actual indication. The result will be the length of the required new load arm.

Formula:

$$\frac{\overset{20,000}{\text{Required Indication}} \times \overset{4''}{\text{Existing Load Arm}}}{\underset{26,000 \text{ lbs.}}{\text{Actual Indication}}} = \underset{\text{Required Load Arm}}{3.076''}$$

This formula is good for both plus and minus errors without any change. At the same time it can be applied to both first and second class levers, providing that in the case of a first class lever, a change in the total length of the pivot distances is permissible.

If the total length of the pivot distances between the power and load pivots cannot be changed, further calculations will be necessary.

The power arm will have to be divided with the new load arm in order to find the new multiple of the lever.

When this is accomplished, the required total length will have to be divided by the new multiple, plus 1. The result will be a lever with a total length identical to the old lever. In this case, the fulcrum pivot will be moved.

Formula:

$$\frac{\overset{40''}{\text{Power Arm}}}{\underset{3.076''}{\text{New Load Arm}}} = \overset{13.003''}{\text{New Multiple}}$$

$$\frac{\overset{44''}{\text{Total Length}}}{\underset{14.003}{\text{New Multiple} + 1}} = \overset{3.142''}{\text{Final Load Arm}}$$

Final power arm: 44'' - 3.142'' = 40.858''.

1.12 ONE SCALE OUT OF TWO

There are occasions when a request is made to match a dial head to the lever system of another scale. More often than not, the multiple of the lever system is incorrect.

Example: The total multiple of the lever system is 10. The pull of the dial head is 25 lb. per revolution. That is to say, a 25 lb. load will be required on the dial connecting rod to move the indicator from zero to full capacity.

The capacity of the dial is 3,000 lbs. In other words, 25 lbs. suspended on the dial connecting rod will indicate 3,000 lbs. on the dial.

The ratio of the understructure being 10:1, a 3,000 lb. load will produce a (3,000 ÷ 10 = 300) 300 lb. pull at the connecting rod of the dial head.

The dial head requires only 25 lbs. for a 3,000 lb. indication, so evidently the 300 lb. pull is too much. The solution is to increase the multiple of the understructure (lever system).

One possibility would be to install a first class multiplying lever. Assume that a total length of 50 inches must be adhered to for proper positioning.

Procedure: Divide the 3,000 lb. chart capacity with the required 25 lb. pull of the dial mechanism. Divide the result with the existing understructure multiple. The final result will be the required additional multiple.

$$\frac{\overset{3,000}{\text{Chart Capacity}}}{\underset{25}{\text{Required Pull (Lbs. per rev.)}}} = \underset{\text{Required Understructure Multiple}}{120}$$

$$\frac{\overset{120}{\text{Required Understructure Mult.}}}{\underset{10}{\text{Existing Multiple}}} = \underset{\text{Required Additional Multiple}}{12}$$

The distance between the lp and the pp has to be 50 inches. The problem is to locate the position of the fp. Because the lever is going to be first class and the total length is predetermined, the total length will have to be divided by the required additional multiple, plus 1.

$$\frac{\overset{50}{\text{Total Length}}}{\text{Required Additional Multiple + 1}} = \underset{\text{Load arm (la)}}{3.83''}$$

Should it be necessary to use a second class lever, the total length would have to be divided by 12 only.

$$\frac{\overset{50''}{\text{Total Length}}}{\underset{12}{\text{Required Additional Multiple}}} = \underset{\text{Load arm}}{4.16''}$$

It may be necessary to use two levers for proper hook up. If that is the case, one lever may be a 1:1 even lever, or the required additional multiple may be split between two levers. One lever may have a multiple of 3 and the other 4, or 2 and 6.

$3 \times 4 \times 10 = 120$ or, $2 \times 6 \times 10 = 120$

There may be an occasion when it is necessary to couple an existing beam to the lever system (understructure) of another scale. The multiple of the understructure does not match the beam and for some reason it is impractical or impossible to repivot the beam. It is possible, however, to install one or two additional levers.

The solution is exactly the same as was in the case of the already explained conversion for the dial head, with the exception that the pull of the beam has to be determined first.

One way is to mount the beam on an adjusting (sealing) stand. Suspend a weight pan on the load pivot loop and balance beam. Move poise to the capacity graduation. Load weight pan till balance is reestablished. The total of the weights on the pan represent the required pull for the beam.

Another way is to remove and weigh one of the poises (preferably the main poise) of the beam. Next, measure the poise run from zero to full capacity. Finally, the length of the load arm should be measured.

Example: Weight of poise (Power): 5 lbs.
Poise run (Power arm): 40"
Load arm: 4"

Using the formula described below, it is easy to establish the pull (load).

$$\frac{\overset{5 \times 40}{P \times pa}}{\underset{4}{la}} = L = 50$$

The result is 50 lbs. This is the load or pull required to balance the beam when the poise is at the full capacity graduation.

After the "pull" has been established, follow the same procedure as described for the dial head. In the place of "chart capacity", use the value of the capacity notch.

Example: Value of the capacity notch is 3,000 lbs. Load or pull per capacity is 50 lbs.

$$\frac{\overset{3{,}000}{\text{Value of Capacity Notch}}}{\underset{50}{\text{Pull}}} =$$

Required Understructure Multiple

If the tip multiple of the beam is definitely known, either because it is stamped on the beam or from a blueprint, then the required additional multiple is very easily established.

Simply divide the tip pivot power arm of the beam (the distance between the fp and the pp) with the load arm, in order to establish the multiple of the beam itself.

The next step is to multiply the multiple of the beam with the total multiple of the understructure and with the result divide the known tip multiple. The result will be the required additional multiple.

Example: $\frac{pa}{la}$ = Multiple of beam

Multiple of beam x Multiple of understructure = Existing Multiple

$$\frac{\text{Tip multiple}}{\text{Existing multiple}} = \text{Required additional multiple.}$$

This accomplished, proceed with the calculation of the additional lever or levers, as has been previously explained.

Up to this point it has been assumed that the multiple was inadequate, and strived to increase it in order to get the correct pull.

Should there be an occasion when the multiple is too great, producing insufficient pull, it will be necessary to install a third class lever.

The required change in the multiple is obtained by exactly the same methods as described previously, but the final results will be in decimals.

From here on the procedure changes. When increasing the multiple, the total length of the new lever is divided by the result of the second division and the result is the load arm of the lever.

In this case the total length of the new lever is multiplied by the result of the second division and the result is the power arm.

1.13 OBTAINING TOTAL MULTIPLE OF UNDERSTRUCTURE WITHOUT MEASURING PIVOT DISTANCES

Balance or zero scale. Hang a one pound weight with a thin thread on the steelyard rod. Move your poises until the beam balances again. Take a reading. Whatever the total reading, it is the multiple of the understructure. For example, if with a one pound weight suspended on the steelyard rod, the poises read 320 lbs., then the total multiple of the understructure will be 320, or ratio 320:1.

On a dial scale this multiple can be read directly off the chart.

A more accurate reading can be obtained by using a 10 lb. weight instead of 1, but in this case, the reading will have to be divided by 10 to get the multiple.

When no weight is available, a wrench or a hammer may be suspended on the steelyard rod. Note reading and weigh the article used in your shop on a fine scale. If possible, use hundredths of pounds for this purpose in order to make the calculation easier. The multiple can be easily calculated now by dividing the beam or dial reading with the weight of the article.

1.14 TIP MULTIPLE

To obtain the tip multiple, which is the total multiple of the whole scale, including the beam at its power (tip) pivot, place a one pound weight on the counterpoise. In some instances there is no counterpoise on the tip loop. When such is the case, a makeshift hanger made out of a piece of wire will serve the purpose.

This accomplished, place a load on the platform big enough to counter balance the one pound weight. On a compound beam with four poises, the tare and its fractional poise may be used to help establish balance. Reserve the main poise and its fractional poise for establishing the multiple.

Remove the one pound weight from the counterpoise or hanger. Do not remove the counterpoise or the hanger. Move the main poise and if necessary its fractional poise till the balance is reestablished. The reading of the main and its fractional poise will be the tip multiple of the scale.

If no load is available, but the beam has a back balance pivot and loop, a load substitute may be suspended on it to establish a balance condition with the one pound weight on the counterpoise.

1.15 "DEAD LOAD PULL"

The "dead load pull" is the applied weight of the understructure lever system and platform on the steelyard rod.

First determine the multiple of the understructure. Next disconnect transverse lever from steelyard rod.

Beam scale: Load back balance loop until beam balances. Connect transverse lever to steelyard rod. Move poises until balance is reestablished. Note reading on the beam. Divide the weight indicated by the poises with the multiple of the understructure and the result will be the "dead load pull".

Formula:

$$\frac{\text{Weight indicated by poise}}{\text{Multiple of understructure}} = \text{Dead load pull}$$

Cabinet Dial or Weightograph: After the transverse lever has been disconnected, disconnect drop weight bucket to help zero the dial. If necessary, attach additional weight to the steelyard rod. Connect transverse lever to steelyard rod. Take a reading on the chart. If chart capacity is insufficient, use capacity and tare poises. If poises were used, sum up the total. The reading on the chart, or the total reading of chart and poises divided by the multiple of the understructure results in the dead load pull.

$$\frac{\text{Total weight as indicated}}{\text{Multiple of Understructure}} = \text{Dead load pull}$$

Another way to obtain the dead load pull is to place a counter scale underneath the transverse lever. Cut a piece of two by four to fit between the transverse lever and the platform of the counter scale. Place the two by four directly underneath the nose iron pivot and disconnect the steelyard rod. Weight the pressure of the transverse lever on the scale. The reading, minus the weight of the two by four, will be the dead load pull.

When a new dial or weightograph is to be installed on an already existing scale, the engineering department of the manufacturer has to have the dead load pull in order to be able to provide adequate balancing facilities. A slight error in the dead load pull can be easily compensated for at the time of the installation.

1.16 CALIBRATED NOSE IRON ADJUSTMENT

A great deal of time and effort can be saved by calculating the nose iron adjustments to correct an error in a scale. To find out how far a nose iron should be moved to correct an error, the power arm of the lever in question should be multiplied by the error in pounds, and the result should be divided by the actual load. The result will be most likely in decimal inches. To make the nose iron adjustment, it is good to use a sliding caliper in thousandths of an inch.

Example: 20,000 lbs. read 20,060 lbs. and the power arm of the lever on which the adjustment is to be made is 96". 96" x 60 = 5,760   5,760 ÷ 20,000 = .288"

The nose iron will have to be moved .288". Because the error was plus, it will have to be moved away from the fulcrum pivot.

1.17 HOW CONSTRUCTION OF A LEVER AFFECTS SENSITIVITY

An important consideration on constructing a lever is the strength of the material to be used. Depending on the class of the lever, the strength of the material varies at the pressure points.

The strongest point of a first class lever is its fulcrum pivot and the area surrounding it. If it is an even lever, then the load and power pivots can be made equal in strength; and the load and power arms can symmetrically taper off toward the ends.

On a first class multiplying lever it is again the fulcrum pivot and its surrounding mass that is the strongest, but the load pivot and its area is stronger than the power pivot and its area. The difference in strength depends on the ratio.

The strongest point of a second class lever is its load pivot and the material surrounding it. The fulcrum pivot and its area is usually stronger than the power pivot and its area. The difference in strength depends upon the ratio.

The power pivot and its area is always the strongest portion of a third class lever. Figures 1.7, 1.8, 1.9, 1.10 and 1.11 clearly illustrate the shaping of the levers.

The proper distribution of mass above and below the pivot line and the effects of "Range" have already been clarified and are subject to slight variations as required by application.

The pivots must be straight, sharp, and hard. They must also run parallel to each other on two planes.

Figure 1.26 is an illustration of pivots in proper "gauge parallel". Figure 1.27 is an illustration showing pivots not in "gauge parallel". The distance "A" is greater than distance "B". This means that the ratio of the lever at distance "A" is smaller than at distance "B".

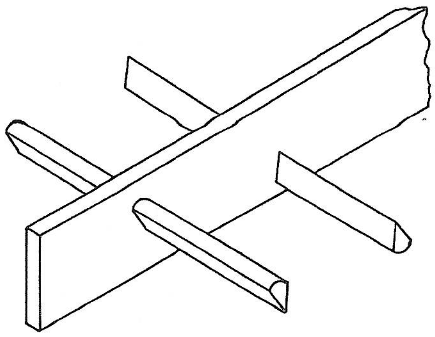

Figure 1.26. Pivots in "Gauge Parallel"

The pivots of a lever must have a certain amount of end play in their bearing to ensure freedom of movement. Having this end play, it is obvious that the levers can change their positions. When this occurs, there will be a change in ratio, with resultant inconsistent indication.

The relationship of the pivots on Figure 1.28 may be called "parallel" in order to differentiate it from "gauge parallel". In this case the pivots are found to be parallel on another plane. Care should be taken to set the pivots in this manner.

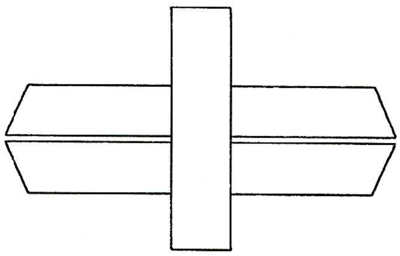

Figure 1.28. Pivots in "Parallel"

The reason for this effect is that when the bearings ride on the high portion of the sloping pivot, the sensitivity of the system will increase. Increased sensitivity will increase the travel of the indicating element per load unit. This, of course, will mean a plus error. On the other hand, when the bearing (or bearings) slide to the low side of the pivot, the opposite effect will take place.

The levers of a dial scale should be made sturdy enough to prevent deflection. The pivot line should be neutral. Should there be a closed range in a lever so that the scale's sensitivity would increase as the scale was being loaded?

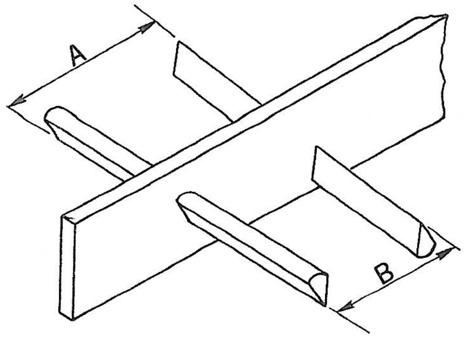

Figure 1.27. Pivots Not in "Gauge Parallel"

Figure 1.29 illustrates a lack of parallel. Pivots set in this manner will cause inconsistency (change). The first reason for inconsistency is that the bearings will have a tendency to slide down on the sloping pivot and jam against the friction points. This jamming will cause friction with a resultant inconsistency.

A more pronounced inconsistency will develop due to lack of parallel in automatic scales with graduated travel. These would be dial scales with either mechanical or optical transmission.

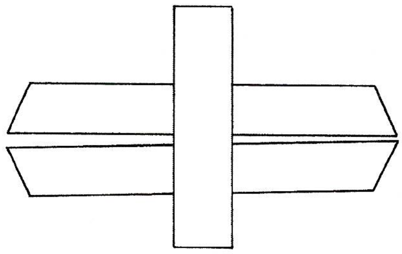

Figure 1.29. Pivots Not in "Parallel"

Increasing sensitivity means increasing travel and this in turn means a plus error. The opposite effect would take place if the lever had an open range.

Of course this effect could be counteracted by regulating the pendulums of the dial; but only if there was no deflection in the levers and if one full revolution or movement of the indicating element covers the total capacity of the scale. In this case the scale should not have any tare or capacity bars, or unit weights.

When the capacity of a dial scale is increased with the use of counterbalance weights (unit weights) and/or with tare and capacity bars, a straight, neutral pivot line is a must, because it is impossible to correct any errors in the second movement of the indicating element with the pendulums without spoiling the adjustments of the first movement.

The purpose of a unit weight is to return the pendulums and the indicating element to zero position; thus, by giving the pendulums a fresh start, the capacity of the scale is increased.

During the first revolution or movement, the unit weight is completely detached from the lever system. It has no influence on the sensitivity.

If the pivot onto which the unit weight is suspended is high, its mass will be applied to the existing mass above the fulcrum pivot; thereby increasing the total mass. This increased mass above the fulcrum will increase the sensitivity of the scale. Increased sensitivity means increased travel. Increased travel means a plus error. If the edge of the unit weight pivot is below the pivot line, the effect will be the opposite. Any attempt to correct these errors by raising or lowering the pendulums will create an error in the first revolution, when the influence of the unit weight is removed. The proper correction is to raise or lower the unit weight pivot. Some manufacturers make their unit weight pivots vertically adjustable.

When sliding capacity poises are used to increase the capacity of the scale, the poise run must be parallel with the pivot line. If the poise run has an upgrade in its relationship to the pivot line, there will be a plus error on the second movement of the indicating element. If the poise run has a down grade, the effect will be opposite. The same applies to tare bars and poises. The scales sensitivity is also effected by the angularity of the lever couplings and steelyard rods.

Figures 1.30, 1.31 and 1.32 illustrate the various effects caused by correct and incorrect couplings. It is important to consider these various effects during the construction, installation, and repair of scales. These effects are especially important when dealing with scales having a predetermined sensitivity, such as dial scales.

Figure 1.30 illustrates the correct neutral position, with the levers horizontal and the coupling vertical; thus forming a 90° angle. In this case the actual load arm (which is the distance between the fp and the lp) is identical with the acting load arm. The lever coupling being at a 90° angle, place the center of gravity of the lever in line with the fulcrum pivot; thereby excluding any pendulum arm effect of the load.

Figure 1.31 illustrates an unstable (top heavy) position. The angle in this case is 60°; as a result a load pendulum arm is created, which is marked with "a" on the illustration. This pendulum arm will place the center of gravity above the fp, to the point "c" where the acting load arm and the pendulum arm intersect each other, with a resulting increase in sensitivity, or beyone that, instability.

Not only will the center of gravity be displaced, but the actual load arm and the acting load arm will lose their identity. Distance "b" is the acting load arm.

Figure 1.32 illustrates a stabilizing position. The angle between the pivot line and the coupling is 120°. This creates a load pendulum arm effect below the fulcrum pivot; thus displacing the center of gravity of the lever below the fulcrum. The center of gravity will be at point "c", where the acting load arm and the pendulum arm intersect each other. This will result in the loss of sensitivity. Distance "b" is again the acting load arm.

To increase or decrease the sensitivity of a scale by utilizing this effect is not advisable particularly in scales with predetermined sensitivity.

Some manufacturers recommend that the errors in the second and third revolutions, etc., of their cabinet dials be corrected by shifting the cabinet. This recommendation is based on the assumption that the pivot line is perfect and the error is caused by angularity.

A non-vertical condition of a coupling also creates a symmetry.

Figure 1.33 illustrates the symmetry in the beams travel when the coupling is vertical (plumb). In this case, when the load is decreased by one ounce, the power arm of the lever will sink 1/2" then if the load is

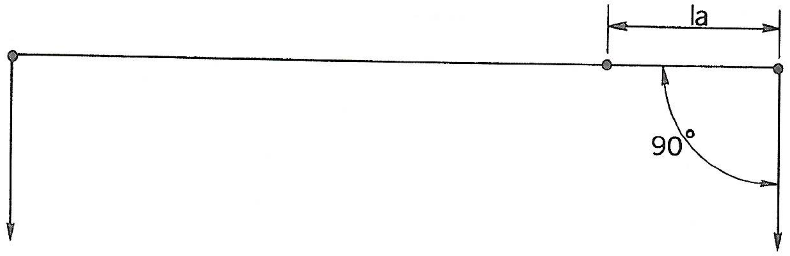

Figure 1.30. Correct Neutral Position

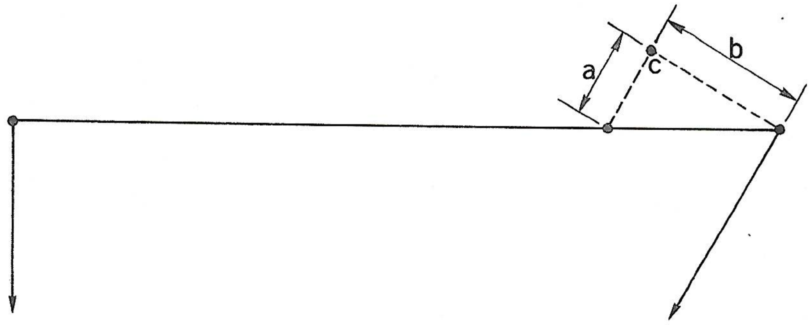

Figure 1.31. Unstable Position

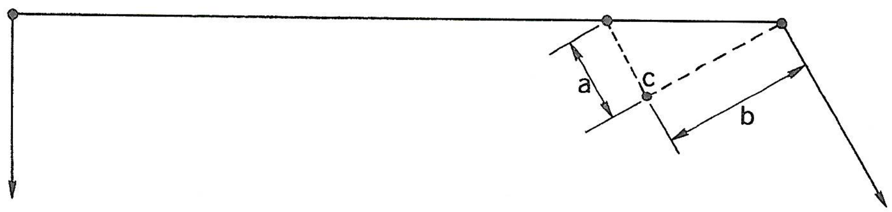

Figure 1.32. Stabilized Position

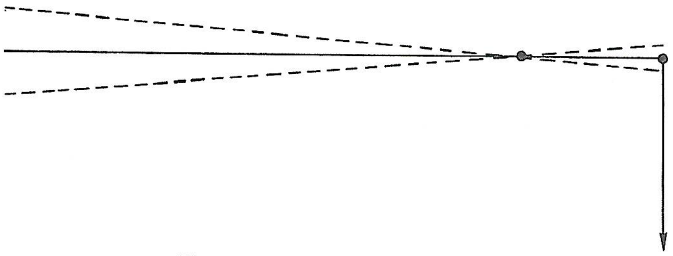

Figure 1.33. Symmetry

increased by one ounce; the lever of the power arm will rise 1/2". The travel will be symmetrical.

Figure 1.34 illustrates the asymmetry in travel when the angle is acute. The overall travel has increased, but there is a greater amount of travel above the horizontal than there is below.

Figure 1.35 shows the opposite effect created by an obtuse angle. This change in sensitivity and lack of symmetry is more pronounced when short couplings are used for obvious reasons. For instance, if we move a nose iron that is connected to a 5" shackle one full inch, the angle will become more acute than if the shackle was 20" long and we moved the nose iron 1".

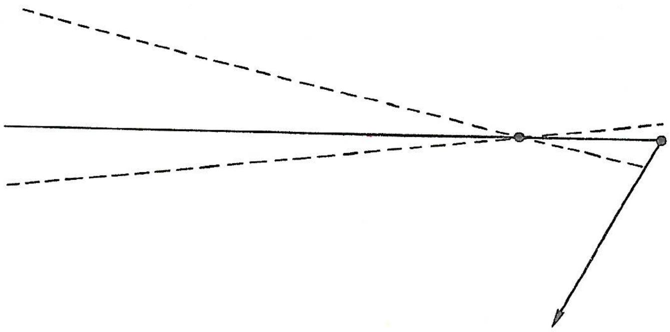

Figure 1.34. Asymmetry

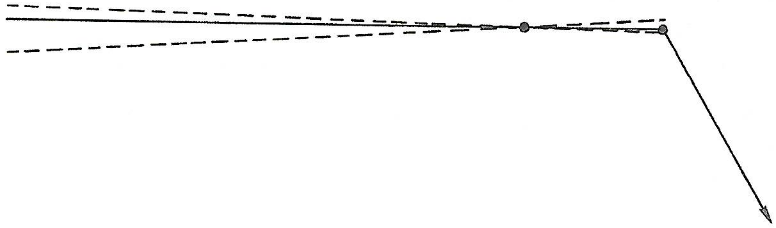

Figure 1.35. Opposite Effect-Obtuse Angle

While a slight lack of symmetry will not seriously affect the usefulness of a beam scale, it is disastrous in a scale with predetermined sensitivity. A very good example is a Motor Truck Scale equipped with a dial or weightograph.

A Motor Truck Scale has a number of nose irons, (adjustable pp), which are of necessity coupled onto the load pivots of the next levers with very short couplings (shackles).

Let us assume that one section of the scale has a minus error, which will have to be adjusted with the aid of a nose iron. The correct procedure is to move the nose iron of the lever that takes care of that section toward its fp. By moving the nose iron, the right angle existing between the lever and shackle is altered. The angle becomes obtuse. This has a stabilizing effect, which in turn decreases sensitivity. A decrease in sensitivity means a decrease in travel, which in turn means a minus error. The result is, that although the ratio adjustment of the lever was correctly made, the travel decreasing effect of the angular coupling will defeat the result of the ratio correction, or even produce an opposing effect. In other words, instead of correcting the minus error, we either get very little result or none at all. In extreme cases, the error will increase.

For this reason, when nose iron adjustments are made, the top side of the shackle has to be moved along on the load pivot of the next lever (onto which it is suspended), the same amount the nose iron has been moved. (This is for pipe lever scales. On straight lever scales, both nose irons have to be moved in the same direction an equal amount to keep the connecting linkage plumb.)

The travel of a dial scale must be symmetrical from half capacity to zero and from half capacity to full capacity. An angular coupling destroys this symmetry. Furthermore, an angular coupling has the tendency to slide to a plumb position after the adjustment has been made and is aided in this tendency by the vibrations created while the scale is being loaded. This, of course, will change the adjustment and create an error.

Another bad effect caused by this tendency to seek plumb position is that the shackle will jam, as has been explained in the section dealing with friction.

Finally, it will cause a one sided wear on the pivot edges and will result in a change in the levers ratio, not to mention the possibility of the pivot side pressing against the side of the bearing.

To summarize the preceding: The sensitivity of a scale is the distance the beam or indicator travels when a certain unit of weight is placed on the platform of the scale.

The Sensitivity Requirement (SR) of a scale is the amount of weight necessary to make the scale travel a certain specified distance.

When we increase or decrease the travel of the indicator we are making a sensitivity adjustment.

The sensitivity of a dial scale is predetermined by its graduations. If one graduation on the chart is 1/8 of an inch wide and represents one pound, then the sensitivity of the scale must be such as to move the indicator 1/8 of an inch when one pound is placed on its platform.

The sensitivity of beam scales, even arm scales and scales equipped with only a zero indication, is not so closely determined. As a matter of fact, the more sensitive these scales are, so much the better. The sensitivity of these scales may vary without impairing their accuracy in a certain sense.

The sensitivity of these scales is important for another reason. If the scale is very sensitive, then the slightest discrepancy in weight will move the beam a definite, easily noticeable amount, and vice versa. In this sense, the more sensitive the scale is, the more accurate it is.

What is a dial scale actually? The dial scale is nothing but a desensitized beam scale. The travel of this scale is mechanically enlarged with the aid of a rack and pinion, or tape driven transmission, and this travel is projected on its chart. The chart of a dial scale is the enlarged projection of its travel.

What is a Weightograph? The Weightograph is also a desensitized scale. The travel of this scale is optically magnified and this magnified travel is projected on a ground glass screen.

As a conclusion to this section, it is advisable to keep in mind that when the travel of a scale is either mechanically or optically amplified, the errors of the scale caused by uncalled for friction and poor workmanship, are amplified. For this reason, if careful workmanship is necessary to make a good beam scale, a good dial scale requires even more painstaking care in the course of its construction or repair.

1.18 FRICTION

Scales are mechanical devices and contain moving parts.

It is a well known fact, that all motion is retarded by friction. In some cases more, and in some less.

To be able to construct a perfect scale it would be necessary to eliminate friction entirely. This is impossible; therefore, it must be admitted that there are no perfect scales in the true sense of the word.

However, friction can and must be eliminated as much as possible, taking into consideration the required accuracy and the abuse the scale is expected to withstand.

The pivots of an analytical scale can be made razor sharp and the anti-friction points and plates eliminated, with the resulting near perfection in friction elimination. This can be done only because of the light load and careful handling these scales will receive.

On the other hand, in the case of commercial and industrial scales, speed of operation is essential. This results with less careful handling and considerably heavier loads. To be able to weigh heavier loads and take more abuse, these scales have to be made sturdier, with longer pivots, longer bearing surfaces, and ways and means to keep the moving parts of the scale in their predetermined positions. This requires the aid of anti-friction points and plates, check rods, etc. The result is a source of unavoidable friction.

With careful construction and workmanship, friction can be decreased to such a degree, that it will not interfere with the practical usefulness of the scale.

Friction can be given two classifications in a scale. One is avoidable and the other - unavoidable friction.

1.19 PIVOTS AND BEARINGS

The unavoidable friction is caused by the surface resistance of the pivots edge as it revolves on the surface of the bearings, and by the friction created at the anti-friction points (commonly referred to as friction points).

It is a well known fact that even the finest razor edge has a surface; although indiscernible to the human eye. If it did not have a surface, it would not have an edge.

No matter how fine the edge of the pivot is, after a certain amount of usage, it will develop a small radius due to the fact that is is unable to withstand the strain of the load for any length of time.

In the course of a certain amount of usage, the pivot edge will wear down to a point, where, if the pivot and its bearing are made of good and properly hardened material, it will be able to withstand the strain for years, without seriously affecting the accuracy of the scale.

Figures 1.36 and 1.37 illustrate, in an exaggerated manner, the pivot before and after a certain amount of usage. Of course the bluntness is greatly exaggerated, but that is how it would look under a magnifying glass. If the pivots actually looked as bad as shown in Figure 1.37 the scale would require an overhaul.

Another source of unavoidable friction is shown under Figures 1.38 and 1.39. Figure 1.38 illustrates the most ideal way of friction elimination. It is best because the point of contact between the end of the pivot and the anti-friction plate is reduced to minimum. This method should be used wherever economy permits it, and it is technically possible.

The method illustrated by Figure 1.39 is commonly used on the understructures of commercial and industrial scales, partly because it is a cheaper method, and partly because the anti-friction plate is backed up by the body of the lever and as a result cannot break off as easily as the plate of Figure 1.38. This is important because these scales are subjected to rough handling. The pivot is also less likely to be damaged. The fine pivot tip and the anti-friction plate of Figure 1.38, tends to break off through rough handling.

The method of Figure 1.39 is also used on the beams of various scales, mostly for economy. Recently, however, the end contact method of Figure 1.38 is more and more frequently used despite the higher cost, for greater dependability.

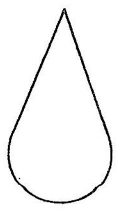

Figure 1.36. Before Use

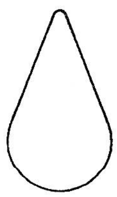

Figure 1.37. After a Certain Amount of Use

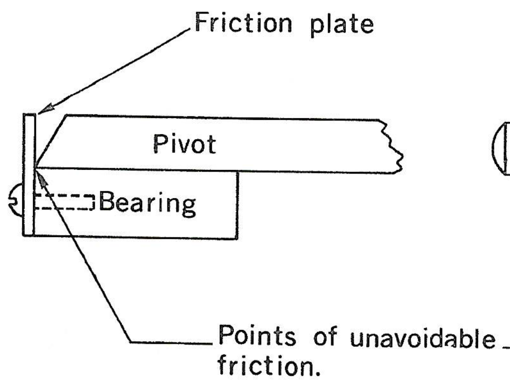

Figure 1.38. Ideal

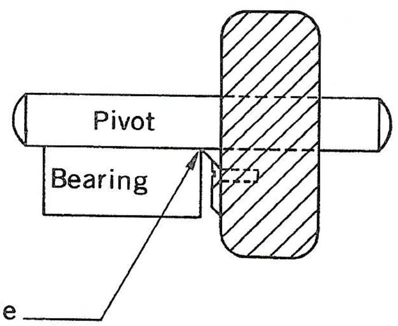

Figure 1.39. Common

There are, of course, a number of other sources of unavoidable friction, such as the surface resistance of ball bearings, the clinging tendency of steel tapes to cams, dashpots, etc., but there is no reason to discuss them in detail as nothing can be done about them.

1.20 AVOIDABLE FRICTION-MAINTENANCE OF FRICTION POINTS

Improperly made, and improperly aligned bearings can be a source of avoidable friction. Figures 1.40, 1.41, 1.42, 1.43, 1.44 and 1.45 illustrate the types of bearings generally used.

"V" bearings, as illustrated by Figure 1.40 should be used on beam assemblies, tare, and shelf levers that are connected directly to automatic or semi-automatic equipment, and scales constructed on the principles of a parallelogram. In these cases, the bottoms of the bearings should be sharp and clear cut.

Very frequently "V" bearings are used in the understructures of various types of scales. These bearings should not be very sharp at the bottom. The sides of these bearings should end in a small (about 1/32") radius at the bottom. (Figure 1.42).

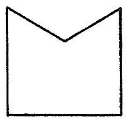

Figure 1.40. Good "V" Bearing

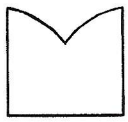

Figure 1.41. Bad "V" Bearing

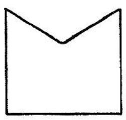

Figure 1.42. Good Round Bottom "V" Bearing

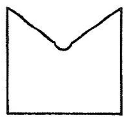

Figure 1.43. Bad Round Bottom "V" Bearing

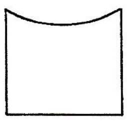

Figure 1.44. Concave Bearing

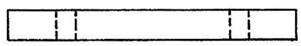

Figure 1.45. Flat Steel Bearing

These bearings are usually self-aligning. If they are not, care should be taken to align them properly.

Should it be necessary to adjust the corners of such a scale, the pivots should be honed on a wide surface. Honing on a short blunt angle is inviting trouble, because a shoulder will be formed on the pivot. Figure 1.46 illustrates the proper way, and Figure 1.47 the improper way to hone. A shoulder on the pivot will lean against the side of the bearings (Figure 1.48) and cause friction.

The advantage of a "V" bearing is that it keeps the lever in position. This is absolutely necessary in automatic and semi-automatic scales because a shift in the position of the lever will change the angle of the steel tapes or connecting rods; thus causing a change in the reading.

The bearing illustrated by Figure 1. 49 is bad because of its convex sides. The side of the pivot will lean against the side of such bearing before the lever completes its travel, creating a great deal of resistance and friction.

When a pivot resets in two bearings, it is important that they are perfectly lined up as illustrated by Figure 1. 50.

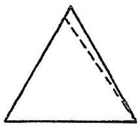

Figure 1. 46. Proper Honing of Pivot

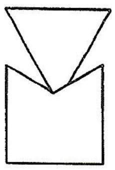

Figure 1. 48. Friction Caused by Improper Honing

Figure 1. 50. Good

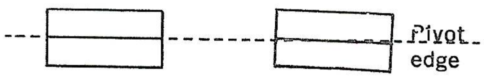

Figure 1. 52. Improperly Set

To be able to make perfectly straight, sharp bottomed, and highly polished "V" bearings, it is best to use a lapping bar with fine grained silicon carbide dust and water. A good mechanic should have several lapping bars handy, each with a different angle.

Lapping bars are usually made of copper, brass, aluminum, or cast iron; but hot rolled low carbon steel will also give good results. The lapping powder will cling to soft metals and as a result cut the hardened steel much faster.

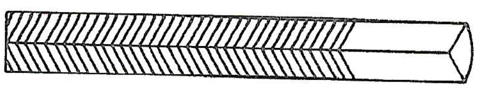

Figure 1. 54. Slant Cut

A straight pivot will reset on four points when the bearings are not aligned, and thus the tendency of the pivot to slide down to the bottom of the groove will cause a strong jamming effect. This in turn will stiffen the action of the scale, with a resultant change in the readings.

Figures 1. 51, 1. 52 and 1. 53 illustrate improperly set bearings.

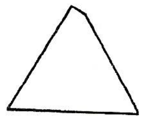

Figure 1. 47. Improper Pivot Honing

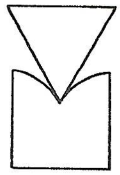

Figure 1. 49. Friction Due to Convex Bearing Sides

Figure 1. 51. Bad

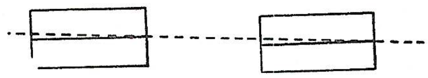

Figure 1. 53. Improperly Set

Take a piece of round stock about a foot long, file the necessary angle on the bar with a slanting cut, as illustrated by Figure 1. 54. Use a bastard file. Leave the surface course. The angle of the lapping bar should be just slightly sharper than the angle of the bearing. The difference in the angle is slightly exaggerated on Figure 1. 55.

Double bearings should be lapped in position in their bracket or stand. The stand, or bracket, should be firmly gripped in a vise, taking care not to deform it with the pressure of the jaws.

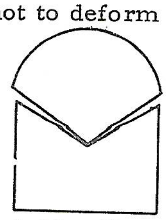

Figure 1. 55. Slant Cut

This is very important, because if the bearings are lapped in a deformed bracket, they will be thrown out of alignment when removed from the vise.

One end of the bar should be marked to avoid reversal and the resultant double cut. Moisten the bar with water, and sprinkle it with carbonrundum powder. Place bar in bearings and start pushing back and forth with an even pressure on both ends. Keep on lapping until the bar is dry and starts to slip easily. Remove the bar.

An inspection of the bearings will show a highly polished surface where the lapping bar contacted the bearing. If the bearings were badly out of alignment, the procedure will have to be repeated. The bearings can be considered finished when the bottom and the side of the bearings have a highly polished surface up to at least 1/16" from the bottom. It would require too much time and work to polish the sides fully and it is unnecessary. If the angle of the bearing is too sharp, not giving the pivot room enough to turn, it will be necessary to anneal the bearings, file them to a broader angle, reharden, and finally lap them.

When the bearings are self-aligning in two directions, it is best to lap the bearings individually, by clamping the bar in the vise and moving the bearings instead of the bar. In this case, wet your bsr and sprinkle it with carbonrundum power. Mark one end of the bearings to avoid reversal.

Loops with "V" bearings should be lapped the same way, with both bearings in place and their normal working position. Do not turn bearings out while lapping. Figure 1.56 shows the correct position while lapping, and Figure 1.57 the incorrect position.

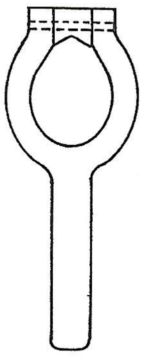

Figure 1.56. Correct

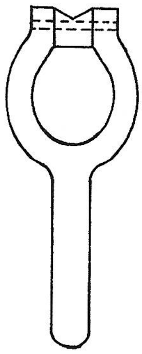

Figure 1.57. Incorrect

The understructure of scales generally receive a severe punishment while being loaded and unloaded; therefore, it is not advisable to use a sharp bottom "B" bearing. The fine edge of the pivot will wear down, and jam into the sharply lapped bottom.

For this reason round bottomed "V" bearings are much more suitable for understructure.

Almost all manufacturers of dial scales use "V" bearings with a light radius at the bottom, to give room for dulling edges.

On the other hand, these scales are equipped with round shank pivots, which enables the mechanic to make necessary adjustments by turning them in the desired direction. Do not hone pivots that work in such a bearing, unless you hone the whole surface of the pivot.

When repairing such a bearing, a piece of fine emery cloth on a three cornered file will do the job in cleaning and removing minor cuts.

Flat (Figure 1.45) and concave (Figure 1.44) bearings are also used in the understructure of the scales. The advantage of these bearings is that they will not interfere with the action of the scale, no matter how obtuse the angle of the pivot is. The disadvantage of these bearings is that they require some sort of checking system to keep the levers and the platform in their predetermined positions. This checking system, in turn, will introduce a great deal of desirable friction.

In such scales, care must be taken that the checks do not permit the platform to shift too much; but, at the same time, they must be loose enough to insure free action. However, in spite of these precautions, during the process of loading and unloading, the platform may be pushed hard against one of the checks and remain in that position, causing considerable friction.

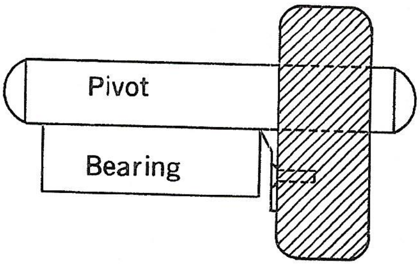

Figure 1.59. Proper

Improperly made pivots are another source of avoidable friction.

When the sharp and pointed end of a pivot is used to minimize friction, an anti-friction plate is needed as a shift limiting agent. A soft, blunt, broken off pivot tip, or a soft, pitted, or coarsely finished anti-friction plate can be a bad source of friction.

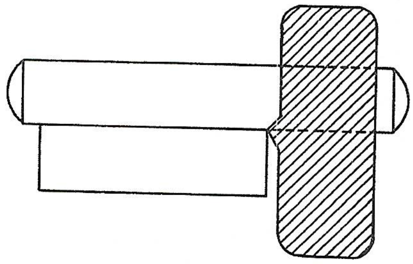

Figure 1.60. Proper

Figure 1.58 shows a pivot with a broken off tip. The protruding point "A" will describe a circle on the anti-friction plate as it turns in its bearing, causing considerable friction.

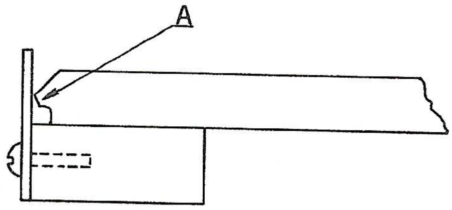

Figure 1.58. Broken Tip

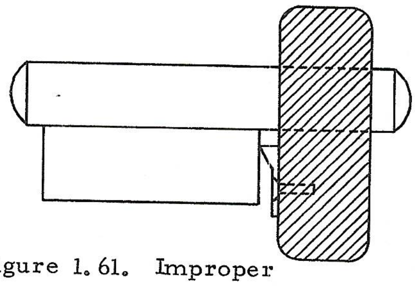

Figure 1.61. Improper

The other methods of minimizing friction use the body of the lever or a hardened steel plate screwed on to it, where the pivot enters the lever. This method may also be a source of avoidable friction.

Figures 1.59 and 1.60 indicate the proper placement of the points of friction. Figures 1.61 and 1.62 illustrate the improper placement.

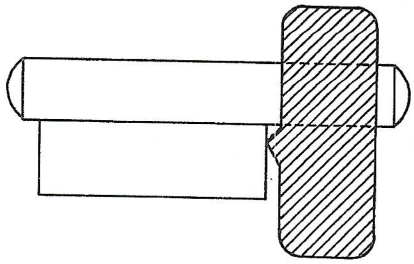

Figure 1.62. Improper

As may be seen on Figure 1.61 the protruding point of the lever body beneath the pivots edge will hit the surface of the bearing below the fulcrum point, and as the lever turns, it will describe a circle on the surface of the bearing; this increases friction.

The third type of friction minimizing is generally used on beams. The points of friction are little bits of steel, placed under the edge of the pivot, protruding past the body of the lever or beam.

Figure 1.63 shows a properly placed friction steel, while Figure 1.64 shows one that is improperly set. As may be seen on Figure 1.63 the friction steel meets the bearing exactly below the edge of the pivot; thereby minimizing friction. In Figure 1.64 the friction steel is placed slightly below the edge of the pivot, resulting in increased friction when it is pushed against the bearing.

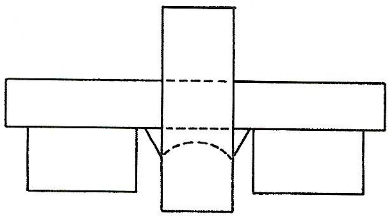

Figure 1.63. Proper

The meeting point of the two tapering sides of a round bottom "V" bearing (Figure 1.42) should never be sharp. It should be rounded off slightly. The reason for this is that in such bearings the pivot can change its position, and might not be exactly above the meeting point of the two tapered surfaces, and undue friction may develop.

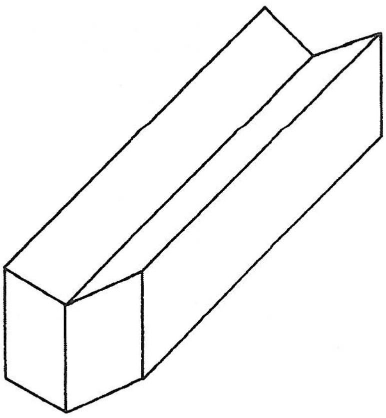

Figure 1.65. Correct

In many cases the ends of the bearings are tapered to a point in order to minimize friction when they are pressed against the surface of the lever's body. In such cases, we must make sure that the surface of the lever is smooth and that there are no cavities below the pivot edge; otherwise, the tapered point of the bearing might dig into such a cavity and cause friction that could be avoided.

The meeting point of the two tapering surfaces should always be at the center of the "V" bottom as shown in Figure 1.65. It should never be off to the side, as on Figure 1.66.

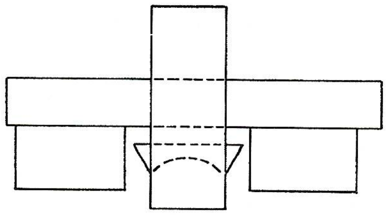

Figure 1.64. Improper

Bearings with tapered surfaces should never be used when friction points are provided on the lever or beam, because the two points might clinch if it should happen that they do not meet perfectly point to point.

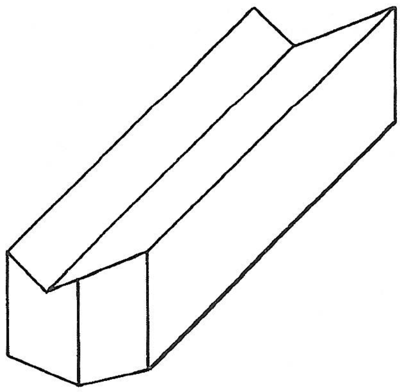

Figure 1.66. Incorrect

A crooked pivot will also cause tension and friction. A simple experiment will prove this to be true. It is impossible to stand a thin, straight, strip of steel on its edge. It will easily fall over on its either side. On the other hand, if this strip is slightly curved or bent, it will stay in an upright position and a slight push will be necessary to make it fall over on its side. The crooked pivot will do the same thing. It will resist against being turned. A soft pivot will also cause friction, because it will become dull very quickly and a soft metal has a clinging tendency.

The same applies to soft bearings. The hard and sharp edge of the pivot steel will cut its way into the soft bearings, and it will not be able to turn freely.

There are many other sources of friction in all scales, such as the steelyard rod rubbing against something, or the platform rubbing against the frame of the scale, etc., but if the following rule is kept in mind, they can be easily eliminated. Keep all moving parts at a safe distance from stationary parts.

When assembling or installing a scale, care should be taken to see that all loops, connecting links, and connecting rods are hanging plumb. It would never do to have them hang in the manner shown on Figures 1.67, 1.68 and 1.69 because the applied load would tend to force them to a vertical position; thus jamming the bearings against the friction points.

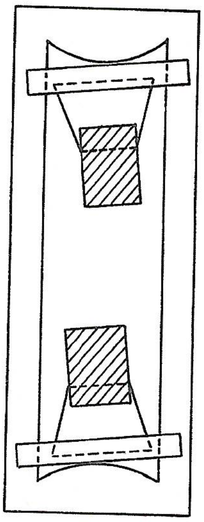

Figure 1.67. Improper Hanging

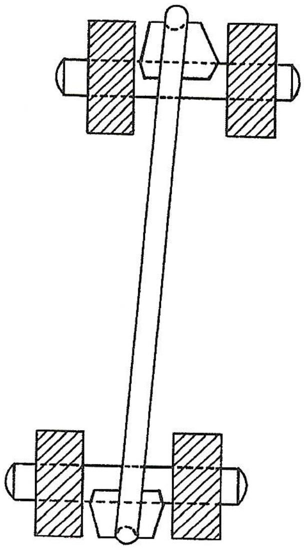

Figure 1.68. Improper Hanging

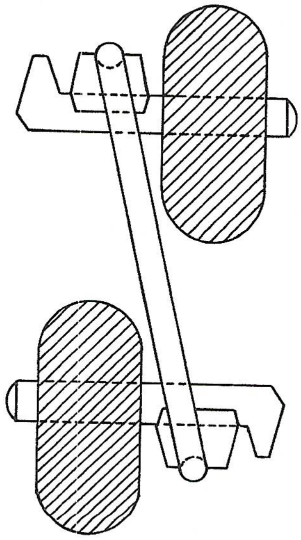

Figure 1.69. Improper Hanging

There are several other sources of friction in dials, and other automatic or semi-automatic scales.

All ball bearings must be absolutely clean, smooth running. Rusty, dirty, and worn bearings will create considerable friction.

To test a ball bearing, it should be removed from its socket. Squeeze the bearing on the end of a pencil and spin the outer ring. If it is noisy when held close to the ear, or comes to a sudden stop, it is either rusty, dirty or worn.

Pour some benzine or pure alcohol in a clean dish. Soak the bearing in the liquid for a while, and then squeeze it on the pencil again. Spin outer ring a number of times in the liquid and test again. If it comes to a slow gradual stop, and no irregular noise is heard, the bearing was only dirty and may be reused. Now take a clean screw driver and dip it into watchmakers oil. Let one drop fall into each bearing for lubrication and rust prevention.

When the bearing is reassembled, care should be taken to allow sufficient end play for the shaft to insure free action. This end play must not be too great, especially in the "Weightograph" pendulum and chart assemblies, because it will affect the clearness of the reading on the ground, glass screen, due to the fact that the chart moves between condensing and focusing lenses, and too much end play for the shaft will have a tendency to cause a change in the focus, resulting in an alternating clear or blurred vision.

All pin screws and pins should be smooth and clean. Even so, they are sources of considerable friction, because of the surface their circumference presents. The same thing applies to the bearings or holes through which they pass.

Pin screws, pins, and holes should be cleaned with a fine emery cloth and finished off with crocus cloth. A drop of fine oil will prevent rust and help decrease friction.

1.21 DASH POTS

A poorly made and poorly finished dashpot can ruin the best scale; therefore, the inner surface of the dashpot wall and the outer rim of the plunger should be polished to the highest possible degree.

There are many dashpots installed in scales with coarse walls, unfinished, unpolished after the lathe cutting tool. In some cases they are polished while on the lathe, with an emery cloth, but not to a fine finish. This is not good enough. Generally plungers have very little clearance and the levers on which they are suspended, describe an arc; therefore, it is almost impossible to prevent the plunger from leaning against the wall of the dashpot at one time or another.

Figure 1.70 illustrates the grooves on the dashpot wall when it has been polished on a lathe. Of course the cuts are exaggerated, but even with much finer cuts, it will introduce unnecessary friction, unless it is polished to a mirror like finish.

To forestall this possibility, the walls of the dashpot should be polished vertically. This can be accomplished with a short length of pipe, round steel, or wood. The diameter should be about 1/8" smaller than the inside diameter of the dashpot.

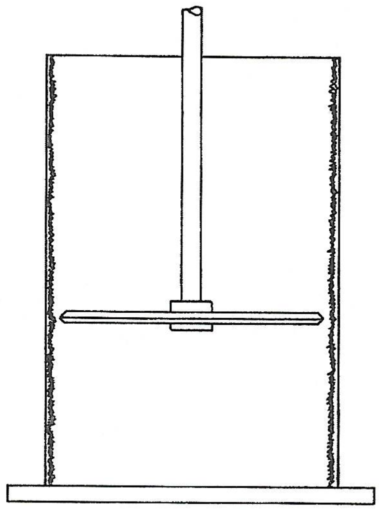

Figure 1.70. Polished Dashpot Wall

Grip one end of the pipe or wood block in a vise. Wrap a fine grade emery cloth around the whole circumference and slip the dashpot over it. Move dashpot back and forth with a piston like motion, maintaining a steady and firm downward pressure, while slowly turning it around. The back and forth motion should be quick, short, and to the bottom of the dashpot.

This should be done until all cross cuts have disappeared. This accomplished, the emery cloth should be replaced with a crocus cloth and the procedure repeated. The final finishing touch may be made with the cloth side of the crocus cloth up.

When the polishing has been completed, the dashpot should be thoroughly washed out with gasoline or kerosene. It is not advisable to wipe the pot out. Install and fill it with oil.

A dashpot polished in this manner will give a much better performance, because all possible bumps and cross cuts have been eliminated, and because minute though they may be, the vertical cuts diminish the area of surface contact, as seen on Figure 1.71. Of course the cuts on Figure 1.71 are also greatly exaggerated.

The rim of the plunger must be highly polished. The plunger under Figure 1.72 is not good, because of the wide, flat surface of its rim.

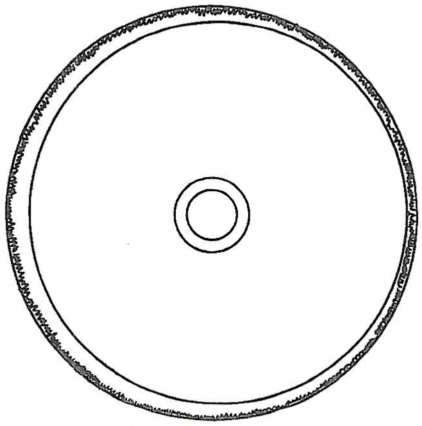

Figure 1.71. Eliminated Bumps & Crosscuts

1.22 INDICATING DEVICES

1.23 THE BEAM

The beam of a scale is the final lever in its lever system. It is the indicating element. In most cases the beam is a first class lever. Occasionally a second class lever may be used for a beam. There are still a large number of earlier model abatoir (overhead rail) scales in operation. The beams of these scales, are ordinary beams with a second class application. The usual load loop is anchored and used as an up-pull bearing for the load pivot now acting as a fulcrum. The power of the understructure is actually the load of the beam and is applied to the pivot that would be normally used as the

Such a plunger can be easily corrected, even in the field, by inserting the stem of the plunger into the chuck of an electric hand drill, drill press, or lathe; and by first rounding the corners off with a smooth file, then finishing it with emery and crocus cloth.

Finally, dirty, sticky, and worn racks and pinions will also cause friction and change.

Excessive and avoidable friction is one of the major causes of inconsistency (changes). There are other equally important causes of inconsistency and these will be explained elsewhere in this manual.

All of this may seem like a great deal of work, and the question of profit may arise, due to the time factor involved. Actually, careful workmanship is a great time saver when the scale is being adjusted and a great money saver because "call backs" are eliminated.

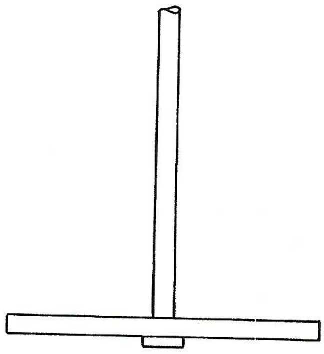

Figure 1.72. Incorrect Plunger

fulcrum. In this application this pivot would become an up-pull load pivot.

The beam illustrated by Figure 1.73 is a normal first class lever. It has two power arms. One of the power arms is the distance between the fp and the pp, and the other is the run of the poise.

The "poise run" of a beam is the distance a poise weight travels from zero to full capacity position.

The distance between the fp and the zero graduation has no bearing on the ratio of the poise run. This distance affects only the empty balance of the scale.

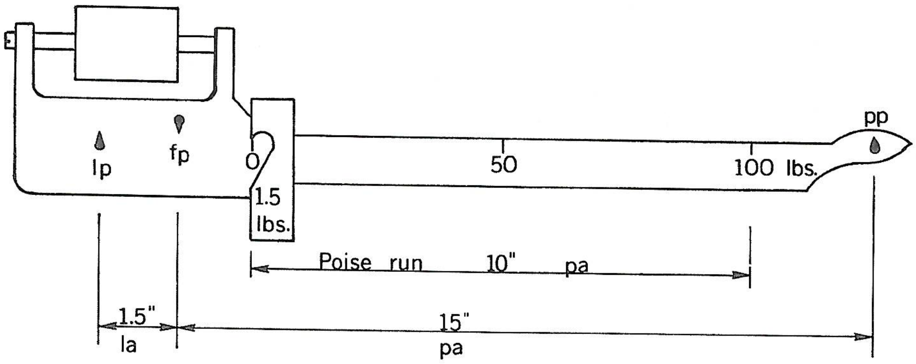

Figure 1.73. First Class Lever Beam

The poise run is a power arm with a variable ratio. As the poise is moved along, the various positions represent various ratios or multiples. The poise at zero position is balanced with the rest of the scale structure. An equilibrium is established. As soon as it is moved away from the zero position it becomes one of the power factors of the scale.

The power arm between the fp and the pp is not variable. The counterpoise that is hung from the pp is part of the scale structures balance. Any weight placed on the counterpoise becomes a power factor. To summarize the preceding: Any and all counterpoise weights or the poise moved away from its zero position becomes the applied power of the scale.

The load (L) that is applied to the load pivot (lp) of the beam is actually the power (P) pressure of the understructure. When applied to the load pivot it becomes the acting load of the beam.

Assume now that the beam illustrated by Figure 1.73 has an understructure with a total multiple of ten. The poise run is ten inches long and represents one hundred pounds. The load arm (la) is 1.5 inches long. The weight of the poise (P) is not known.

If the full capacity value of the poise run is 100 lbs., then it represents a 100 lb. load on the platform. This 100 lb. load is reduced to a 10 lb. power pressure by the 10:1 ratio of the understructure. This 10 lb. power pressure becomes the actual load of the beam. There are now three known factors, the la, the pa, and the L.

$$\frac{L \times la}{pa} = \frac{10 \times 1.5}{10} =$$

1.5 The weight of poise in pounds.

In other words the weight of the poise would have to be 1 pound and 8 ounces.

Figure 1.74 is a compound beam with five power arms. Each power arm and its poise has to be individually calculated and adjusted. The procedure is the same as described for the single beam. During the course of poise adjustments, care should be taken to adjust poise "A" first before attempting to adjust poise "B". The reason for this is that the poise "A" is an integral part of poise "B". Any changes made to poise "A" will also alter the total weight of "B". Poises "C" and "D" are independent of each other.

A complete and detailed description of adjusting procedure for a beam will be found in a section dealing with repairs.

1.24 DIAL INDICATORS

1.25 THE CAM AND THE PENDULUM

Dial scale charts are manufactured with evenly spaced graduations for two reasons. First, because it is much easier to construct an evenly graduated chart than an uneven one. Second, because it is much easier on the eye. It is easier to read and less confusing.

On an evenly graduated chart the weight indications are linear. The distance between the 0 and the 20 lb. graduation is twice as great as the distance between 0 and 10 lbs., and ten times greater at 100 lbs.

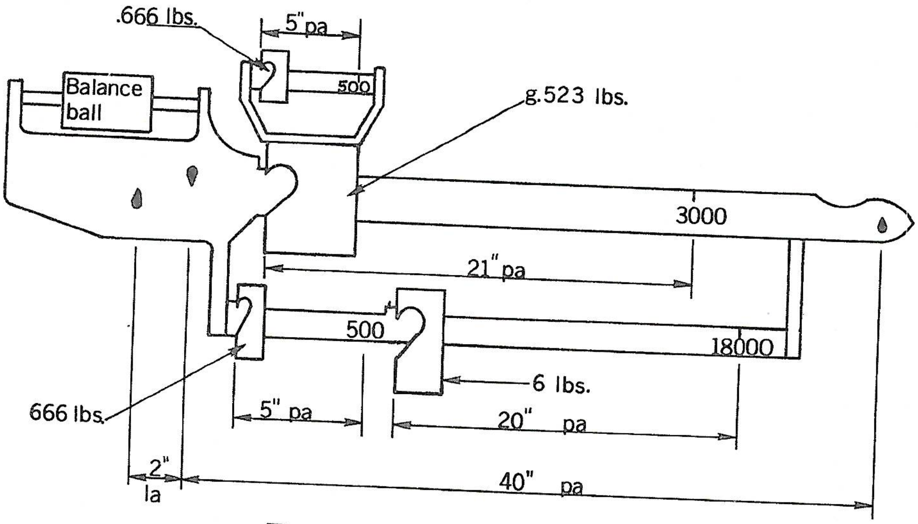

Figure 1.74. Compound Beam

If the chart graduations are linear, then it stands to reason that the travel of the indicator must be linear too. In other words, if our indicator travels 1/8" when 10 lbs. are placed on the platform, then it must travel ten times as much when the load is 100 lbs.

To be able to achieve a linear indicator travel, it is necessary to use a cam for the load arm that activates the pendulum.

The pendulum has a multiplying factor on the horizontal plane, like a poise traveling on a beam, but it is not multiplying on the arc on which it travels, as illustrated by Figure 1.75.

Assuming that the load arm of the pendulum assembly is constant, like a concentric disc with a tape attached to its outer surface (Figure 1.75), it would mean that no matter what position the pendulum takes, the load arm would remain the same.

When four equal units of weight are placed on the load pan consecutively, the pendulum will move four consecutively equal distances on the horizontal plane as indicated by distances a, b, c, and d. The distances a, b, c, and d are the vertical projections of the pendulum positions. On the other hand, these same projections indicate unequal distances on the arc. For instance, distance "e" is much longer than distance "h".

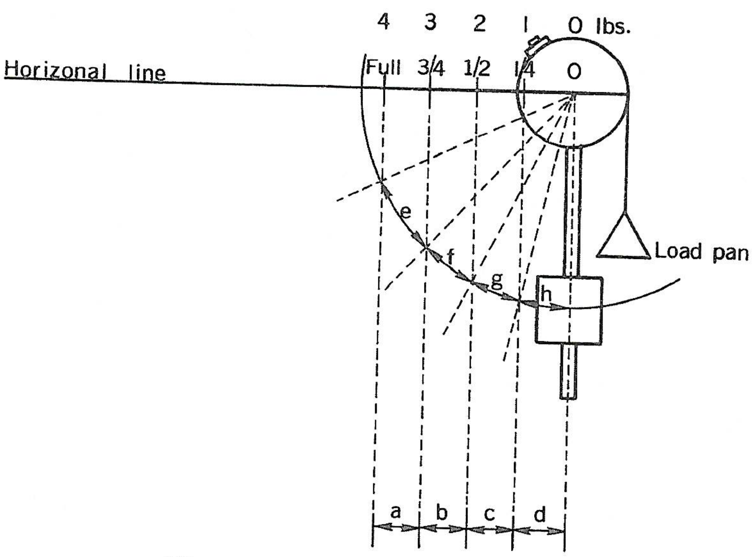

Figure 1.75. Dial Scale Chart

In order to be able to use an evenly graduated (linear) chart, it stands to reason that the indicator travel has to be linear also; therefore, it is essential that the pendulum travels equal distances on the arc, because it is this circular travel of the whole pendulum assembly that is finally transmitted to the indicator with the aid of a rack and a pinion, or a tape driven transmission. For this reason, the distances e, f, g, and h must be equal, as illustrated by Figure 1.76.

When the distances e, f, g, and h are of equal length it will be found that the vertical projections of these distances are unequal, as indicated by a, b, c, and d. These varying distances on the other hand give us the true picture of the necessary variable multiplying factor that will produce a linear travel on the arc.

The combined length of a, b, c, and d is the variable acting power arm of the pendulum at full capacity.

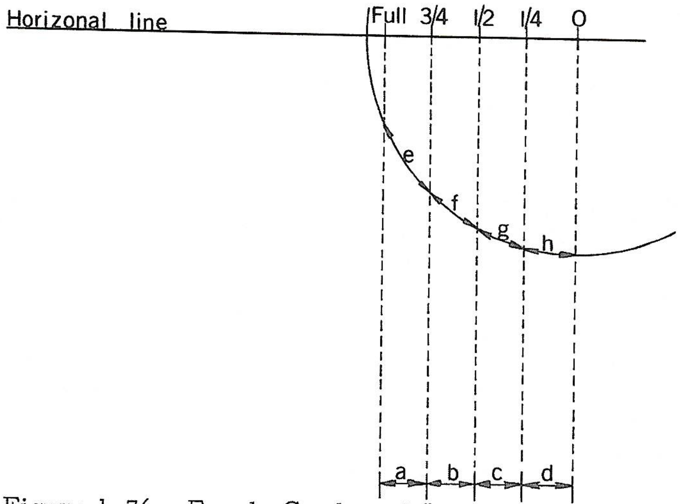

Figure 1.76. Evenly Graduated Linear Chart

Having established that it is necessary to have a decreasing tendency in the power arm in order to insure a linear travel on the arc, consequently it is also necessary to have a varying load arm with a decreasing tendency. In other words, the acting load arm must shorten as the pendulum swings toward capacity. This gradually shortening "acting load arm" cuts down the force exerted against the pendulum by the applied load, and as a result, the pendulum will not travel as much on the horizontal plane toward capacity as it did in the zero vicinity.

To be able to accomplish this, the concentric disc will have to be replaced by an eccentric one; or in other words, a cam will have to be used for the load arm.

Figure 1.77 illustrates a cam action with an eccentric disc. The solid lines of Figure 1.77 indicate the zero position of the cam and pendulum. The broken lines indicate the full capacity position. Note that the distance "a" is longer than distance "b". The distances "a" and "b" are the acting load arms at zero and full capacity.

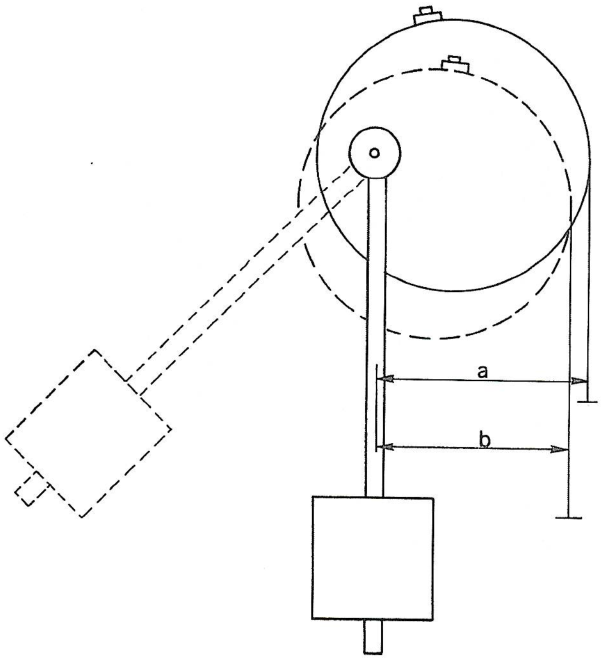

Figure 1.77. Cam Action

As the pendulum swings forward under the power pressure of the connecting lever system acting as load on the cam tape, the acting load arm of the cam gradually shortens; thus decreasing the force of the load. As the force of the load decreases, due to the variable acting load arm of the cam, the pendulum assembly will also move forward less and less on the horizontal plane. The projection of this movement is indicated by a, b, c, and d of Figure 1.76. The distances of e, f, g, and h on the other hand indicate that the travel is linear on the arc. This circular movement is then stepped up with the aid of a rack and pinion, or tape driven transmission from 45 to 355-360 degrees.

The concentric disc of Figure 1.75 does not produce the effects illustrated by Figure 1.76. On the other hand, Figure 1.76 only illustrates the desired effect, but not the method of achieving it.

The concentric disc will produce only the effect as illustrated by a, b, c, and d, and e, f, g, and h of Figure 1.75, which is a linear travel on the horizontal plane and a non-linear travel on the arc.

The distance between the fulcrum and the point where the tape departs from the cam surface is the acting load arm.

The most commonly used cam is the eccentric disc, turned on a lathe, with a hole bored off center to accommodate the fulcrum shaft or pivot of the pendulum assembly. Often only a section of a circular disc is used for a cam with an eccentric fulcrum.

Various methods are used to adjust the eccentricity of a cam that is made of a section of a circle. In some cases the cam section can be moved up or down along finely machined grooves with the aid of adjusting screws, thus changing the eccentricity of the cam.

In other cases a full disc is used. It is made eccentric by boring the hole off center.

A great variety of cams and methods for their adjustments are in use, but the principle is always the same.

Cams made of eccentric circles have certain limitations. They can be used only when the travel of the pendulum is not intended to exceed 45 degrees.

Any pendulum action beyond 45 degrees requires a section of a sine hyperbolic spiral for a cam. These cannot be turned on a lathe, but require very precise mill work.

In conclusion, the principles covered in this section comprise the basis for all mechanical scales, whether a small fan scale or a large railroad track scale. A thorough understanding of these principles and their application can simplify servicing scales.