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THE DESIGN AND CONSTRUCTION OF FLOATING-ELEMENT SKIN-FRICTION BALANCES FOR USE AT 50° TO 150°F

By J. C. Westkaemper

Prepared under Contract No. NAS 1-5572 by
DEFENSE RESEARCH LABORATORY
THE UNIVERSITY OF TEXAS AT AUSTIN
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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SUMMARY

This report presents the results of a study of the use of floating-element skin-friction balances for use over a moderate range of temperature and heating rates. The study included an investigation of the buoyancy forces on the floating element and of the use of pressure orifices in the balance case to indicate element misalignment. The case orifices were found to be too large to give an accurate indication of vertical misalignment; the size also precluded their use to determine the buoyancy forces. The orifice size necessary to accurately indicate buoyancy forces appears to be too small to be mechanically practicable. The main influence of temperature variations appears to be a combination of thermal expansion on thermal strains in the mechanical components of the balance design used. These effects were minimized by selective assembly so that the individual contributions of the components cancelled out in the complete balance. This method proved very tedious and time-consuming, particularly for balances with high sensitivities.

INTRODUCTION

The skin-friction balance based on the floating element principle, shown schematically in Fig. 1, has been in use for over 35 years (Refs. 1, 2, 3, and 4). The main advantage of the device is that it measures the actual friction on a small element of the test surface, while other methods deduce the friction indirectly from pressure or temperature measurements. Despite its attractive principle, the balance method has exhibited some problems in application which are important in high-precision experiments. It has long been evident that the element that floats on leaf springs must be mounted flush with the test surface, but only recently has the error caused by misalignment been studied (Ref. 5). Another area of uncertainty is the buoyancy force that results when the balance is used in a non-uniform pressure field. This force occurs when unbalanced pressures act on the edge of the floating element, and it is customarily minimized by making the edge (lip) dimension as small as practicable. The major problem which has evolved in recent balance applications is a sensitivity to temperature changes such as frequently occur in supersonic wind tunnels. The balance employs en electrical position transducer to sense the deflection of the mounting springs when friction is applied to the floating element. The usual transducer is a linear variable differential transformer, known to be temperature sensitive. In addition, temperature changes also introduce mechanical problems in the form of thermal stresses and expansions, which may introduce errors.

The satisfactory performance of the friction balance in the past has perhaps been the result of an absence of temperature changes plus the care of the investigator. It is estimated that the accuracy under those conditions is approximately ± 5 percent, which may be adequate for exploratory studies. More recently, it has become necessary to predict skin-friction drag to approximately 1 percent accuracy, since the success or failure of some vehicle designs may depend on a small change in the skin-friction component of total drag (Refs. 6 and 7). Several balances, which performed quite satisfactorily for the tests reported in Ref. 8, were loaned to the NASA-Langley Research Center where they were used for several investigations. As noted in Refs. 6 and 7, their performance did not satisfy the more stringent requirements of today. This ultimately led to the present work, the development of balances of improved accuracy, and a study of the use of pressure orifices in the case lip to determine buoyancy forces.

BALANCE DESIGN

The design specifications for the six balances are listed in Table I. As noted earlier, the general accuracy of the balances available prior to the present work was approximately 5 percent. The specifications in Table I require an accuracy of from 1/2 to 2 percent in the various aspects of balance performance. Although these requirements are considerably more stringent than those attained by the earlier balances, it was decided that an improvement in design was more practicable than an entirely new design. A major factor in this decision was the operating temperature range. This range presented no problems in physical deterioration of balance components, and it appeared that the electrical performance of the position transducer could be made acceptable over the range. Bench tests of the earlier balances at room temperatures showed that accuracies of 1/2 percent were consistently obtained. Thus, the primary problem in the present work was the influence of the specified ranges of operating temperature and heat transfer.

The influence of temperature on balance performance may be placed in two categories. The first, is the influence on the electrical characteristics, of the position transducer, and the second is the effect represented by thermal expansion and thermally induced strains in the various mechanical components of the balances. These will be discussed separately in some detail.

The effect of temperature on the linear variable differential transducer is primarily a change in the resistive component of the impedance of the primary. Although there are several ways to treat this problem, the simplest approach appears to be the use of an excitation frequency that is high enough to make the inductive

TABLE I

BALANCE DESIGN SPECIFICATIONS

Shear force range, lb	0.002 0.005 0.012
Operating temperature range, °F	50 to 150
Non-linearity, percent full scale	± 1
Zero drift (8 hr), percent full scale	± 1/2
Thermal zero shift, percent full scale	± 1
Repeatability and agreement among balances from wind tunnel tests, percent full scale	± 2
Heat transfer $(T_{aw} - T)$, °F	± 25

impedance much larger than the resistive component. In this way, the change in resistance becomes a negligible part of the total impedance. Thus, the current remains constant with temperature when a constant supply voltage is applied to the primary. A frequency of 20 kHz was selected to permit use of a commercial carrier amplifier using that frequency. A temperature change also results in a change in phase between the primary and secondary voltages. Therefore, carrier amplifiers operating on a phase-sensing principle may be suitable for exciting the primary, but will not necessarily be free of temperature effects if used to measure the output by phase-sensing. In the present study the excitation was supplied by an audio oscillator and amplifier, selected to be relatively constant in amplitude. The transducer output voltage was measured using an rms voltmeter having a dc proportional output, which was indicated using a dc digital voltmeter. In this manner, the system was made insensitive to the phase between input and output voltages.

The above effect of temperature on the transducer pertains to the gauge factor, i.e. the slope of the curve of output voltage vs displacement. Another possible effect is the change in reading when the transducer core is fixed with respect to the transformer. This second effect is equivalent to a zero shift with temperature. An estimate of the thermal zero shift of the transducer itself was obtained by placing the core in the transformer at the approximate zero position. The threaded shaft which normally supports the core (see Fig.1) was not used in this case. The transformer output under these conditions was monitored while the temperature was varied by 150°F. The zero shift was found to be approximately 1 percent of the reading obtained when the core is traversed a distance equivalent to full scale travel on an assembled balance. Although this test was somewhat crude, the results did indicate that the effect of temperature changes on the transducer zero were comparatively small. It is perhaps appropriate to note at this point that in terms of skin-friction balance displacement 1 percent represents a travel of 45 μin. The increase in length of the core resulting from a 100°F temperature change is approximately four times that amount.

It has previously been noted that the floating element of the skin-friction balance must be very carefully aligned in the test surface. If the element protrudes above, or is recessed into the test surface, erroneous skin-friction readings will result. Since the balances in the present work are intended for operation over a range of temperatures, it was necessary to consider the effects of thermal expansion on the alignment of the floating element with the mounting case. The floating elements were constructed of aluminum to minimize the sprung weight. The flexure springs were made of Elinvar Extra material for reasons discussed later. The supporting members at the balance base were constructed of stainless steel to preclude corrosion. Thus, the balance proper was constructed of three different materials. The combined thermal expansion of the structural elements made from these three materials was determined. A material was then selected for the case, such that its increase in length with temperature would be matched by the increase in length of the balance and its supporting base.

In addition to the effect of temperature on floating-element alignment, there exists the problem of the effect of thermal expansion, in particular thermal strains on balance performance. Any motion of the transformer core which results from thermal effects will appear as a spurious drag indication. It is therefore necessary that such thermal displacements be eliminated. This was found to be the major problem in obtaining satisfactory balance performance. As previously noted, a displacement of 45 μin. represents 1 percent of the full scale travel for which the balances were designed. In the case of the most sensitive balance, i.e. the two having a full scale friction force of 2 mlb, the force necessary to obtain a 1 percent displacement is 20 μlb. Thus, it is evident that the balances are extremely sensitive, and quite small thermal strains can result in large errors in the balance readings. Because of small variations in fabrication and assembly, each balance performed with slight differences from the others. In particular, the variations in flatness of the flexures, resulting from the heat-treating process, proved to be the major source of thermal zero shift. Although considerable care was taken in fabricating and heat treating, some degree of distortion was present in all of the flexures. During assembly of the balances, the least distorted flexures were selected. However, even these resulted in some thermal zero shift. The only practicable solution found involved selectively orienting and interchanging flexures until a pair was obtained which cancelled overall distortions.

Since the transducer output is directly proportional to the input signal, it is evident that variations in input will appear as corresponding variations in the output. Thus if the supply voltage varies, a corresponding shift in the output reading is observed. The long-term stability of the oscillator and amplifier used in the present work was monitored. After a warm-up period of approximately one-half day, the amplitude was found to be stable to within less than 1/2 percent. During these tests, the amplifier was connected to a position transducer to stimulate the normal load of operation. Similar tests were made in which the output of various skin-friction balances was monitored over a period of several days. During these tests, the balances were maintained at room temperature. The results indicated that the long term zero shift of the balances was primarily determined by the stability of the power supply to the transducer. In order to eliminate transients, associated with warm-up of the power supply and read out instrumentation, this equipment was kept turned on continuously. Where practicable, a balance was connected to the power supply to avoid operating it in the unloaded condition. Each time a new balance was attached to the power supply, a warm-up period of the balance itself of 2 to 4 hours was allowed before testing began.

Because of the low level of the output signal from the skin-friction balance, some problems were encountered with shielding. In order to avoid stray pickup, it was necessary to carefully shield all cables in the circuit. In addition, the linear variable differential transformer used as a position transducer in the friction balances was itself shielded. In the final test set-up, the stray voltage level was less than 0.1 percent of the output equivalent to full scale deflection of the balance.

BALANCE CONSTRUCTION

The earlier skin-friction balances employed by Defense Research Laboratory used beryllium-copper flexures. This material was found satisfactory in terms of its low hysteresis and ease of fabrication, as well as being corrosion resistant. The modulus of elasticity of this material is temperature sensitive, however. For a temperature change of 100°F, a reduction in modulus of elasticity of slightly less than 2 percent will occur. Thus, the beryllium copper is not as attractive as a flexure material when balances are operated over varying temperature conditions. For this reason it was decided to employ a material for flexures that had a constant modulus of elasticity with varying temperature. A material which appeared suitable was Elinvar Extra, manufactured by the Hamilton Watch Company. This material has as its primary constituents iron and nickel, hence it is comparatively corrosion resistant. A disadvantage, which became evident after balances using these flexures were assembled, was the low thermal conductivity of the material when compared to beryllium copper. The flexures for the two mlb balances were 0.003-in. thick. The combination of the high sensitivity of these balances and the thin flexures resulted in an extreme sensitivity to thermal strains. This resulted in thermal zero shifts of 20-50 percent of the full scale balance output. Attempts to minimize this shift by selective pairing and orientation of the flexures did not result in zero shifts of less than 5 percent. Therefore, for the two balances having a full scale friction force of 2 mlb, it was necessary to employ beryllium-copper flexures. The results with the beryllium copper were satisfactory, presumably because of its much higher thermal conductivity. The higher conductivity would be expected to reduce temperature gradients and thus thermal strains which might occur in the flexures. The two balances employing beryllium-copper flexures were fitted with cases made from 303 stainless steel, in order to match the overall changes in length of the balance and the case. The balances using the Elinvar flexures were fitted with cases made from 431 stainless steel in order to match the thermal expansion of the Elinvar assembly.

The design specifications call for the inclusion of a thermocouple in the floating element. Since the floating element is spring mounted, the thermocouple wires must be treated as additional springs attached to the element. Thus, it is necessary that the force required to deflect the thermocouple leads be kept small in order to avoid altering the balance spring constant. In addition, the force required to deflect the thermocouple wires must be highly repeatable in order to avoid changes in balance gauge factor. This problem was approached by using a thermocouple made from 0.001-in. diameter iron and constantan wires. The thermocouple bead was staked in a hole in the lower surface of the floating element and the wires were strung somewhat loosely to one side of the transformer holder, as seen in Fig. 2. In this manner, the effect of the wires on the spring constant of the balances was made negligible. The very small diameter of the wires presented some handling problem, however, and extreme care was necessary in making the thermocouple installation.

The specifications also called for an iron constantan thermocouple to monitor the case temperature. This thermocouple was made of teflon-insulated thermocouple wire, spot-welded to the inside of the case as seen in Fig. 3. The thermocouple wires were brought through an opening in the case in such a way that the balance could be removed and installed in the case without disturbing the case thermocouple leads. The spot-weld was covered with a small amount of epoxy glue for insulating purposes and to protect the thermocouple from damage. The case opening through which the thermocouple leads passed was made airtight by filling with epoxy.

In order to minimize the possibility of misalignment of the floating element in the case, due to slippage in connections in assembling the various parts of the balance, the flexures were spot-welded in place. In addition, the floating element was mounted and then pinned in position such that slippage is highly improbable. All the connections involved in mounting and positioning the transducer and its core were also fixed after final adjustment was made by means of epoxy cement.

Since the floating element is a spring-mounted mass, it oscillates readily when subjected to vibrations or to sudden loads. The output from the balance while the floating element is vibrating is not useful for determining drag. Thus, it has been found necessary in the past to provide for damping the vibrations that occur in normal tunnel operation. This was accomplished by means of the arrangment shown in Fig. 4. A small disk was fixed to the lower side of the floating element and a similar adjustable disk was attached to the transformer mount. These two disks formed a flat gap, the thickness of which was adjustable. A small amount of damping fluid was installed in this gap to eliminate the vibration problem. A silicone damping fluid, manufactured for use in Gray phonograph record changers, was used since it has proved satisfactory in earlier applications.

Provisions were made in the balance design to permit adjustment of the position of the transformer with respect to the floating element. This was accomplished by means of the wedge arrangement shown in Fig. 5. When the final adjustment was completed, the transformer housing was cemented to the balance base as noted previously. Thus, the position of the transformer is not adjustable without disassembly of the balance.

The provisions for positioning the balance with respect to the case are shown in Figs. 3 and 4. The rotational alignment of the disk is accomplished by means of adjustment in the screws that attach the balance to the case. The translational adjustment of the balance to position the floating element properly in the gap is accomplished by means of four adjusting screws, accessable from the outside of the balance. The four adjusting screws shown in Fig. 4 actuate wedges which bare against a conical surface on the bottom of the balance. This arrangement permits adjusting the position of the disk in the opening of the case. Thus, realignment of the floating element may be readily accomplished should the need arise without removing the balance from the case.

The influence of the size of the gap surrounding the floating element is not precisely known. However, it is evident that a small gap will create less flow disturbance than a large one. Accordingly the present balances were designed with a diametral clearance of 0.005-in. Since the element must "float" free in the case, precise alignment was necessary. The element was positioned so that it had a very slight clearance from the forward edge of the case opening. The assembled balances were mounted with the element horizontal during alignment, since the force of gravity on the sprung components will otherwise cause large deflections of the springs. The clearance between the element and case was adjusted to approximately 5 percent of full scale travel.

The normal use of a balance is such that there may be a pressure differential acting on the balance. If the balance were installed in a wind tunnel wall, for example, the tunnel static pressure may greatly differ from the ambient pressure outside the tunnel. Such a pressure differential will result in flow through the balance case and the gap around the floating element. Since this flow will disturb the free-stream flow and cause errors in the skin-friction measurements, the balance must be sealed to preclude any "leakage" flow. This was accomplished by using the "O" ring, shown in Fig. 3, and a sealed cap through which the electric leads were routed. The leads were sealed with epoxy-type cement where they passed through the cap. There was sufficient slack left in the leads to permit partial removal of the seal cap without breaking the lead-to-cap seals. To prevent leakage flow along the external surfaces of the case, a seal groove was included, as shown in Fig. 3.

The floating element was positioned flush with the test surface of the case using a linear variable differential transformer to measure the alignment. In this manner it was possible to align the element parallel to the case surface with a deviation of approximately 100 μin. Generally the deviation was that of a recessed element, since it is noted in Ref. 5 that a recess causes less error than an equal protrusion. The tolerance in element flatness was easily observed during alignment and was considered in making the adjustments.

To calculate shearing stresses and buoyancy forces, the diameters and edge thicknesses of the elements are required. These are tabulated in Table II

TABLE II

MEASURED FLOATING-ELEMENT DIMENSIONS

BALANCE	RANGE lb 1000	DIAMETER in.	LIP THICKNESS in.
2A	2	1.000	0.003
2B	2	1.000	0.003
5A	5	1.001	0.004
5B	5	1.001	0.004
12A	12	1.000	0.002
12B	12	1.000	0.004

PERFORMANCE TESTS

The performance tests on the six balances were made using a Hewlett-Packard Model 204B oscillator and Model 465A amplifier to supply input power. The balance output was indicated using a Model 3400A rms voltmeter and Model 3439A digital voltmeter from the same manufacturer. The rms voltmeter served as an ac to dc converter by using its proportional dc output feature. The linear variable differential transformer used in the balances was the Schaevitz Engineering Model 010-MS-LT which requires 180 mW of power. The stable-amplitude oscillator produced 10 mW, hence the need for the amplifier. The balance calibrations were made using a low-friction pulley and weights. The output of the position transducer used in the balances is highly linear when calibrated at a constant temperature.

The major mechanical component of the balances that can influence linearity is the spring system. Since the deflection of the springs is quite small, no hysteresis effect occurs in the springs. Thus, the basic design of the balance is such as to result in excellent linearity. This was confirmed in all cases by the bench tests, where the linearity of the balances at a fixed temperature was well within the specifications of Table I. The non-linearity appeared to be determined more by the care in calibration than by the balance itself. When the input voltage to the balance was kept constant, and care was taken in the calibration procedure, the non-linearity was less than one-quarter of 1 percent.

The zero drift of the balances, when maintained at a constant (room) temperature, appeared to be determined primarily by the stability of the input voltage to the transformer. After an initial warm-up period, the zero drift became quite small. In several instances, the balance zero reading was monitored for two and three day periods with less than a one-quarter percent variation. It should be noted that the initial warm-up includes the heating of the position transducer itself as well as the instrumentation used in conjunction with the balances. Although the power supplied to the transformer is small, there is some change in temperature of the transformer due to the application of power, therefore, it is necessary to include a period for warm-up of the balance itself. The balance warm-up period appeared to be somewhat longer than that required for the power generating equipment to become stable.

As previously noted, the thermal zero shift resulting in change of operating temperature was the most difficult problem in the balance design. The thermal expansions and thermal strains in the mechanical components of the balance were not predictable, hence the need for a trial and error approach to this problem. The balances were very sensitive to non-uniform heating, particularly with respect to the flexures. Thus in making tests to determine the thermal zero shift, considerable care was necessary to ensure that the balance was maintained at a uniform temperature while the temperature changes were made. This was accomplished primarily by using a very low velocity supply of heated air, and changing the heating air temperature slowly. There was from one-quarter to one-half percent scatter in repeating the thermal zero test on any given balance. Under those conditions, it was possible to obtain a thermal zero shift that did not exceed approximately ±1 percent for all but one balance. For balance nb. 12a, the shift was +2 percent and -1 percent. The zero shift was generally linear with temperature within the accuracy of the tests. The tests were made using a temperature range of room to 180°F. This was done to avoid the use of refrigeration equipment to obtain the 50°F figure noted in Table I. The thermal zero shift results are considered applicable to any change in temperature of 100°F which will not physically damage the balance. In this connection, it should be noted that the maximum temperature, which will not cause damage to the balances, is the limitation of the epoxy cement used to secure the adjusting screws in the balance assembly. It is estimated that this cement is capable of withstanding temperatures of 250-300°F without serious damage. The remaining components of the balance including the transformer are designed to withstand at least 400°F.

The effect of heat transfer on the balances was determined by heating the floating element to 25°F above the temperature of the case and balance base. Similarly, the temperature differential in the opposite direction was obtained by heating the balance base and case to a temperature 25°F above the floating element temperature. During these tests, the change in the zero reading of the balance was generally within ±1 percent. This particular test result is believed to be strongly influenced by the temperature gradients introduced in the balance. This aspect is related to the influence of non-uniform heating mentioned earlier. Thus, the main influence of heat transfer during operation of the balances is believed to be the resulting thermal gradients in the balance and the associated expansion and strains which are known to cause zero shifts.

The wind tunnel repeatability and mutual agreement tests were made using a subsonic wind tunnel in order to test two balances simultaneously. The repeatability of each individual balance was very good, generally within one-quarter percent. The mutual agreement between two balances of the same shear force range was also good, being in general within one-half percent. For these tests the two primary windings of the balance transformers were connected in series across the power supply, and a switch was used to alternately read the output of the two balances using the same rms voltmeter and the dc digital voltmeter used in the bench tests.

Two operational aspects of the tunnel tests warrent some discussion. Since the balances include a damping mechanism to prevent vibration, the response to changes in load is correspondingly slow. This is particularly true of the most sensitive balance, where the spring constant is quite small and the restoring forces correspondingly weak. Thus in calibrations on the bench or in the tunnel, it is necessary to tap or vibrate the balances to minimize any damping effect on readings. During operation of the tunnel, the tunnel vibration serves this purpose. Another problem which occurs in the tunnel is in the orientation of the balances. When the balance is mounted in a horizontal surface, it is necessary that the surface be very nearly level. Any inclination of the surface in the drag direction will result in a displacement of the floating element position due to gravity forces acting on the sprung weight. This can be troublesome if the inclination is such as to deflect the balance in the upstream direction, since in this case the floating element may easily touch the case lip.

With respect to both calibration and operation of the balances, it is necessary to insure that foreign matter does not become lodged in the gap around the floating element. This is particularly critical in the upstream region of the gap, since that clearance of the floating element with respect to the case is quite small.

Poor repeatability of the zero reading is generally an indication of either foreign matter in the gap, or a contact between the element and case. The method found most practicable for cleaning the gap is the use of very thin tissue paper, which can be inserted in the gap and drawn around the element to remove the impediments. Care must be taken, however, to insure that the paper thickness is not so large as to bind in the gap. Care should also be taken not to insert the paper too deeply as it is possible to disturb the disk thermocouple wires, which are quite fragile. The position of the disk thermocouple may be deduced from the fact that it is located on the opposite side from the case thermocouple.

PRESSURE FORCES

Tests were made in an effort to determine the buoyancy forces on the floating element caused by slight misalignment of the element with respect to the balance case. A balance case was fabricated having a lip thickness of 0.025 in. Six lip pressure orifices were installed, as shown in Fig. 6, using 0.020 in. tubing cemented in slots in the case test surface. These orifices had an opening of 0.010 in. with the center line of the opening at approximately 0.010 in. below the test surface. The orifices were uniformly spaced at 60 degree increments around the case, starting at the most upstream point of the case. Thus, one orifice was opposite the leading edge and one was opposite the trailing edge of the movable element. The element diameter was approximately 1.0 in., and a difference in diameters between the case and element of 0.006 in. was used. The element lip thickness was 0.005 in. The element was mounted on a traversing mechanism which permitted motion perpendicular to the case test surface. The case was mounted flush with the test-section floor of a 2 x 2 in. variable Mach Number wind tunnel. Case lip pressures were measured with the element position ranging from a 0.001 in. recess (below the test surface) to a 0.001 in. protrusion. These limits were selected from Ref. 5 as covering the maximum tolerable error in balance accuracy due to misalignment. The tests were made at several Mach and Reynolds Numbers. In general, the pressure variation around the case lip, referenced to the most upstream case lip orifice, was less than 1 in. of water. The pressure distribution around the periphery of the case showed no consistent pattern other than a rough proportionality to misalignment. The pressure changes observed as the amount of misalignment was varied were too small to use as a practicable indicator of the degree of misalignment.

The case lip pressure data were used to calculate the buoyancy forces on the element by assuming that they reflected the pressures on the element lip. The results were erratic in that there was no consistent trend with misalignment and the repeatability of the buoyancy forces was poor. In general the indicated buoyancy forces were less than those reported in Ref. 5. The poor results are attributed to the difficulty in making precise measurements of very small pressure changes.

An investigation was made of the effect of translation of the floating element on the balance output reading. An operating balance was constructed and mounted on a traversing mechanism which permitted moving the balance in the thrust-drag direction while maintaining the element flush in the test surface. Tests were made at several Mach and Reynolds Numbers during which the balance was traversed across the full range of element positions. The variation in drag reading with element position ranged up to ±3 percent of the average reading. There was a consistently positive change in the balance output during a traverse for some test conditions and negative for others. Figure 7 shows the trend which was generally observed. Case lip pressures were also recorded during the traverses, but it was not possible to correlate the changes in drag reading with buoyancy forces calculated from the lip pressures. There was some indication that the drag reading changes were the result of buoyancy forces, however.

A final set of tests were made in which a dummy floating element was constructed to incorporate six pressure orifices on the element lip (Fig. 6). These were 0.0035 in. in diameter and were located opposite the case lip orifices. Four pressure orifices were also installed inside the balance case, well below the dummy element, in order to measure the balance "cavity" pressure. A Mach Number of 2.2 was selected to give a minimum pressure gradient on the tunnel floor near the test station. The cavity pressure, the element and case lip pressures, and two floor static pressures were measured for several positions of the dummy element in its annular gap. The four cavity pressure measurements were always identical, although there was some change in level with element position. This was consistent with the generally accepted theory that there are no pressure gradients inside the case except in the immediate vicinity of the gap between the element and case lips. The cavity pressure was also nearly equal to the static pressure on the tunnel floor just ahead of the balance.

Figures 8, 9, and 10 show typical comparisons of lip pressures on the element and the case. There was a favorable pressure gradient of approximately 2.5 in. H_2O over 2 in. length spanning the balance location in the tunnel floor. The element edge thickness was approximately 0.006 in., and the case lip thickness was approximately 0.020 in. The case lip orifice diameters were approximately 0.010 in., with their centerlines about 0.010 in. below the test surface. The element had a slight bevel at Position 3, resulting in erratic readings, which should be disregarded. The gap (difference in case and element diameters) was 0.008 in. The pressures were measured using oil-filled U-tubes. All pressures were referenced to a static-pressure orifice on the tunnel floor at a point 1.5 in. ahead of the center of the dummy "floating" element.

Interpretation of Figs. 8 to 10 may be aided by noting that a linear pressure gradient of 1 in. of water across the 1 in. diameter floating element will induce a buoyancy force of 0.17 millipounds. A gradient of 3/4 in. H_2O would result in an error of 1 percent in the 12 millipound balances of Table 1. Figure 8 shows the results obtained with the element in the fully-rearward position; i.e., touching the aft lip of the case. The pressure differences (ΔP) are quite small at all orifices, and the repeatability errors are similar in magnitude to the pressure differences. Figure 9 shows the results when the element was centered in the gap; i.e., at the mid-point of travel. Again the values of ΔP are small and similar in magnitude to the repeatability errors. Despite this, it is evident that the case lip pressures do not accurately indicate the actual pressure on the element lip. The case lip pressures apparently reflect the cavity pressure to some extent. This is also evident in Fig. 10 which shows data for three element positions between those of Figs. 8 and 9. A precise comparison of the change in balance reading as the element position was varied and the pressure forces obtained from Figs. 8 to 10 is not possible because of some variation in test conditions between the two experiments. An approximate comparison indicates that the element lip pressures gave force changes roughly the same as noted in the balance output during a traverse. The forces obtained from the case lip pressures were approximately half as large as measured during the traverse.

From the above results, which cover the normal operating positions of the balance floating element, it is concluded that the orifices used in the case lip are not suitable for determining misalignment. This is attributed primarily to the fact that the orifice openings are partially open to the case cavity, as shown in Fig. 6. It is customary to use a small element lip thickness to minimize the area on which buoyancy forces may exist. Thus a case lip orifice must be very small if its opening is not to extend below the element lip. Although small orifices were successfully used in the present work, their use was quite difficult. The orifices frequently became plugged during normal handling and some leaks were encountered in the connection between the drilled orifice and the hypodermic tubing leading out of the element. Also, the small drills that worked well in the aluminum element were not successful in harder materials. The use of case lip orifices to determine misalignment or buoyancy forces, therefore, does not appear practicable in balances intended for general use. In addition to the mechanical problems, it has not yet been shown that even a small orifice would be satisfactory, since there may well be pressure gradients in the gap between the case and element lips.

CONCLUSIONS

1. Investigation of the use of pressure orifices in the case lip of a floating element balance indicated that the orifices were not suitable for use in determining either misalignment or the buoyancy forces on the floating element which result from pressure gradients. The primary difficulty was the need for an extremely small orifice, the size being too small to be mechanically practical.

2. It was found that the shear force indication changed by as much as ±3 percent as the location of the floating element was varied over the full operating range. Pressure measurements on the element lip indicated that the buoyancy forces were least when the element was near the rear limit of travel.

3. Balances were constructed having approximately the performance desired. The major problem in attaining satisfactory performance was the thermal expansion and thermal strains which resulted from changing the temperature of the balances. It was found possible to minimize such effects by means of selective assembly of the balance flexures. This method permitted obtaining flexures which resulted in approximate cancellation of the overall temperature effects.

4. In order to minimize the change in modulus of elasticity of the spring material with changes in temperature, the original flexure material selected was Elinvar Extra. This material has a very low change in modulus with temperature. Since it is a nickel-steel material, its thermal conductivity was low compared to beryllium-copper, which is customarily used for flexure material in balances operating at or near constant temperature conditions.

5. The balance having the least full scale shear force capability proved most difficult to design. The high sensitivity made the balance subject to large errors as a result of very small thermally induced forces. To minimize these, it was necessary to use beryllium-copper flexures having a high thermal conductivity.

Defense Research Laboratory
The University of Texas at Austin
Austin, Texas, 19 July 1967

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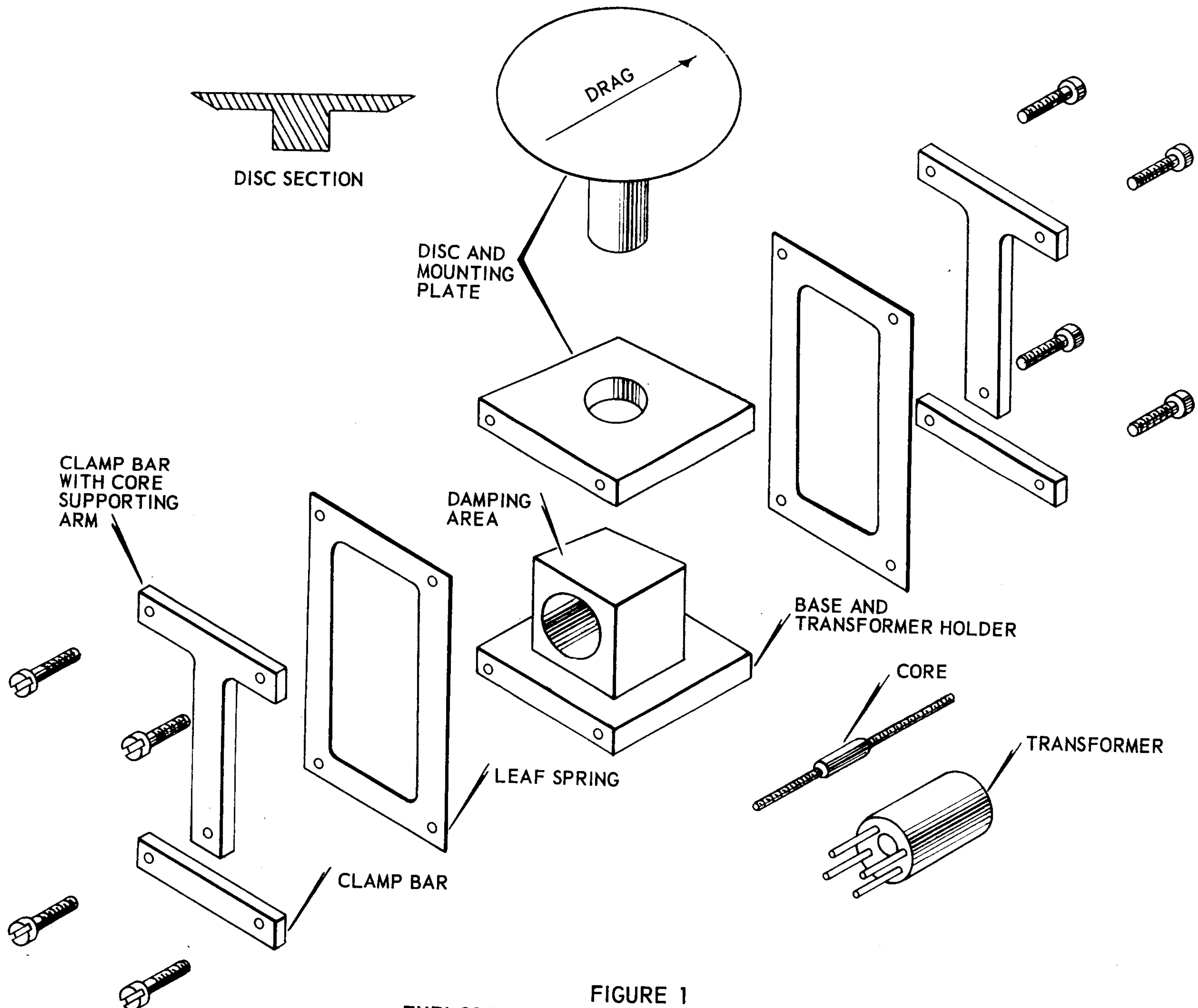

FIGURE 1
EXPLODED VIEW OF SKIN FRICTION BALANCE

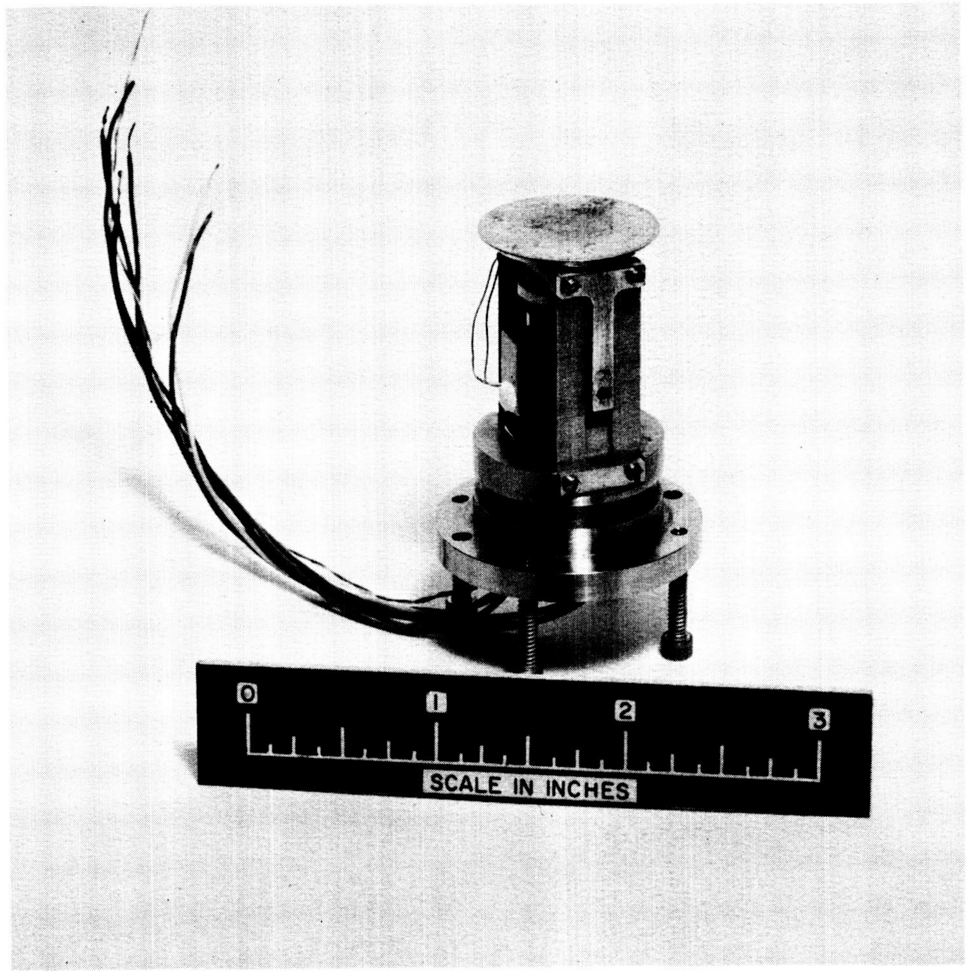

FIGURE 2
ASSEMBLED BALANCE SHOWING THERMOCOUPLE LEADS UNDER LEFT SIDE OF DISK

FIGURE 3
BALANCE AND CASE WITH THERMOCOUPLE INSTALLED

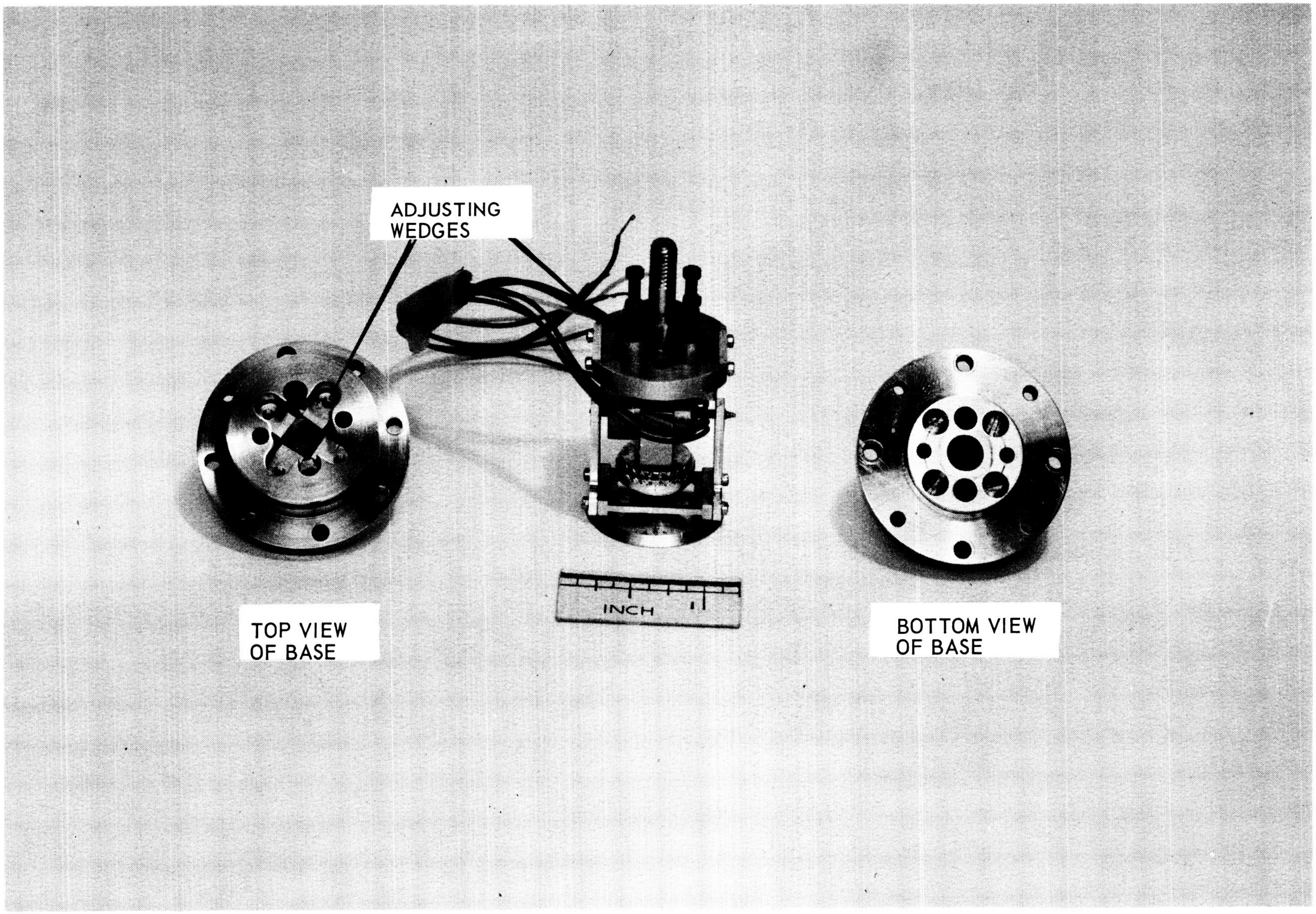

FIGURE 4
BALANCING MECHANISM AND MOUNTING BASE
SHOWING METHOD OF ADJUSTING IN TRANSLATION

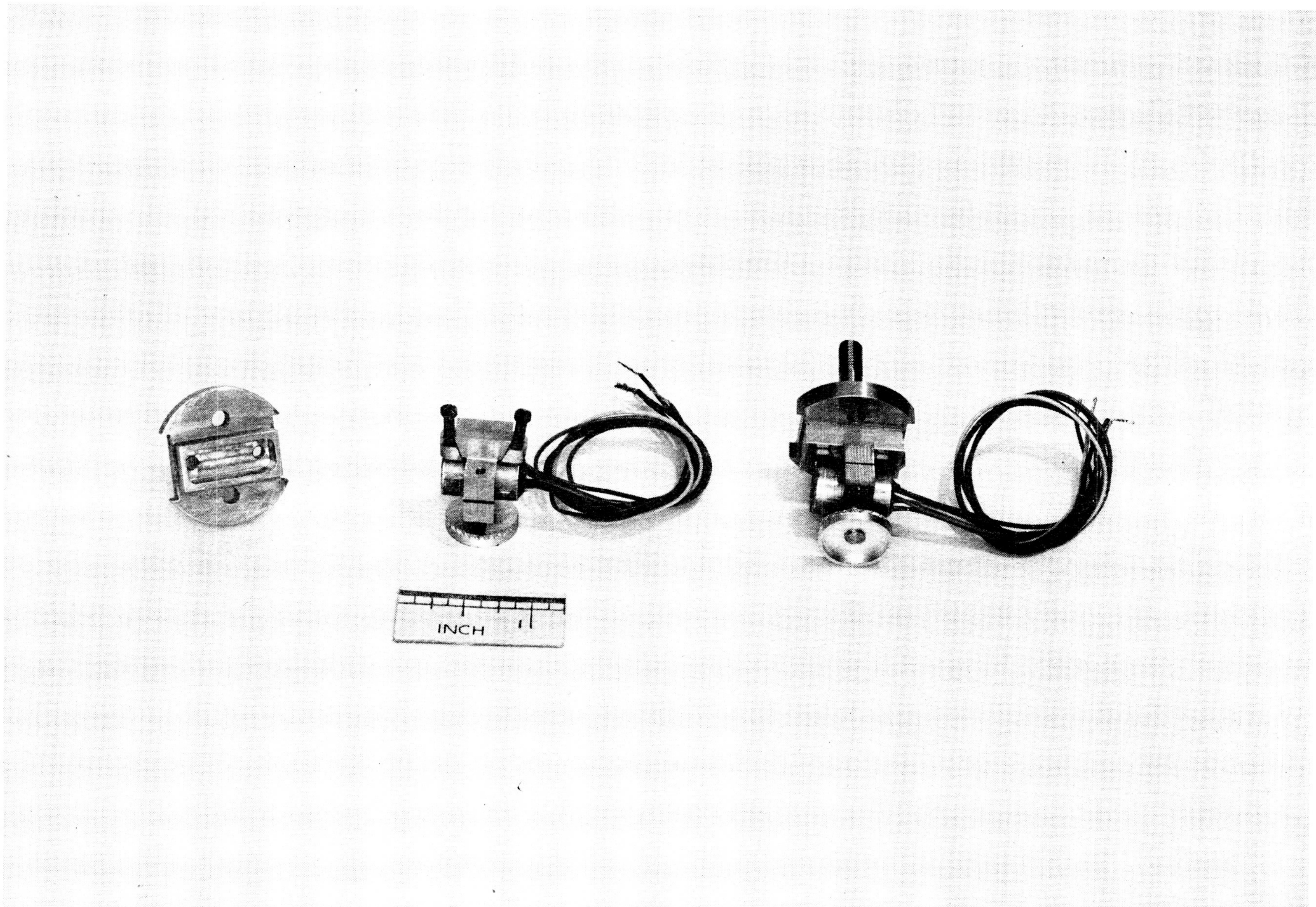

FIGURE 5
TRANSFORMER MOUNTING ASSEMBLY

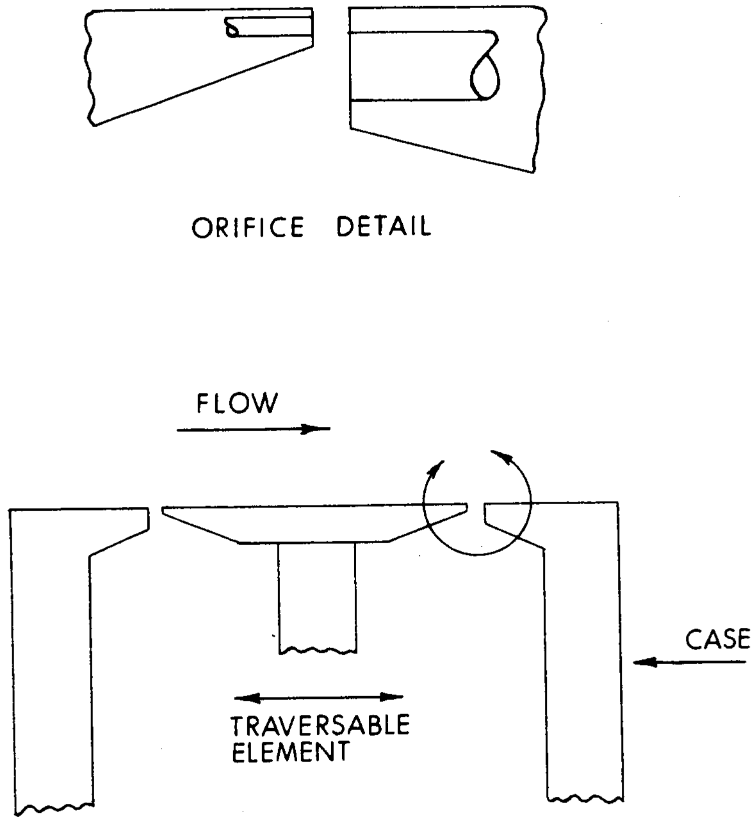

FIGURE 6
SKETCH OF RELATIVE ORIFICE LOCATIONS
IN CASE AND ELEMENT LIPS

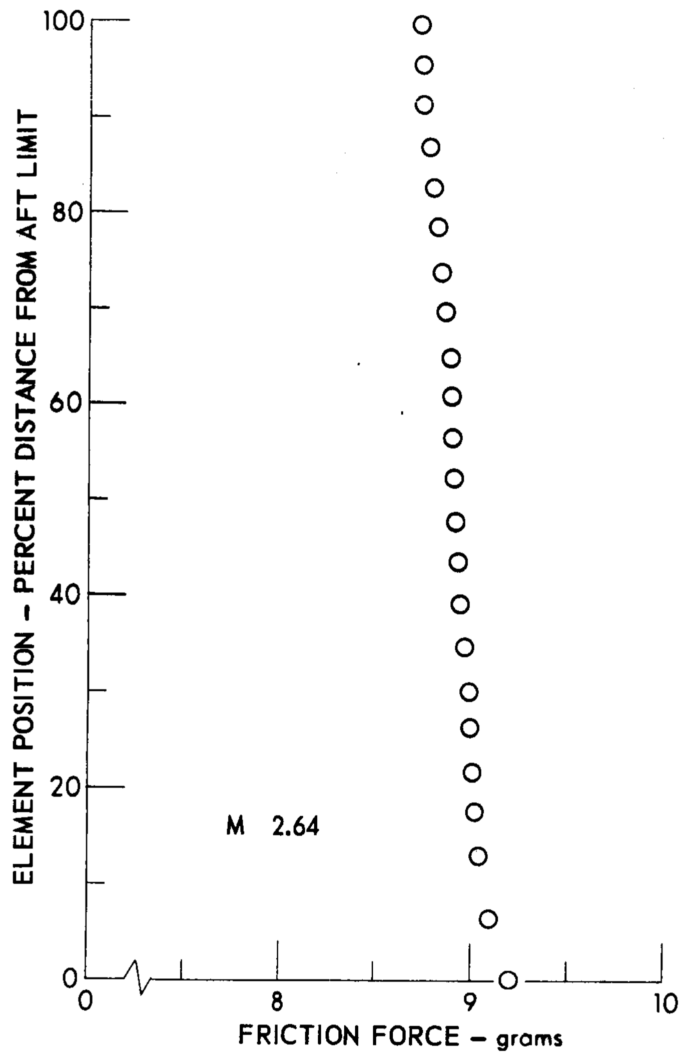

FIGURE 7
EFFECT OF ELEMENT POSITION ON INDICATED DRAG FORCE

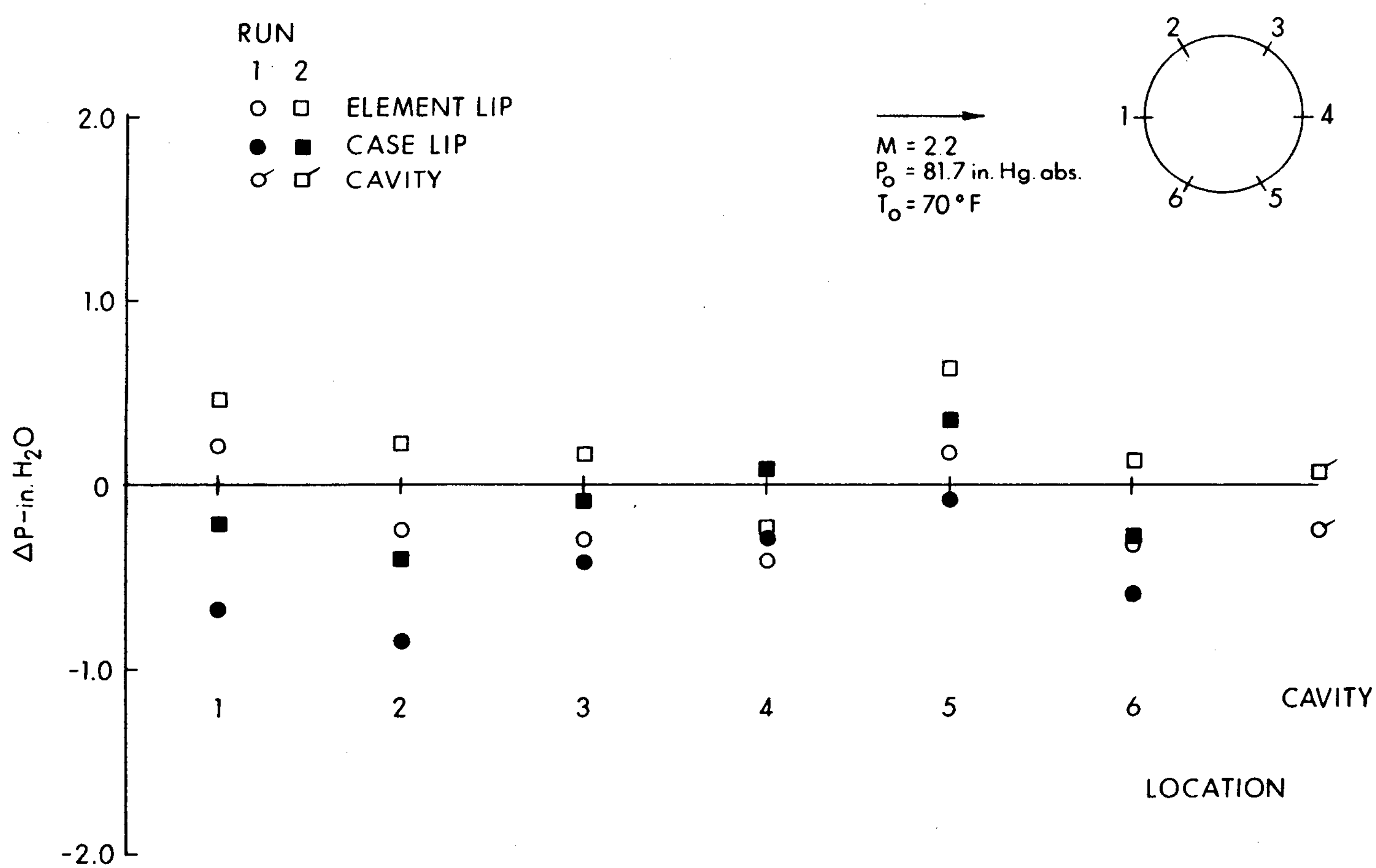

FIGURE 8
CASE AND ELEMENT LIP PRESSURES
FOR ELEMENT FULLY REARWARD

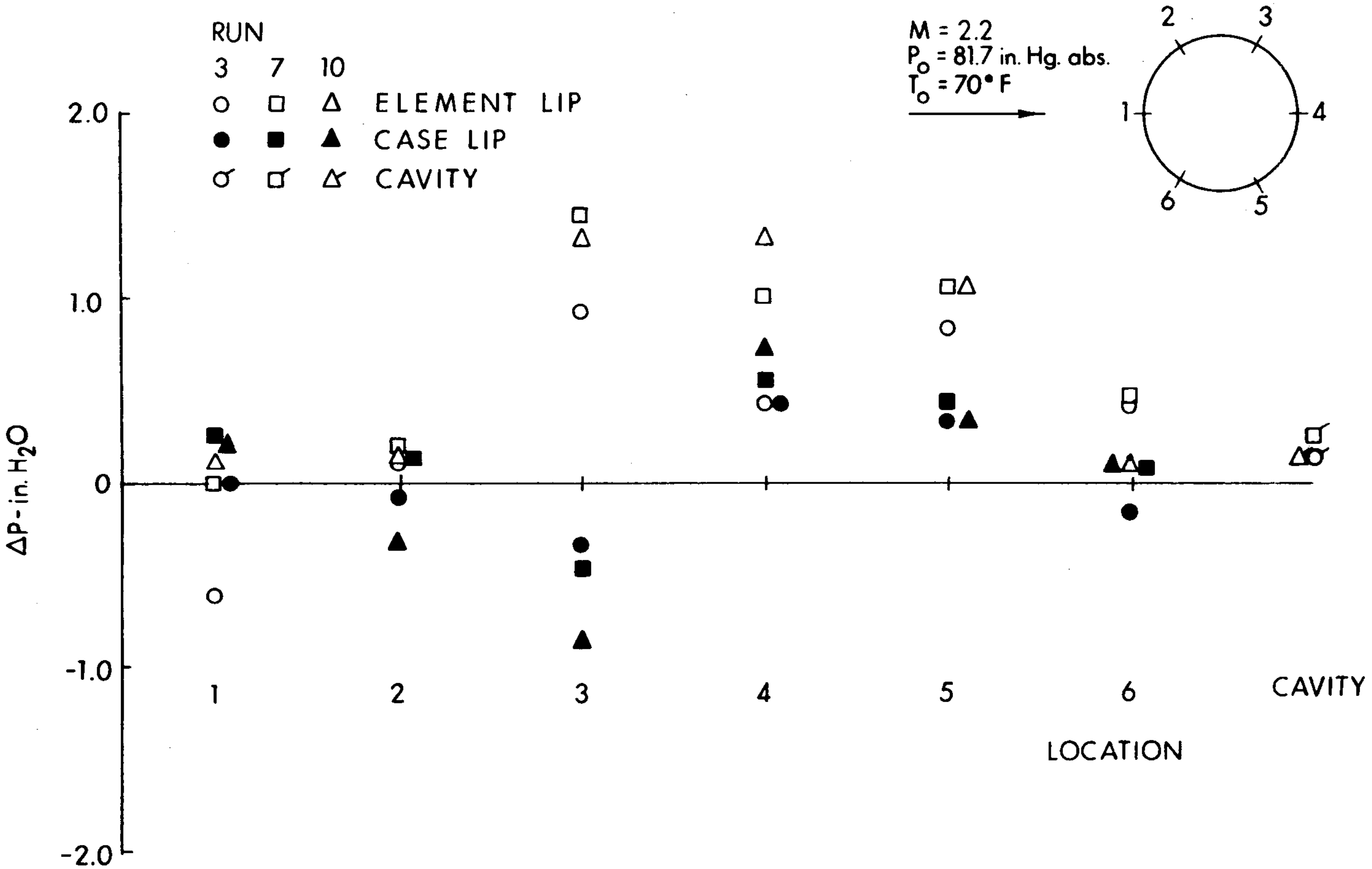

FIGURE 9
CASE AND ELEMENT LIP PRESSURES
FOR ELEMENT AT MID-TRAVEL

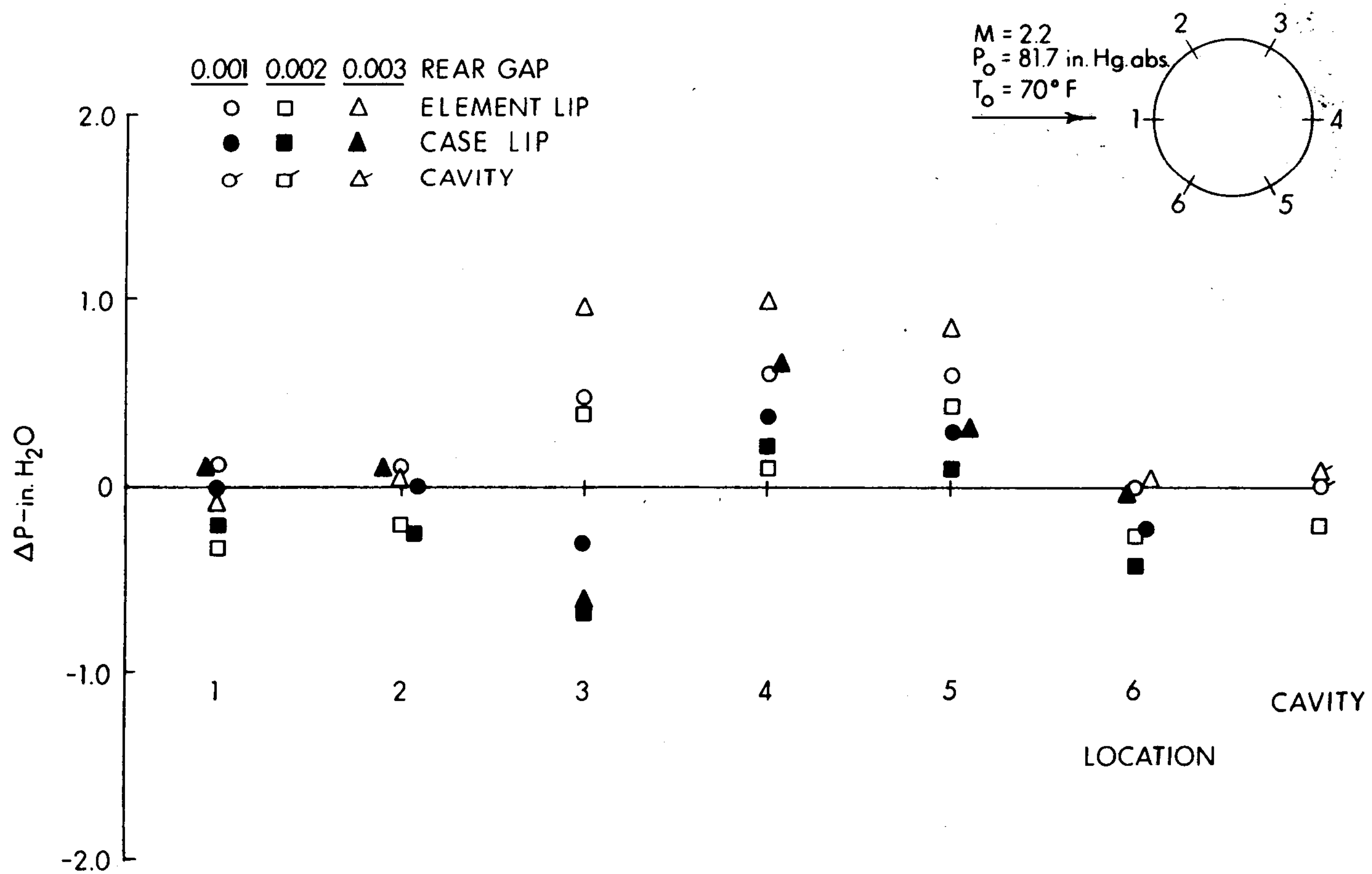

FIGURE 10
CASE AND ELEMENT LIP PRESSURES
FOR THREE REAR-GAP SPACINGS