably the simplest and is very reliable provided that the thermometers are re-calibrated from time to time —more frequently in their early life and for those in the higher temperature range[21]. As an additional check, previously calibrated thermocouples also may be used.

There are several liquids which are suitable and which cover the range up to 300°C. Two examples are 'Super-Hecla' oil—a steam-cylinder type of lubricant—and Di-butyl phthalate. The former is rather too viscous for easy stirring at temperatures below about 50°C, but the latter is satisfactory over the range $-15°$ to $+300$°C, although it begins to bubble over the upper 30° of this range. (The boiling point is 338°C, but vapour is given off at temperatures above about 100°C which, in common with most oily vapours, is unpleasant to breathe.) A fume cupboard is desirable if only to reduce the possible danger of flashing at the higher temperatures. For temperatures below room temperature, crushed solid CO_2 can be placed in the space between the beaker and the vacuum flask (Figure 16) and the required temperature kept steady by the use of the heater. The reference junction temperature is most conveniently kept at 0°C. (*See* 'Other precautions with thermocouples', Section 4.3.)

The calibration curve can be drawn through a large number of measured points, or for some thermocouple materials an equation can be obtained from a calibration at only a few points. For copper-constantan couples a suitable equation is of the form[9] $e = at + bt^2 + ct^3$ where e = e.m.f. in millivolts, t = temperature in °C and a, b and c are constants determined by calibration at about 100, 200 and 300°C. The interpolated values in the range 0 to 300°C will be as accurate as the couple can be relied upon to retain its calibration (about 0·2°C). Chromel-alumel couples give a calibration curve not easily or accurately fitted by equations, while no data were available[9] as to how well the temperature e.m.f. relationship of an iron-constantan couple could be fitted by equations. From experience gained in the past few years, it appears that an equation of the same form as that for copper-constantan can also be used for chromel-constantan with an accuracy within ±0·2°C from 0° to 300°C, the particular equation for one batch of wires being, for example

$$e = 0 \cdot 05572\, t + 0 \cdot 533 \left(\frac{t}{100}\right)^2 - 0 \cdot 0576 \left(\frac{t}{100}\right)^3$$

with e in millivolts and t in °C above 0°C reference temperature. Such an equation for finding a temperature corresponding to a reading of e.m.f. is rather clumsy for manual computation, a calibration curve being much simpler, but, when a fast digital computer is used *ab initio* for data reduction, it may prove more economical to use an equation than to store, read and interpolate a table of values.

(ii) *Dynamic calibration.*

It is necessary to calibrate the complete instrument, consisting of the thermocouple element and its shield, to determine the amount by which the indicated temperature T_i is less than the true total temperature T_{tot}. The measure of the accuracy of the probe under dynamic conditions is often referred to as a 'recovery factor' defined as

$$r = \frac{T_i - T}{T_{tot} - T} \text{ where } T \text{ is the static temperature.}$$

The term recovery factor as defined is not accurately descriptive since it should strictly apply only to the performance of the probe in bringing the fluid adiabatically to stagnation conditions at the thermocouple junction, and should not include errors due to conduction and radiation. However, the various causes of error are not easily separated and moreover the highest factor is obtained when the sum of the residual kinetic energy at the junction and the heat loss is a minimum.

The recovery factor of a thermocouple probe may conveniently be measured using a small wind tunnel, preferably with the air inducted through a faired intake. A reference couple made of the same material and without any shielding is mounted in the very low-velocity air ahead of the contraction and connected in opposition to the couple under test in the working section. For the highest accuracy, or where the wind-tunnel intake is subject to temperature fluctuations—a common occurrence in a heated laboratory—a smoke trace should be used to position the reference junction close to the streamline passing the test couple.

The heat transfer to the junctions, and therefore the recovery factor, is affected by changes of density and in Reference 29 an example shows that the recovery factor drops from 0·985 to 0·96 when the ambient density is reduced from 0·10 to 0·04 lb/cu. ft at a flow Mach number of 0·5. More information on this subject may be found in Reference 23. For this reason, calibration under representative conditions is recommended.

A qualitative check can be made to ensure that conduction errors are small at the immersion and air speed to be used in the rig, by packing crushed solid CO_2 around the probe where it emerges from the wind tunnel, or by heating it.

At the expense of considerable complication this check on conduction errors can be made quantitative.

In Figure 18 is shown the calibration curves for two probes made to the general design of Figure 2. The results are plotted both as a recovery factor and as a temperature error in °C. Also added is the equivalent of the possible potentiometer error and a curve of $T_{tot} - T$.

Sonic pyrometers may be calibrated in the same general way, but without the wind tunnel. The reference thermocouple is mounted close to the pyrometer inlet and a range of suction pressures applied to the instrument. A plot of the resulting correction factor, T_{tot}/T_i, versus the working suction pressure ratio will show the minimum pressure ratio necessary across the instrument to ensure an adequately constant factor.

APPENDIX VI

Sampling and Weighting

Inlet temperature and pressure seldom require very refined sampling techniques since it is usual to go to considerable lengths to ensure that the test rig provides the compressor with a uniform inlet flow. Also it is common practice to explore the inlet flow in new test rigs and this experience can be used to reduce and simplify the instrumentation there without loss of accuracy.

For these reasons, an area-mean inlet total pressure has been adopted since, with reasonably uniform flow, there is little difference between the variously weighted means. At outlet however this is certainly not so and it is found that

$$\bar{P}_{T.M} > \bar{P}_{tA} > \bar{P}_s$$

where

$\bar{P}_{T.M}$ = mass-weighted mean total pressure

$\bar{P}_{tA}$ = area-mean total pressure

$\bar{P}_s$ = mean total pressure derived from static pressure and the continuity equation.

In Reference 39 it is shown that, for incompressible flow

if

$$P_s = p_s + \tfrac{1}{2}\rho\bar{v}^2$$

then

$$P_{tA} = p_s + \tfrac{1}{2}\rho\bar{v}^2(1+\phi)$$

and

$$P_{T.M} = p_s + \tfrac{1}{2}\rho\bar{v}^2(1+K\phi)$$

where ϕ is a 'distribution factor' analogous to the wave-form factor in electrical theory and approximates to $\phi \doteqdot 1/m(m+2)$ where $1/m$ is the power defining the velocity distribution. It is shown also that

$$K \doteqdot \frac{m+2}{m+3}\left(3+\frac{1}{m}\right).$$

The assumptions are rather sweeping in that the hub-tip ratio is taken to be close to unity, while no account is taken of temperature and static-pressure variations around or across the annulus or through blade wakes. However, it is of interest to give an example.

Taking

$$m = 7 \text{ for a compressor inlet}$$

$$K = 2 \cdot 82 \qquad \phi = 0 \cdot 016$$

At a flow Mach number of 0·5 the kinetic pressure $\frac{1}{2}\rho\bar{v}^2$ is little more than one seventh of the total pressure and we get

$$\bar{P}_{T.M} - \bar{P}_s \doteqdot 0 \cdot 65 \text{ per cent of the total pressure and}$$

$$\bar{P}_{T.M} - \bar{P}_{tA} \doteqdot 0 \cdot 45 \text{ per cent of the total pressure.}$$

At the same Mach number and putting $m = 2$ for the compressor outlet

$$K = 2 \cdot 80 \text{ and } \phi = 0 \cdot 125$$

$$\bar{P}_{T.M} - \bar{P}_s \doteqdot 5 \cdot 0 \text{ per cent of the total pressure}$$

$$\bar{P}_{T.M} - \bar{P}_{tA} \doteqdot 3 \cdot 5 \text{ per cent of the total pressure.}$$

These values will be pessimistic since fully developed turbulent flow is seldom found in a compressor intake (i.e. the power law applies to the flow in the boundary layer only) and the outlet velocity profile, although often roughly parabolic away from the walls, is not the 'classical' parabola which terminates at zero velocity at each wall (i.e. the ratio $V_{\text{max}}/V_{\text{mean}}$ is smaller for the practical flow than for the theoretical).

Experience tends to show that the derived mean total pressure $\bar{P}_s$ gives a more consistent result than that obtained by other methods of weighting and it is argued by many (for example, Reference 6), that the compressor should not be given credit for the flow energy associated with a non-uniform outlet velocity (including swirl) and, for this reason, the derived mean total pressure gives a more representative performance.

The final arbiter is practicability; the large amount of instrumentation and analysis work necessary to obtain a mass-weighted mean pressure is unlikely to be acceptable in routine testing, particularly when it is doubtful whether the effort is justifiable. Therefore, from these general considerations, the derived mean total pressure $\bar{P}_s$ is the recommended value for compressor outlet pressure.

With temperature the arguments for a mass-weighted mean cannot be disputed, but fortunately the problem is not so acute since the sensing elements can be sited in a region where some mixing has evened out the temperature profile. (*See* also Appendix III.) Nevertheless, the problems of weighting are the same as for pressure and the recommendation in Section 4.3.2 is a practical solution, leaving sufficient flexibility in interpretation to cover the particular requirements of widely differing test-rig layouts.

APPENDIX VII

Static-Pressure Measurement

In routine compressor testing static pressure is seldom measured by any other means than by wall tappings and the subject will be limited mainly to these. Static pressure is a difficult quantity to measure or even to define in the presence of turbulence and absolute accuracy in interstage measurements cannot be expected when the degree of turbulence, the swirl angle, the effective flow area and the radial and peripheral pressure gradients are not known with precision.

Static-pressure tappings can usually be fitted only in the outer casing of a compressor, although it is sometimes possible to add inner-wall static taps at inlet and outlet. The position of interstage static holes is often governed by the method of stator-blade root fixing and by the casing design. Figure 16 of Reference 15 shows the variation of static pressure over one blade pitch and also the change in the variation with mean flow coefficient. However, not enough is known on this matter to state that one relative position is better than another, the important precaution being to ensure that the same relative peripheral position is maintained through the compressor, thus position error will cancel out except for the first stage.

The static tappings themselves should conform to good practice as far as the mechanical difficulties will permit, and the following recommendations are in order of preference to allow some compromise in difficult circumstances. (*See* Figures 10a and 10b.)

(i)*a* The tapping should have a diameter of from $0\cdot020$ in. to $0\cdot040$ in. It should be normal to the surface and be carefully chamfered (90° included angle) to a depth equal to one half of the hole diameter.

or (i)*b* the tapping should be square edged if accurate chamfering is not possible. Its diameter may, in difficult circumstances, be as much as 1/16 in. and the hole may 'trail' up to 45°, but it must not 'face' upstream. (Figure 10b.)

(From Rayle's data[14], a tapping to recommendation (i)*a* should indicate static pressure with an error of less than $\pm 0\cdot5$ per cent of the kinetic pressure for flow Mach numbers up to $0\cdot6$; a square-edged hole of $\frac{1}{16}$ in. diameter introduces an error of about $+1$ per cent of the dynamic pressure at flow Mach numbers up to $0\cdot4$.)

(ii) The depth of the hole should not be less than twice its diameter.

(iii) The tapping should be free from burrs both internal and external, and situated away from wakes and surface irregularities such as steps in the casing.

For clean, square-edged static holes normal to the surface, Shaw[27] has shown a correlation between the (positive) static-pressure error and the Reynolds number based on the hole diameter and the friction velocity. However, it is unlikely that the wall shear stress, τ_0, will be known in the vicinity of interstage static tappings and, since Shaw's analysis supports Rayle's experimental work, the latter is probably the best practical basis for determining the type of static tapping for interstage measurement.

The turbulence level in a multi-stage compressor could give rise to an error of up to $+1$ per cent of the kinetic pressure.

The only way known to the author of checking static-pressure taps in the absence of other instrumentation and in a reasonably uniform flow is to assume that consistency is the corrollary of accuracy and to measure the differences in pressure between one static hole and the rest, and to discard those showing a pressure different from the mean by more than some arbitrarily chosen proportion of the kinetic pressure. It follows that the number of static tappings provided in any one measuring plane should be as large as practicable and never less than four.

Static-pressure probes, although seldom used for fixed interstage measurements, are shown in Figure 11.

APPENDIX VIII

Miscellaneous

(A) *Traversing instruments.*

Although detailed testing and traversing was outside the terms of reference of this survey, there is a general interest in the wedge-type traversing instrument for measuring both total and static pressures as well as the two-dimensional direction of flow[36, 37]. Experience with this instrument is increasing and it is already adequate to show that the wedge pitot-static yawmeter is a most useful traversing instrument, and one which could well be adopted as a standard. A dimensioned sketch is given on Figure 19.

The original criticism that the differential pressure coefficient is less than that of arrowhead (Conrad) or claw-type instruments for the same angle of yaw remains true, but is seldom found to be a serious disadvantage. At low air speeds, the response time is long, but the advantage of measuring total pressure, static pressure and yaw angle in a single traverse outweighs both of these short-comings. It was suggested[30] that if the leading edge of the wedge was positioned at the centre of rotation of the instrument, then the remainder of the indicated yaw variation through a pressure gradient might be eliminated. (*See* Figure 7 of Reference 36.) This suggestion has not yet been tried out, but it is hoped to do so in the near future.

(B) *Digital computers and compressor testing.*

The use of digital electronic computers for processing compressor test data has spread considerably since the original survey[30], but the general observations made then remain much the same.

The main contribution that can be made by a fast digital computer is towards increased accuracy by dealing efficiently with parameters such as variable specific heat and compressible flow and by eliminating random human errors, providing that the input data are thoroughly checked. (Reference 25 quotes an experiment showing that human errors in film reading led to a scatter of 17 per cent.)

There is no advantage in using the computer to average large numbers of readings, unless these can be fed in without intermediate interpretation from the output of automatic instrumentation. Otherwise, if the readings have to be processed manually for input, and checked for accuracy, the time is better spent in averaging the readings on a desk computer, when their reliability can be examined, before preparing the input data for the digital computer. For this reason, it remains worthwhile whenever possible to develop instrumentation which can provide averaged quantities easily without awkward correction factors. If automatic measuring and recording instruments become general in routine compressor testing, it will be even more important to develop a system which does at least some or the averaging at or between the sensing element and the measuring instrument, since each measuring and recording channel is costly and scanning mechanisms are expensive also.

(C) *Instruments for engine or component testing.*

Ideally, engine instruments would be used additionally during component rig tests and the component-test instrumentation would be added in an engine test. In this way a calibration of the limited engine instruments would result and so would differences in component performance (outlet velocity profiles for example) between the rig build and the engine configuration.

There is a growing tendency to add rig instruments to engines on test, and to manufacture the engine casings with bosses suitable for additional instruments. This is to be encouraged and recommended.

(D) *Centrifugal compressors.*

The problems of instrumentation in centrifugal compressors are the same in principle as those in an axial compressor, but they are aggravated by mechanical complexity and the different airflow conditions, particularly in two (or more) stage centrifugal compressors.

Passages are usually narrow and strongly curved where most easily accessible, and this imposes a short immersion length and a velocity gradient upon thermometer probes, and this may necessitate the use of sonic suction thermocouple probes[26]. The variation in air speed through the compressor is large, as are static-pressure gradients; it is difficult to measure the separate performances of rotor and stator and to compare one stage with another sufficiently accurately.

The detailed mechanical design often differs greatly from one compressor to another and, since this is the largest single factor which governs the type and positioning of the instrumentation, standardisation is almost impossible. It can only be emphasised that the type of individual instruments should be chosen to suit the local flow conditions and the principles of good practice compromised as little as possible.

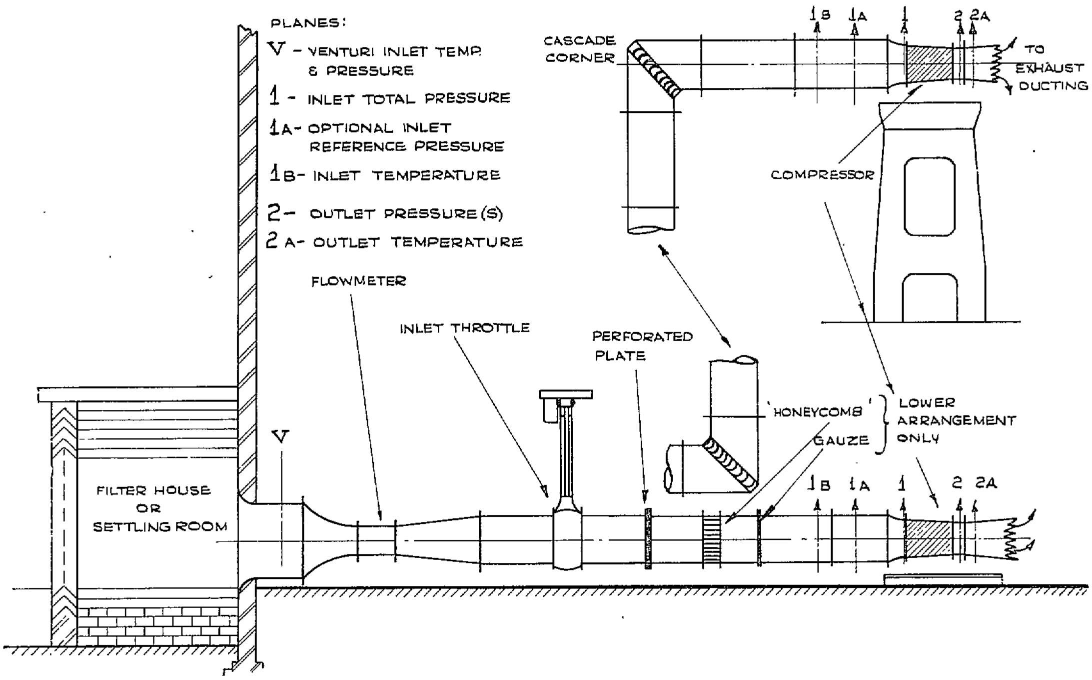

FIG. 1. Inlet ducting arrangement and measuring planes.

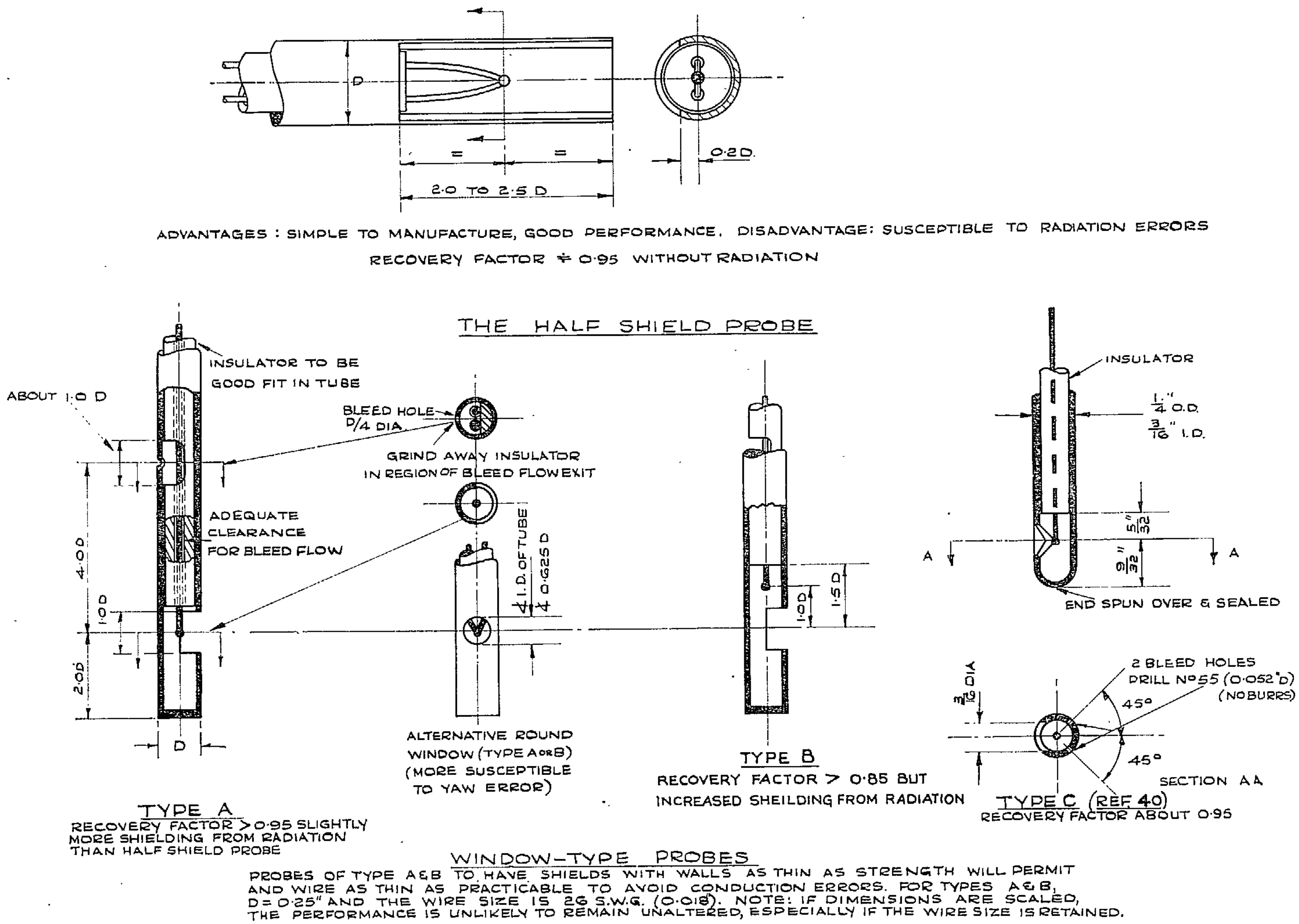

FIG. 2. Thermocouple probes.

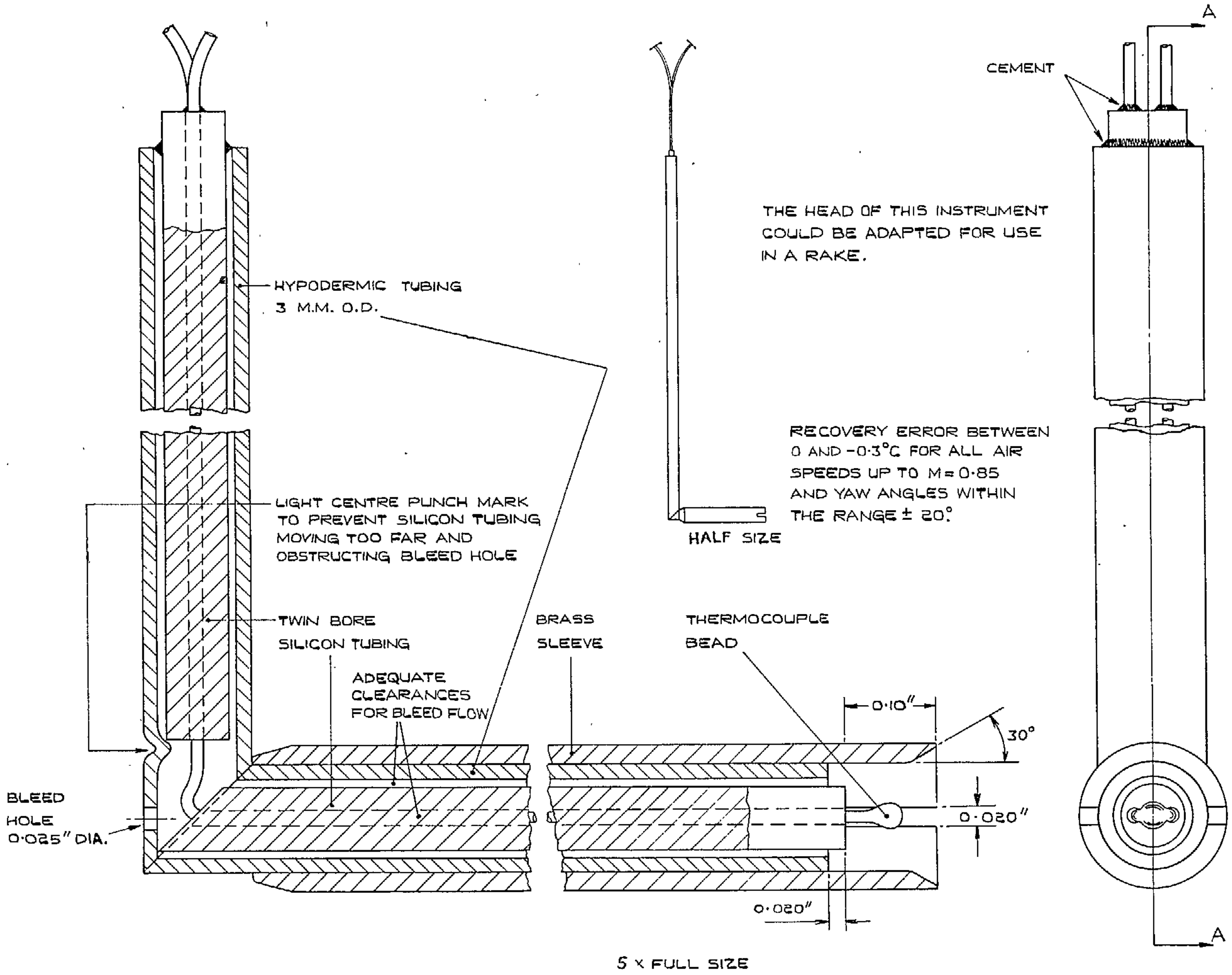

FIG. 3. 'Slit pitot' type thermocouple shield. (Ref. 23—modified version.)

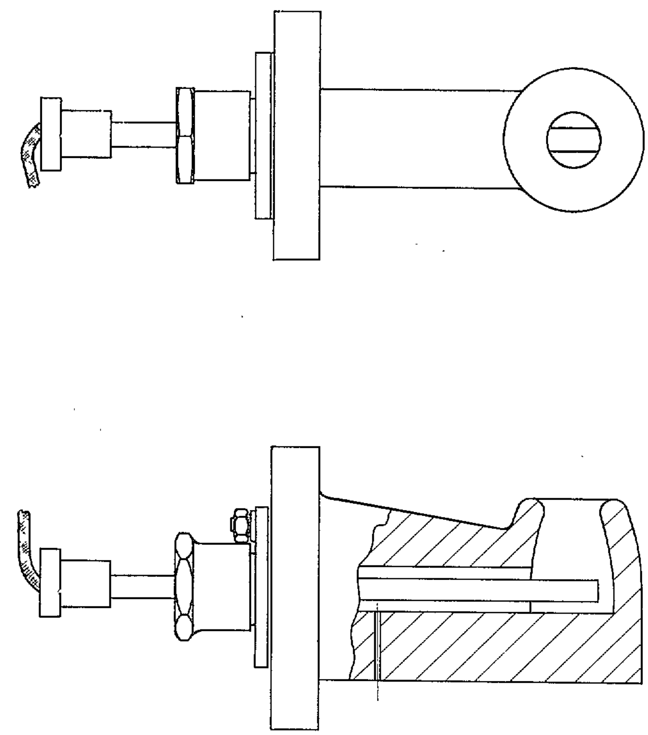

FIG. 4. Stagnation pocket for de Gussa element.

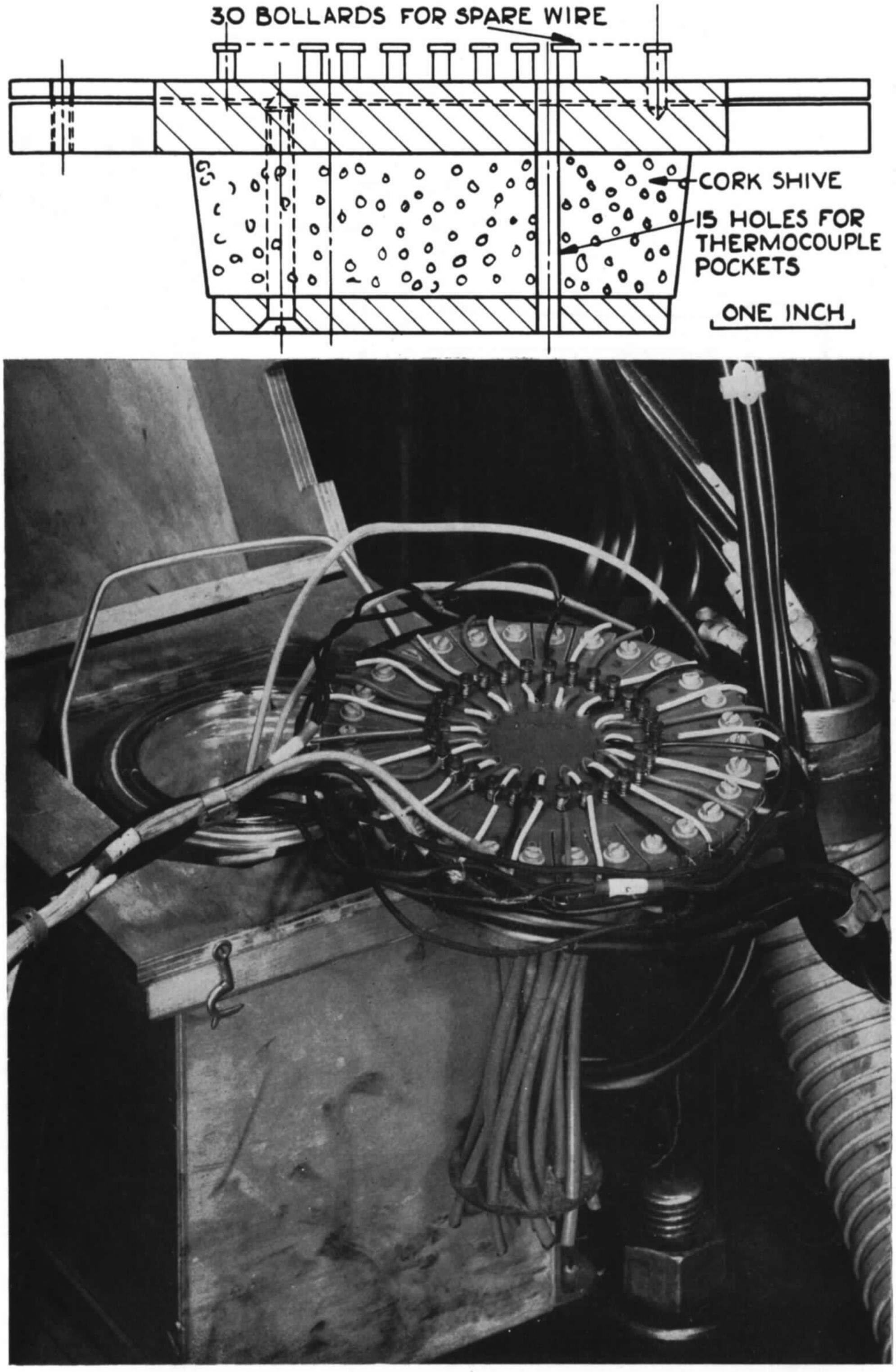

FIG. 5. Thermocouple cold-junction assembly.

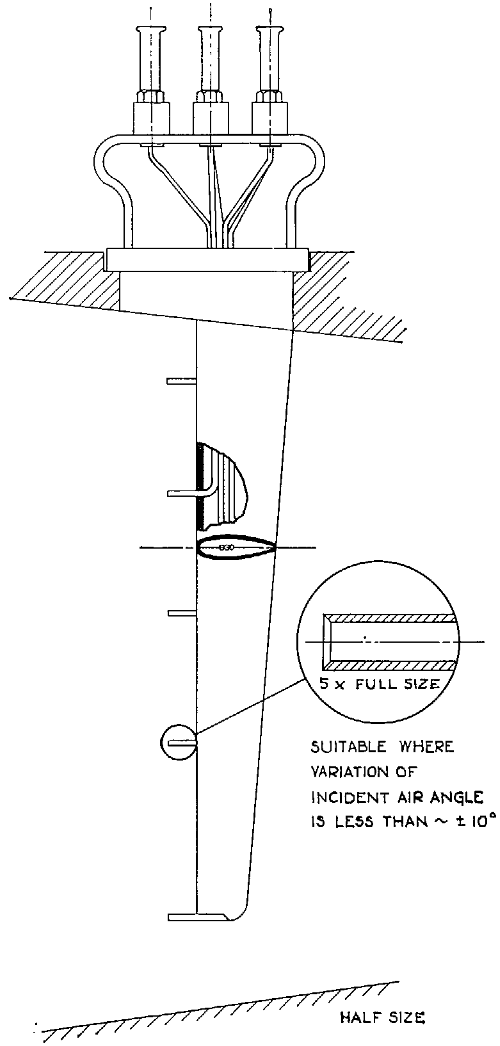

FIG. 6. Typical inlet pitot comb.

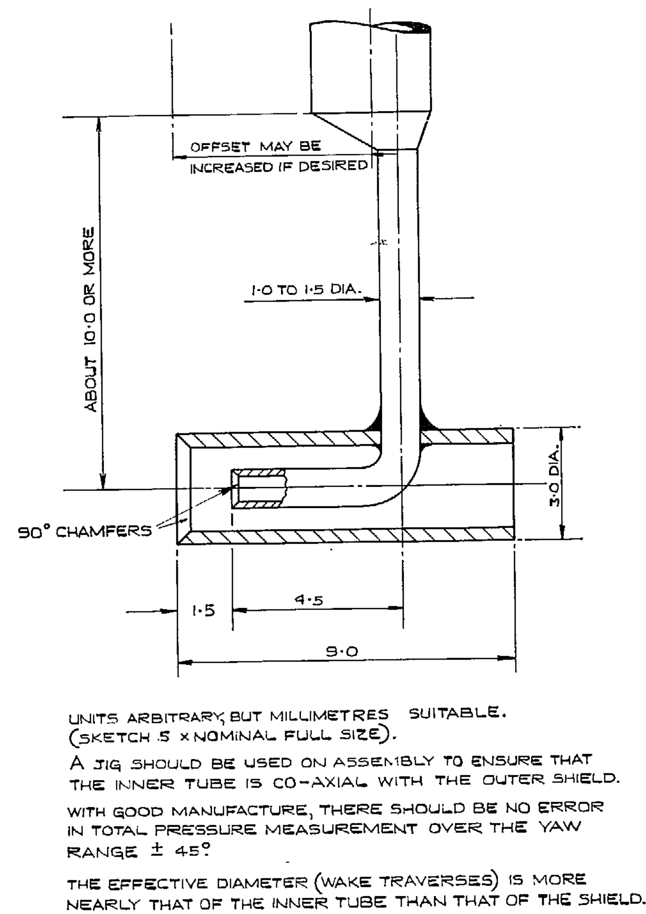

FIG. 7. Simplified 'Kiel' probe.

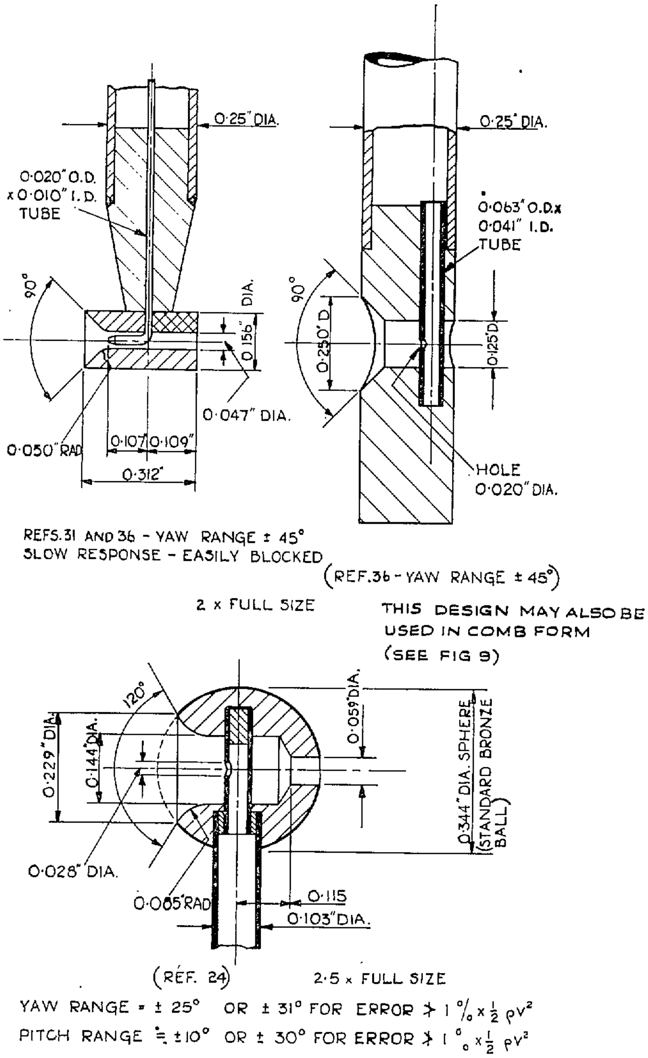

FIG. 8. Other types of simplified 'Kiel' probe.

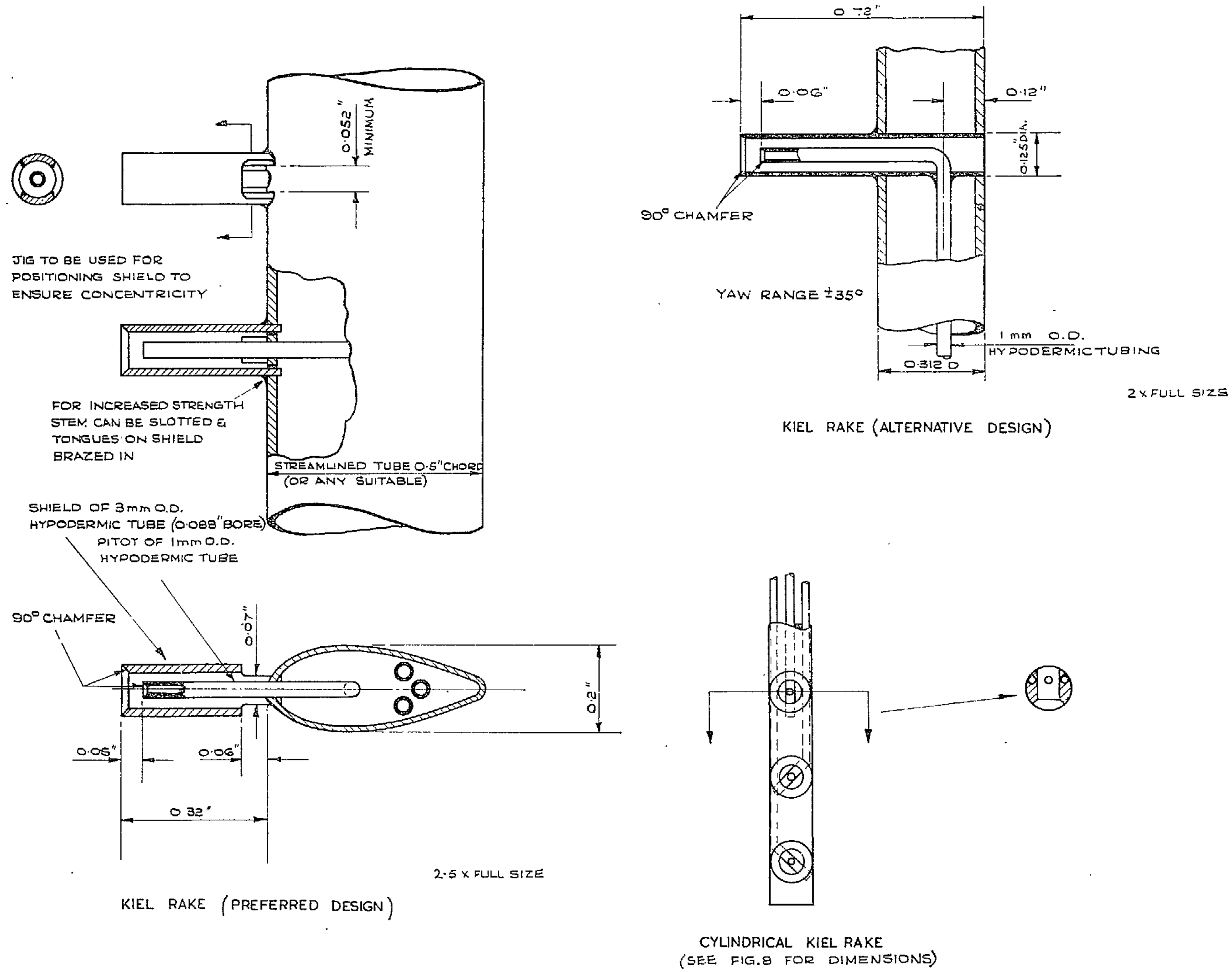

FIG. 9. The 'Kiel' rake.

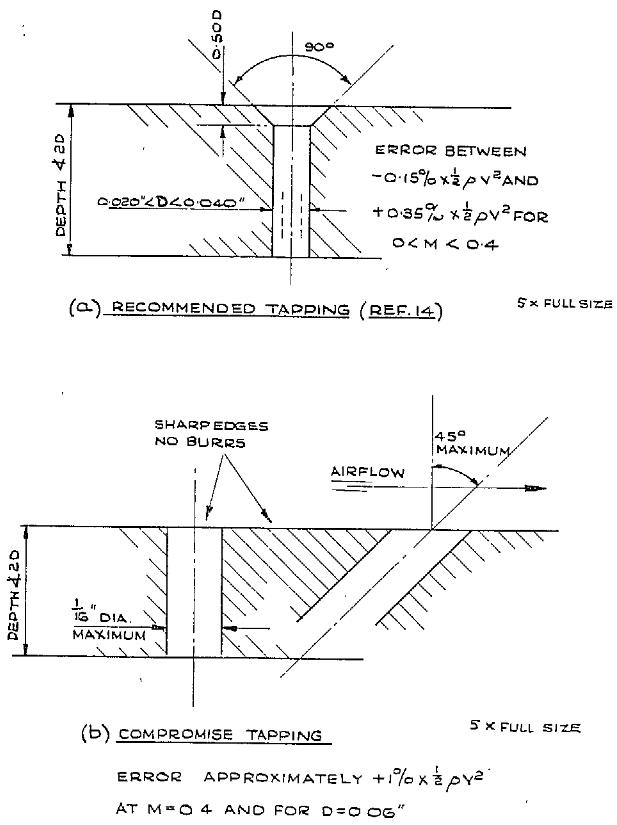

FIG. 10. Static-pressure wall tappings.

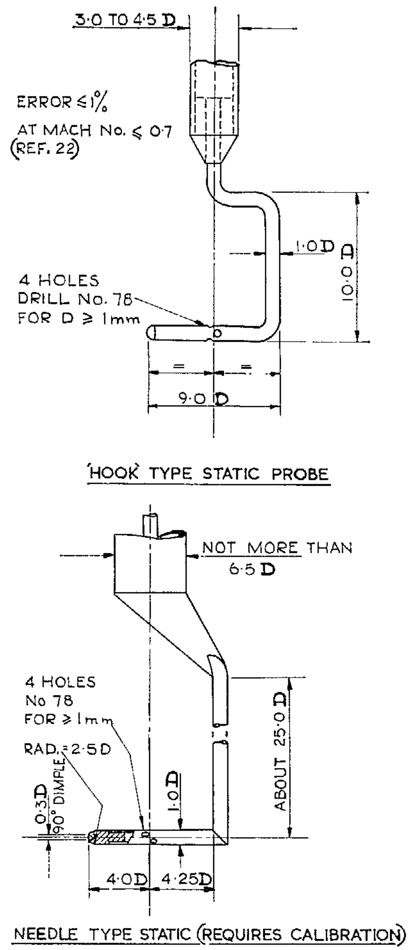

FIG. 11. Static-pressure probes. (Dimensions in arbitrary units—millimetres are suitable.)

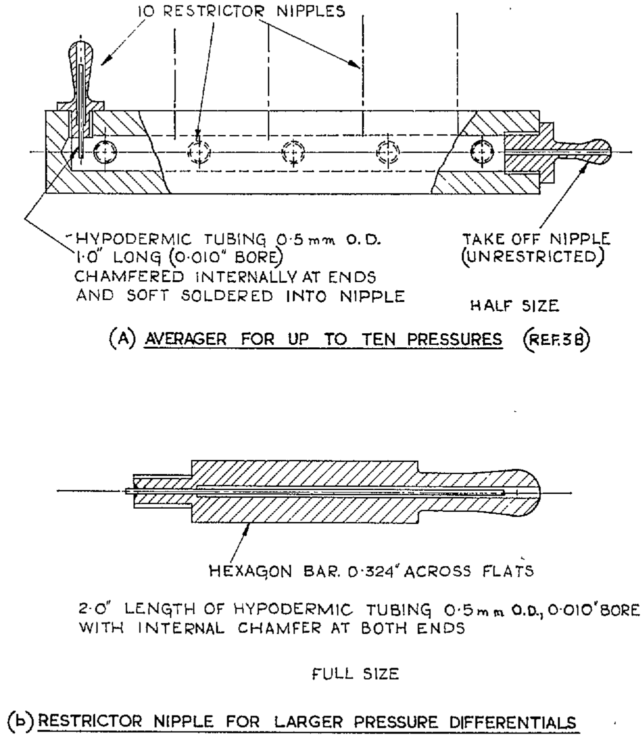

FIG. 12. Pressure averager.

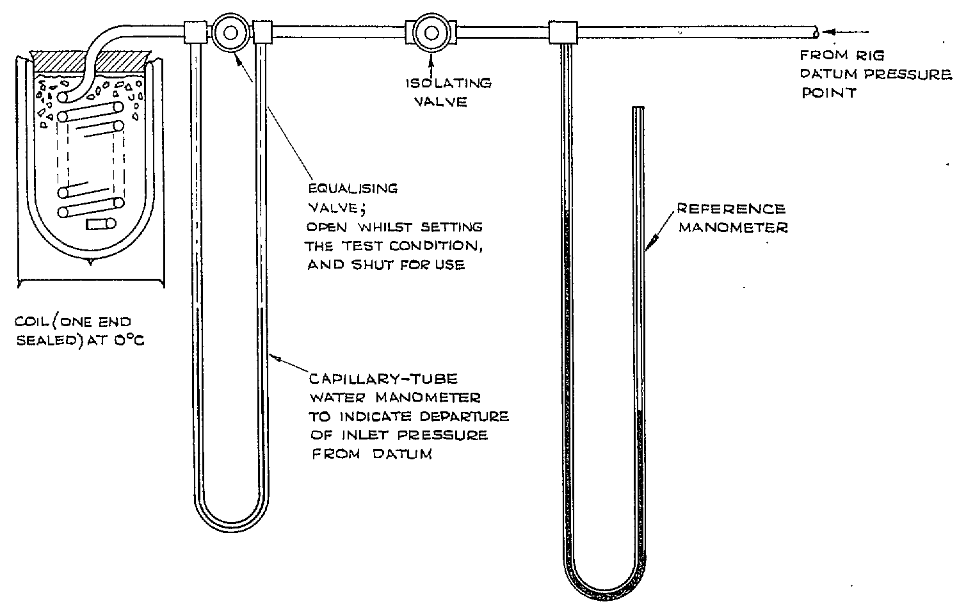

FIG. 13. Inlet datum pressure gauge.

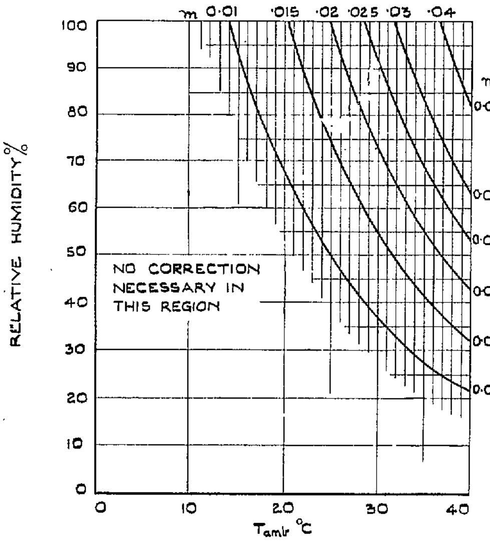

FIG. 14a. Moisture content of air at standard atmospheric pressure for a given ambient temperature and relative humidity.

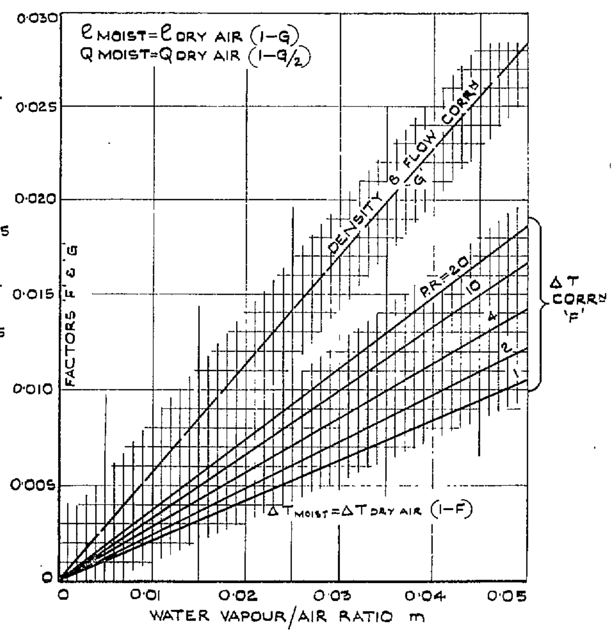

FIG. 14b. Correction factors for moist air. (From Ref. 5.)

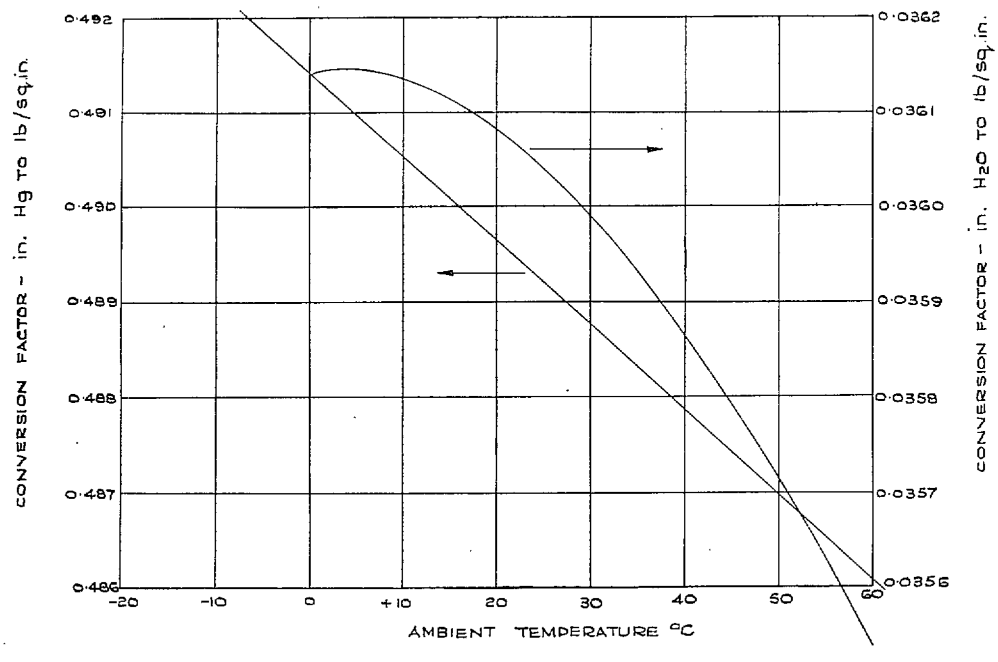

FIG. 15. Conversion factors. (in. H_2O and in. Hg. to lb/sq. in. at local gravity $g = 981 \cdot 183$ cm/sec^2)

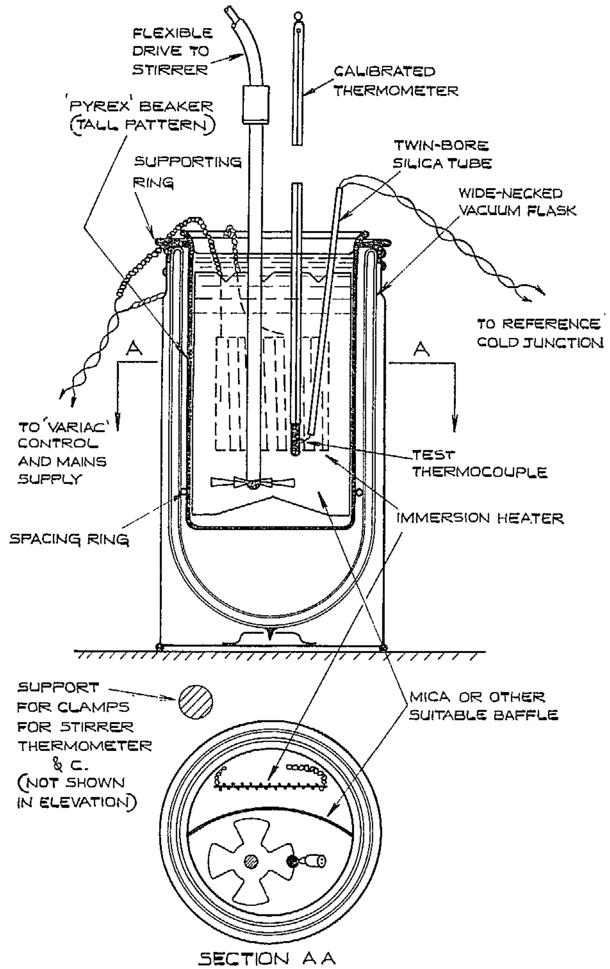

FIG. 16. Simple calibration bath for thermocouples.

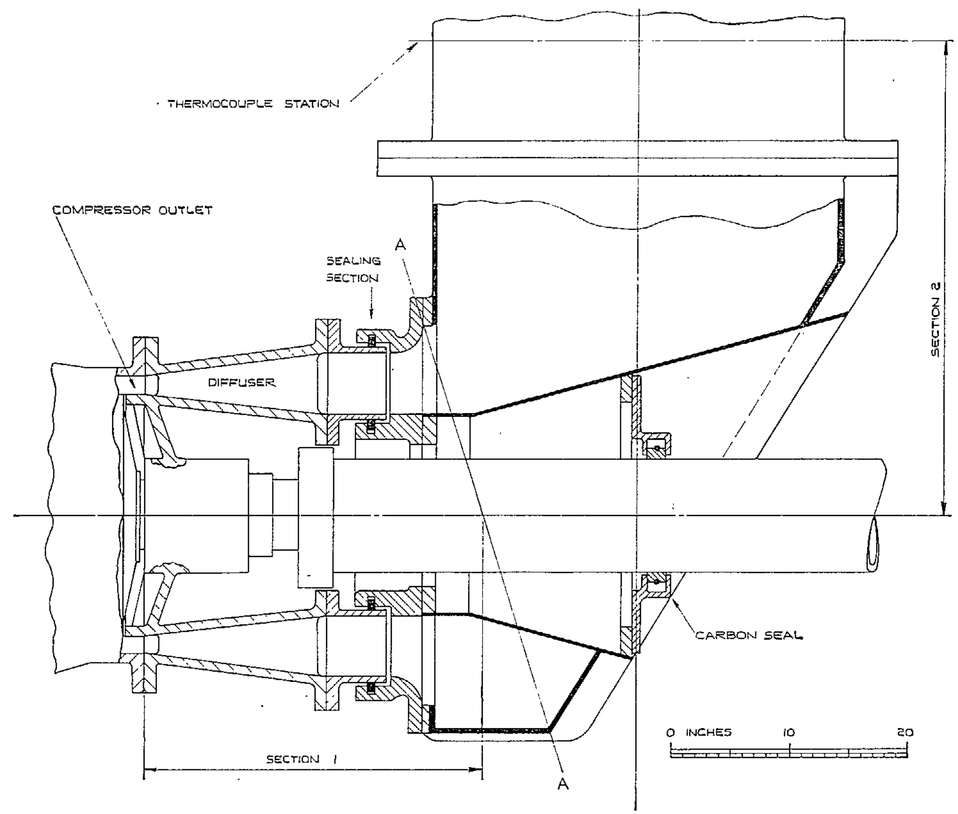

FIG. 17. Compressor outlet ducting.

87836) Wt. 65/1418 K.5 11/63 Hw.

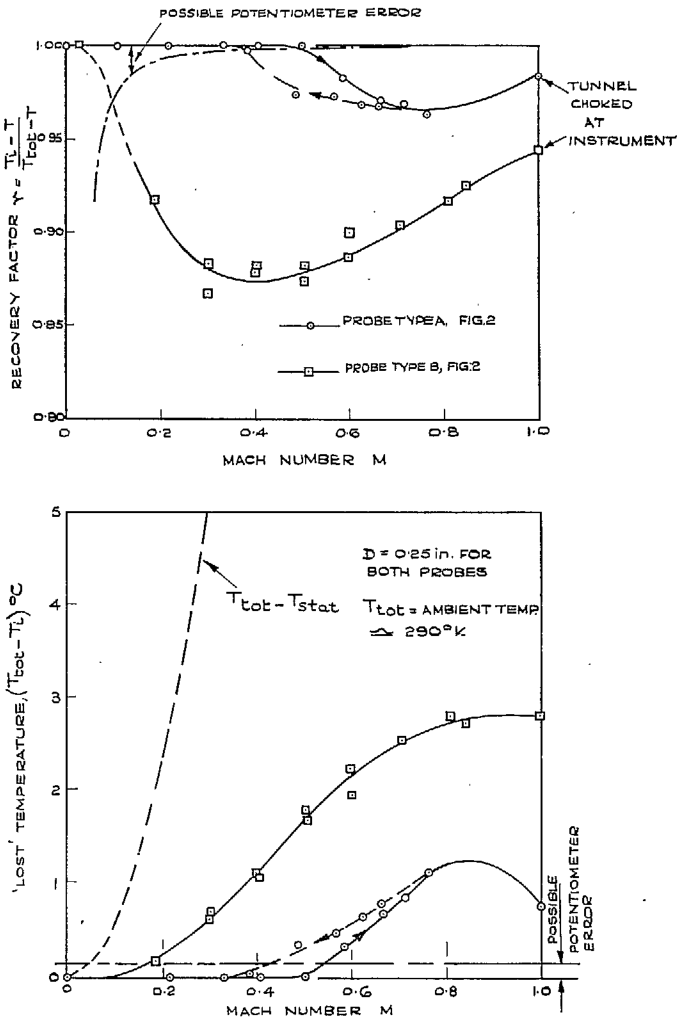

FIG. 18. Calibration curves for two thermocouple probes.

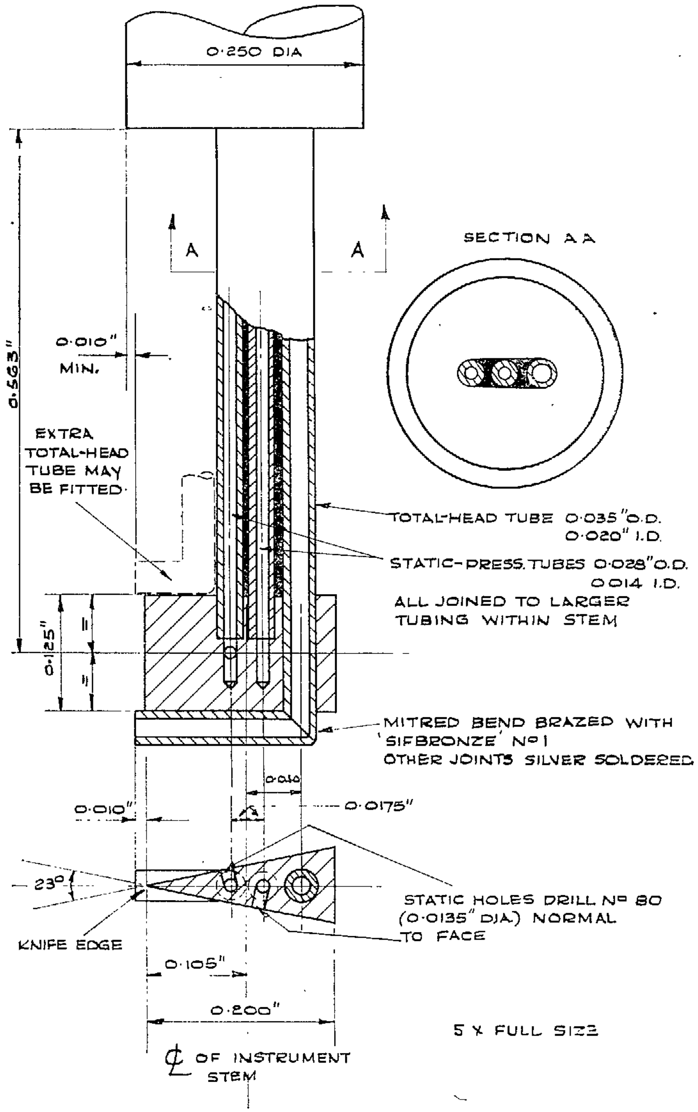

FIG. 19. Combined wedge traversing probe Ref. 37.

Publications of the Aeronautical Research Council

ANNUAL TECHNICAL REPORTS OF THE AERONAUTICAL RESEARCH COUNCIL (BOUND VOLUMES)

1942 Vol. I. Aero and Hydrodynamics, Aerofoils, Airscrews, Engines. 75*s.* (post 2*s.* 9*d.*)
Vol. II. Noise, Parachutes, Stability and Control, Structures, Vibration, Wind Tunnels. 47*s.* 6*d.* (post 2*s.* 3*d.*)

1943 Vol. I. Aerodynamics, Aerofoils, Airscrews. 80*s.* (post 2*s.* 6*d.*)
Vol. II. Engines, Flutter, Materials, Parachutes, Performance, Stability and Control, Structures. 90*s.* (post 2*s.* 9*d.*)

1944 Vol. I. Aero and Hydrodynamics, Aerofoils, Aircraft, Airscrews, Controls. 84*s.* (post 3*s.*)
Vol. II. Flutter and Vibration, Materials, Miscellaneous, Navigation, Parachutes, Performance, Plates and Panels, Stability, Structures, Test Equipment, Wind Tunnels. 84*s.* (post 3*s.*)

1945 Vol. I. Aero and Hydrodynamics, Aerofoils. 130*s.* (post 3*s.* 6*d.*)
Vol. II. Aircraft, Airscrews, Controls. 130*s.* (post 3*s.* 6*d.*)
Vol. III. Flutter and Vibration, Instruments, Miscellaneous, Parachutes, Plates and Panels, Propulsion. 130*s.* (post 3*s.* 3*d.*)
Vol. IV. Stability, Structures, Wind Tunnels, Wind Tunnel Technique. 130*s.* (post 3*s.* 3*d.*)

1946 Vol. I. Accidents, Aerodynamics, Aerofoils and Hydrofoils. 168*s.* (post 3*s.* 9*d.*)
Vol. II. Airscrews, Cabin Cooling, Chemical Hazards, Controls, Flames, Flutter, Helicopters, Instruments and Instrumentation, Interference, Jets, Miscellaneous, Parachutes. 168*s.* (post 3*s.* 3*d.*)
Vol. III. Performance, Propulsion, Seaplanes, Stability, Structures, Wind Tunnels. 168*s.* (post 3*s.* 6*d.*)

1947 Vol. I. Aerodynamics, Aerofoils, Aircraft. 168*s.* (post 3*s.* 9*d.*)
Vol. II. Airscrews and Rotors, Controls, Flutter, Materials, Miscellaneous, Parachutes, Propulsion, Seaplanes, Stability, Structures, Take-off and Landing. 168*s.* (post 3*s.* 9*d.*)

1948 Vol. I. Aerodynamics, Aerofoils, Aircraft, Airscrews, Controls, Flutter and Vibration, Helicopters, Instruments, Propulsion, Seaplane, Stability, Structures, Wind Tunnels. 130*s.* (post 3*s.* 3*d.*)
Vol. II. Aerodynamics, Aerofoils, Aircraft, Airscrews, Controls, Flutter and Vibration, Helicopters, Instruments, Propulsion, Seaplane, Stability, Structures, Wind Tunnels. 110*s.* (post 3*s.* 3*d.*)

Special Volumes

Vol. I. Aero and Hydrodynamics, Aerofoils, Controls, Flutter, Kites, Parachutes, Performance, Propulsion, Stability. 126*s.* (post 3*s.*)
Vol. II. Aero and Hydrodynamics, Aerofoils, Airscrews, Controls, Flutter, Materials, Miscellaneous, Parachutes, Propulsion, Stability, Structures. 147*s.* (post 3*s.*)
Vol. III. Aero and Hydrodynamics, Aerofoils, Airscrews, Controls, Flutter, Kites, Miscellaneous, Parachutes, Propulsion, Seaplanes, Stability, Structures, Test Equipment. 189*s.* (post 3*s.* 9*d.*)

Reviews of the Aeronautical Research Council

1939–48 3*s.* (post 6*d.*)
1949–54 5*s.* (post 5*d.*)

Index to all Reports and Memoranda published in the Annual Technical Reports

1909–1947 R. & M. 2600 (out of print)

Indexes to the Reports and Memoranda of the Aeronautical Research Council

Between Nos. 2351–2449	R. & M. No. 2450	2*s.* (post 3*d.*)
Between Nos. 2451–2549	R. & M. No. 2550	2*s.* 6*d.* (post 3*d.*)
Between Nos. 2551–2649	R. & M. No. 2650	2*s.* 6*d.* (post 3*d.*)
Between Nos. 2651–2749	R. & M. No. 2750	2*s.* 6*d.* (post 3*d.*)
Between Nos. 2751–2849	R. & M. No. 2850	2*s.* 6*d.* (post 3*d.*)
Between Nos. 2851–2949	R. & M. No. 2950	3*s.* (post 3*d.*)
Between Nos. 2951–3049	R. & M. No. 3050	3*s.* 6*d.* (post 3*d.*)
Between Nos. 3051–3149	R. & M. No. 3150	3*s.* 6*d.* (post 3*d.*)

HER MAJESTY'S STATIONERY OFFICE

from the addresses overleaf

R. & M. No. 3337

Printed and published by
HER MAJESTY'S STATIONERY OFFICE

To be purchased from
York House, Kingsway, London W.C.2
423 Oxford Street, London W.1
13A Castle Street, Edinburgh 2
109 St. Mary Street, Cardiff
39 King Street, Manchester 2
50 Fairfax Street, Bristol 1
35 Smallbrook, Ringway, Birmingham 5
80 Chichester Street, Belfast 1
or through any bookseller

Printed in England

R. & M. No. 3337

S.O. Code No. 23-3337