SECTION 4

SUMMARY: A SIMPLIFIED NETWORK

Now that all of the basic mechanisms have been described individually, it is time to see, in a general way, how they work together to solve a fire control problem.

In order to avoid the complexities of any actual computer, Section 4 shows how an imaginary, simplified network of mechanisms might be put together. This network is sufficiently similar to the networks in actual computers to show how mechanisms can be connected to solve mathematical problems mechanically.

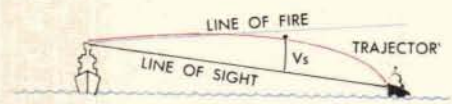
The imaginary network is also sufficiently realistic to demonstrate the general procedures for setting any network of mechanisms.

	Page
A Simplified Network	300
Setting the Network	310

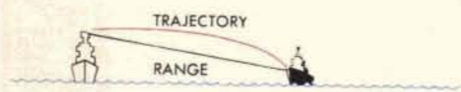
A SIMPLIFIED NETWORK

Each basic mechanism in a computer does one particular job. These mechanisms are combined into networks. Each network solves mathematical equations to arrive at a value that is needed in solving the fire control problem.

In this section an imaginary simplified network is described in order to show how the basic mechanisms are combined into computing networks and how such networks are set. This simplified network is not taken from any actual instrument but was made up specially for this section in order that certain principles could be shown more easily.

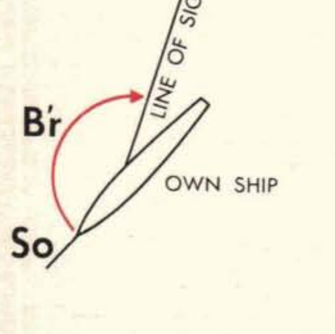


The network roughly approximates a solution of part of the fire control problem for a surface target. It finds Sight Angle, *Vs*, the amount that the guns must be elevated above the Line of Sight to allow for the drop of the shell. This drop is greater for long ranges, which makes it necessary for the guns to be elevated more for long ranges than for short ranges.



Since the range is changing as the ship and target move, and since the shell takes an appreciable time to reach the target, the value of sight angle used in firing the guns must be chosen for the value of range at the end of the time of flight of the shell, that is, Advance Range.

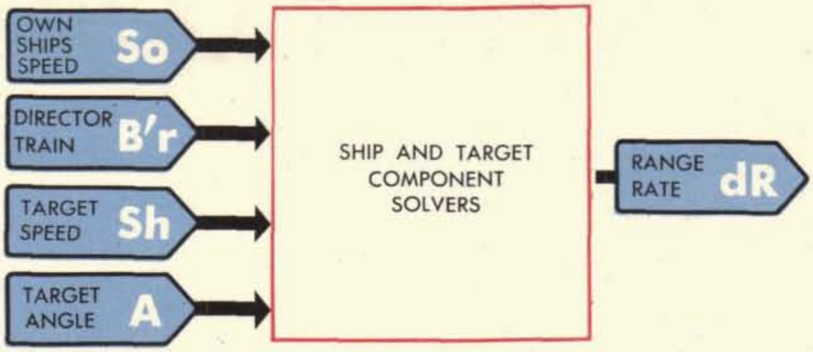
Advance Range in this problem is the sum of the Present Range and the predicted change in Range caused by relative motion between Own Ship and the Target, during time of flight of the shell.



In order to find Sight Angle, *Vs*, these six quantities are used as inputs to the network:

1. Own Ship Speed, *So*, which is assumed to come in automatically from a pitometer log, by synchro transmission.
2. Director Train, *B'r*, which could come in automatically from a director, also by synchro transmission.
3. Target Speed, *Sh*, which might be estimated at the director and be phoned down to the Computer and set in by hand.
4. Target Angle, *A*, which might also be estimated at the director and be phoned to the Computer and set in by hand.
5. Initial Range setting, *jR*, which could be phoned down from the Director and set into the Computer by hand.
6. Time, *T*, which is put in automatically by the time motor.

Component solvers figure RANGE RATE

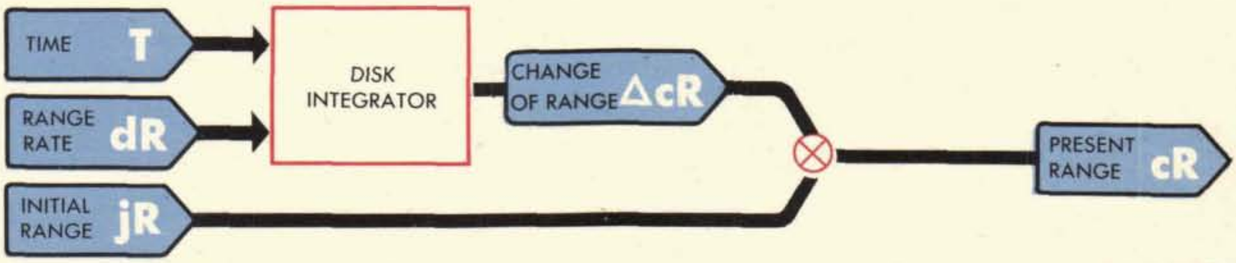


The first thing to do is to find out the rate at which the range between own ship and target is changing. This depends on the speed of both ships and their direction of travel in relation to the line of sight.

Own Ship Speed, Director Train, Target Speed, and Target Angle, are inputs to two component solvers.

The sum of the components of Ship Speed and Target Speed *along* the Line of Sight is the Range Rate, *dR*. Range Rate is the *rate* at which Present Range is changing.

An integrator finds PRESENT RANGE

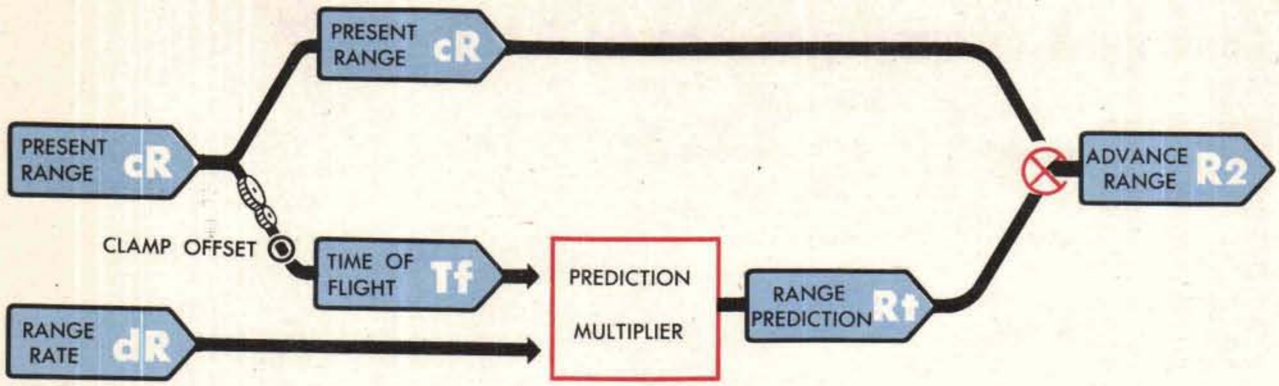


Range Rate, *dR,* now becomes an input to a disk integrator, together with Time, *T*, from the time motor.

The Time Motor drives the disk at a constant speed, and *dR* positions the carriage to control the speed of the output roller. The output drives one side of a differential in the range line and changes the value of Range in the machine as the actual range between the Ship and Target changes.

At any time after the start of the problem, the output of the integrator will have turned an amount corresponding to the amount that Range has changed since the problem started. The output of the integrator is referred to as "Generated Changes of Range," $\triangle cR$. Generated Changes of Range, $\triangle cR$, are continuously added in a differential to Initial Range Setting, *jR,* to give the computed Present Range, *cR,* at every moment.

PRESENT RANGE



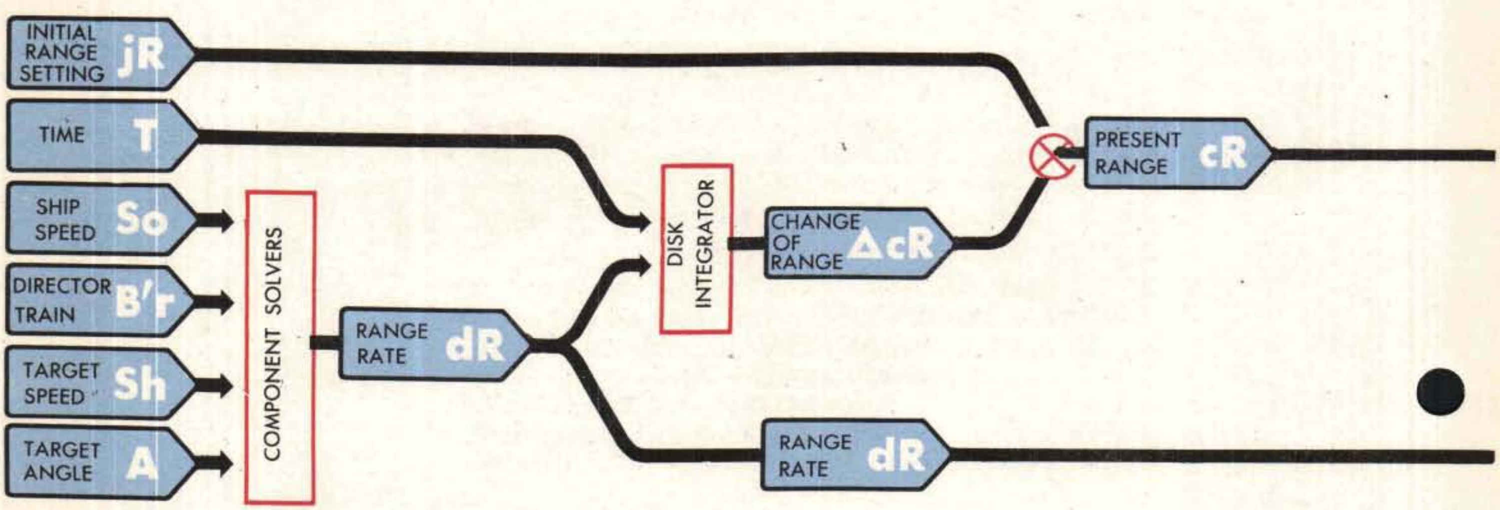
If Present Range, cR, were used for firing the guns, the shell would not hit the target because the Target and Own Ship are moving while the shell is in flight. This change in range during the time of flight is Range Prediction, Rt.

In order to simplify this network, it has been assumed that there is a fairly constant ratio between range and time of flight for any particular shell. A gear ratio is used to multiply the range by a constant. A clamp offset is used to add another constant to the result to give a rough value of Time of Flight, Tf, for any given value of Present Range.

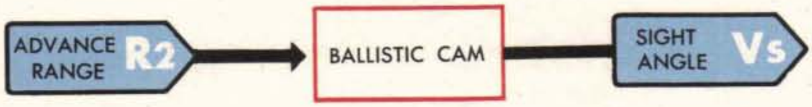
In this network, $Tf = K(cR) + c$

Time of Flight, Tf, multiplied by the rate at which range is changing results in Range Prediction, Rt. Tf becomes an input to a multiplier together with Range Rate, dR. The output is Range Prediction, Rt.

The Present Range, cR, added to Range Prediction, Rt, equals the Advance Range, $R2$. $R2$ in this network is the range at the end of the time of flight of the shell.



COMPUTING SIGHT ANGLE



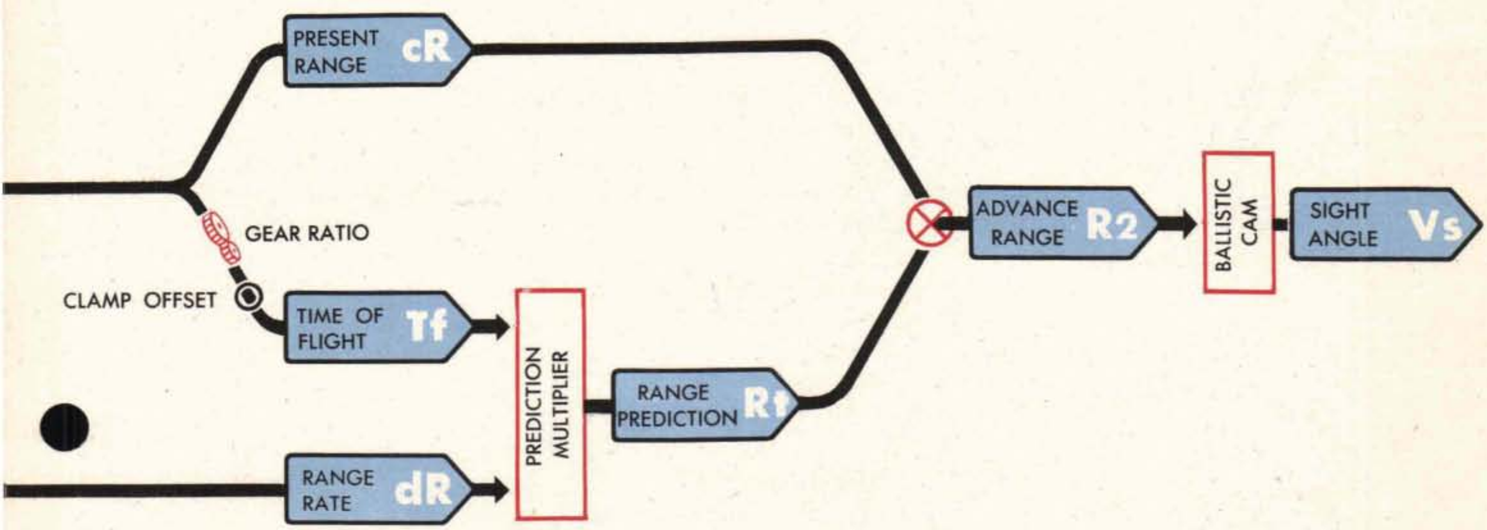
Now Advance Range, *R2*, goes to a flat ballistic cam. This cam is designed so that its output is the Sight Angle, *Vs*, for whatever range is set into it.

The value of *R2* is the input, the output is the value of *Vs* for that value of Advance Range.

THE WHOLE NETWORK

Here's what the whole imaginary network looks like.

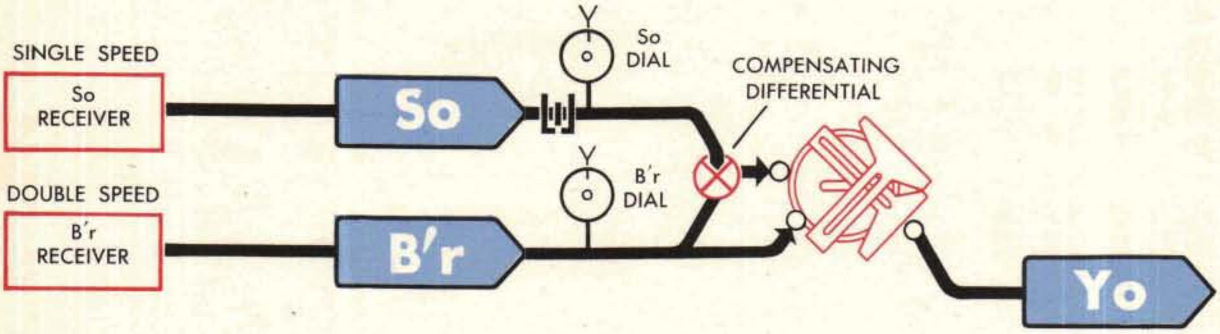
Notice that as the inputs are introduced at the start of the problem, the quantities feed through the mechanism and a corresponding sight angle is immediately computed. Then as the ship and target move with respect to each other, and the inputs to the component solvers change, the network continuously computes a changing value of sight angle.



How the MECHANISMS solve the problem

Here's the same problem of finding sight angle. But in working it out this time, just what happens in each mechanism is shown.

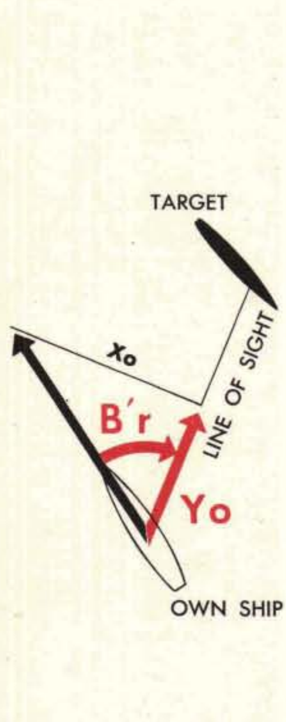
OWN SHIP component solver



Ship Speed, *So*, is assumed to be received electrically from the ship speed transmitter in the pitometer log. The transmitter sends an electrical signal to the *So* receiver and positions its rotor.

The receiver positions the *So* shaft. This shaft positions the *So* dials and the cam of the Ship Component Solver.

There is a limit stop on the *So* line. The limit stop prevents the receiver motor output shaft from jamming the cam follower pin into the end of the cam groove if it overruns the signal.



The fine and coarse motors in the *B'r* receiver accurately position the *B'r* shaft and the vector gear of the component solver. *B'r* also positions the *B'r* dial. In this network *B'r* is not corrected for the effect of deck inclination.

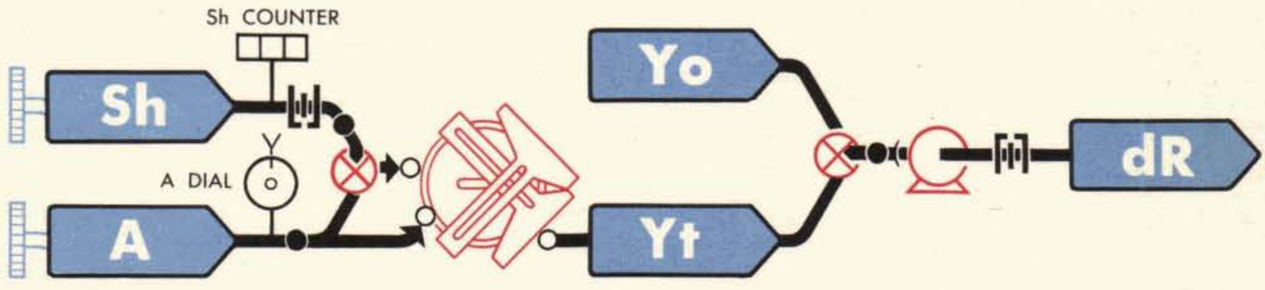
There is a compensating differential on the cam input line of the component solver which prevents any rotation of the vector gear from affecting the radial position of the pin. The compensating differential is explained in the chapter on Component Solvers.

The component solver is positioned by Ship Speed, *So*, and Director Train, *B'r*. The cam and vector gear position the pin and the two output racks. The racks position the output gears.

The only output of interest in this problem is *Yo*, the component of Own Ship Speed *along* the Line of Sight. The other output goes to another network.

$$Yo = -So \cos B'r$$

TARGET component solver



Target Speed, *Sh,* is phoned down from the director. It is set into the Computer by a hand crank. The crank turns a shaft that positions the cam of the Target Component Solver and the Target Speed Counter.

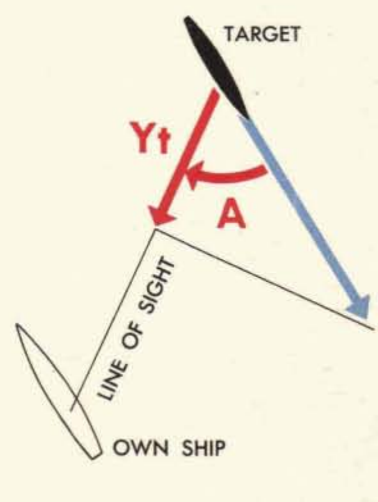
If the *Sh* hand crank should turn past the zero position of the cam, the cam follower would be jammed against the end of the cam groove. A limit stop is put on the *Sh* line to prevent this damage.

Target Angle, *A,* is also phoned down from the Director and is set in by hand crank. The crank turns a shaft that positions the vector gear of the Target Component Solver and the Target Angle Dial.

There is a compensating differential on the cam line.

The output of interest here is *Yt,* the component of target speed *along* the line of sight. The other output goes to another network.

$$Yt = -Sh \cos A$$



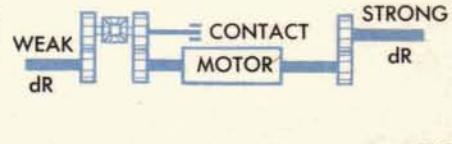
Now the outputs from the component solvers are combined. The sum of the components of ship and target speed, *along* the line of sight is the range rate.

$$dR = Yo + Yt$$

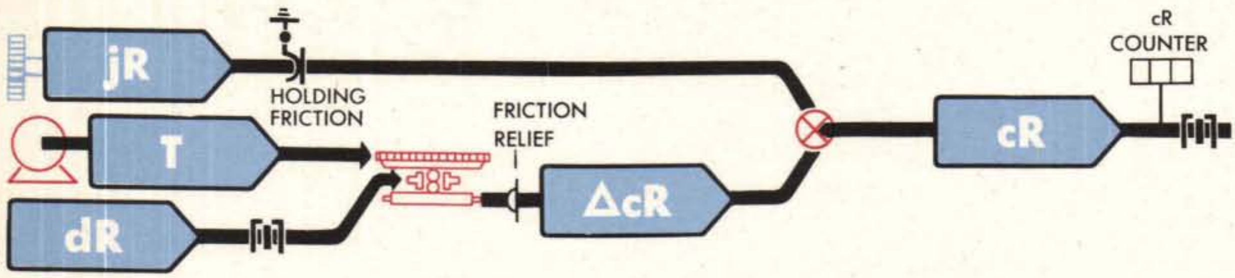
The *Yo* output from the ship component solver positions one side of a differential. The *Yt* output from the target component solver positions the other side. The output on the spider is the sum of the two inputs, Direct Range Rate, *dR,* the rate at which the range between the target and own ship is changing.

The output from the differential is precise, but weak. It has to be made powerful enough to drive the next mechanism in the network. This is done by having the output shaft of the differential drive the input to a servo motor follow-up control.

The *dR* shaft positions one side of the differential in the follow-up. The motor drives the other side of the differential to keep the error shaft on zero and so puts the output line in exactly the same position as the input line. The output from the motor is powerful enough to drive the next mechanism.



The DISK INTEGRATOR



The dR shaft positions the carriage of a disk integrator.

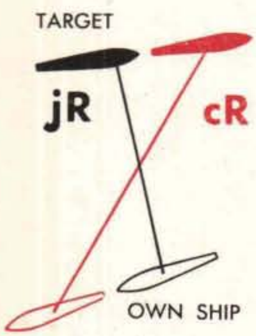
The input to the integrator carriage must be held within the limits of travel of the carriage. Therefore a limit stop is put in the dR line between the follow-up control and the integrator.

The Time Motor, whose speed is kept constant by a motor regulator, turns the disk of the integrator. This input to the disk is Time. The output on the roller represents Generated Changes in Range, ΔcR.

Initial Range Setting, jR, added to Generated Changes in Range, ΔcR, gives Present Range, cR.

$$cR = jR + \Delta cR$$

In this example, initial range is set in the computer by turning the jR handle until the present range counter reads the value of present range which has been phoned down from the director.

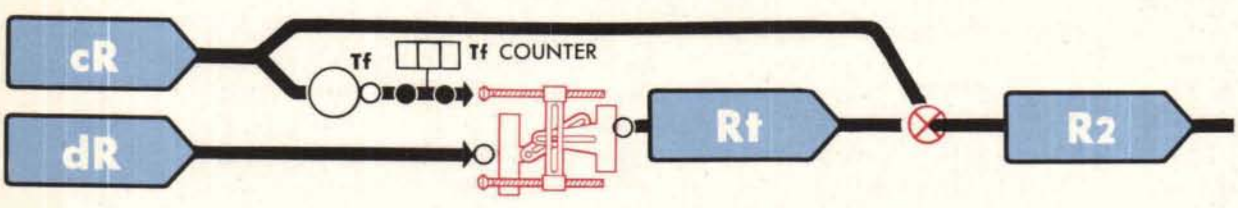


There is a holding friction on the jR line so that as changes of Range are generated, this output of the integrator will drive into the cR line without upsetting the initial setting of the jR line.

There is a limit stop in the cR line to protect the mechanisms driven by that line.

The output line from the integrator has a friction relief. When the value of Present Range reaches its limit, and the integrator output is stopped by the limit stop on the cR line, the friction will slip allowing the integrator roller to turn and so preventing excessive wear in the integrator.

The MULTIPLIER



Time of Flight, Tf, is assumed to be approximately proportional to Present Range. A gear ratio and clamp offset are used to solve this equation:

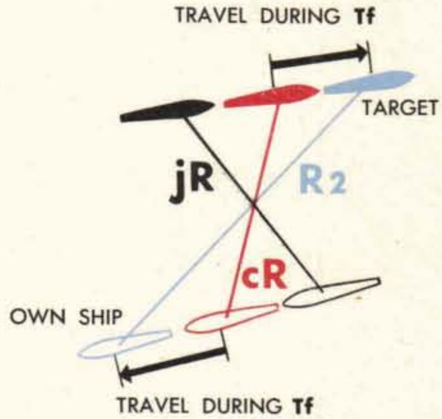
$$Tf = K(cR) + c$$

A gear ratio multiplies Present Range, cR, by a constant, K. The output of the gear ratio is then $K(cR)$.

A clamp offset adds another constant, c. This solves the equation and the result is an approximate Time of Flight, *Tf,* which positions the screw of the prediction multiplier.

The Range Rate, *dR,* coming from the follow-up output line moves the input rack of the multiplier.

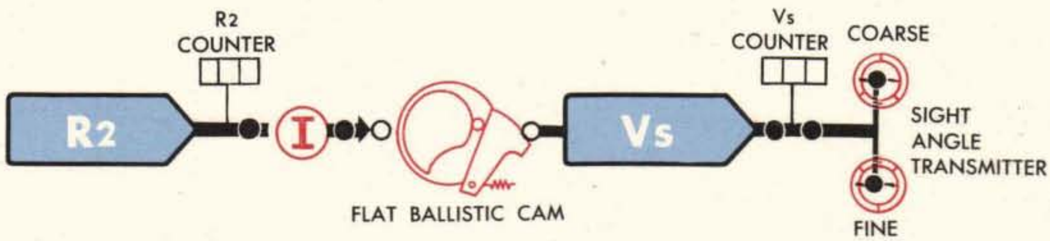
The output from the multiplier is proportional to the Range Prediction, *Rt*. Range Prediction is the change in the range during Time of Flight.



Then the Range Prediction shaft positions one side of a differential. The other side is turned by a branch of the Present Range line. The output on the spider is Advance Range, which is the sum of *cR* and *Rt*.

$$R2 = cR + Rt$$

The BALLISTIC CAM



The *R2* line goes to the ballistic cam. There is an Intermittent Drive in this line because the ballistic cam is designed to handle only range values within certain limits.

The Intermittent Drive cuts out when advance range is below the lower limit or above the upper limit of the cam. As a result the Advance Range shaft on the output side of the Intermittent Drive transmits only ranges within the limits of the cam.

The *R2* shaft puts a reading into the *R2* counter and positions the input gear of the flat ballistic cam. This cam is designed so that for every input of advance range, it gives the correct Sight Angle, *Vs.*

$$Vs = f(R2)$$

As the input gear puts the Advance Range into the cam, the output gear, meshed with the sector arm, is positioned to give the value of Sight Angle for that range.

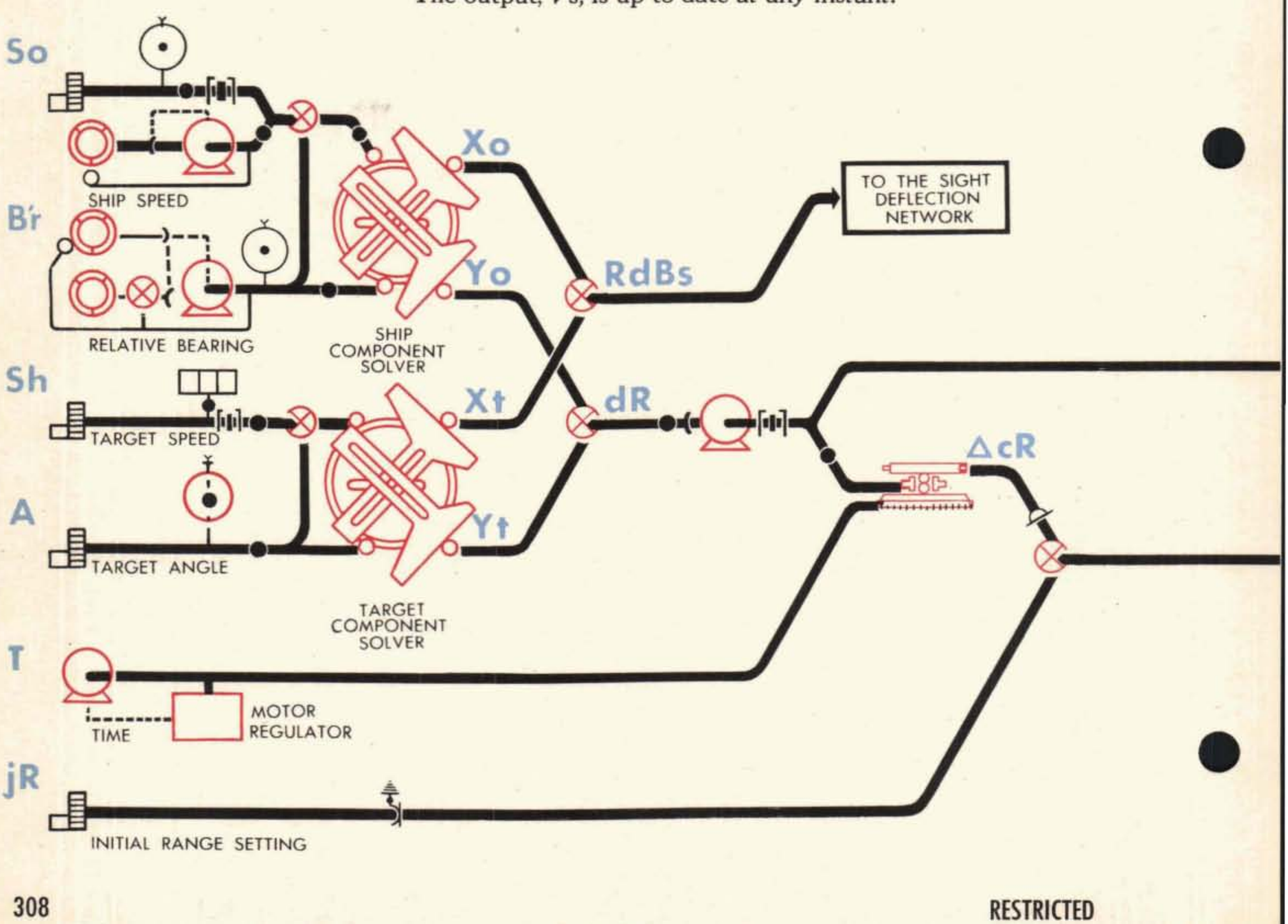
The *Vs* output shaft puts this Sight Angle value into the *Vs* counter and at the same time positions the rotors of the fine and coarse *Vs* transmitter generators. The *Vs* transmitter sends an accurate electrical signal to the synchro receivers in the turrets.

Here's the complete network with the

The inputs to the networks keep changing all the time that the target is being tracked. Each change of the initial inputs to the network immediately changes the values all along the line, including the final output—Sight Angle. In this way the network keeps on computing Sight Angle for the changing conditions.

It should be noted that the output value is *instantaneously* computed. That is, the output always reads the correct value for the inputs in the network at that instant. The time line—through the integrator—keeps present range on the correct value while the value of *dR* is continually corrected by changes in the component solver inputs during tracking.

The output, *Vs*, is up to date *at any instant.*



MECHANISMS shown

Although this is a simplified network, the equation it continuously solves illustrates most of the mathematics needed to solve fire control problems by mechanical means.

$Vs = f\,(R2)$ where $R2 = Rt + cR$
$Vs = f\,(Rt + cR)$ where $Rt = dR \times Tf$
$Vs = f\,[(dR \times Tf) + cR]$ where $dR = Yo + Yt$
$Vs = f\,\{[(Yo + Yt) \times Tf] + cR\}$ where $Tf = K \times cR + c$
$Vs = f\,\{[(Yo + Yt) \times (K \times cR + c)] + cR\}$ where $cR = \triangle cR + jR$
$Vs = f\,\big\{\{(Yo + Yt) \times [K(\triangle cR + jR) + c]\} + \{\triangle cR + jR\}\big\}$

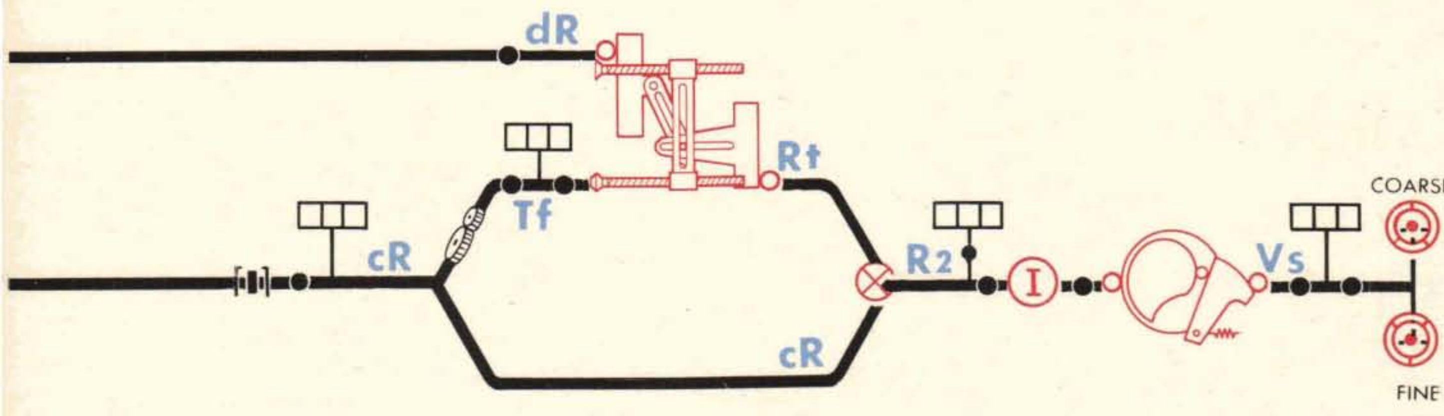
where $\triangle cR$, the accumulated increments or changes of range, is represented this way:

$$\int dR \times \triangle T$$

and $Yo + Yt$ is represented this way:

$$-So \cos B'r - Sh \cos A$$

$$Vs = f\,\big\{\{[-So \cos B'r - Sh \cos A] \times [K(\int dR \times \triangle T + jR) + c]\} + \{\int dR \times \triangle T + jR\}\big\}$$

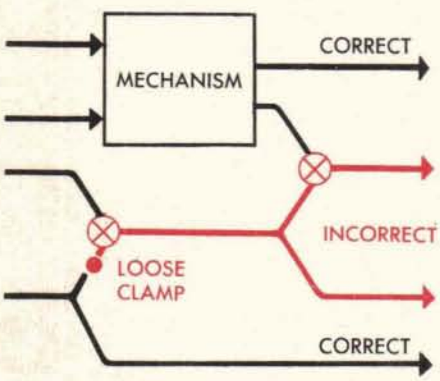


SETTING the NETWORK

A setting method for each basic mechanism considered separately has already been described. Now it is necessary to take into consideration the setting of a *network* of mechanisms. The mechanisms within a network have to be set to each other. Then a network acts as a unit and computes the correct output for the values put in.

In the description of the setting of the basic mechanisms it was assumed that there was a counter on each input and output shaft. In an actual computer this is not so. Counters are placed only where they are needed as reference points.

The order of settings is established to create a "progressive reference point." This means that as each setting is made, its result is used as the basis for the next setting.



THE LINES AFFECTED BY THE LOOSE CLAMP ARE SHOWN IN RED

If any clamp in a network becomes loose, or is improperly set, it upsets the entire network from that point on. All the mechanisms that follow the upset clamp receive incorrect inputs. This acts as a constant offset and introduces an error to all the outputs of the network after the upset clamp. The mechanisms before the upset clamp are not affected.

In setting a network, the input to each mechanism is set to the output of the mechanism before it. In this way, each setting requires that the network up to that point has been correctly set. Putting an error into one clamp to compensate for the error in a previous clamp is to be avoided. The only remedy is to locate the clamp that is upset and correct it.

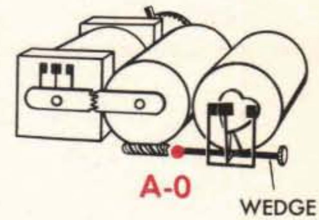
To show the principles of setting a network, these instructions are given for setting the imaginary network that has just been described. The clamp numbers (*A-0, A-1,* etc.) used are for convenience only and do not refer to any specific computer. They were chosen only as examples. ALL adjustment clamps must be loose before starting to set the network.

A-0 COARSE AND FINE SYNCHROS in B'r RECEIVER

Set the fine synchro at electrical zero. Turn the servo output until the fine contacts are centralized, slip-tighten the clamp and wedge the output shaft.

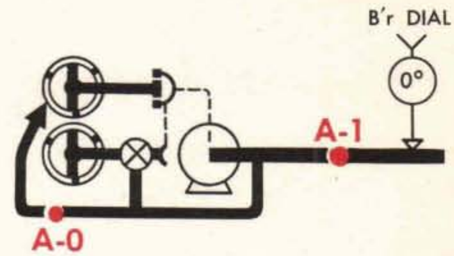
Set the coarse synchro at electrical zero and slip the clamp until the coarse contacts are centralized.

Tighten *A-0* clamp.



A-1 B'r DIAL to B'r RECEIVER

With the receiver output shaft still wedged on 0° set the *B'r* dial to 0°. Tighten *A-1* clamp, and remove the wedge.



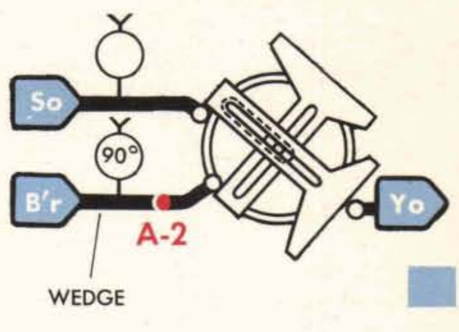
A-2 VECTOR GEAR to B'r DIAL

With *B'r* dial at 90°, wedge the line. Position the vector gear slot parallel to the *Yo* rack slot and pointed away from the *Yo* output. Slip-tighten *A-2* clamp.

Mark the position of the *Yo* output gear on the rack with a pencil line.

Moving the *So* cam should produce *no motion* of the *Yo* rack. This is a rough setting, using a mark on the rack as an indicator. The fine setting is done later on.

Remove the wedge.



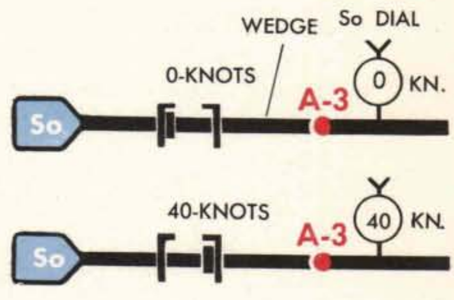
A-3 So DIAL to LIMIT STOP

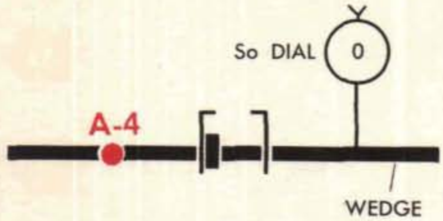
Slip-tighten *A-3* clamp. Turn the *So* line in a decreasing direction to the end of the limit stop. Wedge the line.

Loosen *A-3*. Put the *So* dial at zero knots. Tighten *A-3* clamp.

Remove the wedge.

Run the *So* line until the limit stop block is at the other end. The *So* dial should read 40 knots.



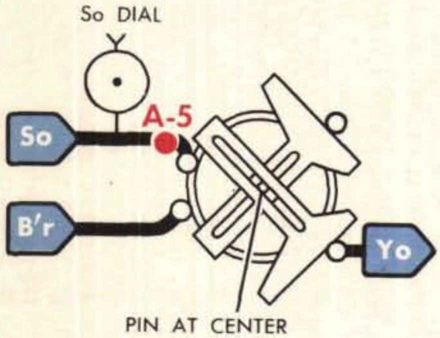


A-4 So DIAL to So RECEIVER

With the *So* dial wedged at zero knots, put the synchro motor at electrical zero.

Synchronize the follow-up by hand with the power OFF. Then energize the servo motor.

Tighten *A-4* clamp.



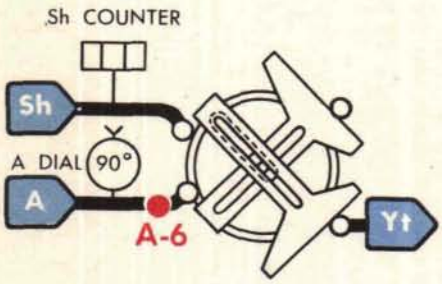
A-5 So CAM to DIAL

Wedge the *So* dial at zero knots. Position the cam so that the pin is in the center. Slip-tighten *A-5* clamp.

Movement of the *B'r* vector gear should produce no motion of the *Yo* rack. This is a rough setting, using a mark on the rack as an indicator. The fine setting is done later.

Now the ship's component solver has been approximately set.

Remove the wedge.

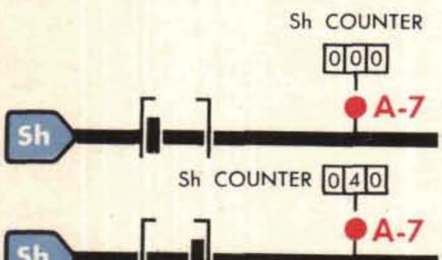


A-6 VECTOR GEAR to A DIAL

Wedge the *A* dial at 90°. Position the vector slot so that it is parallel to the slot in the *Yt* rack and pointed away from the *Yt* output. Tighten *A-6* clamp.

With the *A* dial at 90°, mark the position of the *Yt* output gear with a pencil line. Run *Sh* line. There should be no output on the *Yt* rack. This is a rough setting, using the rack as an indicator.

This setting is refined later.



A-7 Sh COUNTER to LIMIT STOP

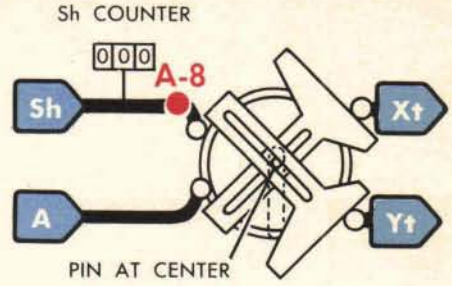
Slip-tighten *A-7*. Run the *Sh* line in a decreasing direction to the end of the limit stop. Wedge the *Sh* line.

Loosen *A-7*. Put the *Sh* counter at zero knots. Tighten *A-7*.

Remove the wedge.

Run *Sh* to the other end of the limit stop. The *Sh* counter should read 40 knots.

A-8 Sh CAM to COUNTER



Wedge the *Sh* counter at zero knots. Position the cam so that the pin is in the center. Slip-tighten *A-8* clamp.

Movement of the *A* vector gear should produce no motion of either output rack. This setting is refined later.

Now both the ship and target component solvers are approximately set.

A-9 dR FOLLOW-UP to the D-I DIFFERENTIAL

Wedge the *Sh* counter and the *So* dial at zero knots. This gives a zero output from the differential because both inputs are zero.

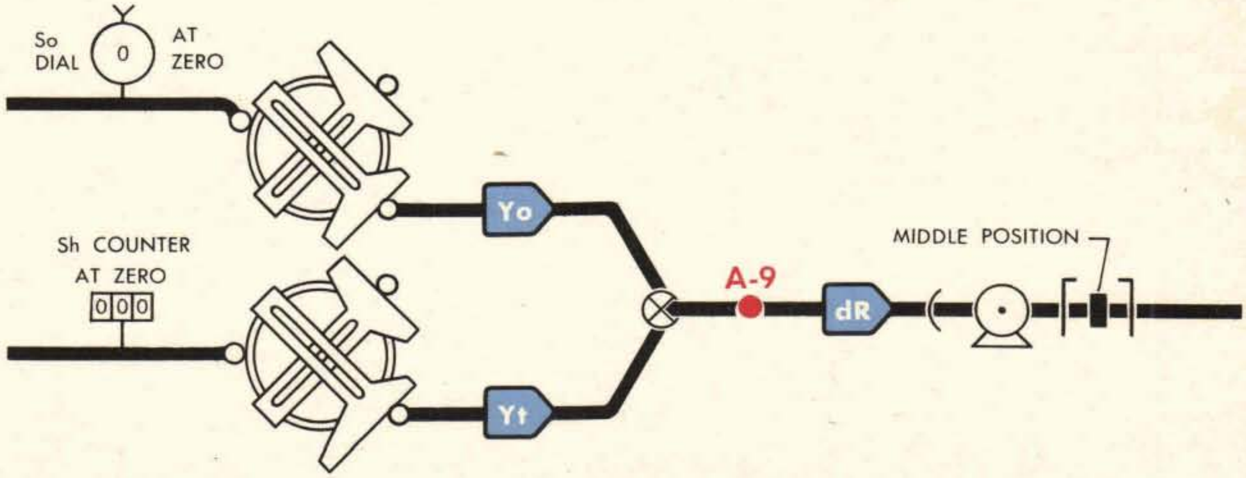
Run the follow-up output to both ends of the limit stop by hand. This is to make sure that mechanisms farther along the line are not upset in such a way that the running of the motor could damage them.

Position the traveling nut of the limit stop on the *dR* line about in the center.

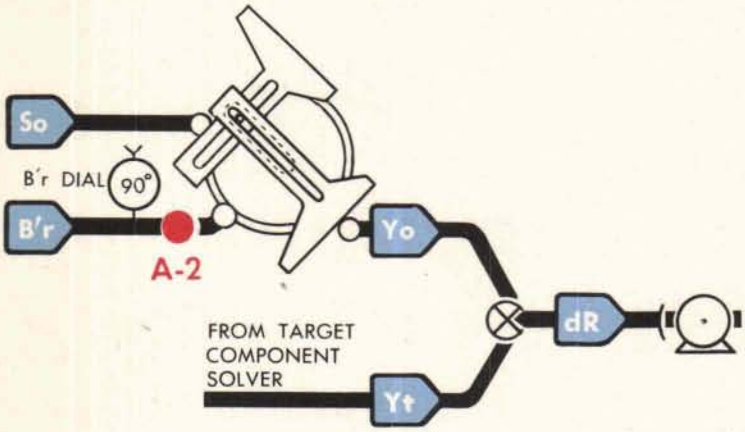
Synchronize the follow-up. When the follow-up is energized, the traveling nut in the limit stop should remain in the middle position. Tighten *A-9* clamp. Remove the wedges.

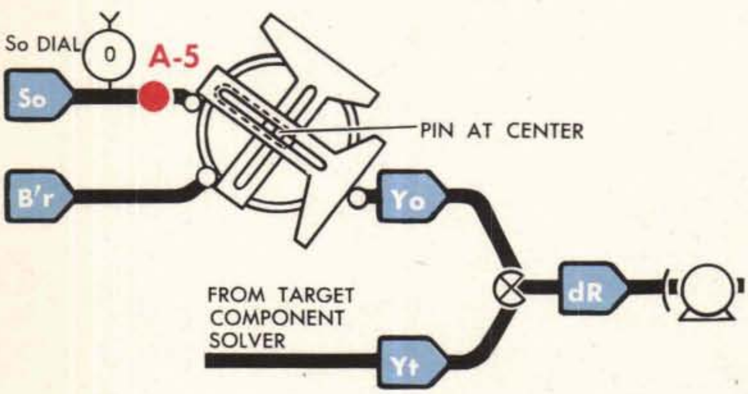
This is a temporary setting. It will have to be done over again after the setting of the component solvers has been refined.

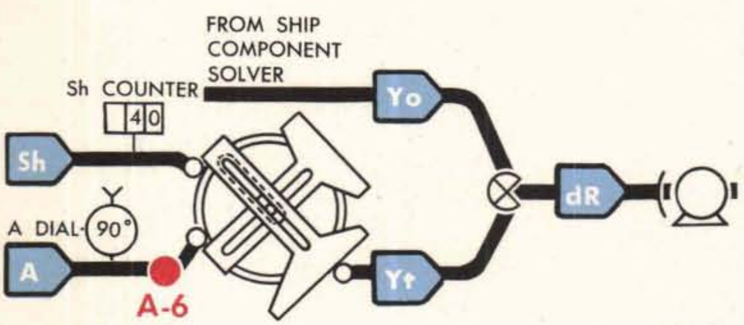
> Another way to set this section of network would be to set the component solvers with a shadow stick or dial indicator. This would give an accurate setting at once. Then the refined settings would not be necessary, and the *A-9* clamp would require only one setting.
>
> If the solvers are approximately set first and then refined, the follow-up setting has to be done twice. However, this gives a more accurate setting because the follow-up acts to remove any lost motion in the mechanisms.



REFINING the SETTING of the COMPONENT SOLVERS







A-2 VECTOR of SHIP SOLVER to B'r DIAL

Set the *B'r* dial at 90°.

Wedge the *Sh* and *A* lines to hold the *Yt* side of the *dR* differential, so that no motion can back out through that side.

Turning the *So* cam with the *dR* follow-up energized should produce no output from the follow-up. If any correction has to be made, it should be done with *So* at 40 knots. Slip-tighten *A-2* clamp. Wedge *B'r* dial and adjust the vector gear until there is no output.

Tighten the *A-2* clamp. Remove the wedge on the *B'r* line.

A-5 CAM of SHIP SOLVER to So DIAL

Put the *So* dial at zero knots. Wedge the *Sh* and *A* lines to hold the *Yt* side of the *dR* differential so that no motion can back out that side. Turning the *B'r* vector should produce no output from the *dR* follow-up. Wedge *So* dial at 0 knots. Slip through *A-5* clamp to adjust the cam position.

Tighten the *A-5* clamp. Remove all wedges.

A-6 VECTOR of the TARGET SOLVER to A DIAL

Set the *A* dial at 90°. Wedge the *So* and *B'r* lines to hold the *Yo* side of the *dR* differential so that no motion can back out of that side.

Turning the *Sh* line should produce no output from the *dR* follow-up. Any correction that is required should be made with *Sh* at 40 knots. Wedge the *A* dial at 90°. Slip through clamp *A-6* to position the vector gear.

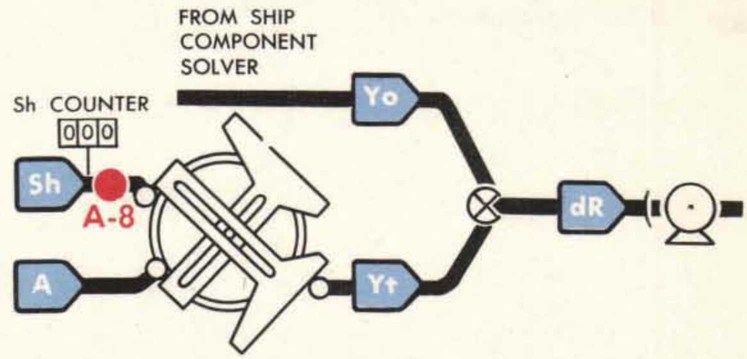
Tighten the *A-6* clamp. Remove the wedge on the *A* line.

A-8 CAM of TARGET SOLVER to Sh COUNTER

Run the *Sh* counter to zero knots. Wedge the *So* and *B′r* lines to hold the *Yo* side of the *dR* differential so that no motion can back out of that side.

Turning the *A* vector should produce no output from the *dR* follow-up. Wedge the *Sh* counter. Slip through *A-8* clamp to position the cam.

Tighten the clamp. Remove all wedges. Now that the settings for the component solvers have been refined, adjustment *A-9* of the *dR* follow-up has to be repeated. The previous setting was only a temporary one. This time it is a final setting.



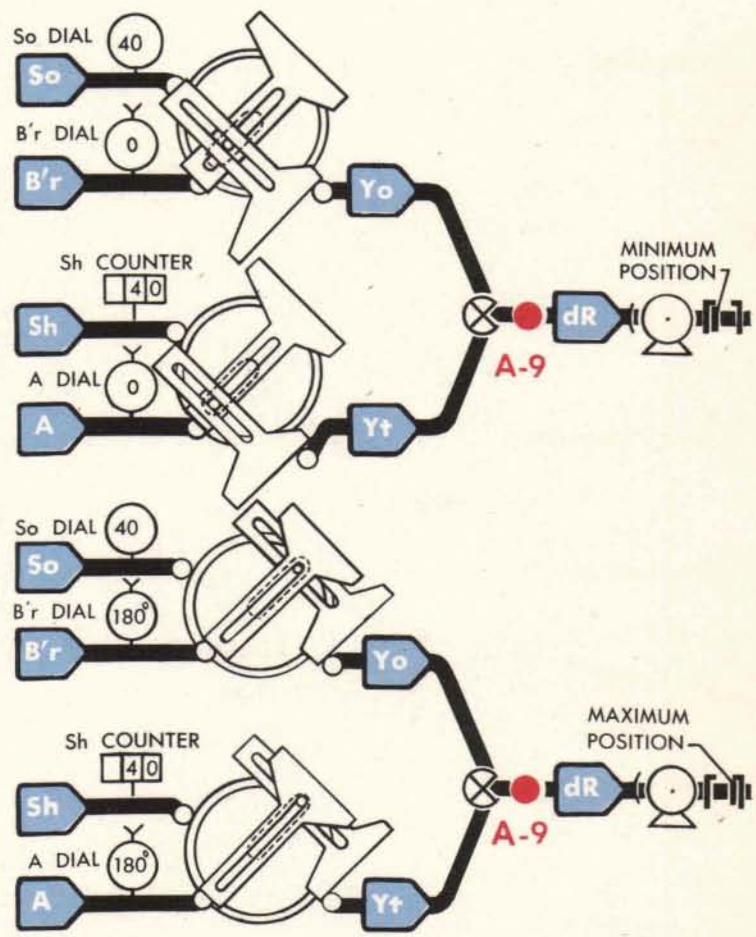
A-9 FINAL SETTING of the dR FOLLOW-UP

With *A* and *B′r* at 0°, put *So* and *Sh* at 40 knots. Synchronize the *dR* follow-up so that the *dR* limit stop traveling nut just touches the minimum end of the stop.

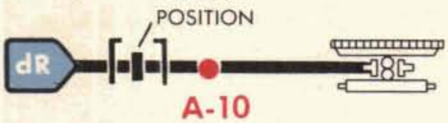
Run *A* and *B′r* to 180°. The follower block should just touch the maximum end of the stop. If there is any error, it should be split at both ends of the stop.

Then tighten the *A-9* clamp.

In general, if one limit of a limit stop is at a zero value, the stop is set exactly to this value to give a definite reference point. When the limits are above and below zero, as in the case of *dR*, the error should be "split" so that there is equal error at both ends.



A-10 CARRIAGE OF DISK INTEGRATOR TO dR FOLLOW-UP

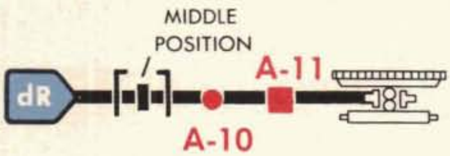


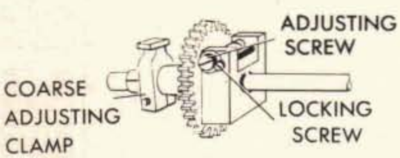
Put *So* and *Sh* at zero knots with the follow-up energized. This makes $dR = 0$.

Position the integrator carriage approximately in the center of the disk. The traveling nut in the limit stop will be at its middle position.

Tighten *A-10* coarse clamp. This is a rough setting.

A-11 VERNIER TO REFINE INTEGRATOR CARRIAGE SETTING



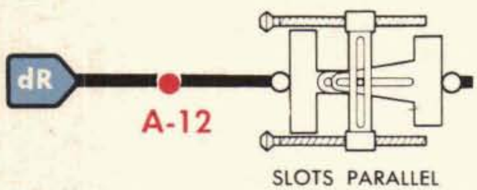


To refine the setting of the carriage, turn the adjusting screw of the *A-11* vernier clamp to put the carriage in the exact center of the disk.

Check the setting by running the disk. Adjust through the adjusting screw of the vernier until there is no output from the integrator roller.

Then carefully tighten the locking screw of the vernier. Now check the setting to make sure that tightening the locking screw has not upset the adjustment of the vernier.

A-12 INPUT RACK OF THE PREDICTION MULTIPLIER



With *dR* at zero, position the input rack until its slot is parallel to the slot in the output rack.

Position the input slide so that the multiplier pin is approximately over the stationary pin. Slip-tighten the *A-12* clamp.

Position a shadow stick between the output gear teeth. Run the slide to the farther end of the screw. There should be no movement of the shadow.

If motion is observed, hold the line, *dR*, and push the input rack until the shadow stick returns to its original position.

Repeat the test until there is no movement shown by the shadow stick.

Always correct the setting with the slide at its maximum distance from the fixed pin.

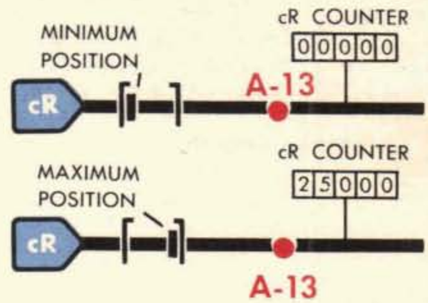
Tighten *A-12* clamp and remove the shadow stick.

A-13 cR LIMIT STOP

Run *cR* in a decreasing direction to the end of the limit stop and wedge the line.

Put *cR* counter at zero yards. Tighten *A-13* clamp and remove the wedge.

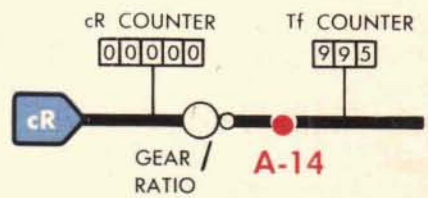
Run the *cR* counter to 25,000 yards. The block in the limit stop should be at maximum position.



A-14 Tf COUNTER TO cR COUNTER

With *cR* counter at zero yards, wedge the line. Set *Tf* counter at minus five seconds. Zero for this counter is 000. Minus five would appear as 995.

In this network the value of *Tf* is only an approximation. A five second offset is put on the line so that errors in *Tf* will be as small as possible.



A-15 MULTIPLIER PIN TO Tf COUNTER

With the *Tf* counter at zero seconds, wedge the line. Set the multiplier screw input at zero, over the stationary pin. Then the multiplier pin is in the middle of the rack travel since *dR* going to the rack can be plus or minus.

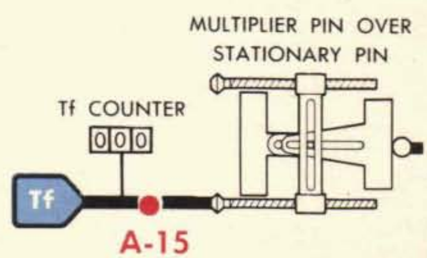
Since the stationary pin is in the center, only half the error has to be corrected when this error is read as movement of the output rack during the full travel of the input rack.

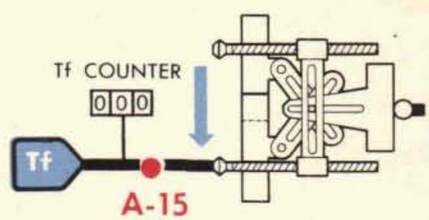
Position a dial indicator on one end of the output rack if it is easy to reach. Otherwise, use a shadow stick to measure the output rack movement.

Move the input rack all the way from one side to the other. Then run the input rack all the way in the other direction. There should be no motion on the indicator. If there is, the multiplier pin is not directly over the stationary pin.

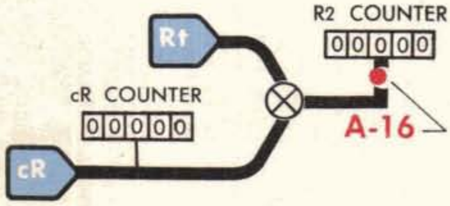
Then with the input rack at one end, re-position the multiplier pin. Run the input rack all the way in the other direction again. When there is no motion shown on the indicator, tighten *A-15* clamp.

Rt is now set to *Tf*.





A-16 R2 COUNTER TO cR COUNTER



Set the *So* dial and *Sh* counter at zero. This makes *dR* zero. When *dR* is zero, *Rt* will be zero.

Set the *cR* counter at zero. Then because *Rt* is zero and *cR* is also zero, *R2* is zero since *R2* = *cR* plus *Rt*.

Set the *R2* counter at zero and tighten *A-16* clamp.

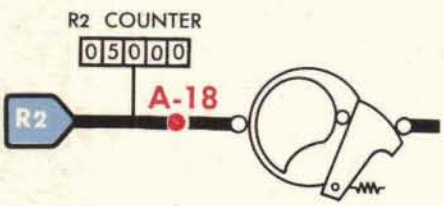
A-17 INTERMITTENT DRIVE to R2 COUNTER

Set the *R2* counter at zero. This is the cut-in reading for the lower limit of the intermittent drive. Run the intermittent drive until it cuts in at the lower limit. Slip-tighten *A-17* clamp.

Run the *R2* counter to 25,000 yards. This is the cut-out reading for the upper limit of the intermittent drive. Slip the line to the intermittent drive to increase the input slightly and check that the unit cuts out immediately. Tighten the clamp and check the lower limit.

When checking the setting of the intermittent drive, put *So* at 40 knots and have *B'r* at 0° for checking the lower limit and *B'r* at 180° for checking the upper limit. Then for either limit, *R2* will reach its cut-out value BEFORE *cR* reaches its limit and turning the *cR* line will carry the *R2* line past its cut-out value. Then if the intermittent drive is set to cut out at some point beyond the limit, this fact can be observed.

A-18 BALLISTIC CAM TO R2 COUNTER



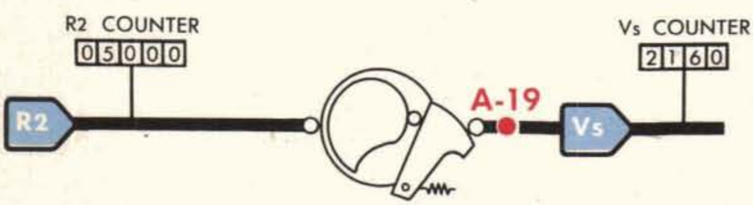
Set *R2* counter at 5000 yards. Insert the ballistic cam setting rod in the cam.

Tighten the *A-18* clamp. Leave the setting rod in the cam.

A-19 Vs COUNTER TO R2 COUNTER

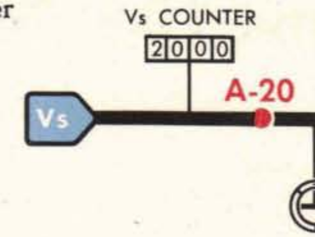
With the *R2* counter at 5000 yards and the setting rod still in the cam, set *Vs* counter at 2160 minutes.

Tighten *A-19* clamp and remove the setting rod from the cam.



A-20 FINE TRANSMITTER TO Vs COUNTER

With the *Vs* counter at 2000 minutes, set the fine transmitter at electrical zero and tighten *A-20* clamp.

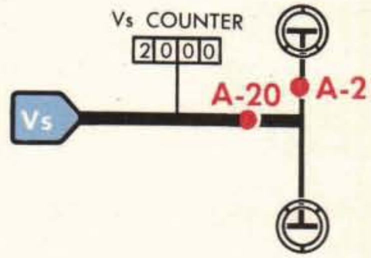


A-21 FINE TO COARSE TRANSMITTER

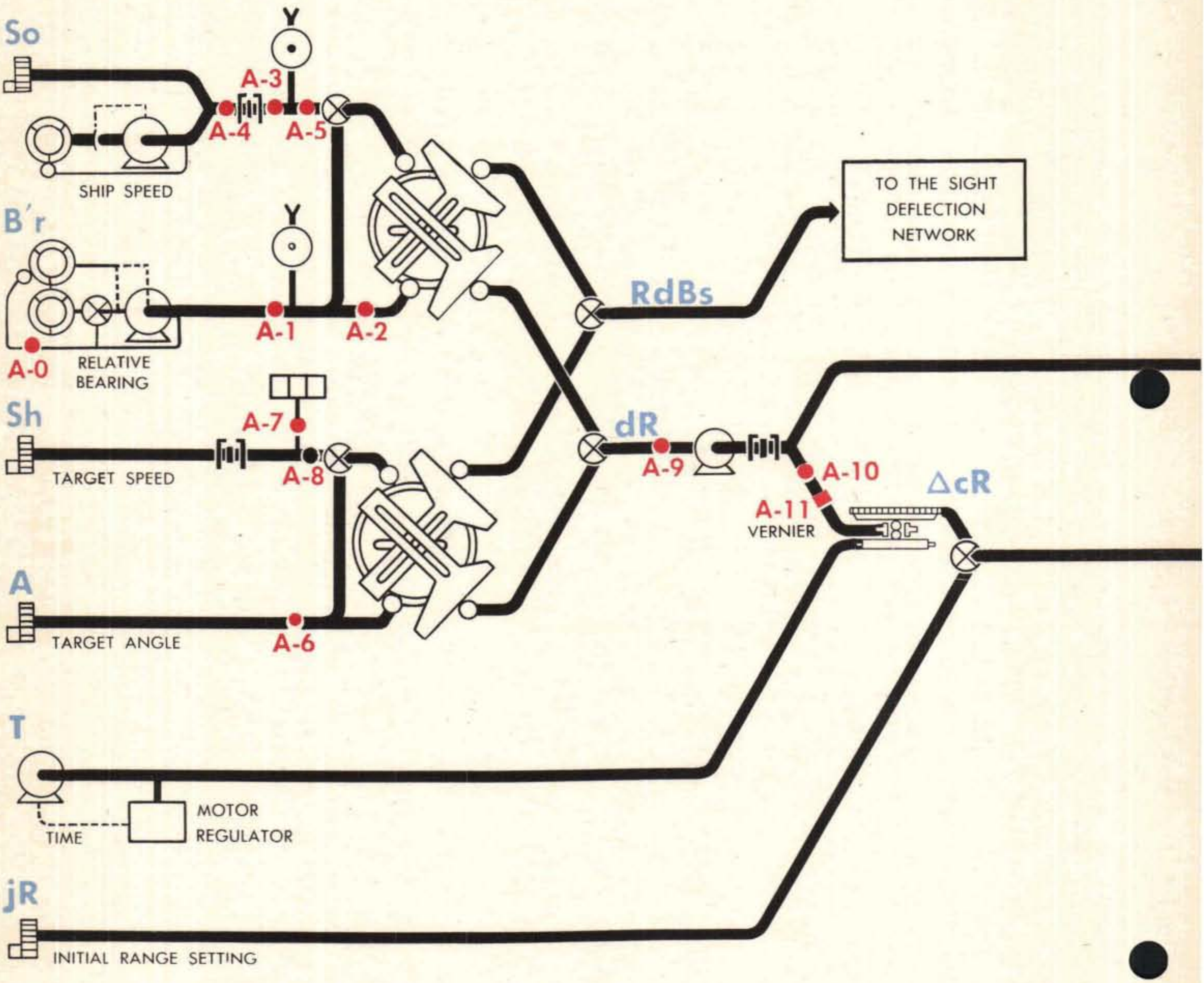
With fine transmitter at electrical zero and *Vs* at 2000 set coarse transmitter to electrical zero.

Tighten the *A-21* fine-to-coarse clamp.

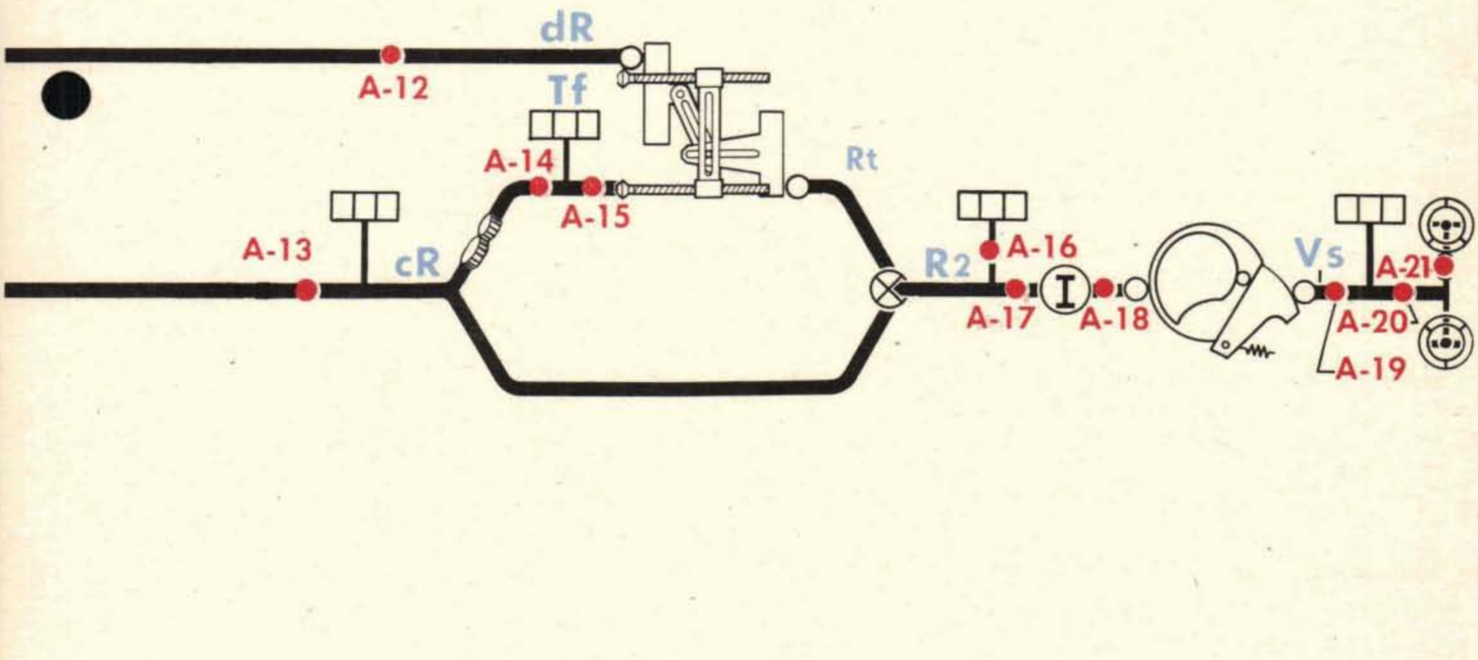
Now the entire network has been set. For any given set of input values, the computed value of Sight Angle will appear on the *Vs* counter and be transmitted to the guns.



Here's the network showing all



the SETTING CLAMPS



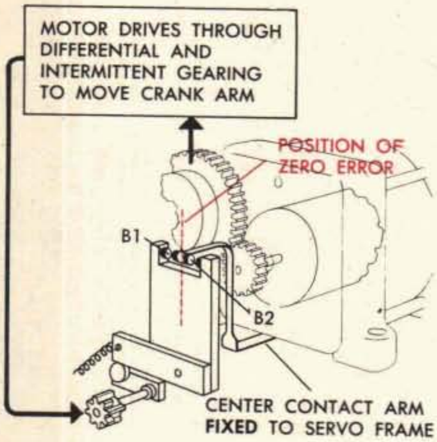
SECTION 5

TECHNICAL APPENDIX

This technical appendix contains descriptive material which was thought to be either too lengthy or too technical to be included in the regular chapters. Most of the appendix is devoted to three devices that are often used in follow-up controls: the Magnetic Drag, the Magnetic Damper, and the Compensator. An explanation of how the Servo Motor works and what the Capacitor does is also included.

	Page
Magnetic Drag	324
Magnetic Damper	334
Compensator	338
Servo Motor	342
Capacitor	345

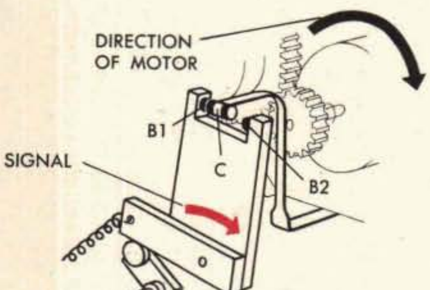
MAGNETIC DRAG



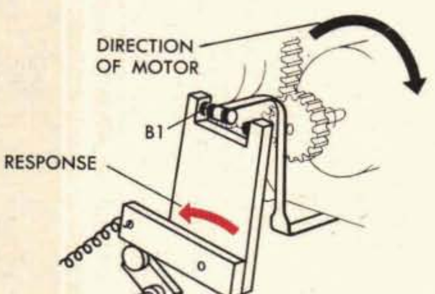
It has been noted that the center contact arm of a follow-up control may be mounted on a shaft rotated, within certain limits, by the core of a magnetic drag, and that such an arrangement is used to bring the servo motor quickly to rest at the point of synchronism.

To understand why such a device is needed, suppose for a moment that a follow-up has no damper or magnetic drag and that the center contact arm is fixed to the frame of the servo in a vertical position, that is in the position of zero error.

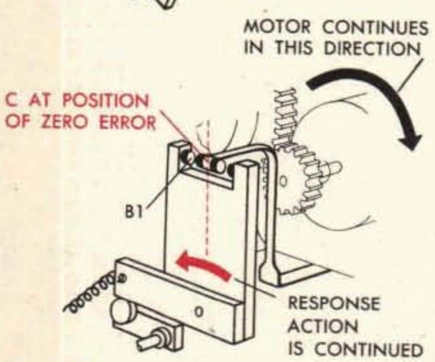
Action of servo without magnetic drag



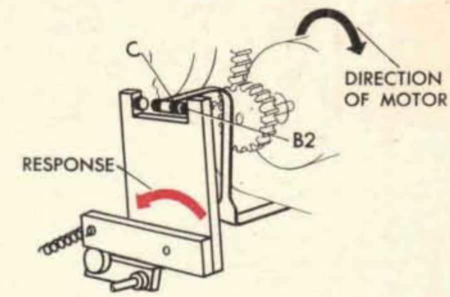
An input from a signal will cause the outer contact arm to rotate and bring an outer contact against the center contact–say outer contact *B1* against center contact C, as shown here.



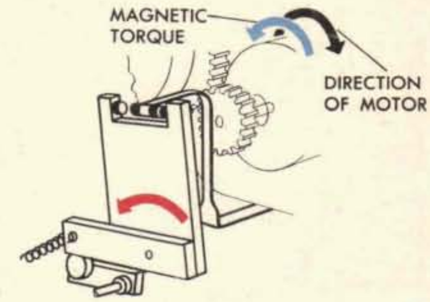
The servo will start, and quickly pick up speed. The response, acting through the differential, will drive *B1* back until the point of zero error is reached.



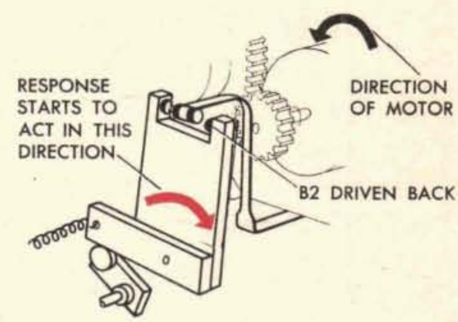
At this point *B1* will be separated from C. However, the servo motor continues to drive in its original direction because of the inertia *of the rotor and shafting even after the current supply is cut off*. This continued rotation of the rotor results in response action being continued.



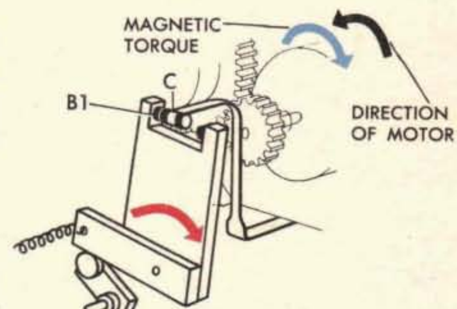
The response continues to rotate the outer contact arm and the other outer contact, *B2*, is brought against the center contact *C*.



Action of the magnetic field in the motor is now reversed, producing a torque acting on the rotor in the opposite direction. This torque slows down the rotor and then reverses it.



When the rotor reverses, contact *B2* is driven back as the response turns the outer contact arm in the opposite direction.



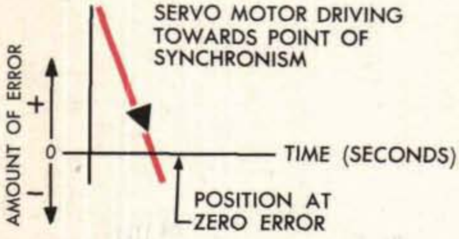
Once more the motor picks up speed and, again because of the inertia in the rotor and connected shafting, overruns the point of synchronism after the circuit is broken. Contacts *B1* and *C* are again brought together, the magnetic torque is applied to the rotor, and the cycle is repeated.

This tendency of the rotor and connected shafting to rotate after the current supply is cut off results in a series of oscillations. In this simplified setup, with the center contact arm fixed to the servo frame, such oscillations would cause the motor to reverse direction and overrun the signal each time an outer contact is swung against a center contact.

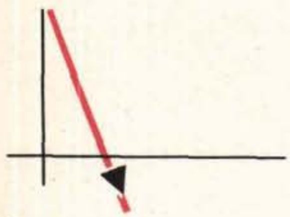
Such oscillations may be repeated many times, although they will finally be reduced to zero as friction and the load on the output shafting bring the rotor to a standstill.

It is the purpose of the magnetic drag to reduce quickly the extent and number of these oscillations.

Servo rotor drives past the point of synchronism many times

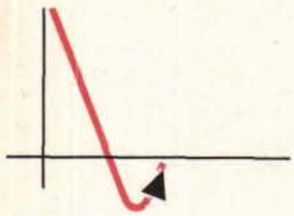


The action of the servo motor, in synchronizing to a fixed signal, can be followed in detail if it is represented in a short series of graphs. By using the vertical axis of the graph to represent the amount the rotor turns after reaching the point of synchronism (i.e., the displacement of the rotor from the value of the signal) and the horizontal axis to represent *time,* a picture is obtained of the rotor's displacement from zero position at any given moment.

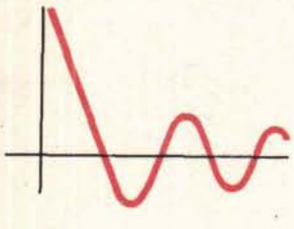


As the current is cut off at the point of zero error (which lies in the horizontal axis of the graph) the servo continues to run at high speed, due to the momentum of its rotor.

The servo thus crosses the horizontal axis at high speed.



As the opposite contact is closed, the current is reversed and this tends to drive the motor in the opposite direction, back to the zero position. The rotor slows, stops, and starts back towards zero. Without drag or damper, it crosses the zero line again with almost the same speed as it had the first time.



In this way, the servo will continue to cross and re-cross the horizontal axis for a long time, due to the oscillations between the contacts of the follow-up control, until friction and the load on the powered shafting finally compel the rotor to come to rest.

Reduction of oscillations

When a magnetic drag is geared to the servo, the tendency of the motor to overrun the point of synchronism is counteracted in such a manner that the oscillations are greatly reduced, resulting in the motor being brought quickly to rest at the point of zero error.

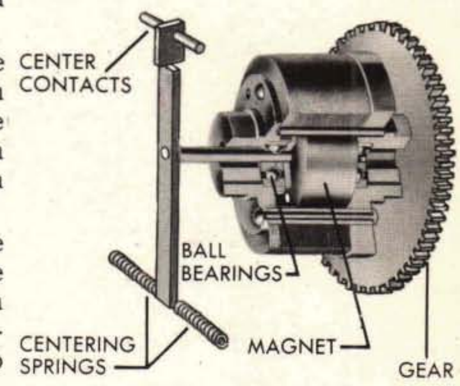
In order to understand how the drag works, it is necessary first to examine its construction and principle of operation.

Construction of the magnetic drag

The magnetic drag consists of two major parts: a frame, and a magnetized core.

The frame is a cylindrical case of laminated construction, made up of thin steel rings held together with copper bars. At each end is a copper end ring into which the bars are fitted. The frame, therefore, is of "squirrel cage" design. It is mounted in bearings at each end and carries a gear which is driven by a servo motor.

The core consists of a solid, magnetized steel cylinder. This core forms a permanent magnet, and is hereafter referred to as "the magnet." This magnet is fixed to a shaft which extends through the center of the case, and to this shaft is fixed the arm carrying the center contacts. Two centering springs are attached to the bottom of this arm.



Principle of operation

The core of the drag, being a permanent magnet, is surrounded by a magnetic field. When the frame is rotated by the servo motor, it carries the copper bars through the field produced by the magnet. A current is induced in the bars, and this current creates its own magnetic field.

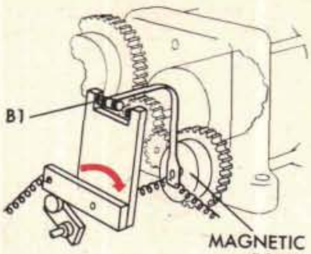
The stator, attached to the frame, tends to pull the magnet around in the direction in which the frame is rotating.

The center contact arm, fixed to the magnet shaft, rotates when the magnet rotates. Two centering springs, however, tend to hold the center contact arm in its zero position, and prevent it from rotating.

When the frame rotates, it must pull the magnet and the center contact arm around against the action of these springs. Because of this, the magnet is able to rotate only a certain distance, offsetting the contact arm from its centered position by a corresponding distance.

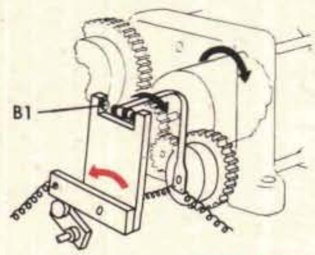
The construction of the drag is such that the torque exerted by the rotating frame is almost proportional to the speed of the frame, and the displacement of the centering springs is proportional to the force applied to them, resulting in the center arm being rotated an amount proportional to the speed of the frame. The speed of the frame depends upon the speed of the servo motor. Thus *the center contact arm is pulled around (or displaced from center) an amount proportional to the speed of the motor.*

The drag in action

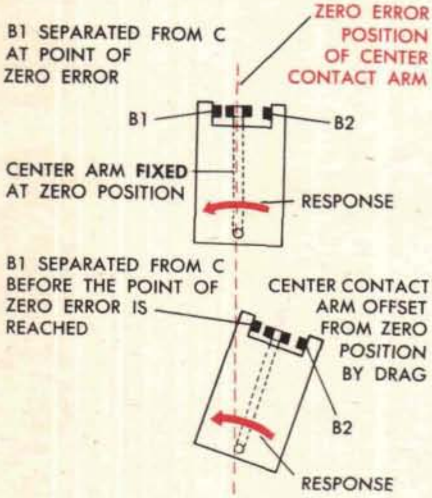


When the drag is used in a follow-up, this is what happens:

The input from a signal causes an outer contact *B1* to be brought against the center contact C, energizing the motor.



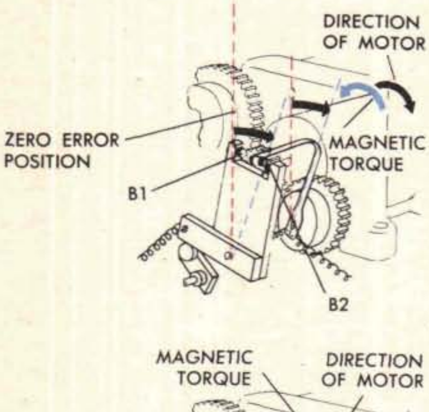
As the motor drives, two things happen: First, the magnetic drag tends to rotate the center contact arm away from *B1*. Then the motor response drives the contact *B1* back (as shown) resulting in the circuit's being broken before the point of synchronism is reached.



If the center contact arm had *remained fixed,* it is obvious that, as the response drove *B1* back, the contacts would have broken at the blue line—the point of zero error.

But when the center contact arm is turned by the drag, the contacts break at the red line, which means that they break *before* the motor reaches the point of zero error—or synchronism. The action of the drag, therefore, can be said to "anticipate" the point of synchronism, causing the contacts to be separated, before the point of zero error is reached.

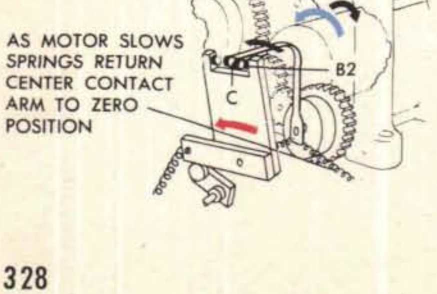
As *B1* is driven back by the response, *B2* is rotated against a center contact. This means that the action of the magnetic fields in the motor is reversed *before* the point of synchronism is reached.



The motor continues to run in the original direction, because of the inertia of its rotor and connected shafting, and continues to rotate the drag frame in the same direction.

But due to the reversed action of the magnetic fields, a torque is applied to the rotor which tends to pull the rotor around in the opposite direction.

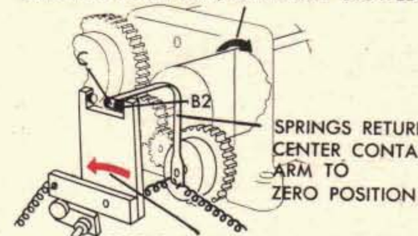
This magnetic torque acts as a brake on the rotor, slowing it down. Thus the speed of the servo motor is decreased before *the point of synchronism is reached.*



As the speed of the servo decreases, the speed of the drag frame, which is geared to the servo, also decreases. Consequently, the strength of the torque coupling between the drag frame and the drag magnet decreases. As this happens, the centering springs act to return the center contact arm to its center position, the position of zero error.

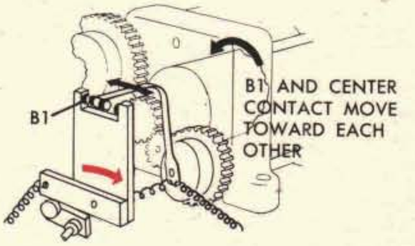
Since the speed of the response decreases as the motor speed decreases, the centering springs move the center contact arm to zero position faster than *B2* is moved by the response, causing the contact between C and *B2* to be broken *before* the point of zero error is reached.

The inertia of the rotor enables it to continue rotating although the action of the magnetic torque has slowed it down. Because of this, response action is sufficient to bring *B2* against the center contact.

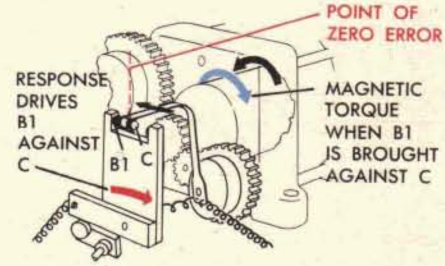


This causes the motor to be energized and to run in the opposite direction.

As the motor picks up speed, the drag turns the center contact toward *B1*, and the response turns *B1* toward the center. *B1* and the center contact, therefore, move toward each other.



As the center contact and *B1* touch, on the opposite side of the position of zero error, magnetic torque is applied to the rotor of the motor. The motor speed and drag speed are immediately decreased, and the center contact arm is returned by the springs to the zero position *faster* than the response can move *B1* toward center.



Contact between *B1* and the center contact is therefore broken *before* the point of zero error is reached.

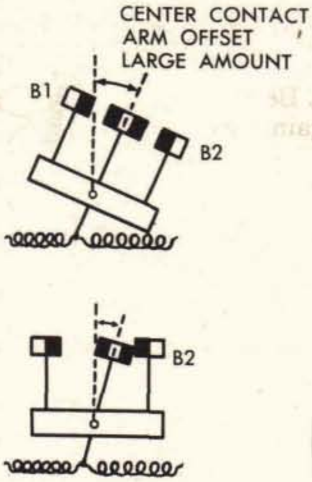
As the rotor of the servo continues rotating, even though the magnetic torque has slowed it down, the drag is turned toward *B2*. At the same time, response action is sufficient to turn *B2* toward center. The cycle described above is therefore repeated. By rotating the center contact arm away from the position of zero error, the magnetic drag causes:

1 the current supply to the motor to be cut off, slowing the rotor, *before* the point of zero error (synchronism) is reached;

2 magnetic torque to be applied, slowing the rotor, *before* the point of zero error (synchronism) is reached.

The action of the magnetic drag can therefore be considered as "anticipating" the point of synchronism by slowing the motor prematurely.

Since the speed of the motor is diminished considerably at each cycle, the amount the center contact arm is rotated at each cycle is considerably diminished also. In other words, the size of the oscillations diminish rapidly, and the motor is brought quickly to rest at point of synchronism.



The gear ratio between the servo and the frame of the magnetic drag (which regulates the speed at which the frame is driven) and the strength of the centering springs can be so regulated that the servo can be stopped practically without oscillations. For instance, if the gear ratio results in the drag frame being rotated at very high speed, and the springs used are weak, the center contact arm can be displaced a considerable amount when the servo runs.

This will result in outer contact *B1* being separated from the center contact while the center contact arm is still at an appreciable distance from its position of zero error.

As the motor overruns the point of synchronism, therefore, *B2* is brought against a center contact, and the current is reversed in the motor coils, while the center contact arm is still some distance from its position of zero error.

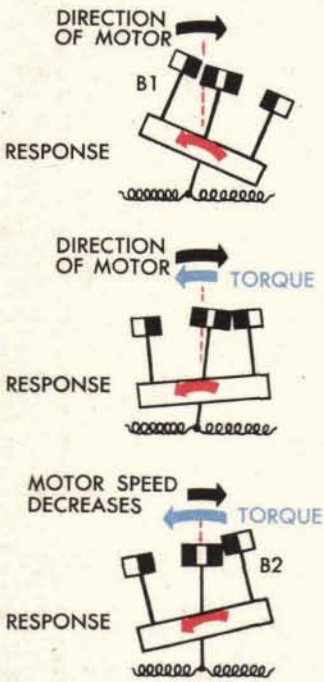
By the time the center contact arm is centered by the springs, the torque has acted sufficiently long on the rotor to bring the rotor to a standstill.

At first glance, this arrangement would appear ideal, as the motor is brought to rest without oscillation.

There is a factor, however, which makes this arrangement undesirable.

When the center contact arm is offset a considerable amount from its center position by the drag, the value of the output lags considerably behind that of the input while following a *moving* signal. The contacts open long before the point of zero error is reached and the response lags behind the signal as long as the signal moves. How this condition is avoided in some follow-up controls is explained later under "The Compensator."

Reducing velocity lag error



To reduce lag error, a compromise is made. The speed at which the drag frame is driven and the strength of the centering springs are so chosen that the center contact arm will not be displaced very far from its center zero position for a given speed.

Under such conditions, the contacts are separated when only a comparatively short distance from the point of zero error— as indicated here, where *B1* is just being driven back from a center contact.

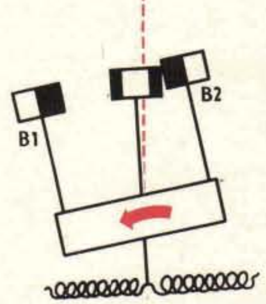
As *B2* is swung over and meets the opposite center contact, current in the motor coils is reversed and a torque is applied to the rotor.

The rotor's speed rapidly decreases, and the centering springs act to bring the center contact back to the zero position.

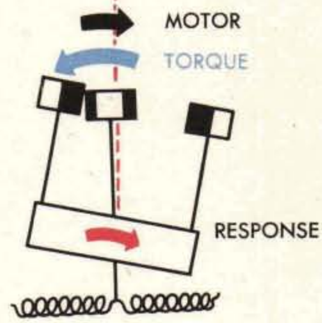
However, in this case, the center contact arm has been returned to its zero position *before the torque has had time to bring the rotor to rest.*

Consequently the motor is still driving in the original direction, although at greatly reduced speed, and this keeps contact *B2* against the center contact.

Under continued application of the torque, the rotor slows down completely and then reverses its direction of rotation.



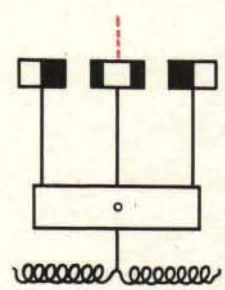
Accordingly, the center contact arm is offset in the opposite direction. Because of the decreased speed of the motor, however, the distance the contact arm is now displaced is *less* than the distance it was displaced initially.



Before the motor has time to pick up any appreciable speed, the servo response, acting through the differential, separates the contacts.

B2 is driven back and the outer contact *B1* is swung over once more. Current is again reversed in the motor coils, and the resulting torque which is applied to the rotor again acts like a brake and slows the rotor down.

Each time the motor crosses the point of synchronism it does so at greatly decreased speed. As the speed of the motor rapidly decreases, the amount the motor overruns the point of synchronism also rapidly decreases.



SPRINGS HOLD ARM CENTERED

With the decrease in speed of the motor, the displacement of the center contact arm becomes less and less. As the motor comes to rest, the centering springs hold the center contact arm at the position of zero error.

When the center contact arm is allowed to move only a comparatively short distance away from the position of zero error, some oscillations will occur between the contacts of the follow-up control, when synchronizing to a fixed signal.

However, since these oscillations rapidly decrease in size (amplitude) and duration, bringing the motor quickly to rest, there is no interference with the efficient working of the mechanism, and lag error in following a changing signal is reduced.

VELOCITY LAG ERROR

In the case of a fixed signal, the action of a magnetic drag introduces no error between the input and the output, for the powered shafting is merely positioned in accordance with the fixed value of the input.

If the powered shafting drifts off its correct position, because of the load, the rotor of the servo will be turned. This will result in the servo response acting through the differential and closing the contacts so that the motor drives the powered shafting back to its correct position.

Output and input are thus kept synchronized at all times during the transmission of a fixed signal.

However, if the signal keeps changing, the action of the drag introduces a difference, or error, between the input and the output while the signal is moving.

This happens because the center contact arm is offset away from the point of zero error by the drag.

If readings are taken at any given instant during transmission of the signal, the reading on the receiver dial will not correspond with the reading on the transmitter dial. The reading on the receiver dial will always be found to "lag" behind that on the transmitter dial.

In other words, the output lags behind the input.

The more the center contact arm is offset from the point of zero error, the greater will be this error, or lag of the output behind the input.

Now, the amount the center contact arm is offset is proportional to the torque exerted by the magnetic drag. This torque depends upon the velocity of the motor.

The velocity of the motor is proportional to the velocity of the incoming signal.

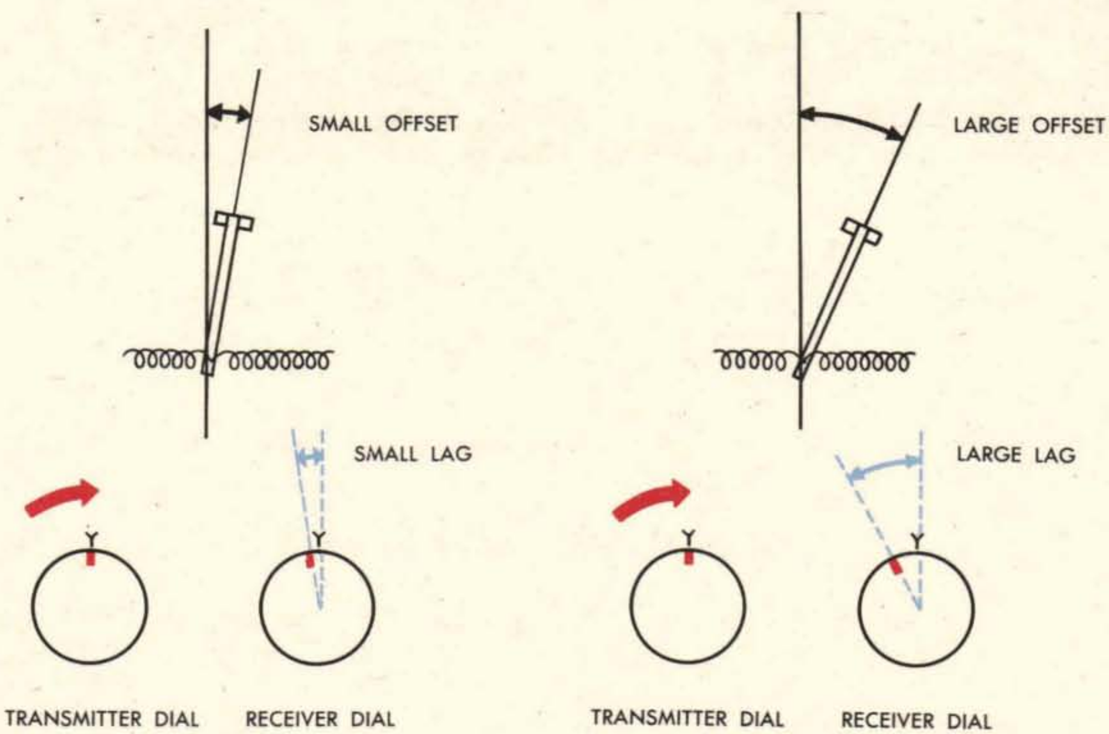
Therefore the error between the input and the output, while the signal is changing, is proportional to the velocity of the signal.

This error is known as the "velocity lag error." Follow-ups having no provision for correcting velocity lag error are called velocity lag type follow-ups.

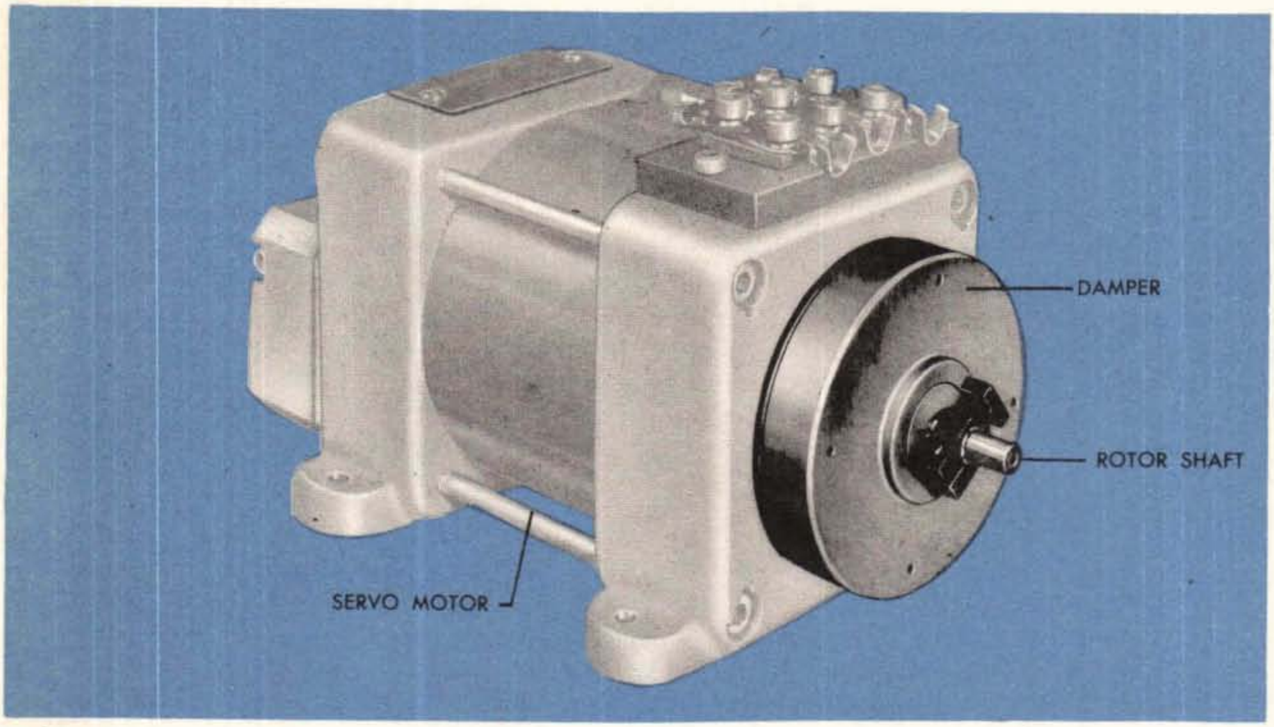
Such an error can be disregarded where the incoming signal reaches a certain value quickly and remains fixed for a considerable length of time, as it does in the case of Own Ship Speed, *So*.

The error can also be disregarded in cases where a constantly changing signal changes slowly, and the center contact arm is offset by only a small amount.

When the incoming signal changes at high velocity, and output readings must correspond exactly with input at every instant, as in the case of Relative Bearing, *B'r*, velocity lag error cannot be tolerated. To eliminate velocity lag, the magnetic drag is incorporated into a device known as a "compensator." Follow-ups having such a compensator are called compensated type follow-ups.



THE MAGNETIC DAMPER



In the section on Follow-up Controls it was mentioned that a magnetic damper is used to reduce the oscillations which occur when the motor reaches the point of synchronism.

The part played by a damper is best understood by considering why such a device is necessary at all. To do this, the nature of the oscillations between the contact points must be examined.

Oscillations between contact points

It has already been shown, in describing the magnetic drag, that because of a servo motor's inclination to keep turning once it has started to turn, the motor causes first one outer contact and then the other to swing over and touch the center contact after position of zero error has been reached. It has also been shown that the purpose of the drag is to limit this swinging to as few oscillations as possible (consistent with the general efficiency of the mechanism) in order to bring the motor to rest quickly.

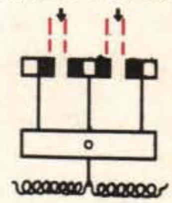
Theoretically, there should be no further movement of the servo motor after the drag has slowed it down, and the centering springs have brought the center contact arm to the position of zero error.

The motor should come completely to rest and no further oscillations should occur.

In practice, however, oscillations between the contacts are *not always completely eliminated* by action of the drag.

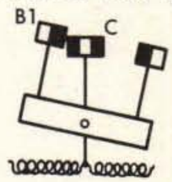
When the motor reaches the point of synchronism the first time, it overruns the point at high speed, and hence has a large amount of energy stored in its rotor, due to angular momentum. Upon being reversed, it crosses the point of synchronism with considerably less speed, and with considerably less energy in its rotor. This process is repeated. Finally, the rotor loses so much momentum that it has only sufficient energy to *drift* past the point of synchronism.

ONLY SMALL SPACE BETWEEN CONTACTS



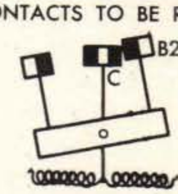
But it is only necessary for the rotor to rotate an outer contact arm slightly (just sufficiently to move it across the small space that normally separates an outer contact from a center contact) in order to cause the electric supply circuit to the motor to close. This amount of rotation the rotor is able to accomplish, even when it only drifts past the point of synchronism. When the opposite outer contact is brought up to the center, the motor drives again, in the opposite direction.

ROTOR TURNING VERY SLOWLY



The process may be repeated indefinitely, resulting in a series of very small oscillations just big enough to cross the space between the outer and center contacts.

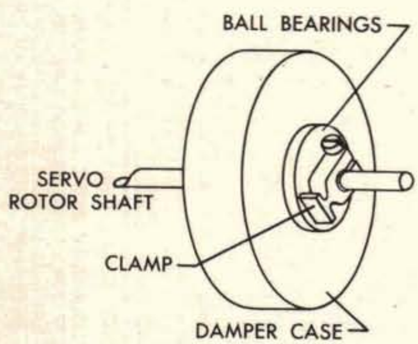
CONTACT B1 TOUCHES C AND ROTOR IS ENERGIZED JUST SUFFICIENTLY FOR CONTACTS TO BE REVERSED



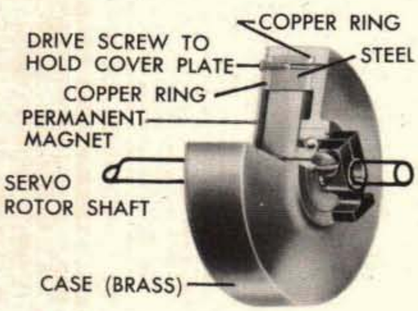
In practice, these small oscillations occur in fairly rapid succession, and can best be described as a form of "jitters."

It is the business of the magnetic damper to supplement the action of the drag and to reduce this jittering. The combined efforts of both drag and damper are finally responsible for bringing the motor to a standstill at the point of synchronism, with oscillation between the contact points practically eliminated.

The magnetic damper consists of:



A brass case which is ball-bearing-mounted on the rotor shaft and *free to turn.* Inside the case is attached a heavy steel ring, on each side of which is a copper ring. When this weighted case is rotated, it has the inclination to keep turning, and therefore is called the "inertia weight" of the damper.



A permanent magnet (inside the case) which is *fixed* to the rotor shaft of the servo motor.

How it operates

1. When the servo rotor revolves, it rotates the permanent magnet.
2. The magnetic field around the magnet is rotated.
3. The lines of force in this magnetic field cut through the steel ring as the field moves past.
4. When this happens, a current is induced in the steel ring, and the two copper rings help to distribute this current evenly throughout the steel.
5. The induced current creates a magnetic field of its own.

As the rotor turns, the magnet field and the field of the steel ring react together in such a way that the ring tends to be pulled around in the direction in which the magnet is moved. And since the ring is attached to the case, the whole case turns. In other words, the rotating magnet exerts a torque on the case which tends to make the case revolve in the same direction as the magnet.

A magnetic force coupling between the magnet and the case is therefore produced by turning the magnet.

The slightest movement of the magnet will set up this magnetic couple.

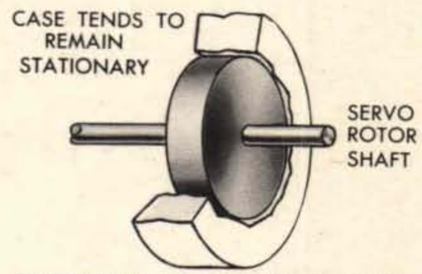
Elimination of small oscillations

Whenever the rotor speed has been so reduced that it has only sufficient energy to drift across the point of synchronism, the case of the damper has become stationary.

The damper case, being comparatively heavy and stationary, is not easily rotated. The inclination of the case to remain at rest once it has stopped acts as a brake on the rotor whenever the rotor starts to turn.

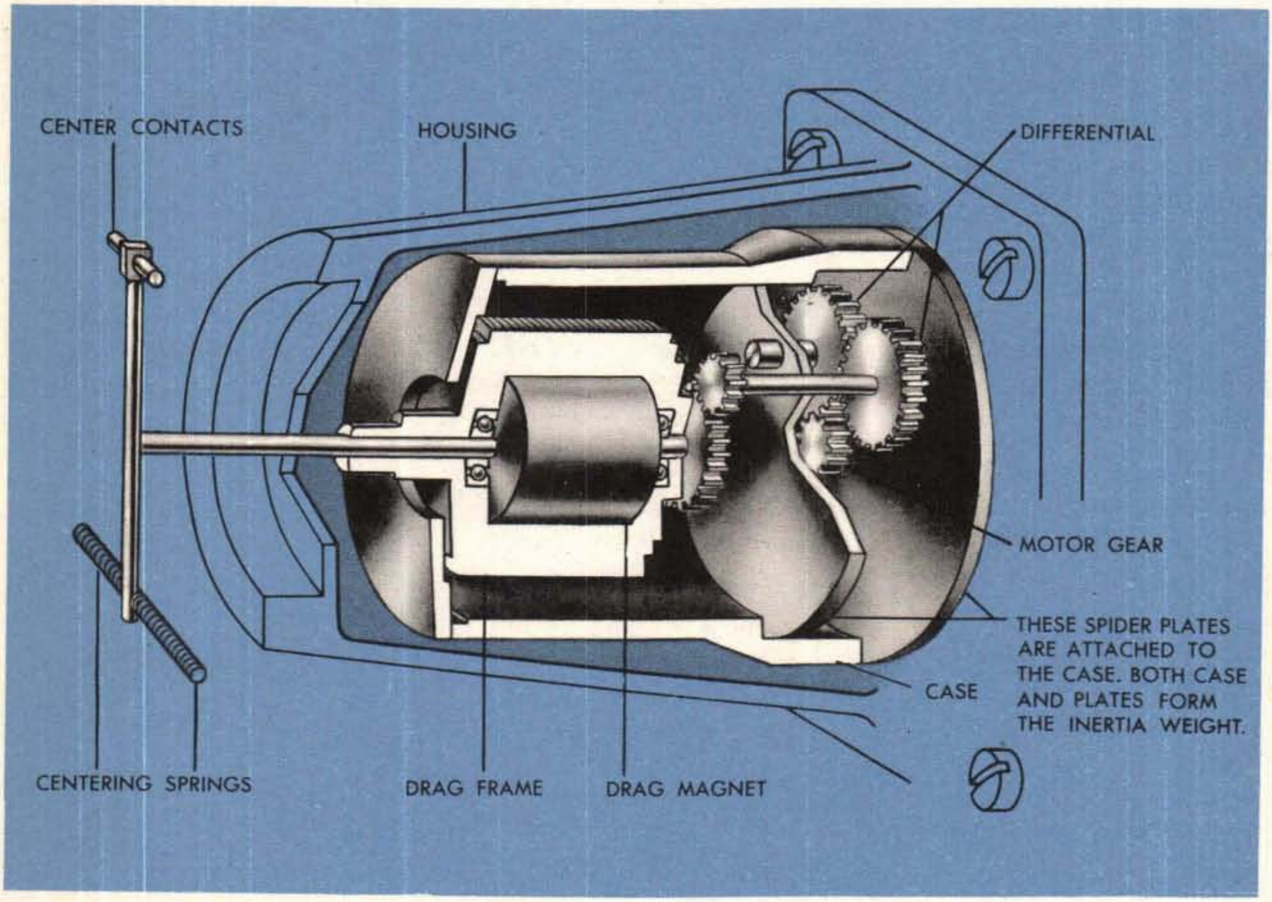
The small amount of energy made available to the rotor, when the contacts momentarily touch as they drift across the point of synchronism, has to overcome the tendency of the damper case to remain stationary, and the rotor comes to a standstill quickly.

All oscillations, therefore, soon cease and the motor is brought to *rest at the point of synchronism,* the outer contact arm being at the position of zero error.



STATIONARY CASE TENDS TO PREVENT MAGNET BEING ROTATED

THE COMPENSATOR



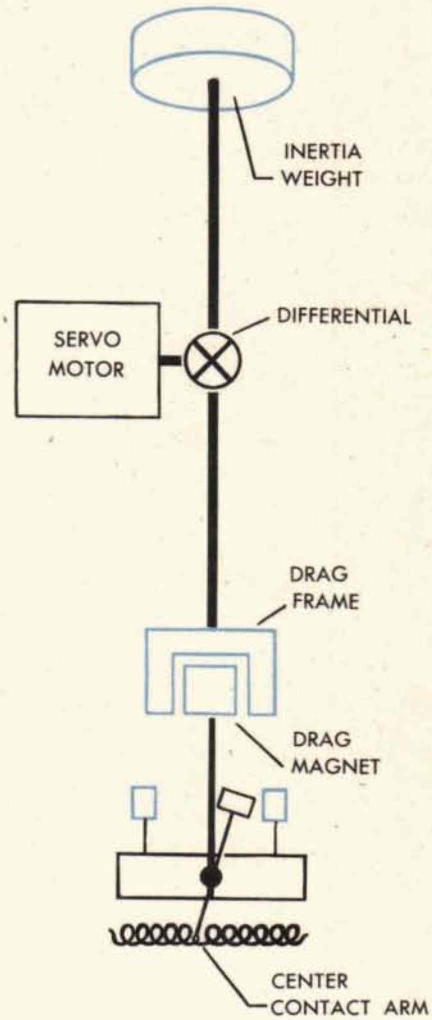
It has been shown that the use of a magnetic drag introduces a velocity lag error between the input and the output of a follow-up control while the signal is changing, and that such an error becomes too great to be permitted when a signal comes in at a high velocity.

The velocity lag error is eliminated by a "compensator," which consists of a magnetic drag and an inertia weight, both of which are driven by the servo motor through a differential.

The motor drives one end gear of the differential and the drag is driven by the other end gear. The spider of the differential, is attached to a comparatively heavy case to form the inertia weight.

Operation

The compensator can be represented schematically as shown here.



When the servo motor starts to drive, the inertia weight tends to remain at rest. One side of the differential rotates the frame of the drag, and this rotation causes the drag frame to be coupled to the magnet by a magnetic force coupling.

The magnetic force coupling applies a torque to the drag magnet, and since the magnet cannot be pulled around more than a few degrees, there is a resistance to the rotation of the frame.

The side of the differential which rotates the frame of the drag must drive against this resistance.

This being so, the other side of the differential must drive with an equal force. And since there is nothing holding the inertia weight stationary, this force starts the weight rotating.

The inertia weight gathers speed under this applied torque, and if the servo motor continues to drive at constant speed, the inertia weight gaining momentum quickly approaches the speed of the motor.

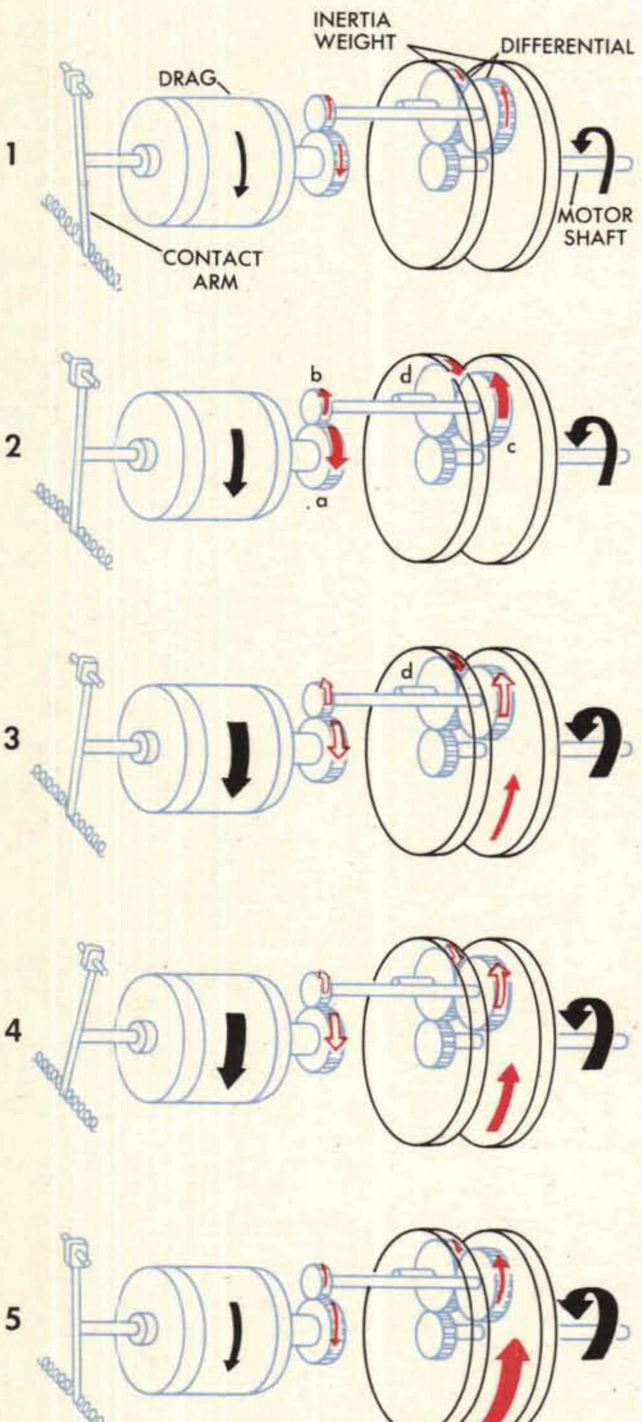
Now the total output of the differential can only equal the input. That is, the speed of the drag frame *plus* the speed of the inertia weight can only equal the speed of the motor. Therefore, as the inertia weight gains speed, the drag frame must lose speed. This it does, until it comes almost to rest.

As the drag frame loses speed, the torque exerted on the drag magnet decreases.

Immediately, the centering springs act to bring the center contact arm back to its center position, and the contact assembly comes back to the position of zero error.

THEREFORE, WHILE A SIGNAL IS CHANGING AT CONSTANT SPEED, THIS FOLLOW-UP CONTROL WILL OPERATE WITHOUT LAG ERROR.

COMPENSATOR and CONSTANT SPEED SIGNAL



The action of the compensator, during the course of a signal which changes at constant speed consists of five steps:

1 The motor starts to drive, and driving through the differential rotates the frame of the drag.

2 As the motor picks up speed, the drag frame picks up speed and the torque exerted by the frame on the drag magnet increases. The drag magnet is prevented from rotating beyond a limited distance by the pull of the centering springs. The effect of the increased torque on the drag magnet therefore reacts back through the drag in the form of a force which tends to reduce rotation of gears *a, b, c,* and *d*.

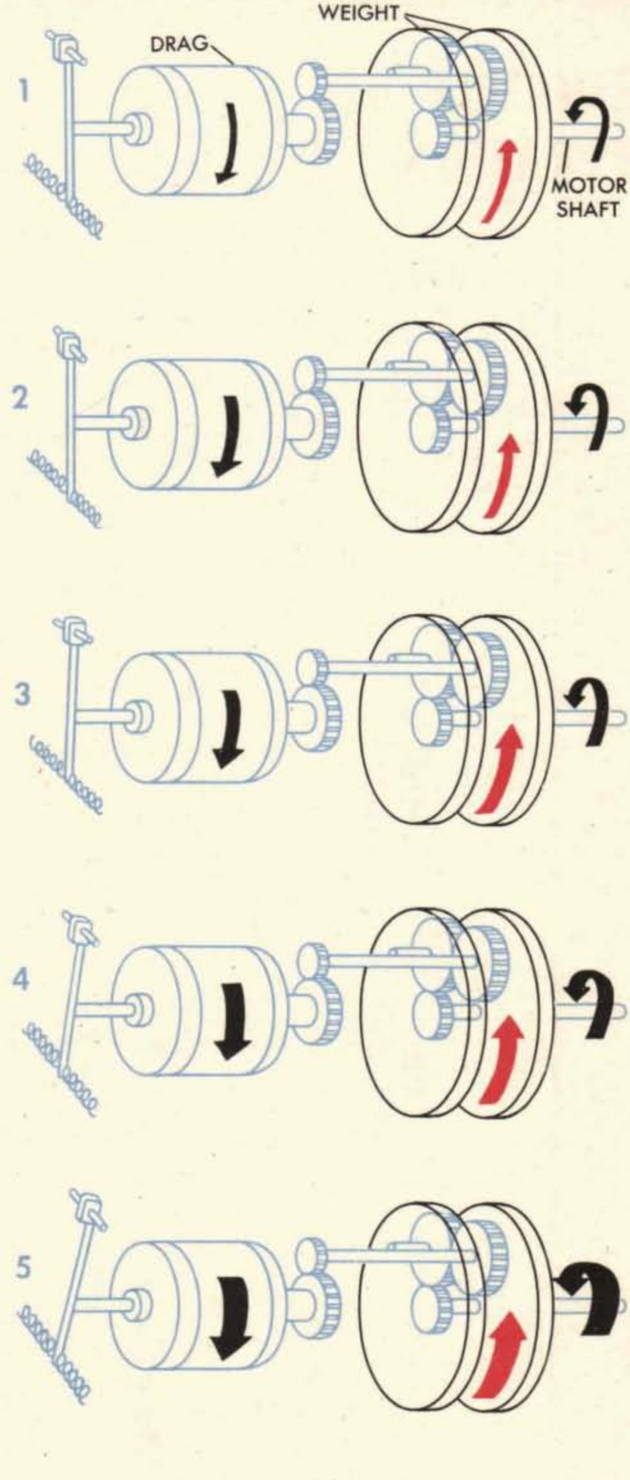
3 The motor is driving at constant speed. When the drag acts to slow down the rotation of the gearing, gear *d*, being in mesh with the motor gear, is carried around with the motor gear instead of rolling on it. Since the shaft of gear *d* is mounted in the inertia weight, this action causes the weight to be rotated.

4 As the motor continues to drive, the inertia weight picks up speed, and tends to reach a speed corresponding to the speed of the motor. As this occurs, the speed of rotation of the gearing, and hence of the drag frame, decreases.

5 With the reduction in speed of the drag frame, and a corresponding reduction in the torque exerted on the drag magnet, the centering springs bring the center contact arm to position of zero error.

COMPENSATOR and ACCELERATING SIGNAL

If the signal does not change at a constant speed but *accelerates* as it comes into the follow-up control, the action of the compensator is somewhat different:



1, 2 Since the signal accelerates, the servo motor accelerates. The inertia weight, however, being comparatively heavy, tends to hold to its own speed, and is not easily speeded up.

As the differential output must equal the input, the increase in speed of the motor acts through the side of the differential which drives the drag frame, making the frame rotate faster.

3, 4, 5 Because of the tendency of the inertia weight to "take its own time," the continually increased speed of the motor (due to acceleration) operates where it finds least resistance: on the frame of the drag.

The inertia weight never attains the speed necessary to allow the drag to slow down sufficiently to permit the center contact arm being brought to point of zero error.

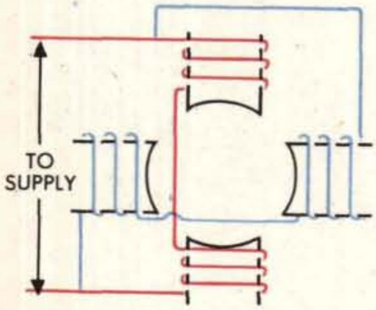
As a result, the contact arm remains displaced from center throughout the period of the signal acceleration, the displacement being proportional to the acceleration of the drive. This, of course, introduces an error into the position of the powered shafting (output) proportional to the acceleration of the drive.

Proper proportioning of the parts of the compensator reduces this error to a point where it becomes negligible.

SERVO THEORY

PRINCIPLE OF OPERATION

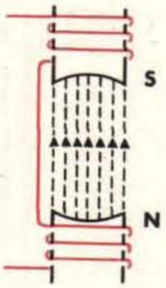
Magnetic flux



Servo motors used in the fire control instruments are of the induction type, designed to operate on single phase, 60 cycle, alternating current.

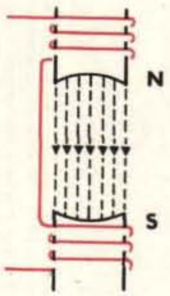
To understand how the rotor in this type of motor is made to revolve, several factors must be considered:

The stator, or field winding, has four poles.

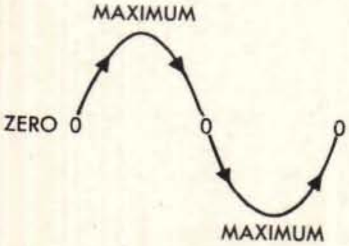


When current is passed through the stator coils, lines of force (magnetic flux) pass between opposite poles.

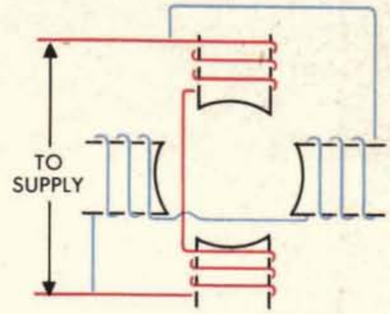
Considering only one of the coils, the direction of the flux at a particular instant can be shown as passing from the *N* pole to the *S* pole.



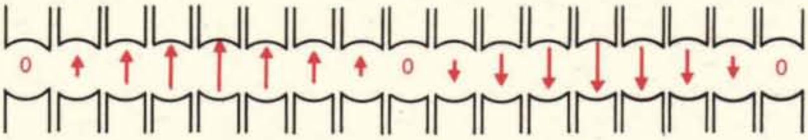
When the alternating current reverses direction, the poles are reversed, and the direction of the flux reverses.



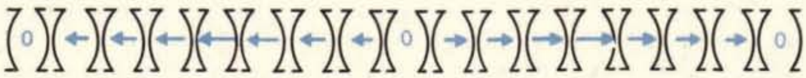
Now, although the current reverses direction rapidly (120 times per second), it is incorrect to consider it as reversing in a series of sharp, clear-cut jerks. It is more correct to consider it as growing in intensity, in one direction, and then fading out to zero, growing in intensity in the opposite direction, and again fading out to zero—as indicated in the graph. The magnetic field can be considered as growing in intensity in one direction and fading gradually to zero, and then doing the same thing in the opposite direction.



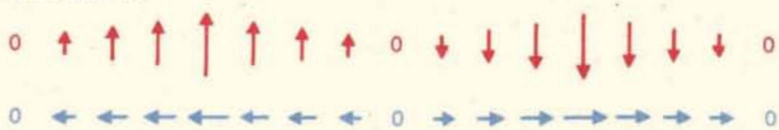
Schematically these pulsations may be represented by a series of arrows between the poles:



Pulsations between the second set of poles (which lie at right angles to the pair just considered) may be shown in the same way:

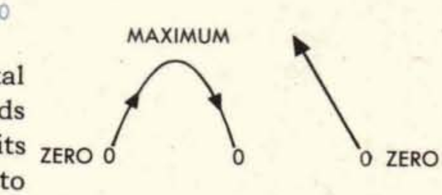


Considering the two sets of poles together with both coils supplied from the same source, the pulsations in the magnetic field between the four poles of the servo motor may be shown as follows:

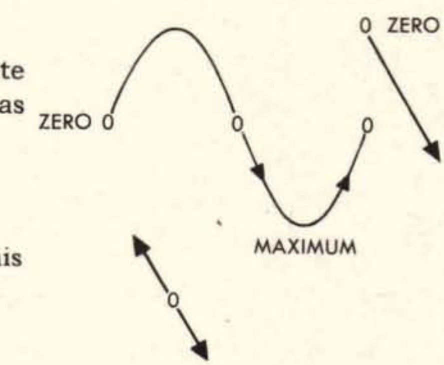


The direction in which the resultant effect of these two fields operates, at any instant, can be shown by adding any two of the components shown here, and tracing the diagonal of the parallelogram obtained:

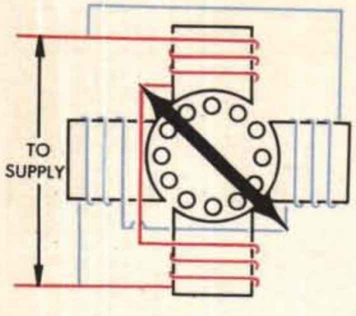
Adding these resultants together, it is found that the total effect of the current in the coils is to create a field which builds up in intensity in one direction, as the current builds up to its maximum, and then fades to zero as the current decreases to zero - - -



- - - - after which the field builds up in intensity in the opposite direction, as the current reverses, and again fades to zero as the current completes the cycle.

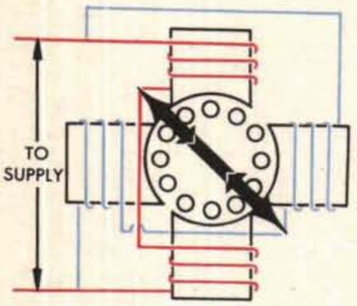


The lines of force, or flux, act along a straight line, and this stator field can be termed an oscillating, or pulsating, field.



How the stator field affects the rotor

When a rotor is placed in a pulsating field, the lines of force cut through the conductors on the rotor, first in one direction and then in the opposite. This varying flux has the same effect as if the rotor conductors themselves were moved through a magnetic field. An electrical pressure is induced in the conductors. Since the ends of the conductors are joined to common end rings, currents flow around the closed circuits thus provided.



The rotor field

Now, according to Lenz's law of physics, voltage induced in a conductor sets up a current whose magnetic field always opposes any change in the existing magnetic field. The magnetic field set up by the current in the conductors of the rotor therefore opposes changes in the magnetic field set up by the current in the stator coils of the motor. In other words, each time the action of the field due to current flowing through the stator coils can be represented as building up in one direction, the action of the field due to current induced in the rotor can be represented as building up in the opposite direction.

Characteristics of the two fields

It will be noted that both stator and rotor fields build up (and collapse) along a straight line, which passes through the center of the rotor.

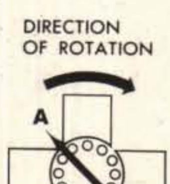
Another characteristic of these fields is that the stator field and the rotor field do not build up evenly at the same time. The field due to current in the stator coils, reaches peak intensity, and is actually collapsing, before the field due to current induced in the rotor reaches peak intensity. That is, the induced current lags the applied current.

Since the resultant flux in both the stator and rotor fields acts along the same straight line through the center of the rotor, however, no force is produced in the form of a torque which will make the rotor turn.

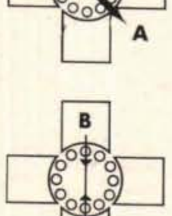
DIRECTION OF ROTATION

Making the rotor turn

Suppose the rotor of the motor is now given a spin by hand in the direction shown. The path taken by the resultant flux of the stator field remains in the same position—indicated here by the line *AA*. But since the induced current lags the applied current, it means that the current induced in conductors of the rotor, as these conductors move through the path taken by the flux of the stator field, does not reach peak until the conductors have *passed* line *AA*.



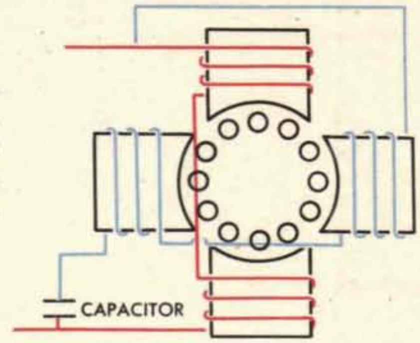
Here the induced current in conductors which have just passed through *AA*, reaches peak intensity at *BB*. The resultant flux of the rotor field, therefore, no longer acts along the same straight line as taken by the flux of the stator field, but along a line which lies at an angle to it.



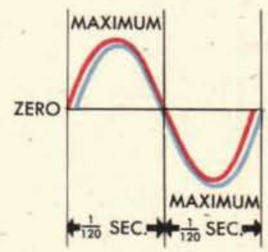
Under such conditions, attraction occurs between the poles of the two fields, as indicated here, and this results in a torque being applied to the rotor, causing it to keep turning.

The effect of the capacitor

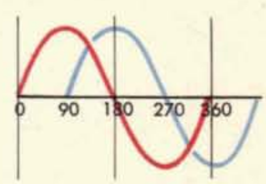
A capacitor is employed in order to make the rotor of an induction type motor, such as a servo, start turning. That is, a capacitor is used to do the job done by hand in the instance just described. To understand how a capacitor can cause a motor to start, consider the action of the lines of force, or flux, in the stator when no capacitor is present.



Without a capacitor, the current in the two stator coils builds up in the same direction simultaneously, reaching a peak value at the same instant. That is, the current in one stator coil is "in phase" with the current in the second stator coil, with respect to time.

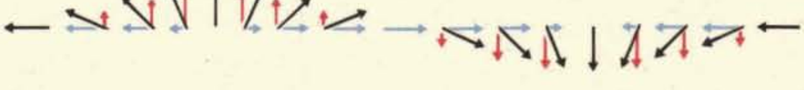


When a capacitor is used, the current in one coil reaches a peak *before* the current in the second coil, in the manner indicated here. That is, the currents are approximately 90° "out of phase" with respect to time.



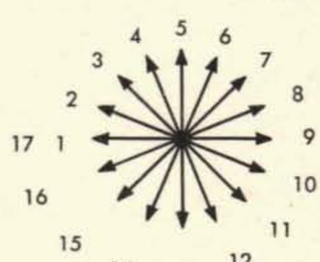
Using arrows again, the pulsations between the poles of the stator coils can now be schematically represented as follows:

Now note that when the current is a zero in one coil, it is at a maximum in the other coil. By adding components and tracing resultants, the following picture is obtained of the action of the stator field during the period of one current cycle (1/60 sec.).



1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17

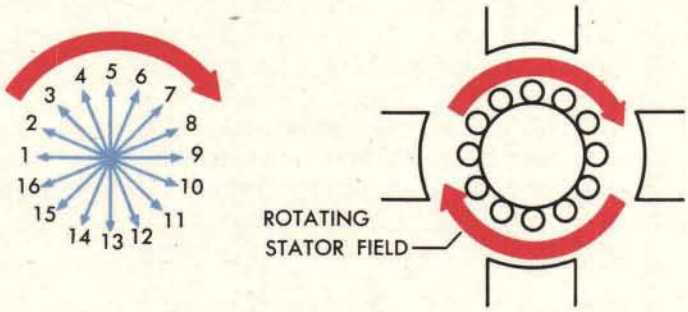
These resultants indicate how the path taken by the resultant flux of the stator field *changes position* from instant to instant. The stator field still remains an oscillating field, pulsating along a straight line which passes through the center of the rotor, but the position of the line now keeps changing.



For instance, the path along which the resultant flux acts, can first be indicated in position (1), and then in position (2), and so forth. Taking these positions in succession, it is found that they rotate through 360°. In other words, the stator field now not only oscillates along a straight line through the center of the rotor, but *also rotates.*

The rotating stator field and the induced current lag both act to make the rotor turn

Placing a capacitor in the circuit, therefore, has caused the stator field to rotate. In this way, a torque is developed which starts the motor running.



As the field rotates, the lines of force cut through the conductors of the rotor, inducing voltages in the conductors. And since the conductors are joined to common end rings, currents flow in these conductors.

Also, since current induced in the conductors at any instant lags the current in the stator coils, it means that at each instant a condition exists such as described previously, where the resultant flux of the induced field acts at an angle to the path taken by the resultant flux of the stator field. That is, at the proper instant, each conductor has a force acting upon it (due to attraction between the poles of the stator and rotor fields) which takes the form of a torque, pulling the conductor around.

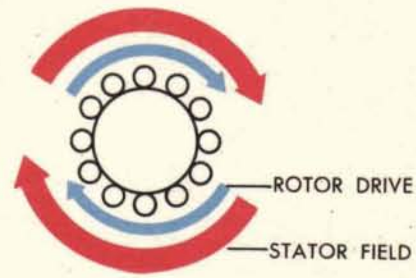
Once started, induction-type motors quickly develop maximum torque under normal load.

If too small a capacitor is used for a motor of a given size, sufficient starting torque is not developed under normal load. If too large a capacitor is used, the starting torque will be large, but torque will diminish when the motor picks up speed. This is because too powerful a starting torque results in the poles of the stator and rotor fields becoming separated by a considerable distance, and this, in turn, results in the repulsion between the poles being diminished.

With any given motor, use only the capacitor specified.

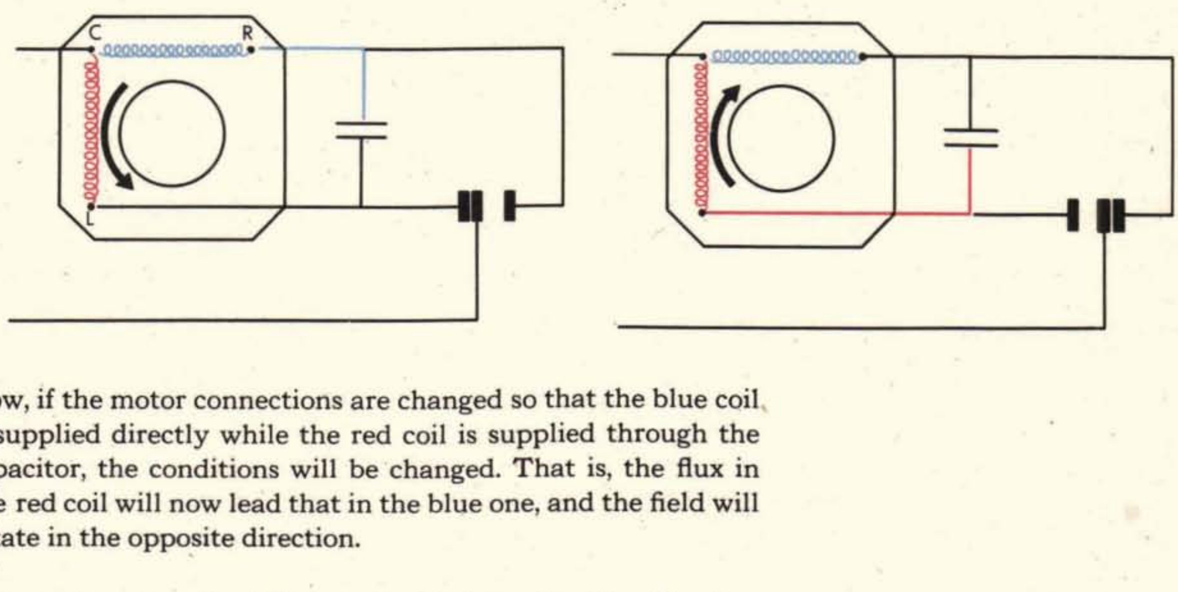
How the rotor revolves

The other requirement of the follow-up motor is that it will drive in the direction required.



The force exerted on the conductors always acts to pull the conductors in the direction in which the stator field is moving. By controlling the direction of rotation of the stator field, the direction of rotation of the rotor is controlled.

Looking back to page 345, it will be seen that with the capacitor in the circuit as shown, the flux of the blue coil reaches a maximum value at 1, while the flux in the red coil does not reach its maximum value until 5. In other words, the flux in the blue coil is "leading" that in the red coil.



Now, if the motor connections are changed so that the blue coil is supplied directly while the red coil is supplied through the capacitor, the conditions will be changed. That is, the flux in the red coil will now lead that in the blue one, and the field will rotate in the opposite direction.

This explains how the follow-up is able to control the direction of rotation of the servo motor by shifting the capacitor from one part of the circuit to the other.

SECTION 6

ARMA MECHANISMS

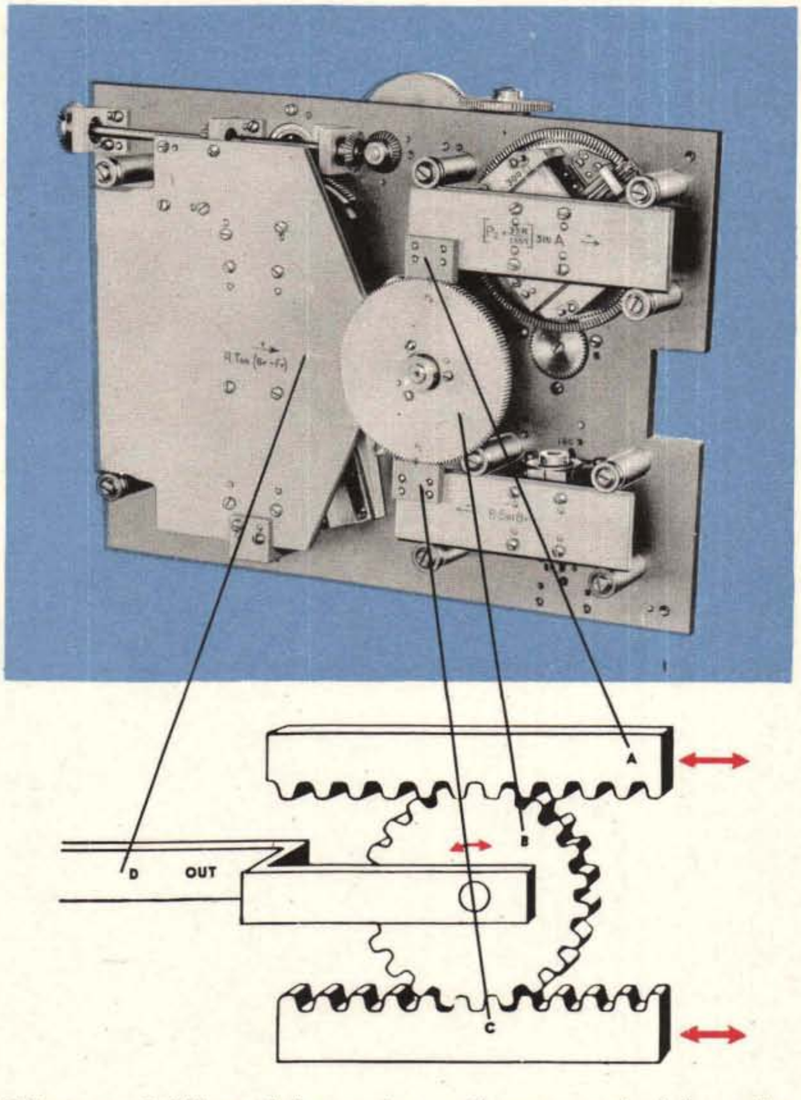
The Units described in this section are used in the ARMA Torpedo Data Computers.

The principles of operation of several of the units have already been explained in detail in describing such Ford mechanisms as differentials, integrators, and component solvers. The reader is, therefore, assumed to be fairly familiar with these principles, and the descriptions in this section have been made as brief as possible.

To a great extent the concise explanations published in the technical literature of the manufacturer have been used.

DIFFERENTIALS

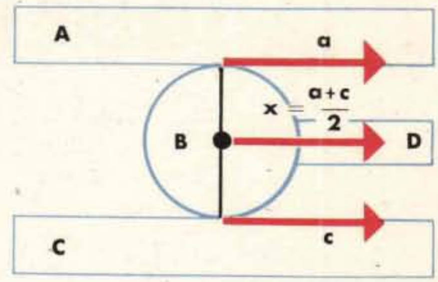
LINEAR RACK AND PINION differential



This type of differential operates on the same principle as the bevel gear differential.

Inputs to this differential, however, are not made by rotation of end gears, but by straight line movement of the two racks, *A* and *C*.

The output is proportional to the linear movement of pinion *B*.



Suppose both racks are moved the same distance, in the same direction, at the same time. The pinion B will not roll, because the torque applied by rack A to the top of the pinion is equal and opposite to that applied by rack C to the bottom of the pinion.

Since the pinion does not turn, it is carried forward the same distance as the racks are moved.

Suppose rack A is moved by an input a, and rack C is moved by an input c. The total input to the differential will be $a + c$. Pinion B will move a distance x. If a and c are equal, x will be equal to both a and c, or $x = \frac{a + c}{2}$.

Movement of pinion B therefore represents half the sum of the two input movements.

Pinion B positions an output mechanism which transmits the output motion to carriage D.

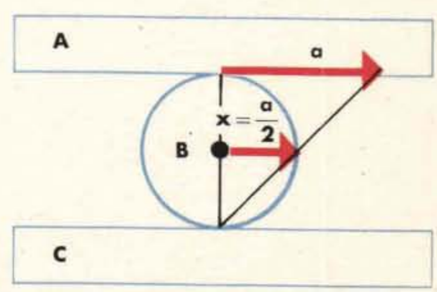
With 1:1 gear ratio between the output mechanism and the gear which actually moves carriage D, one half the sum of the inputs $\frac{a + c}{2}$ positions the carriage.

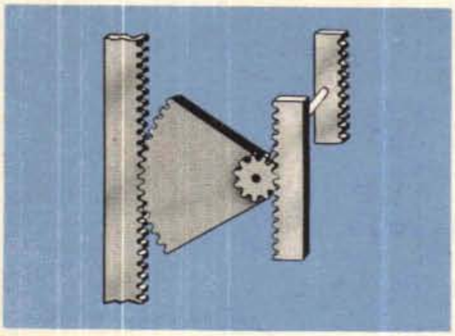
By using a 2:1 gear ratio, however, the sum total of the inputs $a + c$ positions the carriage.

If rack A moves an amount a in the direction shown and rack C remains stationary, pinion B will roll on rack C, and its center will move half as far as rack A is moved. This is explained by the ruler and cylindrical drinking glass experiment shown on page 41.

B will move through a distance x, which is equal to $\frac{a}{2}$.

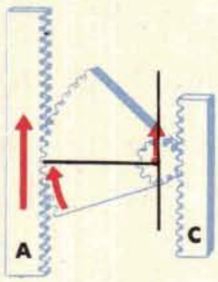
With 2:1 gearing, the output can be doubled, so that movement $\frac{a}{2}$ by the pinion results in an output value a positioning carriage D.





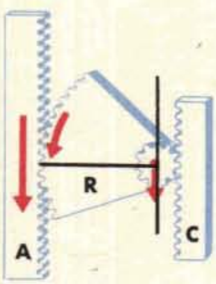
LINEAR RACK AND

In this differential, as in the linear rack and pinion differential, inputs are represented by straight-line movements of two racks. Instead of the output corresponding to the movement of a pinion, however, it corresponds to the movement of a special type of follower which consists of a sector gear carrying a pinion on one side. The sector gear and pinion are in one piece; the pinion does not revolve by itself. Any movement of the sector gear moves the pinion, and vice versa.



If rack *C* remains stationary, and rack *A* moves upward, two things occur:

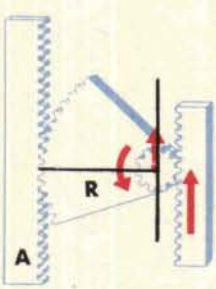
1. The sector gear rotates in an upward direction, following the movement of the rack.
2. The pinion, rotated by the movement of the sector gear, also moves upward—rolling on rack *C*.



Conversely, if rack *A* is moved downward, the pinion will again roll on rack *C*, this time downward.

No matter how far the pinion is moved along rack *C*, it will always remain at the same distance from rack *A*, because the teeth of the sector gear are on the circumference of a circle whose center is also the center of the pinion. The distance between the center of the pinion and rack *A* is always equal to the radius, *R*, of this circle.

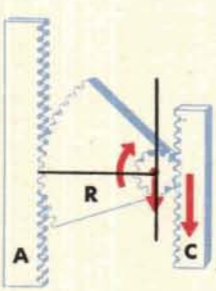
Any movement of rack A, therefore, results in a straight line movement of the pinion in the same direction as the rack is moved.



If rack *A* is held stationary and rack *C* is moved upward, the sector gear will be rotated counterclockwise, but the pinion will be moved upward in the same direction taken by the rack.

In the same way, if rack *C* is moved downward, the pinion will be moved downward.

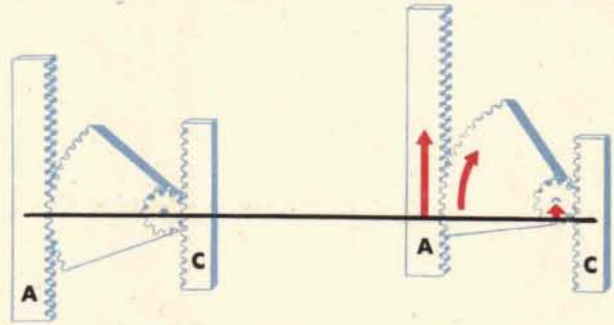
Any movement of rack C, therefore, results in a straight line movement of the pinion in the same direction as the rack is moved.



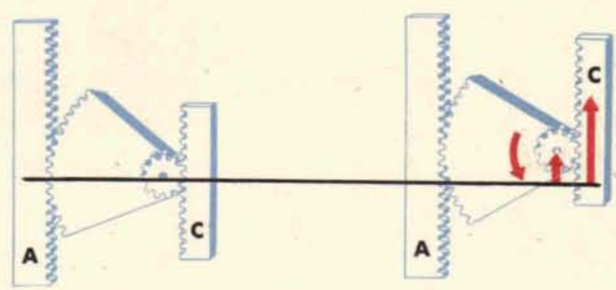
SECTOR GEAR differential

The motions of rack *A* and rack *C* do not affect the movement of the pinion equally. A given amount of movement on *A*, for example input *X*, may only cause the pinion to move a short distance, while the same amount of movement on *C* will cause the pinion to move through a comparatively long distance.

However, the distance the pinion is moved by rack *A* is *proportional* to the input on *A*, and the distance the pinion is moved by rack *C* is *proportional* to the input on *C*.



INPUT MOVES ONLY RACK A

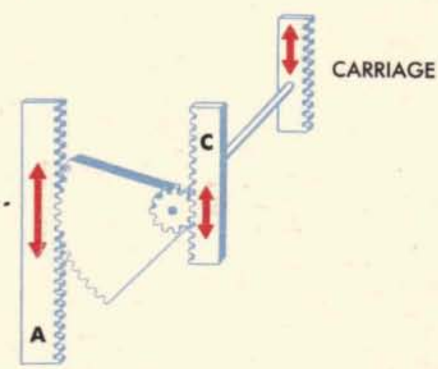


INPUT MOVES ONLY RACK C

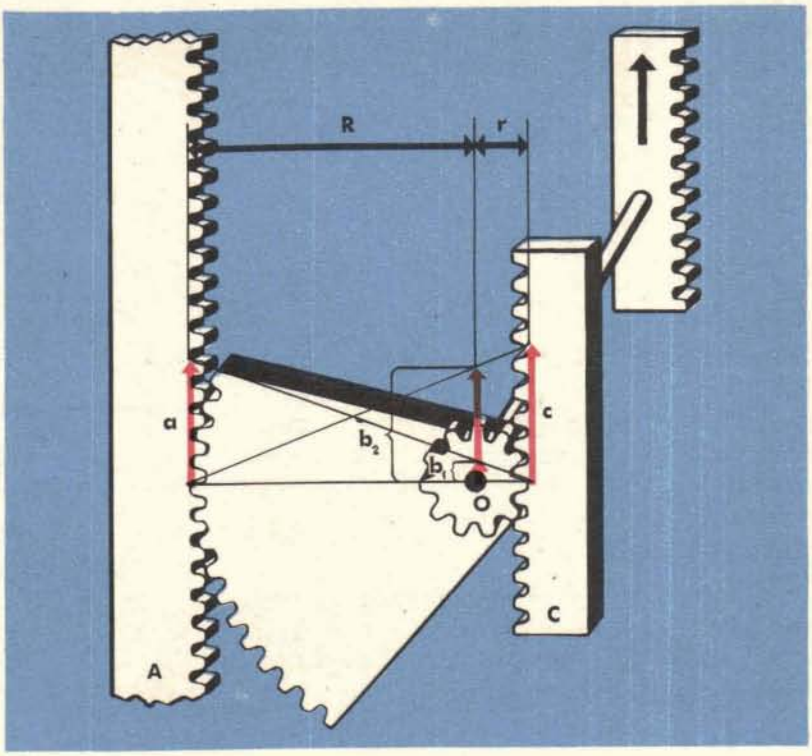
When both rack *A* and rack *C* are moved, therefore, the amount the pinion is moved is proportional to the sum or difference of the inputs on *A* and *C*.

In this way, movement of the pinion corresponds to the output of the differential.

If the pinion is attached to a carriage, any movement of the pinion will move the carriage. The carriage will move an amount proportional to the sum of the inputs of the differential.



Movement of follower in the linear sector gear differential



The amount which the pinion is moved when the racks are moved is computed as follows:

Let O be the center of the pinion. Let R be the radius of the sector gear, and r the radius of the pinion.

Assume that the movement of O due to the displacement a of rack A is b_1; and that the movement of O due to displacement c of rack C is b_2. The total movement of O is equal to the sum of the two components, or $b_1 + b_2$.

To find the values of b_1 and b_2, consider the two pairs of triangles shown. One pair consists of the triangle with side a and base $R + r$, and the triangle with side b_1 and base r; the other pair consists of the triangle with side c and base $R + r$, and the triangle with side b_2 and base R.

In the first pair of triangles, $\frac{b_1}{r} = \frac{a}{R+r}$, or $b_1 = \frac{ar}{R+r}$

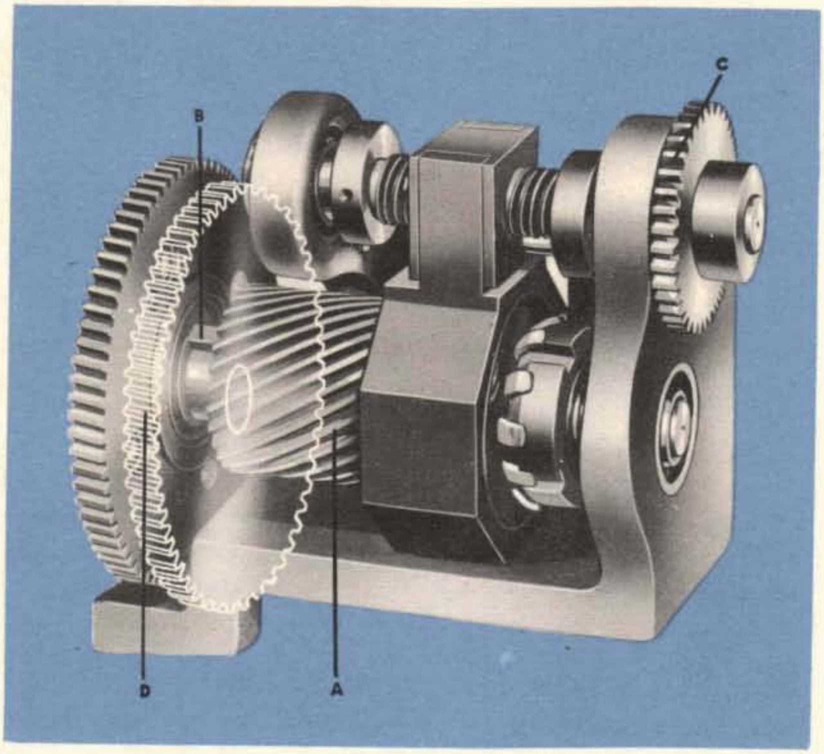
In the second pair of triangles, $\frac{b_2}{R} = \frac{c}{R+r}$, or $b_2 = \frac{cR}{R+r}$

Adding the two equations gives:

$$b_1 + b_2 = \frac{ar}{R+r} + \frac{cR}{R+r} = \frac{ar + cR}{R+r}$$

Since both R and r are constants, the output of the differential, $b_1 + b_2$, is proportional to the sum of the inputs a and c — using the proper proportionality constants.

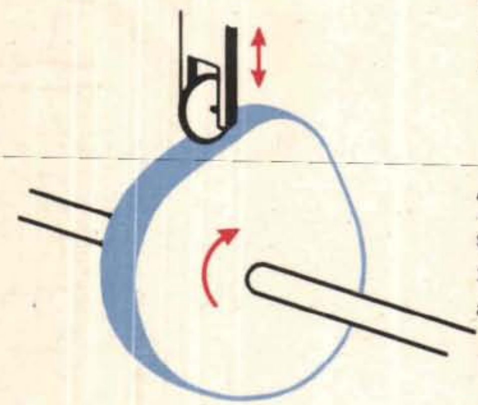
HELICAL GEAR differential



Here is an unusual application of the differential principle. The helical pinion *A* can be moved axially on shaft *B*. This movement is accomplished by turning the input gear *C* which rotates a threaded shaft on which is mounted a carriage. This carriage is fastened to the pinion *A* and carries the pinion with it as it moves along the threaded shaft.

The pinion *A* meshes with a helical gear *D* so that as the pinion moves axially, it is also turned an amount which depends upon the magnitude of the axial movement. In this respect it acts like a screw. In addition the pinion receives a rotary input from the helical gear *D*. The output shaft *B* receives the added motions. The output on shaft *B* is always proportional to the input at gear *C* plus the input at gear *D*.

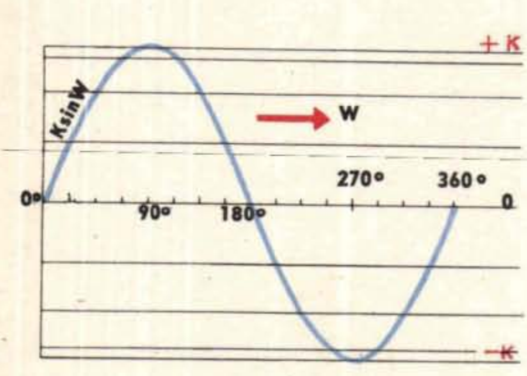
CAMS

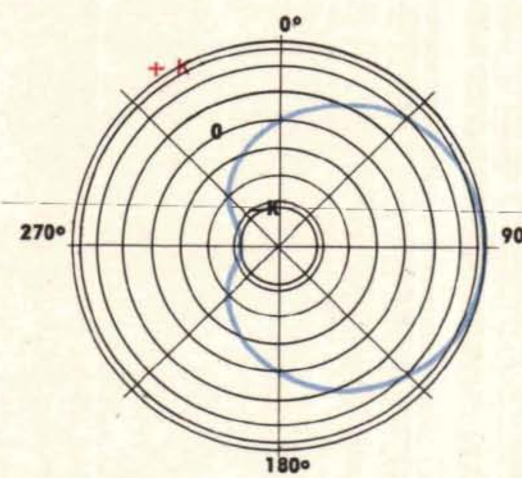


The simplest form of cam is a wheel with some sort of irregularity on its outside edge (periphery). A follower rides on the wheel and reproduces the irregularity in a reciprocating motion. A cam of this type is shown here.

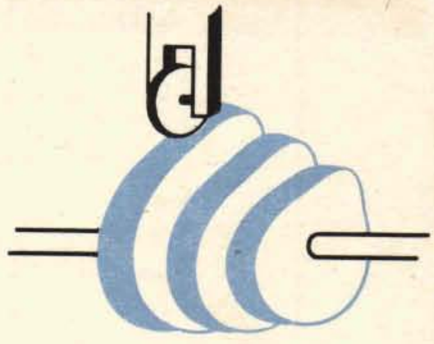
The cam can be designed so that the follower will move to satisfy a mathematical function of the cam input. For example, it is possible to obtain a follower movement equal to $K \sin W$, as represented in the graph below, if K is a constant.

Such a cam is constructed with a gradually changing radius, so that the periphery of the cam causes the follower to move up and down with the correct motion in relation to the turning of the cam. This cam is shown in the figure below. It is nothing more than the curve of the graph wrapped around a circle.

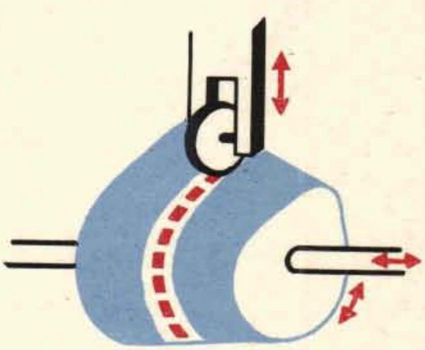




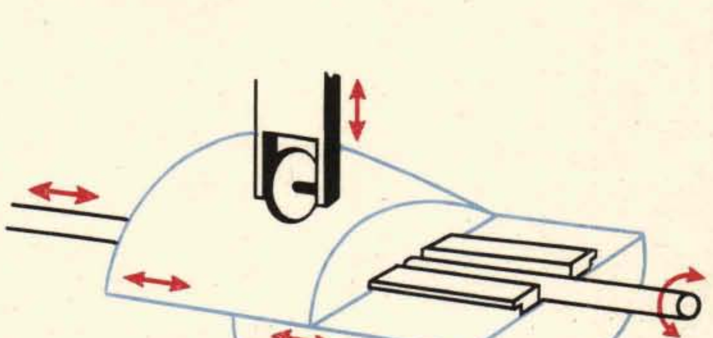
If a cam follower is to travel along different paths with changes in setting, several cams may be provided, so that selection of the proper cam produces the desired follower motion. A set of cams permits the use of another variable quantity.



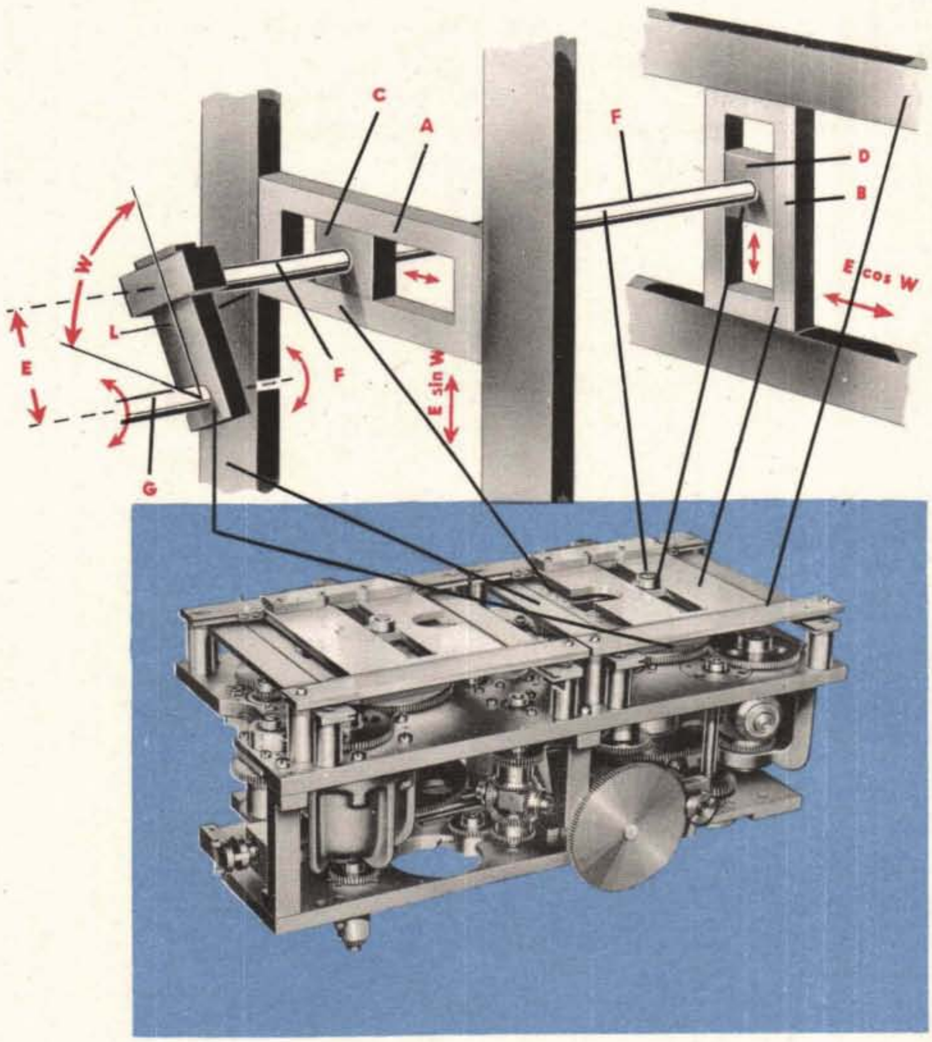
A single three-dimensional cam, as shown here, is an improvement on the set. The contour in contact with the roller is determined by the longitudinal position of the cam, instead of by changing cams. The third variable is introduced here by the axial positioning of the shaft.



The three-dimensional cam is sometimes used in a special form. The cam is split so that the two halves may be moved relative to each other. The split introduces an adjustment which is equivalent to providing a set of three-dimensional cams, any one of which may be selected depending upon the position of the two cam halves.



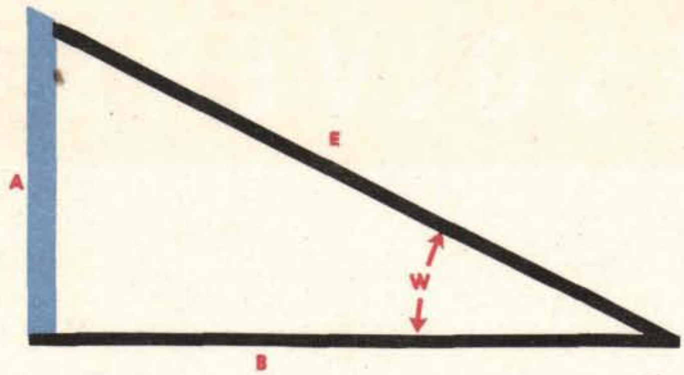
RESOLVERS



The inputs to this resolver consist of a given quantity and an angle which set up a vector. It is the job of the resolver to break down, or "resolve" the vector into two components at right angles to each other.

Let E be the given quantity. This can be set into the mechanism by varying the length of arm L.

Let W be the angle. This can be set in by rotating arm L the required number of degrees from the horizontal.



The resolver has two sliding carriages, A and B, which move in directions at right angles to each other. That is, carriage A moves up and down, whereas carriage B moves from side to side. The position which each carriage takes depends upon the position of the pin F and hence the blocks C and D. The pin is free to rotate in each of the blocks C and D which may travel in the slots of carriages A and B.

When the input shaft G rotates, pin F moves along an arc whose radius is E. Due to movement of pin F, the block C moves to a new location. However, in moving the block C, it is found that it can slide from side to side freely in the carriage A. Although it is not restricted in its side-to-side motion, the up-and-down motion of the block causes the carriage A to move up and down.

The block D also has the same movement as block C but it is permitted to slide up and down in the carriage B. Therefore, since the side-to-side motion of D is restricted, the carriage B must move from side to side with the block.

From the right triangle formed by E, A and B:

$$\sin W = \frac{A}{E} \qquad \text{or } E \sin W = A$$

$$\cos W = \frac{B}{E} \qquad \text{or } E \cos W = B$$

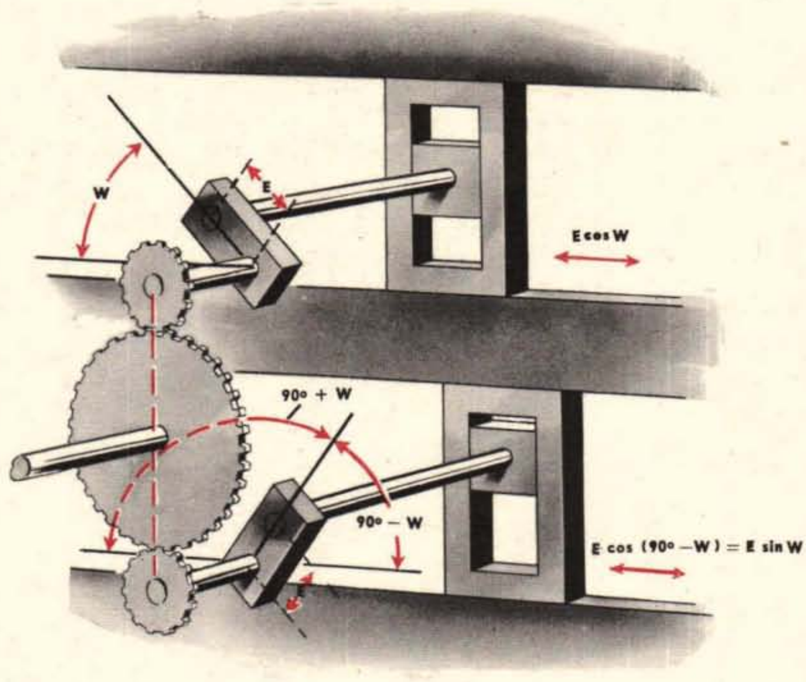
Any position of the arm L of the resolver determines some angle W. The carriage A moving up or down gives a position which determines $E \sin W$, while the motion of carriage B moving from side to side gives $E \cos W$.

A simplified form of resolver has an arm E which is not adjustable. This variation changes the outputs to $\cos W$ and $\sin W$ if the design is such that E may be used as equal to unity.

OTHER TYPES of RESOLVERS

A variation of the resolver is shown here. In this device there are two separate arms, which are always in positions 90° apart, and a carriage for each which moves from side to side. Each carriage will produce the cosine component of the angle between the arm and the horizontal. However, when the upper arm is at an angle W the lower arm is at the angle $90°-W$, as shown.

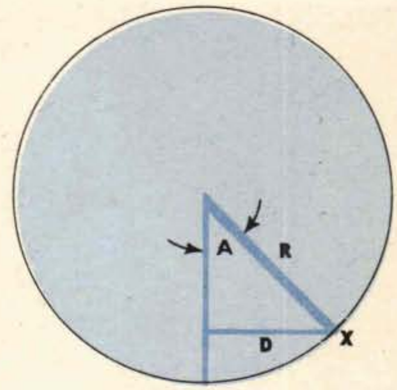
Since $\sin W = \cos(90°-W)$, the lower carriage not only gives $E \cos(90°-W)$, but also $E \sin W$. The upper carriage gives $E \cos W$. This variation of the resolver is used when it is desired to have the two carriages move in the same direction.



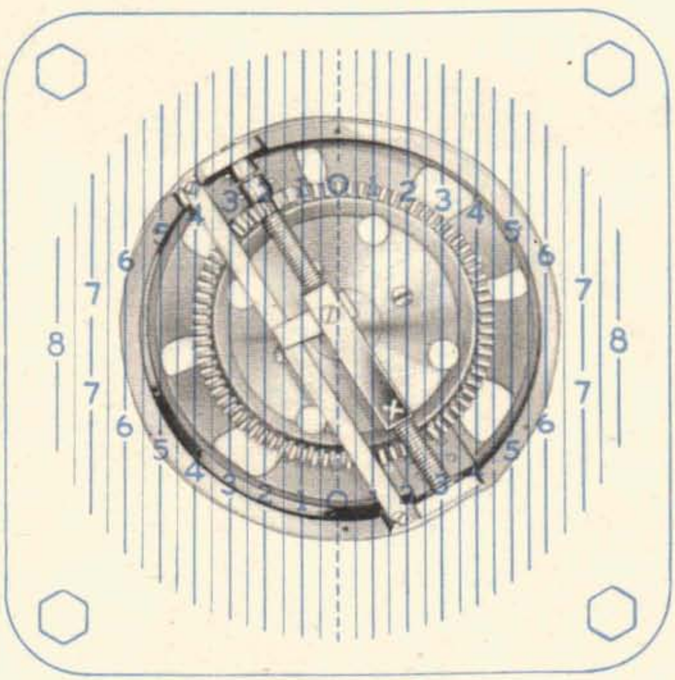
Another variation of a resolver uses the simple trigonometry equation:

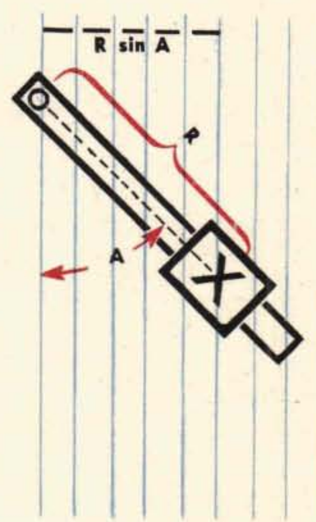
$$\text{Sine of an angle} = \frac{\text{opposite side}}{\text{hypotenuse}}$$

For example, in this triangle, $\sin A = \frac{D}{R}$, or $D = R \sin A$



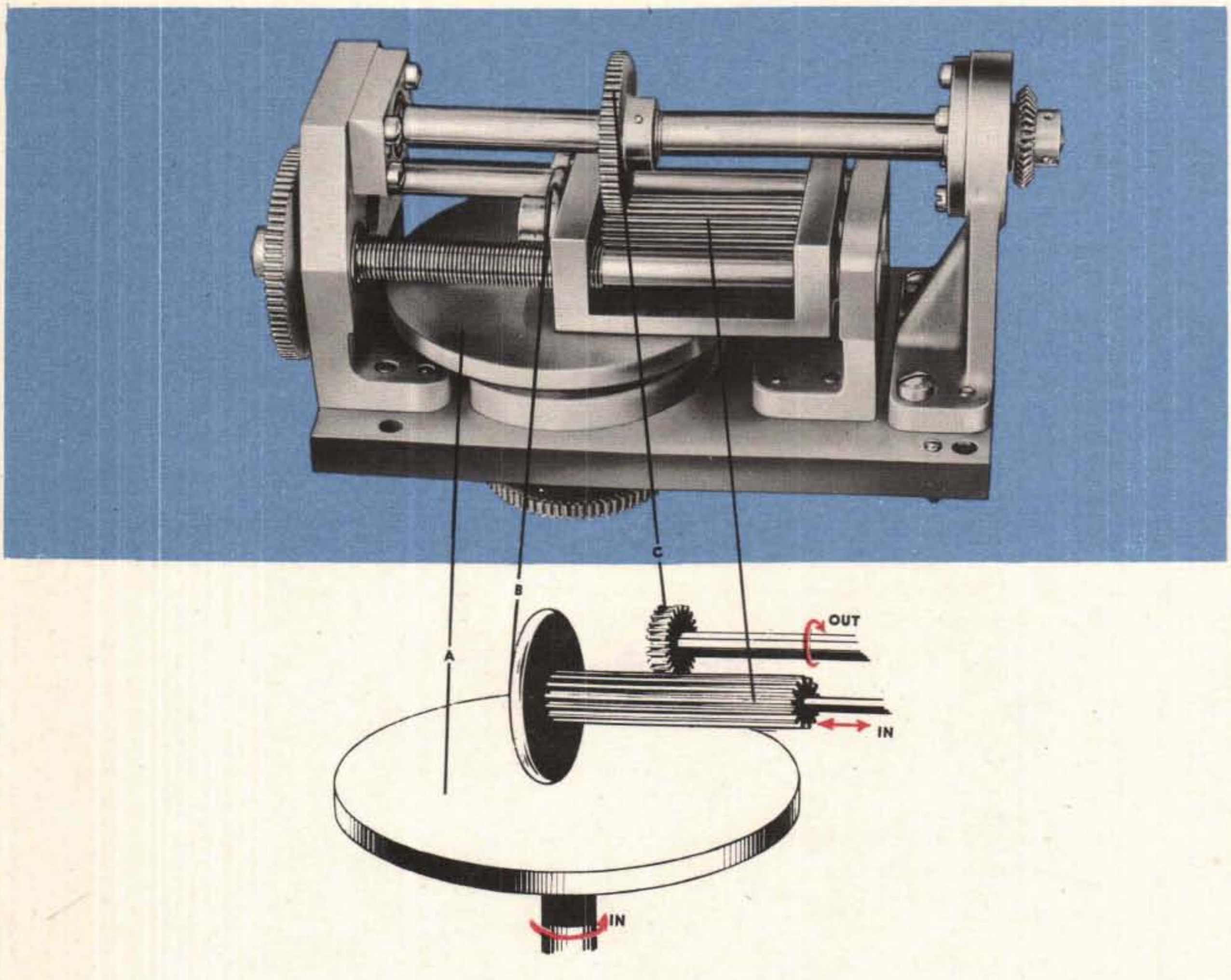
For different values of angle A, the length D will change and the point X will be various distances above the base of the triangle. Hence a resolver may be constructed to give values of $R \sin A$.





In this resolver a nut travels on a threaded rod or arm. The angular position of the arm is determined by the angle A. The value of R is the distance from the pivot to the X mark on the nut and is determined by the rotation of the rod about its own axis. The nut is prevented from rotating, and as the rod rotates, the nut travels axially along it. $R \sin A$ is read from the intersection of the X mark with the vertical $R \sin A$ graduations. $R \sin A$ may be varied either by changing angle A, changing the position of the nut on the arm, or both.

INTEGRATORS



This integrator is used for multiplying two values, each of which may vary continually. The values are represented by inputs and the integrator gives the product of these inputs at each instant.

If the integrator is connected to a counter, these instantaneous products will be set into the counter as they are obtained. That is, the quantity shown on the counter at any instant will represent the sum of all the instantaneous products obtained up to the moment a reading is taken.

The integrator, therefore, not only multiplies, but sums up the instantaneous products of the two inputs as these inputs vary.

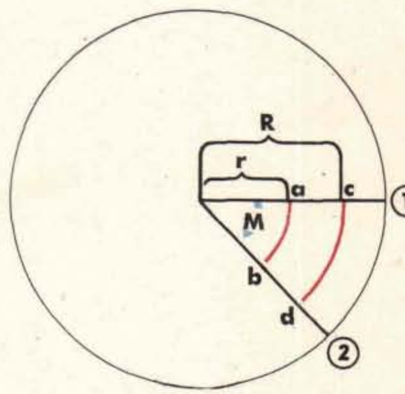
When used in conjunction with a motor and a follow-up head, this unit may also be used for division. See page 370.

One of the inputs to the integrator rotates the large disk *A*, and the other input consists of locating the follower roller *B* at various radii on the disk. As the disk revolves, it causes the roller to rotate and transmit the motion to the output pinion C.

The amount of rotation of *B* depends upon its distance from the center of disk *A*, and upon the rotation of *A*. If disk *A* is rotated and *B* is at the center of *A*, there will be no rotation of *B*. But if *B* is moved away from the center of *A*, *B* will start to rotate, and its speed of rotation will increase the nearer it is brought to the edge of the disk.

In other words, for a given speed of the disk, the rotation of *B* is zero when the point of contact is at the center of the disk, and rotation of *B* is maximum when the point of contact is at the edge of the disk.

Here motion of the disk from position 1 to position 2 through the angle *M* is shown. If a point *a* is at the radius *r*, this point moves through the arc during the rotation.



1 arc $ab = M \times r$

where the angle *M* is expressed in radians. When another point c is at a radius *R*, its movement equals the arc *cd*.

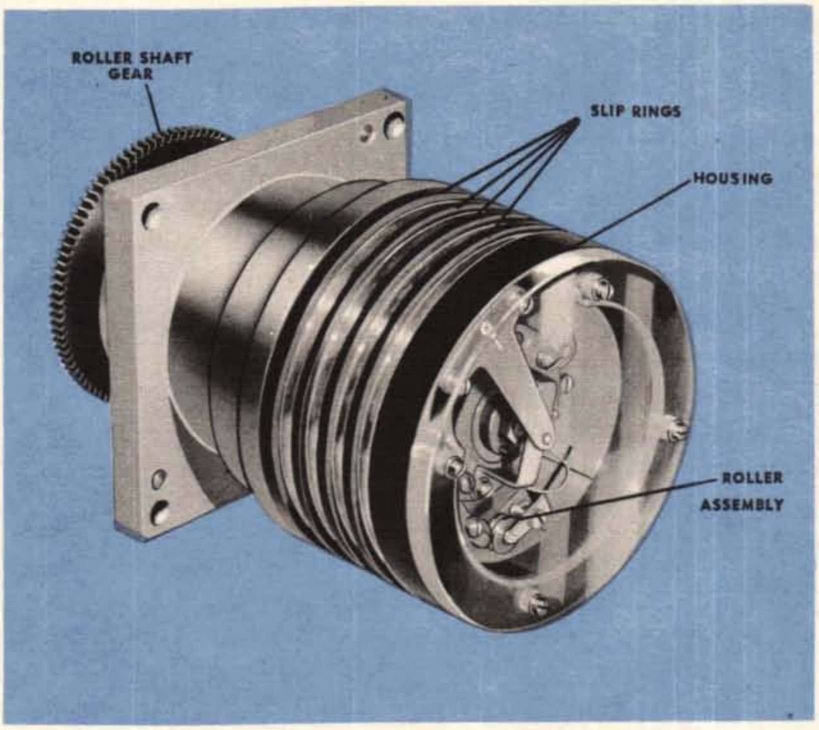
2 arc $cd = M \times R$

where the angle *M* is again expressed in radians. Then from equations 1 and 2

3 $\dfrac{\text{arc } ab}{\text{arc } cd} = \dfrac{Mr}{MR} = \dfrac{r}{R}$

If the follower rolls on the disk without slipping, it rolls an amount equal to the arc *ab* when the point of contact is a distance *r* from the center and the rotation of the follower equals the amount of $M \times r$. Also it can be seen from Equation 3 that the amount of movement is directly proportional to the radius. When the radius is zero or when *M* is zero, $M \times r = 0$ and the follower does not move. Since the rotation of the follower equals $M \times r$ the mechanism creates the sum of instantaneous products of *M* and *r* as either or both of these quantities vary. When neither *M* nor *r* vary, the integrator operates as a simple gear ratio and the result is pure multiplication.

FOLLOW-UP HEADS

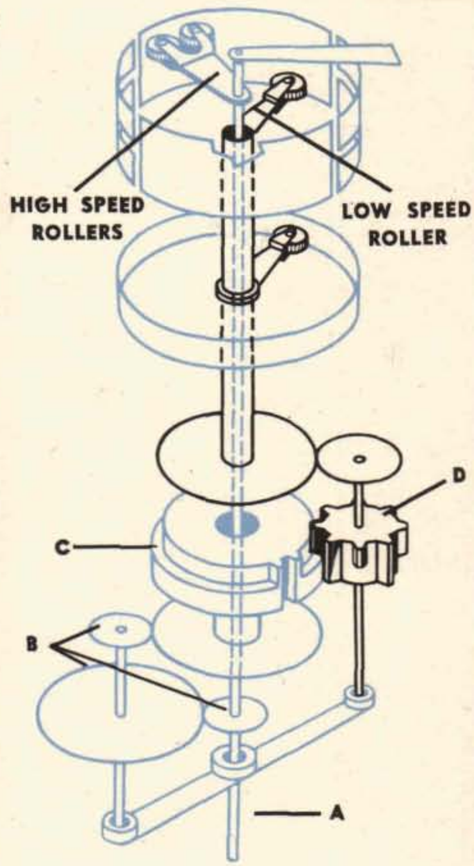


The follow-up head shown here has two uses. First, it may be used in a mechanical circuit to control electrically the direction and amount of rotation of a follow-up motor to increase the driving power in the system. The follow-up head may also be used for matching mathematical quantities.

In most applications, the input drive displaces the head of the unit, while the response drive from the follow-up motor enters directly (or indirectly through an element in the mechanical circuit) and matches the input drive by displacing the rollers. However, it can be used with the response drive connected to the head and the input connected to the rollers.

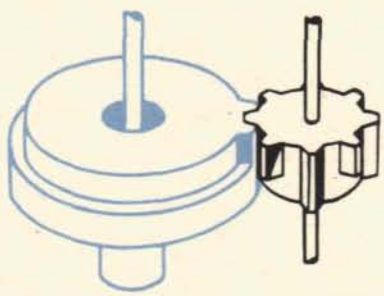
The parts and how they work together

Here is the mechanism shown diagrammatically. The main driving shaft *A* is directly connected to the high-speed rollers and also to a reduction gear train *B*, which, through a transfer gear assembly, drives a low-speed roller.

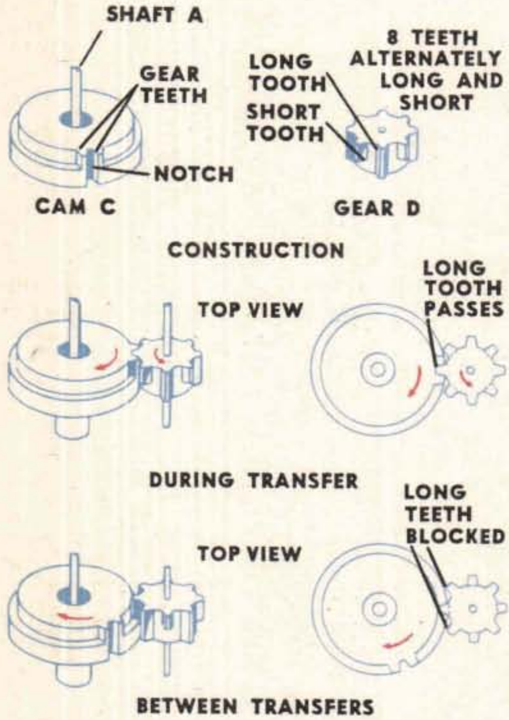


The gears *B* provide a 9 to 1 reduction in speed which is delivered to the locking cam *C* and the intermittent gear *D*. The upper half of cam *C* is a portion of a gear made with a pitch diameter for 20 teeth, only two of which are left on the periphery, as shown. Therefore, the two teeth require 2/20 or 0.1 revolution for their action period, being inactive then for the remaining 0.9 of each revolution.

Since *C* only moves 1/9 of the speed of *A*, the shaft *A* must turn 9×0.1 or 0.9 revolution in order to make the two teeth on cam C go through a complete transfer. When the transfer mechanism is in the position shown in the figure below, the intermittent gear is halfway through a transfer motion, and it will require only 0.45 revolution of *A* in either direction to complete the movement. After a transfer has been completed, shaft *A* must turn through 9×0.9 or 8.1 revolutions before a transfer begins again. The low-speed roller is geared so that it moves 45° during a complete transfer. It is prevented from moving during 8.1 revolutions of shaft *A* by the locking cam *C*. During the transfer motion a notch in the cam permits passage of one long tooth on the intermittent gear, but further motion of the intermittent gear, which drives the low-speed roller, is prevented by the lack of notches and the absence of driving gear teeth on the top half of cam *C*.



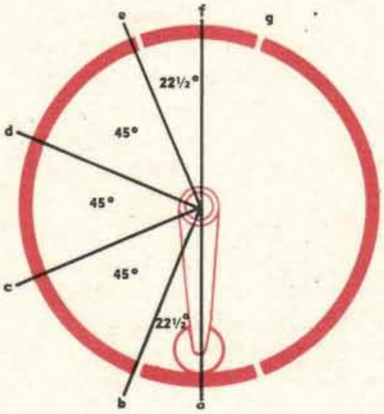
The INTERMITTENT GEARING



In this illustration the constructional details of the cam C and gear D, as well as the locking action between transfers, are shown. The single notch in the lower portion of cam C is a continuation of the "valley" between the two driving teeth on the upper portion of the cam. The diameter of the lower part of the cam is the same as the major diameter of the gear which forms the upper portion of the cam. The diameter of the greater part of the upper portion is the same as the minor (root) diameter of the two gear teeth cut on it. When the cam and gear are in the transfer position, the two teeth on the upper part of cam C engage and move one of the long teeth on the gear D, while the notch in the lower part of the cam allows this tooth free movement to complete a transfer, producing a one-quarter turn of the intermittent gear. As the tooth leaves the notch, another long tooth approaches and makes contact with the lower peripheral surface of the cam to be in position to enter the notch during its next time around. During this waiting period, therefore, two long teeth on the intermittent gear are in contact with the notchless surface of the cam, thus locking the gear in this position until the notch in the cam again arrives at one of these two teeth. During this time also, one of the short teeth of the gear D rests in the clearance space provided by the missing teeth on the upper portion of the cam, and awaits contact by one of the two teeth on the cam. The four short teeth, alternately spaced between the long teeth of the gear D, are merely for the purpose of starting motion of the intermittent gear when the cam notch has approached the position required for passage of one of the long teeth.

This type of follow-up head has a "storage" capacity of 35.55 revolutions of shaft A, obtained as follows: Since the low-speed roller only moves the amount provided by half a transfer before coming to a stop due to locking of the cam, it will move $22\frac{1}{2}°$ (starting with the tooth in the notch). During the $22\frac{1}{2}°$ motion, shaft A (and the high-speed roller) moves 0.45 revolution. The successive positions of the low-speed roller are shown on the next page.

The high-speed and low-speed rollers



In the "matched position" of the head, which occurs when the response drive has equalled the input drive and the follow-up motor actuated by the head is no longer energized, the low-speed roller is at *a*. If now the response drive remains de-energized while an input is applied to the follow-up head rollers, the following motions of the low-speed roller occur.

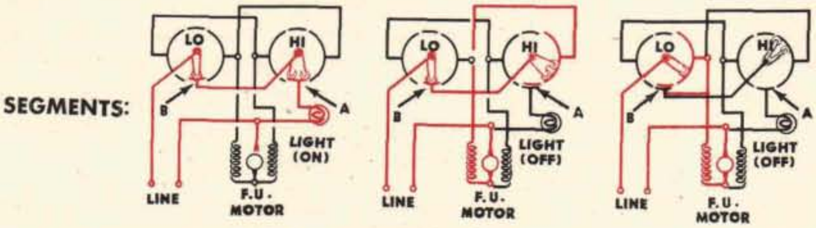
After the first one-half transfer, the point *b* is reached where the roller remains for the following 8.1 revolutions of *A*. Then, during the next 0.9 revolution of *A*, the roller travels to the point *c* where it remains for another 8.1 revolutions of *A*. Traveling to point *d* requires another 0.9 revolution of *A*, and after remaining there for 8.1 revolutions of *A*, another 0.9 revolution carries the roller to *e*. Shaft *A* can rotate another 8.1 revolutions before it starts to move the roller again, but any further movement of the roller will bring it to a dead segment at point *f*. The second half of the transfer would carry the roller to the contact at *g* which would connect the opposite field winding of the follow-up motor and cause the motor to turn in the wrong direction for synchronizing if the response drive were re-energized. Therefore, the total of all motion of *A* up to the beginning of the transfer of the roller from *e* to *f* adds up to 35.55 revolutions. This is the amount of displacement of shaft *A* of which the follow-up head is able to keep account. In other words, if there is any displacement of the shaft *A* up to a maximum of 35.55 turns, the follow-up will synchronize in the proper direction to match the input quantity.

This storage capacity is obtained in the same amount (35.55 turns) in either direction from the matched position. The addition of another transfer gear and intermittent gear could be made to multiply the storage capacity of the head by approximately ten to give a capacity of 359.55 turns in either direction.

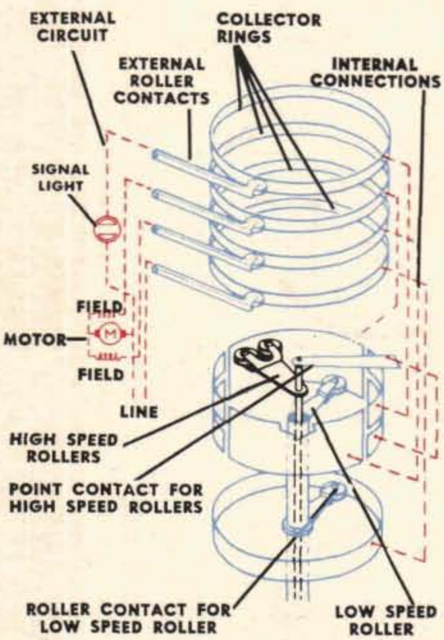
The circuits in high-speed and low-speed control

Movement of the high-speed rollers normally controls the follow-up motor without causing the low-speed roller to leave its matched position segment. Follow-up is continuous as long as the circuit is energized. However, if the circuit is open for any reason, the follow-up motor will not run, and under such conditions the ability of the follow-up head to store up a number of revolutions is of value. When the circuit is again closed, the motor can start in the proper direction and catch up to the driving shaft.

Three conditions may exist in the follow-up action.



MATCHED (A) HIGH SPEED ROLLER CONTROL (B) (C)



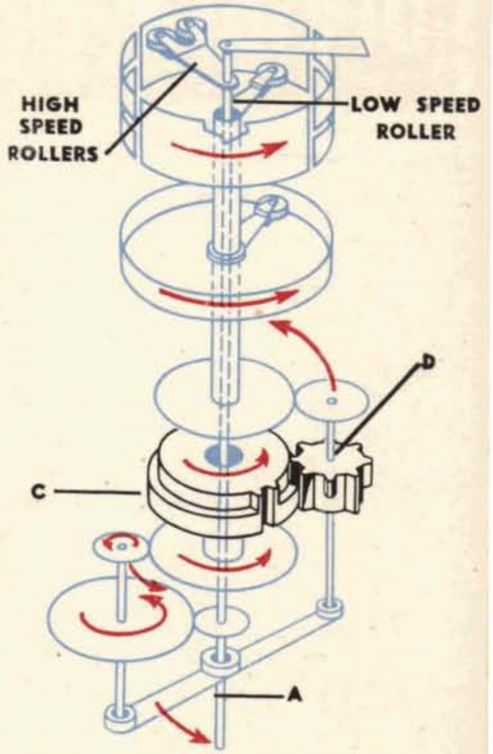
In the circuit diagrams, the red-colored leads indicate energized circuits. In sketch *A* the follow-up head is matched; the circuit to the follow-up motor is incomplete to both fields, and the indicating lamp is lighted. In sketch *B* the high-speed rollers have been displaced a few degrees, but not enough to cause transfer of the low-speed roller to controlling position. Now the motor is energized by completing the circuit through the proper field to cause it to run in the direction to restore the original relationship between the roller and segment *A*. Until this matching occurs the motor runs and the indicator light is off. In sketch *C* the head has stored up a few turns and the low-speed roller is in control of the motor. The high-speed roller is no longer energized and the indicating lamp is off. As the motor runs to restore the original matched relationship, the high-speed roller (or the head around the rollers) moves until a transfer places the low-speed roller on segment *B*, thus allowing the high-speed roller to assume control and bring the follow-up head into synchronism.

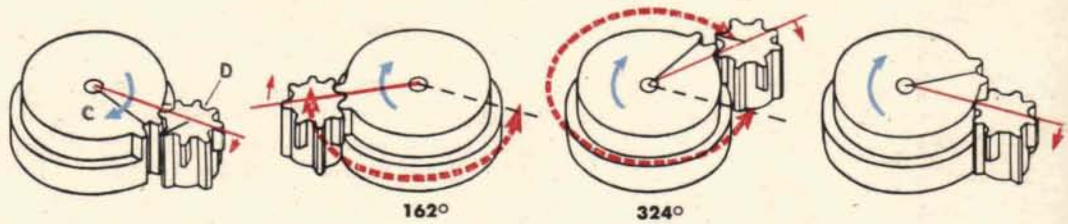
How the matched position is restored

If an input to the roller assembly has caused the head to store up a few turns, the action in restoring the matched condition is as follows: The shaft *A* and the high-speed rollers are assumed to be stationary at any position, and the follow-up motor is turning the head in the proper direction to "catch up" to the matched position. The head carries with it the gears and bearings of the transfer mechanism; therefore the low-speed roller and intermittent gear *D* will be carried along until the gear reaches the two teeth on the cam *C* and goes through a transfer. From previous explanations, it was seen that if the follow-up head were held stationary at this point and the shaft *A* turned 0.9 revolution, a transfer would take place, moving the low-speed roller 45°.

Similarly, at this point, if input shaft *A* is stationary and the head is turned 0.9 revolution, the low-speed roller will transfer 45° *with respect to its former position in the head.* However, this transfer takes place while the head and roller are still in motion, after which the low-speed roller and head travel together but in different relative positions until the next transfer. The roller loses ⅛ revolution in the head at each transfer. Since the 9 to 1 gear reduction mechanism is also traveling with the head, the cam *C* is traveling in the same direction, at a speed of 8/9 that of the intermittent gear and its supports; therefore, it will take nine revolutions of the head in order for the gear *D* to catch up to the transfer teeth of cam *C* and complete another transfer. (See illustration below.)

This sequence of transfers continues until a final transfer brings the low-speed roller onto the segment which transfers control to the high-speed rollers. The high-speed rollers then bring the follow-up head to its matched position and continue to keep it matched.

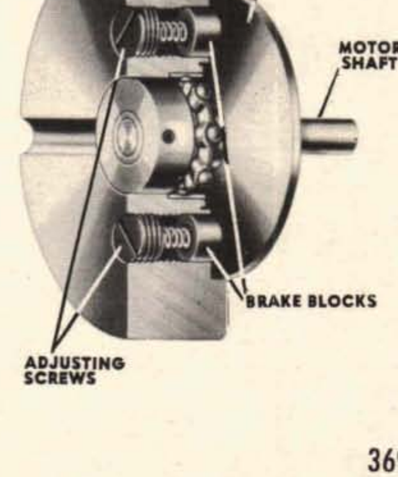




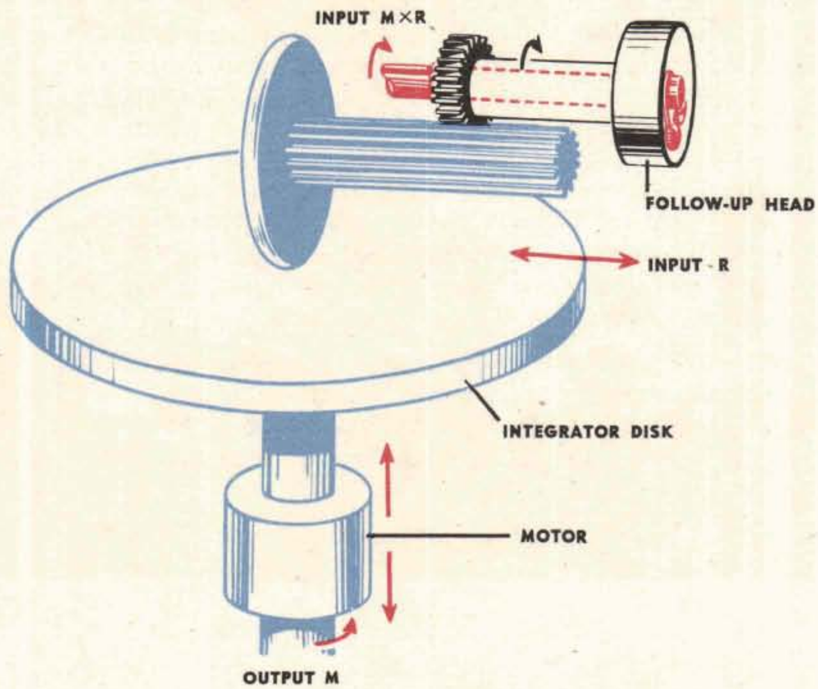
INTERMITTENT GEAR D GOING THROUGH TRANSFER MOTION

MECHANICAL DAMPERS

Mechanical dampers (inertia-type oscillation-damping devices) are employed with the follow-up motors to prevent the system from "hunting" as the matched position is reached. The damper shown consists of a flywheel carrying two brake-blocks which contact a disk pinned to the motor shaft. The only connection between the motor and the flywheel is by friction between the brake-blocks and the disk. As the motor tends to accelerate or decelerate, part of the inertia of the flywheel will attempt to prevent speed changes. The amount of flywheel effect may be varied by changing the amount of friction. This is accomplished by turning the adjusting screws which change the compression in the brake-block springs.



DIVIDER WITH FOLLOW-UP HEAD

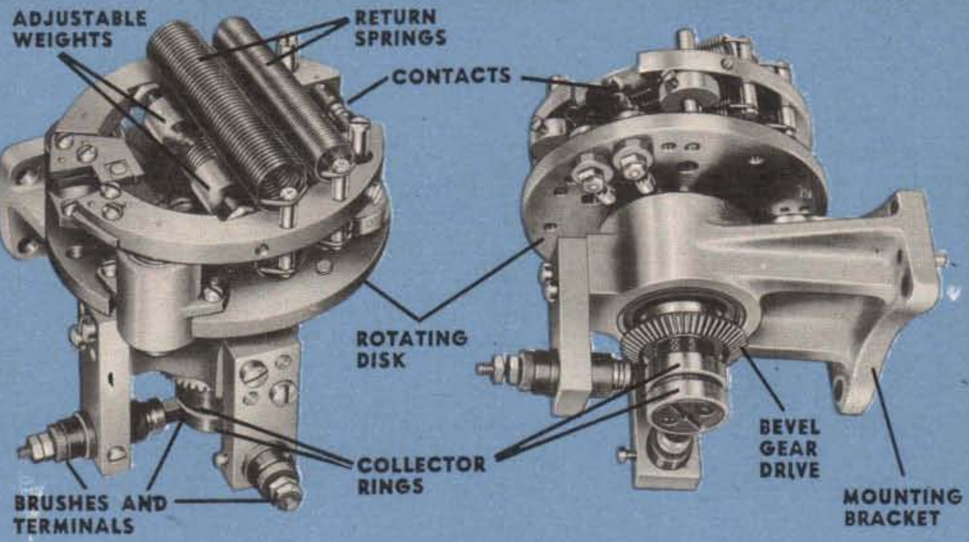


The follow-up head is used to give power to signals from other mechanisms. The follow-up motor will drive a load the same number of revolutions as the follow-up head is driven by a weak mechanism, which is itself unable to drive the load.

The follow-up head is also used for matching mathematical quantities in, for example, the integrator of the divider unit when performing division.

The divider unit is made up of three parts: an integrator, a follow-up head, and a follow-up motor. The integrator roller is positioned by one input (quantity R) and the other input (quantity $M \times R$) operates the follow-up head rollers. The motor which drives the integrator disk is controlled by the follow-up head in such a way that the disk rotates and causes the roller to turn the correct amount $M \times R$ to make the body of the follow-up head continuously match the follow-up head rollers. This disk rotation must be the quantity M, which then may be taken from the motor shaft for other uses. The follow-up head makes it possible, therefore, to match the given quantities R and $M \times R$ to obtain the desired quantity M as an output.

TIME MOTOR GOVERNOR



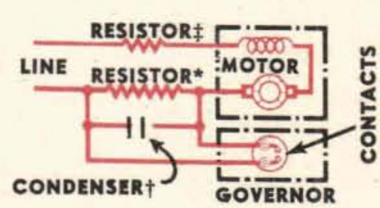
The cast aluminum mounting bracket which supports the governor assembly is secured so that the bevel gear engages a similar bevel gear on the output mechanism of the Time Motor. The bevel gear drives the rotating disk upon which are mounted two ball-bearing pivoted arms, each of which carries a contact and an adjustable weight. Collector rings and brushes serve to connect the contacts to the external circuit.

Four coil springs act to hold the contacts together. These springs are sufficiently strong to keep the two arms positioned with the contacts closed, so long as the motor does not exceed 1200 R.P.M.

When the motor runs at over 1200 R.P.M., the speed at which the governor assembly is rotated results in sufficient centrifugal force to overcome the action of the springs, causing both arms to be thrown outward. When this occurs, the contacts are separated. The current supply to the Time Motor is cut off, and the motor slows down.

The reduction in speed causes the centrifugal force to diminish and allows the springs to close the contacts again. This action continues several times per second and holds the motor to an average of 1200 R.P.M.

The fundamental circuit for this type of control is shown here.



SECTION 7
LINKAGE MECHANISMS

Other sections of this OP describe basic mechanisms consisting of gears, cams, and other rotating parts. Computations made by these basic units in the solution of a fire control problem are also described.

These same computations can be made with linkage mechanisms. Linkages are designed to accomplish the work of gearing units but with fewer moving parts. Linkage mechanisms contain links, levers, or cranks as essential computing elements. Like gearing units, linkage mechanisms may also include gears or slides. However, these gears or slides are always combined with purely linkage elements to form the complete linkage mechanism. In linkage mechanisms, values are usually represented by linear movements of links rather than by angular movements of gears.

This section describes the basic linkage mechanisms that are used for solving fire control problems. Adjustments that apply particularly to linkage mechanisms are illustrated.

This section also describes mechanisms that do not compute, but serve some other purpose such as limiting the movement of a linkage.

Finally, this section includes a description of a simplified network of linkage mechanisms, illustrating how they work together to solve a fire control problem.

Accuracy of Linkage Mechanisms

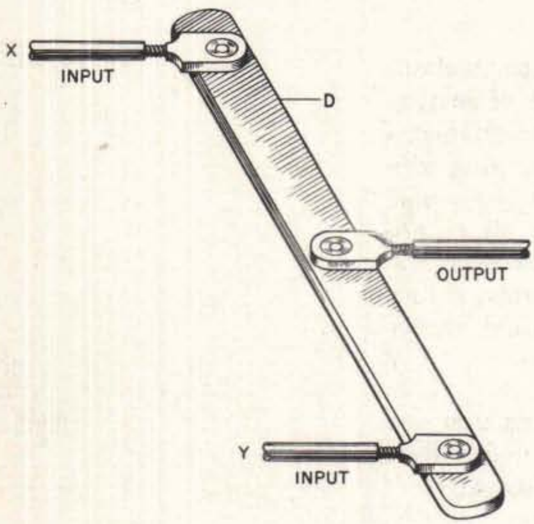
In illustrating the operation of linkage mechanisms, movements of links have been exaggerated for the sake of clearness. This creates the illusion that these linkages are inaccurate. Actually, the links are so long in proportion to their movements that any error introduced in their operation is negligible.

Further information concerning accuracy can be found under the heading DISTORTION, included in the description of each mechanism.

DIFFERENTIALS

LINKAGE TYPE DIFFERENTIALS

The linkage type differential has the same function as the conventional gear type differential; that is, the addition or subtraction of two quantities. In this differential, however, the inputs and outputs are represented by linear movements of links rather than by rotation of gears.

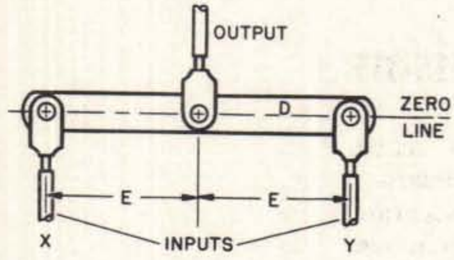


LINKAGE TYPE DIFFERENTIAL

The differential consists of a bar D to which three links, $X, Y, Z,$ are connected by means of pivot pins. For the present, it is assumed that the two end links are the inputs and the middle link is the output. Later it will be shown that either end link can be made an output and that the inputs can be applied to the other two links.

If the output link is pivoted on bar D midway between the two input links, the value of the output is proportional to the sum or difference of the inputs. Any other location of the output link will result in a sum or difference of the form $(x+Ky)$ or $(x-Ky)$. The value of the constant, K, depends upon the location of the output link relative to the two input links.

Adding: x + y

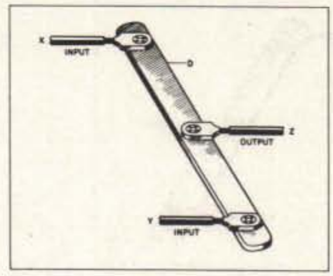


ZERO POSITION

The addition of two quantities is accomplished by moving both input links in the same direction, provided the output link is located between the input links.

Assume a differential with input links X and Y located at the ends of a bar D and an output link Z located at the center of the bar. With inputs X and Y at zero, bar D will be positioned along the zero line.

If input X is moved upward a distance x, while input Y remains stationary, the output moves upward a distance a, an amount depending upon the relative distance between pivots on bar D. The distances x and a form the sides of two similar triangles. (For a description of similar triangles, see page 28.) In two similar triangles, the ratio between any two sides of one triangle is equal to the ratio between the corresponding two sides of the other triangle. Therefore:



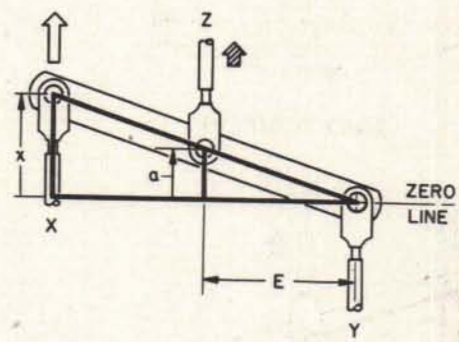
$$\frac{a}{E}=\frac{x}{2E}$$

where E is the distance from output Z to inputs X and Y. Multiplying both sides by E:

$$\frac{Ea}{E}=\frac{Ex}{2E}$$

Then:

$$a=\frac{x}{2}$$



x INTRODUCED

If input X remains fixed at its new position, and Y is moved upward a distance y, the output moves upward a distance b. Making use of similar triangles again:

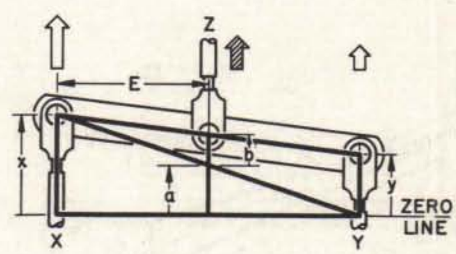
$$\frac{b}{E}=\frac{y}{2E}$$

Then:

$$b=\frac{y}{2}$$

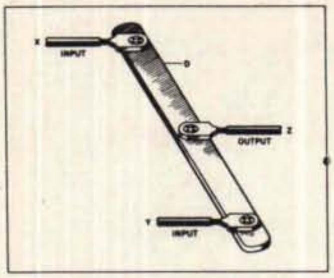
The total movement of the output link Z is $(a+b)$, or z.

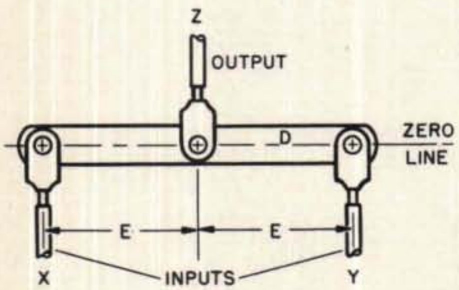
$$z=a+b=\frac{x}{2}+\frac{y}{2}=\frac{x+y}{2}$$



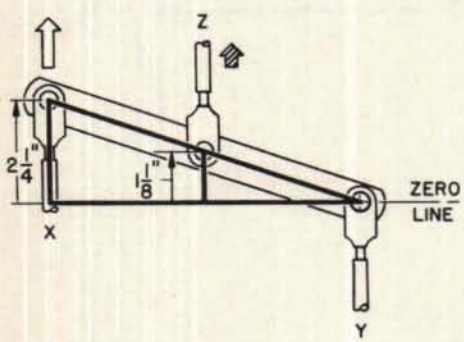
y ADDED TO x

Although the actual movement of the output link equals *one-half* the sum of the input movements, the output movement nevertheless *represents* their sum, but to a different scale; that is, to a different value per linear inch of link movement. If necessary, the scale of the output movement can be converted to the scale of the input movements by means of a simple linkage multiplier, described later.

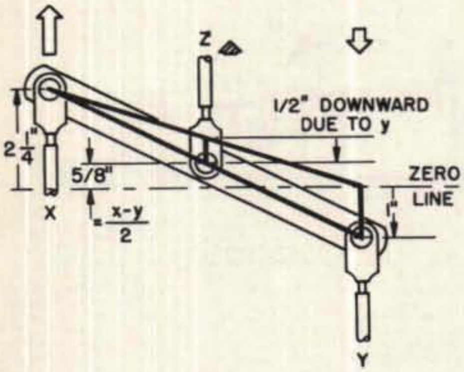




ZERO POSITION



x INTRODUCED



y SUBTRACTED FROM x

Subtracting: x − y

The same differential may be used for subtraction by moving the input links in opposite directions. In this case, the total output equals one-half the difference of the inputs.

If X is moved upward $2\frac{1}{4}$ inches, while Y remains fixed, Z moves upward $1\frac{1}{8}$ inches.

If X is fixed at its new position and Y is moved downward one inch, link Z moves downward $\frac{1}{2}$ inch.

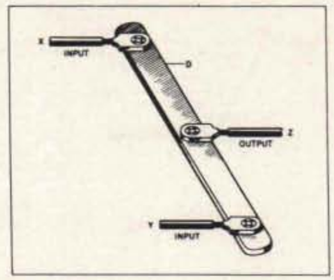
The total movement of Z from its initial position at the zero line is:

$$z = \frac{2\frac{1}{4}'' - 1''}{2} = \frac{5''}{8} \text{ upward}$$

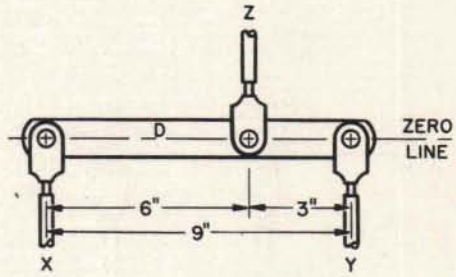
or

$$z = \frac{x}{2} - \frac{y}{2} = \frac{x-y}{2}$$

Therefore, the total movement of the output link is equal to one-half the difference between the movements of the two inputs. This net movement, nevertheless, represents the difference between the two quantities, but to another scale.



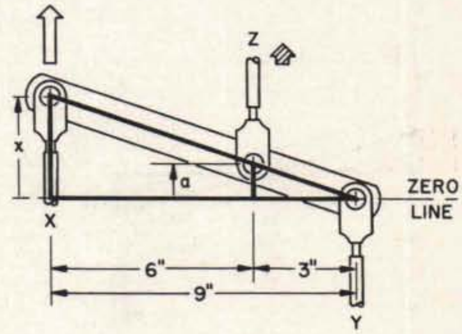
Adding: x + Ky



ZERO POSITION

By locating the pivot point of output link Z off-center on bar D, a sum of the form $(x + Ky)$ may be obtained.

Assume bar D to be nine inches long and link Z to be connected three inches from link Y. If X is moved upward, with Y fixed, Z will move a distance a.



x INTRODUCED

Making use of similar triangles:

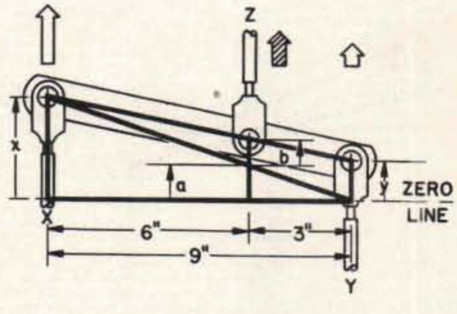
$$\frac{a}{3} = \frac{x}{9}, \quad \text{or} \quad a = \frac{x}{3}$$

If X is fixed at its new position and input Y is moved upward a distance y, output Z moves a distance b. Then:

$$\frac{b}{6} = \frac{y}{9}, \quad \text{or} \quad b = \frac{2}{3}y$$

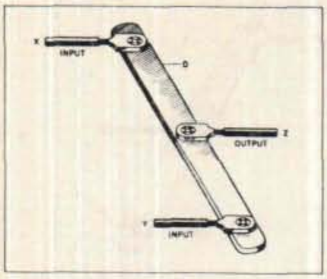
The total movement of Z is $(a + b)$. Therefore:

$$z = a + b = \frac{x}{3} + \frac{2y}{3} = \frac{x + 2y}{3}$$

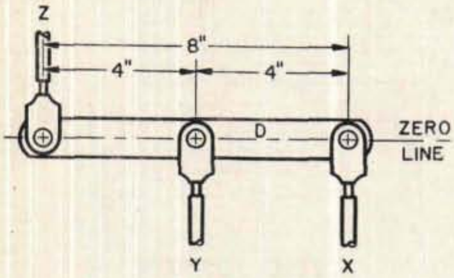


Ky ADDED TO *x*

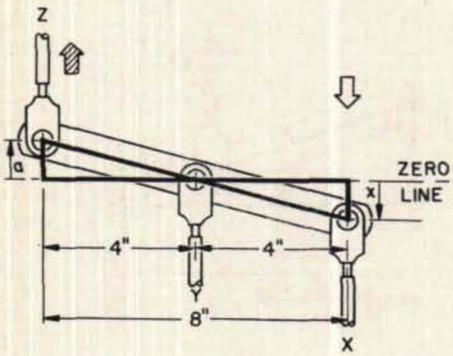
In this case, the value of the constant K is two, and the total movement of the output link is equal to one-third the sum of x and $2y$.



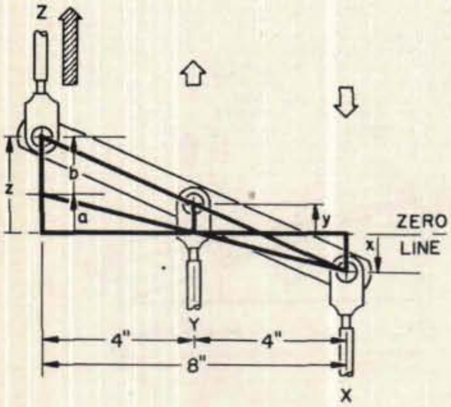
Other Arrangements



ANOTHER ARRANGEMENT



x INTRODUCED



Ky ADDED TO x

Any two links of the differential can be used as inputs, and the remaining one can be used as the output, depending upon which arrangement proves most convenient mechanically. However, if one of the end links is used as the output, the addition of two quantities is accomplished by moving the input links in opposite directions. Since, in this arrangement, the output link is not equidistant from both input links, sums of the form $(x+Ky)$ are obtained.

Assume that the output link *Z* is located at the left end of bar *D*, that input *X* is at the right end, and that input *Y* is at the center of the bar.

If input *X* is moved downward a distance *x* while input *Y* remains fixed, output *Z* will move upward a distance *a*. From the similar triangles formed:

$$\frac{x}{4}=\frac{a}{4} \quad \text{or} \quad a=x$$

If *X* is fixed at its new position and input *Y* is moved upward a distance *y*, output *Z* moves upward a distance *b*. Therefore:

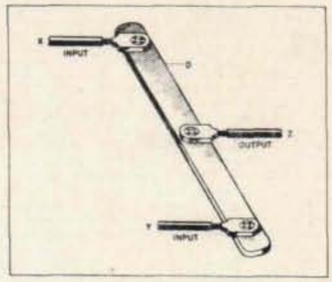
$$\frac{y}{4}=\frac{b}{8} \quad \text{or} \quad b=2y$$

The total movement of the output link *Z* is $(a+b)$, or *z*.

$$z=a+b=x+2y$$

The value of the constant *K* is two.

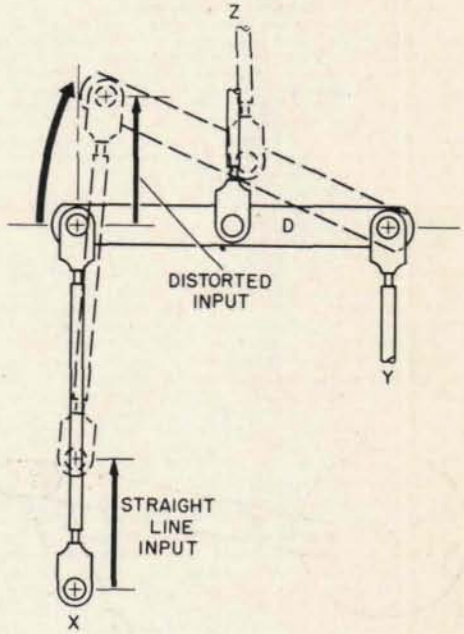
Subtraction in this arrangement is accomplished by moving the two input links in the same direction. Differences of the form $(x-Ky)$ are obtained.



Distortion of Output

Since the length of bar *D* is fixed, the inputs will not actually be applied in a straight line even though one end of each input link moves in a straight line. This distortion causes errors to be introduced in the output.

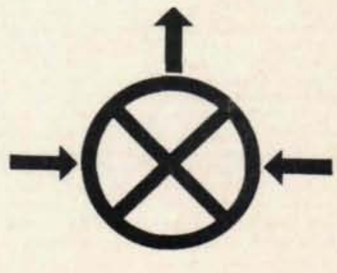
Suppose input link *X* is moved upward while input *Y* remains fixed. The upper end of link *X*, which is connected to bar *D*, moves in an arc although the lower end moves in a straight line. The input to bar *D* is therefore slightly less than the actual input supplied to link *X*. This error, however, may be kept insignificant by limiting the angular motion of bar *D*.

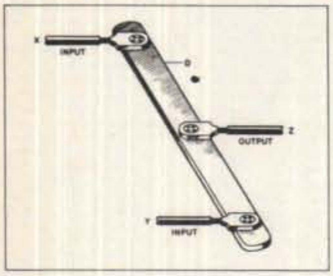


DISTORTION ERROR

Schematic Symbol

The schematic symbol for the linkage type differential is similar to that for the gear type differential. The arrows pointing inward are the inputs, and the arrow pointing outward is the output.

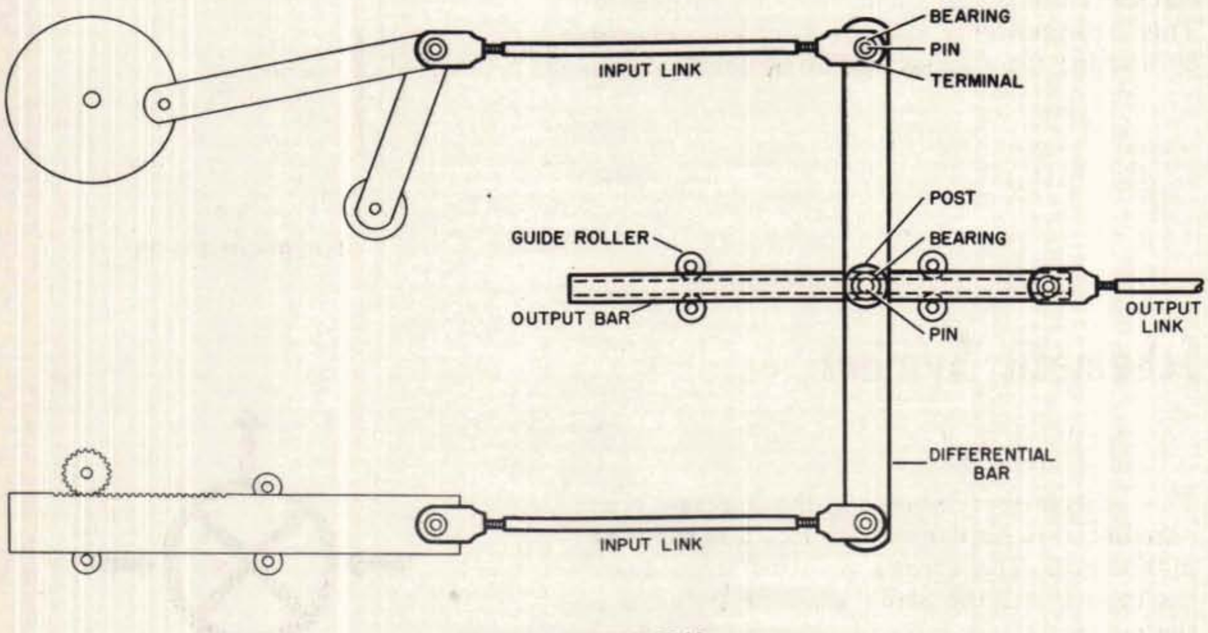




Construction of a Typical Unit

Here is a typical linkage type differential. Each link assembly consists of a rod and two terminals. The ends of the rod are screwed into these terminals. The terminals hold the pivot bearings. One terminal is ball bearing mounted on a pivot pin fixed to the top of the differential bar or output bar. The other terminal is ball bearing mounted on another linkage mechanism.

The differential bar is ball bearing mounted on a pivot pin fixed to the top of an output bar. The output bar, in turn, is held by guide rollers which allow it to move in a straight line parallel to the input links. Its straight line movement is imparted to the output link through a pivot pin which supports a terminal of the output link.



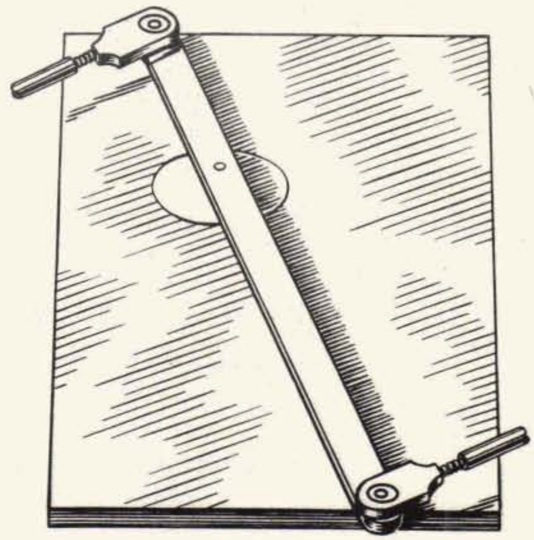
TYPICAL DIFFERENTIAL

MULTIPLIERS

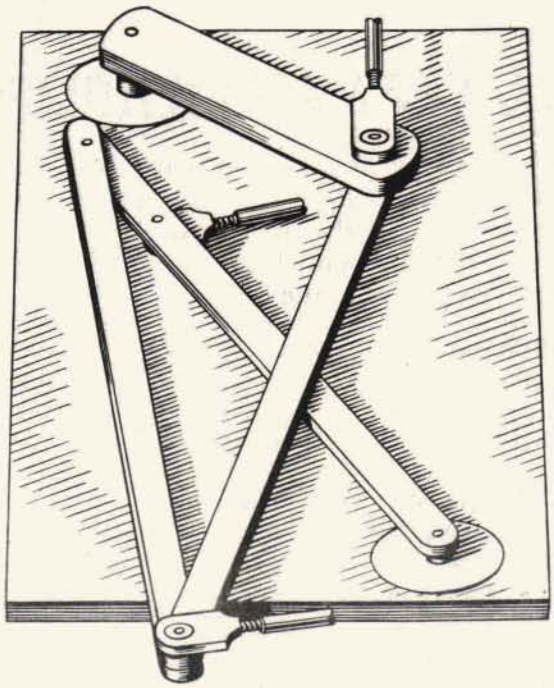
LINKAGE TYPE MULTIPLIERS

There are two general types of linkage multiplier. One type multiplies a variable quantity by a constant. Its function is similar to that of a gear ratio for changing the value per revolution of a shaft line in a gearing unit. The lever multiplier and the bell crank multiplier are of this type.

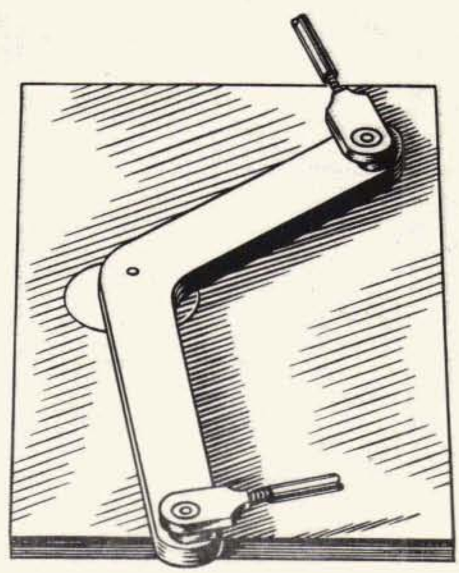
The other general type of linkage multiplier is used to multiply two variable quantities. The output of this multiplier is directly proportional to the product of two inputs. This multiplier is known as the *XY* linkage multiplier.



LEVER TYPE MULTIPLIER

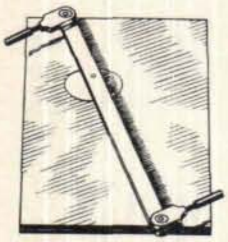


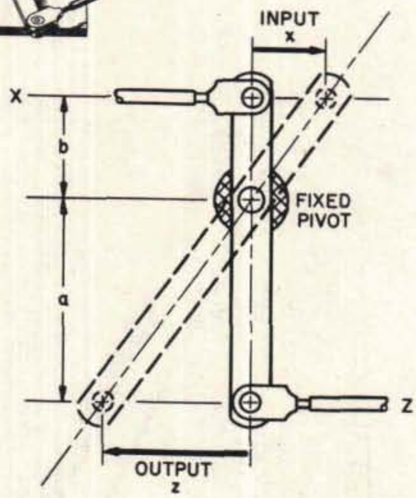
XY TYPE MULTIPLIER



BELL CRANK TYPE MULTIPLIER

THE LEVER MULTIPLIER





LEVER MULTIPLIER

The lever multiplier is simply a straight bar which has a fixed pivot at some point in its length. Connected to the bar are two links; an input link and an output link. The direction of motion of the input link is parallel to the direction of motion of the output link.

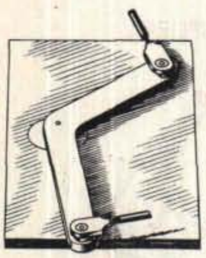
The output of this multiplier is equal to its input multiplied by a constant. The value of the constant depends upon the location of the fixed pivot on the bar relative to the pivots for the input and output links.

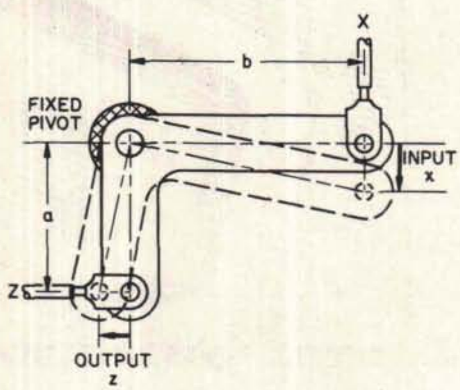
If the input link is moved a distance x, the output link will move a distance z. From the two similar triangles formed:

$$\frac{z}{a}=\frac{x}{b} \quad \text{or} \quad z=\frac{a}{b}x$$

Therefore the output is equal to the input multiplied by a constant, the value of the constant being a/b. The fixed pivot is located along the bar at a position that establishes the required ratio of a to b.

THE BELL CRANK MULTIPLIER





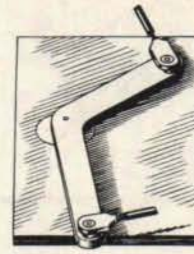
BELL CRANK MULTIPLIER

The bell crank multiplier is a bar whose two arms form an angle and whose fixed pivot is at the vertex of the angle. An input link is connected to one arm of the bar and an output link is connected to the other.

As in the case of the lever multiplier, the output is equal to the input multiplied by the ratio a/b, where a is the length of the output arm and b is the length of the input arm.

Unlike the lever multiplier, however, the direction of motion of the output is not parallel to the direction of motion of the input. The input and output motions are directed at right angles to the arms to which the links are connected. The angle between the two motions therefore depends upon the angle between the two arms.

Distortion

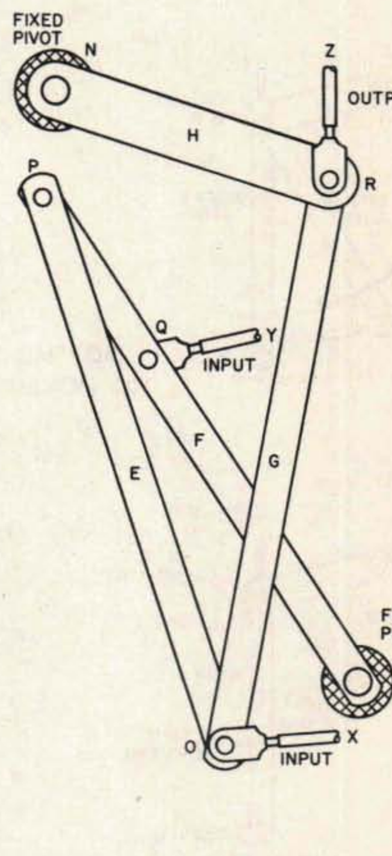


Both the lever multiplier and the bell crank multiplier are subject to distortions of the same sort as the linkage differential. However, errors in the output can be kept small by limiting the motion of the inputs.

Schematic Symbol

The schematic symbol for the lever and the bell crank multipliers, which are of the type having one input, is a multiplication sign enclosed in a circle, with the input arrow pointing toward the center of the circle.

THE XY LINKAGE MULTIPLIER



Ky ADDED TO *x*

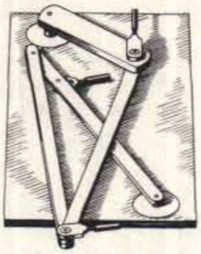
The *XY* linkage multiplier consists fundamentally of three links *E*, *F*, and *G*, all of equal length. A fourth link, *H*, merely guides link *G* and output link *Z*.

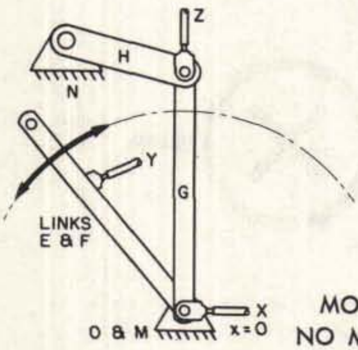
Input link *X* is connected to links *E* and *G* through pivot *O*. Input link *Y* is connected to link *F* through pivot *Q*. Output link *Z* is connected to links *G* and *H* through pivot *R*. Fixed pivots *M* and *N* are centers of rotation for links *F* and *H*, respectively. Pivot *P* joins links *E* and *F*.

How It Multiplies

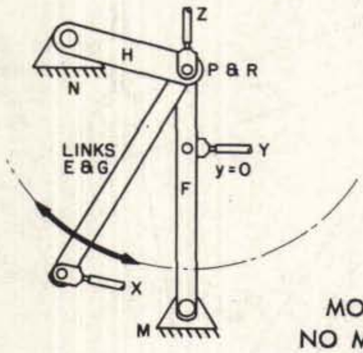
The output of the *XY* multiplier is proportional to the product of the two inputs.

It will be recalled that the output of a bell crank multiplier is equal to the product of the input and the fixed ratio a/b, where a is the length of the output arm and b is the length of the input arm. If the ratio a/b could be changed, the value of the output relative to the input would change proportionally.

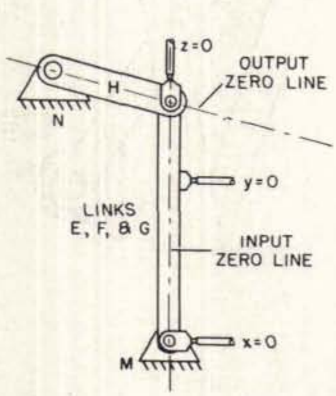




MOVEMENT OF Y
NO MOVEMENT OF Z



MOVEMENT OF X
NO MOVEMENT OF Z



ZERO POSITION

In the *XY* linkage multiplier the value of the ratio a/b can be varied by an input. Thus the output is made proportional to the product of the two inputs.

Determining the zero position

The zero position of the multiplier is important as a starting point from which to gain information about the operation of the multiplier. It is therefore desirable to establish the zero position before consideration is given to the multiplying action.

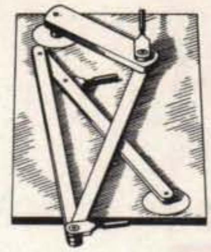
The zero position of the multiplier may be defined as the position of the multiplier linkage when both inputs are at zero. Also, one input may be said to be at zero if the other input can be changed without moving the output. From this fact, the zero position of each input can be determined. Then both inputs can be placed at zero to establish the zero position of the multiplier.

Assume that input link *X* is positioned so that links *E* and *F* are aligned. As a result, pivot *O* coincides with fixed pivot *M*. In this position, movement of input link *Y* merely rotates links *E* and *F* about pivots *O* and *M*, respectively. Since pivot *O* is also the point at which link *G* is connected to link *E*, neither links *G* nor *Z* are affected by movement of link *Y*. The zero position of input *X* is therefore that position which aligns link *E* with link *F*.

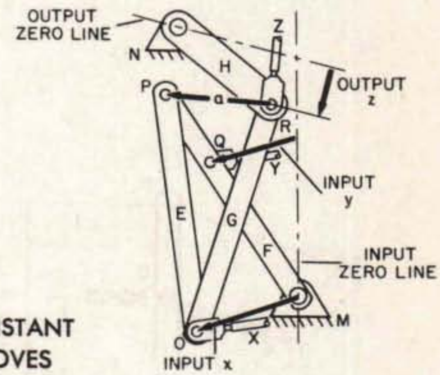
Similarly, assume that input link *Y* is positioned so that links *E* and *G* are aligned. Pivot *P* coincides with pivot *R*. Movement of input link *X* rotates links *E* and *G* about pivots *P* and *R* but causes no movement of output link *Z*. The zero position of link *Y* is therefore that position which aligns link *E* with link *G*.

With both input links at their zero positions, all three links *E*, *F*, and *G*, are aligned along a common input zero line; link *H* defines an output zero line; and the multiplier is at its zero position.

Operation as an equivalent bell crank multiplier

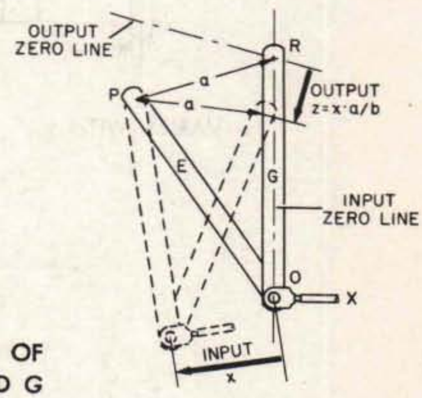


If input link Y is held at a fixed distance y from the input zero line and if movements of link X are small, the XY multiplier operates essentially the same as that of the simpler bell crank multiplier. In this case, its operation is as follows:



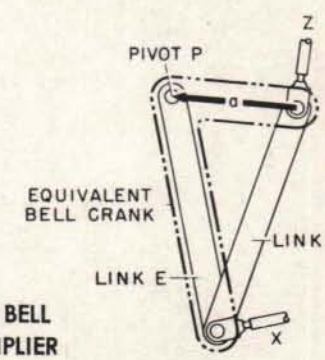
Y CONSTANT
X MOVES

Assume that link X moves through distance x. Pivot P is stationary since link F is restrained by fixed pivot M and the fixed input y. While x changes, link E rotates about pivot P. Link G, connected to the lower end of link E, causes pivot R to move down. Because link H holds pivot R at a constant radius from fixed pivot N, pivot R remains at a nearly constant distance from stationary pivot P. Therefore, a, the distance between pivots P and R, remains *substantially* constant.



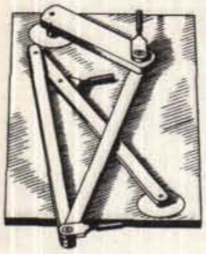
MOVEMENTS OF
LINKS E AND G

With a constant, links E and G form a rigid frame because their lower ends are connected through pivot O and their upper ends are a constant distance, a, apart. This rigid frame operates as an equivalent bell crank, turning about stationary pivot P. Distance a may be considered as one arm of a bell crank multiplier. It is nearly perpendicular to output link Z. The other arm of this multiplier may be considered to be link E, which is nearly perpendicular to input link X. If the length of link E is denoted as b, then, by comparison with the bell crank multiplier described previously, output z equals x multiplied by the ratio a/b, or:



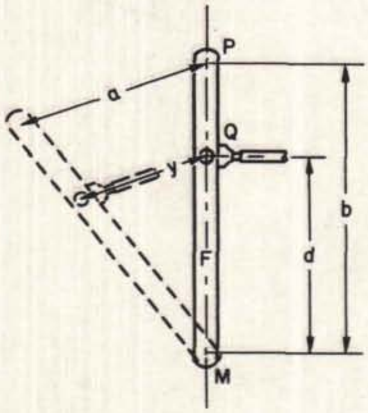
EQUIVALENT BELL
CRANK MULTIPLIER

$$z = (a/b)x$$



Multiplication of two variable quantities

The product xy of two variables, x and y, is obtained by making the length of one arm of the bell crank multiplier vary with y. From the previous equation, the output z of the bell crank multiplier equals $(a/b)x$, where x is the input and a and b are the lengths of the crank arms. A second input, y, is introduced by varying a with y. This causes a corresponding change in the ratio (a/b) because b is constant, b being the length of link E. Since (a/b) varies with y and since z equals $(a/b)x$, the output z of the XY multiplier represents the product xy.



a VARIES WITH *y*

The value of a depends upon two factors. One is the magnitude of y and the other is the distance d of link Y from fixed pivot M. In any multiplier, distance d is constant. This leaves y as the only variable affecting a.

As y increases, a increases in direct proportion because distances y and a are corresponding sides of similar triangles. The exact expression for a may be derived as follows:

Since link F acts as a lever multiplier with its fixed pivot at M, the value of a is equal to the product of the movement y of the input link and the ratio b/d, where b is the length of link F. (Previously the length of link E was designated b. The length of link F, equal to that of link E, may also be designated b.) Therefore:

$$a = \frac{b}{d}y$$

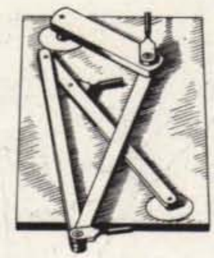
Substituting this expression for a in the equation derived previously for the output of the multiplier:

$$z = \frac{x}{b} \times y \times \frac{b}{d}$$

or:

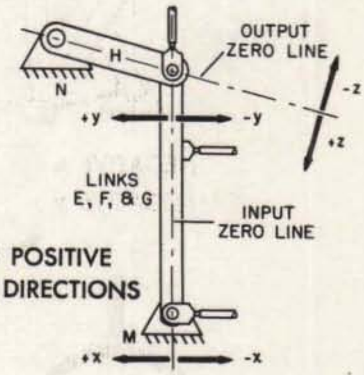
$$z = \frac{xy}{d}$$

Since d is a fixed distance for any multiplier, it is a constant that can be removed by means of a bell crank in the output, or simply by assigning a different value per unit of travel for the output link at the time of designing the instrument.



Negative Values

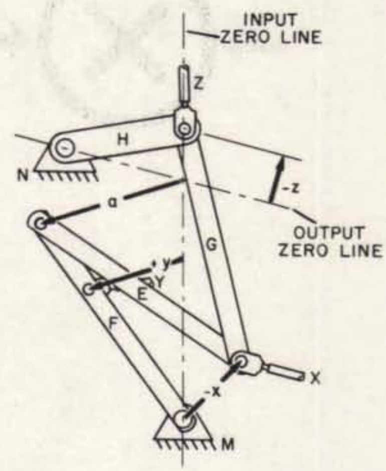
If movements of the two inputs to the left of the input zero line are considered to be positive values, negative inputs of x and y can be introduced by moving the respective links to the right of the input zero line.



ESTABLISHING POSITIVE AND NEGATIVE DIRECTIONS

Such a selection of positive and negative directions for the inputs will cause movements of Z below the output zero line to be positive, and movements of Z above the output zero line to be negative.

Assume that a negative x is to be multiplied by a positive y. Input link Y is moved a distance $+y$ to the left for a positive input. Input link X is moved a distance $-x$ to the right for a negative input. This causes output link Z to move above the output zero line a distance $-z$ for a negative output.



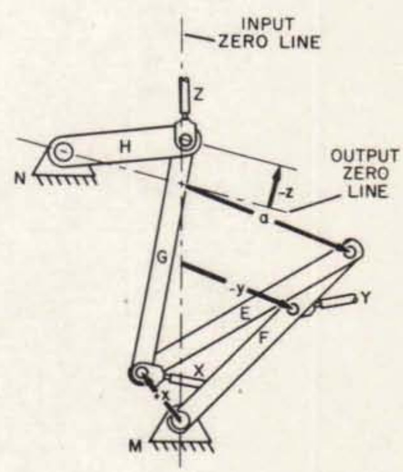
NEGATIVE x
POSITIVE y

Through the same analysis as described previously for the multiplication of positive values, it can be shown that the output, $-z$ in this case, is proportional to the product of $+y$ and $-x$. Stated mathematically:

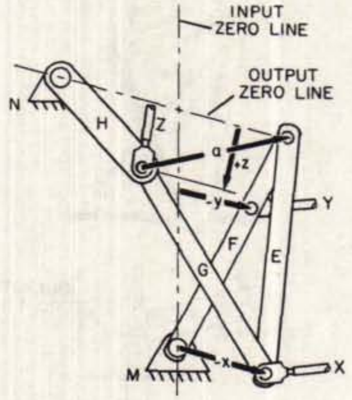
$$z = \frac{(-x)\ (+y)}{d}$$

Similarly, a negative y can be multiplied by a positive x. A negative y input moves input link Y to the right of the input zero line. Input link X moves to the left for a positive input, and output link Z moves above the output zero line for a negative output. The output is:

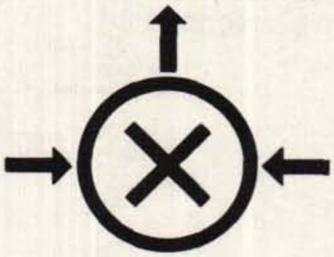
$$z = \frac{(+x)\ (-y)}{d}$$



POSITIVE x
NEGATIVE y



NEGATIVE x
NEGATIVE y



Since the product of two negative quantities is positive, negative inputs of both x and y will result in a positive output of z. Moving input links X and Y to the right for negative inputs will cause the output link Z to move below the output zero line for a positive output. The output in this case is:

$$z = \frac{(-x)\ (-y)}{d}$$

Distortion

Errors caused by distortion can be kept small by limiting the range of movement of the inputs.

Schematic Symbol

The schematic symbol for the XY linkage multiplier is a multiplication sign enclosed within a circle.

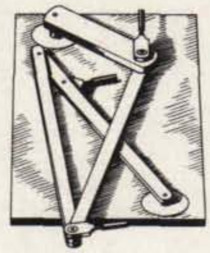
USING A MULTIPLIER AS A DIVIDER

The XY linkage multiplier can be employed as a divider. It has been shown that the XY linkage multiplier solves the following formula for z:

$$z = \frac{xy}{d}$$

This formula can be rearranged to give the following division equations, whose right-hand members express the division of one variable by another:

$$x = \frac{zd}{y} \quad \text{and} \quad y = \frac{zd}{x}$$



By reversal of input and output connections, the XY linkage can be made to solve either of these division equations.

For example, assume that the following division is to be performed:

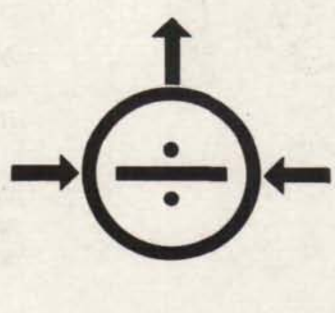
$$x = \frac{zd}{y}$$

Link Z becomes an input link of the divider. It is positioned by the quantity to be divided (that is, the dividend). The other input, the divisor, positions link Y. The output, or quotient, is taken from link X.

Comparison with Other Dividers

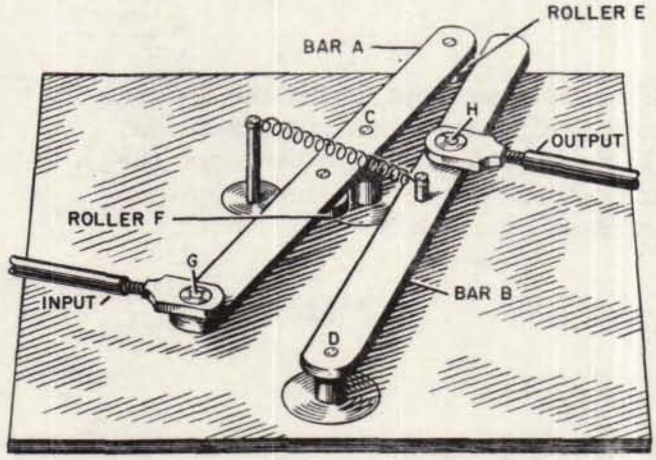
The XY multiplier, used as a divider, is unusual in respect to its simplicity. It is made of simple, easily constructed members with only pivot joints. It does not have sliding or screw members, or reciprocal cams. If the quantity used as the divisor does not approach zero, it requires no follow-up since its members are sufficiently rugged to withstand the driving forces. However, because of distortion errors, it is not an exact divider.

Schematic Symbol

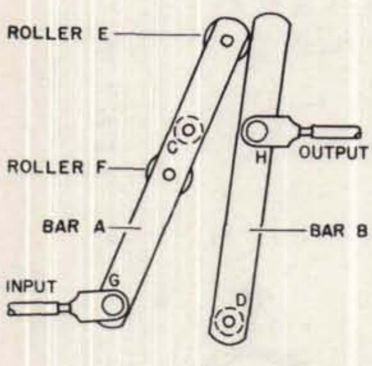


The schematic symbol for the linkage type divider is a division sign enclosed within a circle.

THE ABSOLUTER



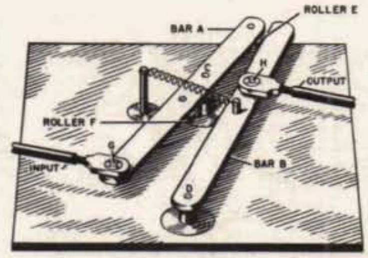
The absoluter consists of two bars, *A* and *B*, with pivots and links, somewhat resembling lever multipliers. Bar *A* has a fixed pivot at *C*, and bar *B* has a fixed pivot at *D*. Rollers *E* and *F* transfer motion from bar *A* to bar *B* by pushing against the side of bar *B*. A coil spring maintains continuous contact between bar *B* and the rollers on bar *A*. An input link and an output link are connected at points *G* and *H*, respectively.



The absoluter is a linkage mechanism that computes the absolute value, or magnitude, of a quantity, regardless of its sign. For example, the cardinal number 2 is the absolute value of $+2$ or -2. In the absoluter, negative and positive inputs cause the output to move in the same direction.

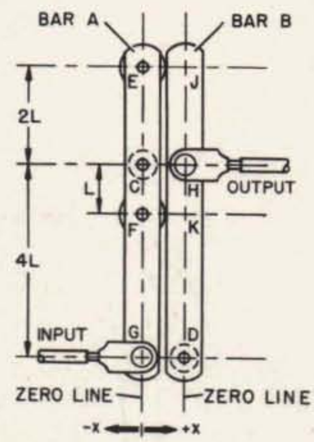
Motion of the input link to the left causes roller *E* to push against bar *B*, thereby moving the output link to the right a distance which is proportional to the input.

Motion of the input link to the right causes roller *F* to push against bar *B*, and likewise moves the output link to the right a distance which is proportional to the input.



How It Works

The absoluter may be considered as two lever multipliers working in series. Bar A is one of the levers, and bar B is the other. The two bars are of equal length.



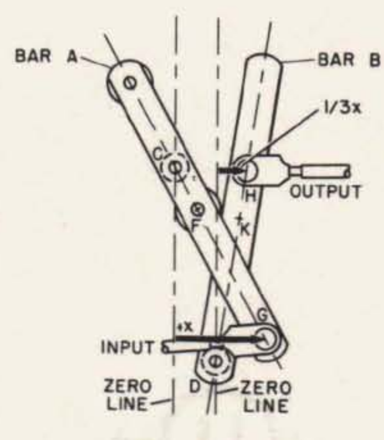
ZERO POSITION

Certain relations must exist among the distances between various points on these bars in order that input x shall have the same effect on the output for movement in either direction. Let L be the distance CF between pivot C and roller F. In the absoluter illustrated here, the distances between points on bars A and B have the following values in terms of the distance L. (Other absoluters may have different values but the relations among these values are the same.):

Bar A	Bar B
$CF = L$	$HK = L$
$CE = 2L$	$HJ = 2L$
$CG = 4L$	$HD = 4L$

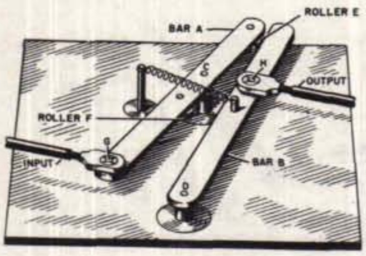
When the absoluter is at its zero position, the two bars, A and B, are parallel to each other, and the centers of all pivots and rollers lie on the input and output zero lines.

Assume that movements of the input link to the right of the input zero line represent positive values and that movements to the left are negative.



POSITIVE INPUT

To explain the operation of the absoluter, suppose that the input link is moved a distance $+x$ to the right of the zero line. Since bar A is pivoted at C, roller F pushes against bar B, causing the output link to move to the right.



Bar A forms a lever multiplier with arms CG and CF. A movement of $+x$ at point G causes point F to move a distance equal to $x \times \frac{CF}{CG}$.

Since: $CF = L$
And: $CG = 4L$

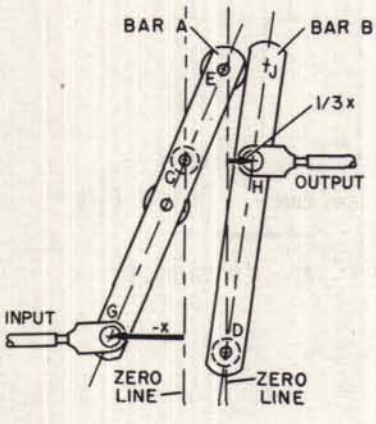
Then: movement at $F = x \times \frac{L}{4L} = \frac{x}{4}$

Quantity $\frac{x}{4}$, representing movement at F, forms the input to bar B at point K. Bar B forms a second lever multiplier with arms DH and DK. Therefore, movement of the output link is $\frac{x}{4} \times \frac{DH}{DK}$.

Since: $DH = 4L$
And: $DK = 4L - L = 3L$

Then: movement at $H = \frac{x}{4} \times \frac{4L}{3L} = \frac{x}{3}$

Consequently, the output of the absoluter equals one-third of the input, for positive values of the input.



NEGATIVE INPUT

Assume that the input link is moved a distance $-x$ to the left of the zero line. Roller E pushes against bar B, and the output link again moves to the right. In this case, the first lever multiplier is formed by arms CE and CG, and the movement at E is equal to $x \times \frac{2L}{4L}$, or $\frac{x}{2}$. Quantity $\frac{x}{2}$ is the input to the second lever multiplier formed by arms DH and DJ. The output of the absoluter is $\frac{x}{2} \times \frac{4L}{6L} = \frac{x}{3}$.

The output of the absoluter is therefore equal to one-third of the input in both cases.

Distortion

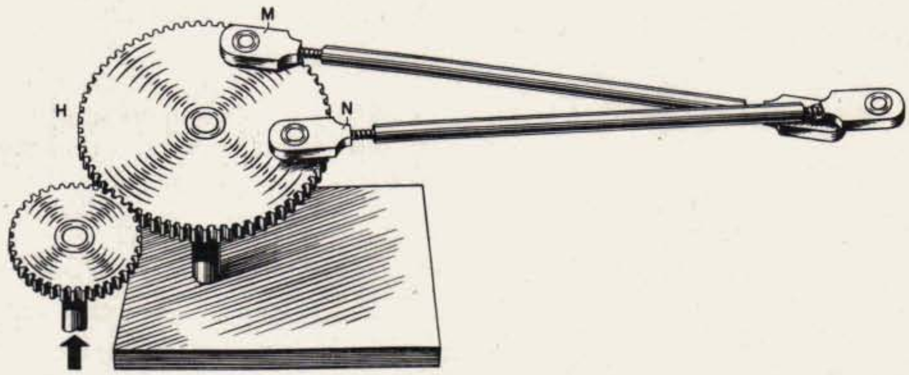
Distortion can be held to a minimum by limiting the range of movement of the input.

Schematic Symbol

The schematic symbol for the absoluter consists of two vertical lines enclosed within a circle.

ANGLE RESOLVERS

An angle resolver is a mechanism which computes the sine and the cosine of an angle.

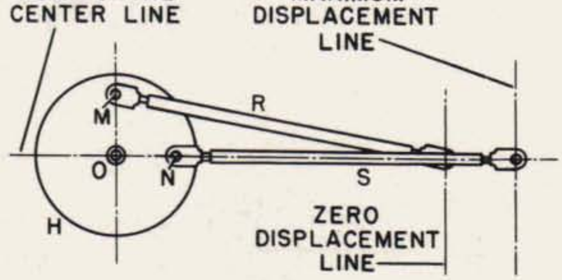


SIMPLE ANGLE RESOLVER

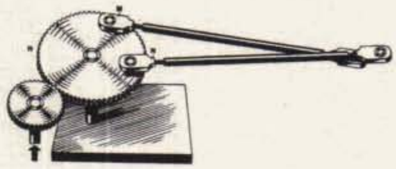
A simple resolver consists of a gear *H* on which two pins, *M* and *N*, are mounted 90 degrees apart and equidistant from the center of the gear. Attached to each pin is an output link which transmits the horizontal component of the motion of the pin as the gear rotates. The horizontal component of the displacement of pin *M* is proportional to the sine of the angle through which gear *H* rotates, and the horizontal component of the displacement of pin *N* is proportional to the cosine of the same angle.

How It Works

Assume that radius *OM* is perpendicular to the horizontal center line and that radius *ON* is on the horizontal center line. This is the zero position of the resolver; it is the position assumed by the mechanism when the angle to be resolved is zero.



ZERO POSITION



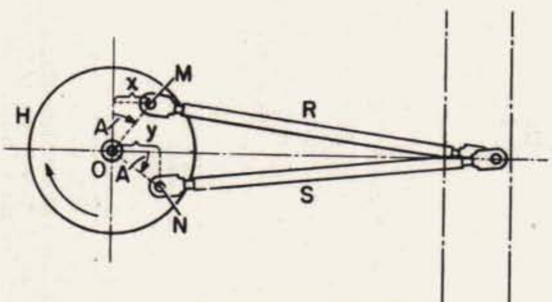
At the zero position, the sine of the angle is equal to zero and output link *R* is at its zero displacement position. Similarly, the cosine of the angle is equal to unity and output link *S* is at its maximum displacement.

If gear *H* is rotated in a clockwise direction through angle *A*, link *R* moves to the right and link *S* moves to the left.

The horizontal component of the displacement of pin *M* is distance *x*. From the right triangle formed with *x* as one side:

$$\sin A = \frac{x}{OM}$$

$$\text{or:} \quad x = OM \sin A$$



SINE AND COSINE OUTPUTS

Since *OM* is a constant, the distance *x* is proportional to sin *A*. Hence the displacement of output link *R* is proportional to sin *A*.

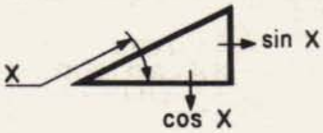
The horizontal component of radius *ON* is distance *y*. From the right triangle:

$$\cos A = \frac{y}{ON}$$

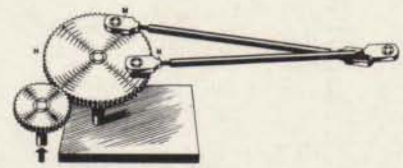
$$\text{or:} \quad y = ON \cos A$$

Therefore, the displacement of output *S* from the zero line is proportional to cos A.

Schematic Symbol



The schematic symbol for the angle resolver consists of a right triangle with an arrow across the angle being resolved. An arrow pointing outward from the opposite side represents the sine output, and an arrow pointing outward from the adjacent side represents the cosine output.



Distortion

If each output link were always parallel to the horizontal center line, motion of the output end of the link would be strictly proportional to the sine or cosine of angle *A*. However, this is not the case. Movement of the output end of each link, therefore, is not the same as the horizontal displacement of pin *M* or *N*.

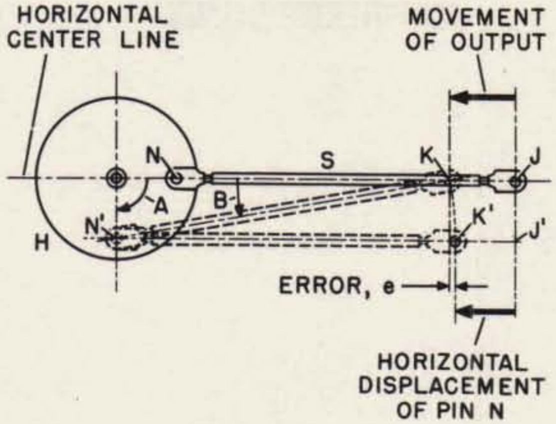
Let pin *N* be at the horizontal center line. Then output link *S* is parallel to the horizontal center line, and its output end is at *J*.

Now let angle *A* be increased so that pin *N* moves to *N'*. The output link will be positioned at angle *B* from the horizontal center line and its output end will be at *K*.

If the output end of the link is dropped from point *K* to point *K'*, so that the link is parallel to the horizontal center line, the horizontal displacement, *K'J'*, of the link will be equal to that of pin *N*. However, *K'J'* is less than the movement, *KJ*, of the output end when it travels along the horizontal center line. Thus the movement *KJ* is in error by an amount *e*. This is the maximum value; it decreases as angle *B* decreases, and is never very large if the proper relationship between the sizes of gear and link is selected.

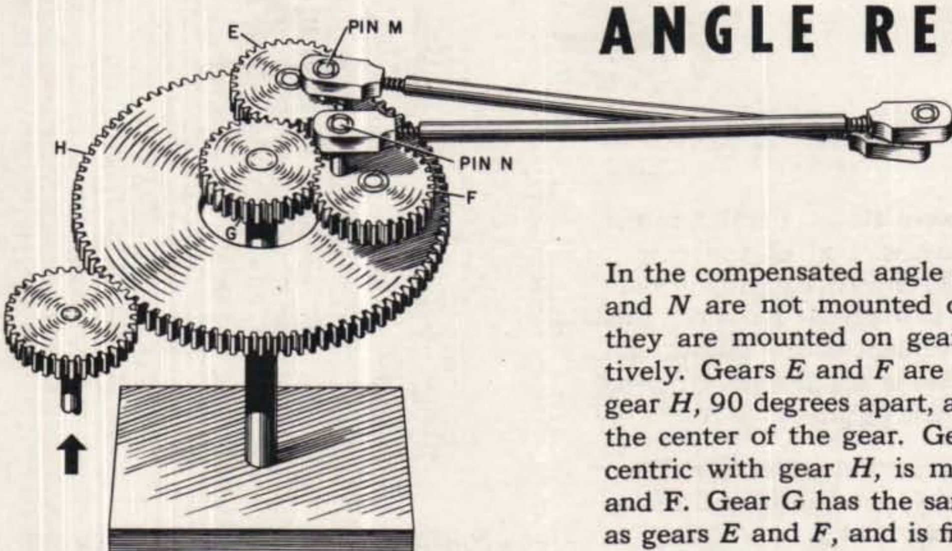
As angle *A* increases, the angle between the horizontal center line and the sine output link decreases, while the angle between the horizontal center line and the cosine output link increases. Therefore it can be stated that: the error in the sine output decreases and the error in the cosine output increases as angle *A* moves from zero to ninety degrees.

In the compensated angle resolver, these errors are reduced to a negligible value for the computation performed.



DISTORTION ERROR

THE COMPENSATED ANGLE RESOLVER

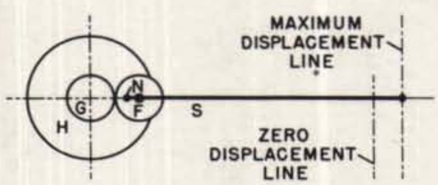


In the compensated angle resolver, the pins *M* and *N* are not mounted on gear *H*. Instead, they are mounted on gears *E* and *F*, respectively. Gears *E* and *F* are bearing mounted on gear *H*, 90 degrees apart, and equidistant from the center of the gear. Gear *G*, which is concentric with gear *H*, is meshed with gears *E* and F. Gear *G* has the same number of teeth as gears *E* and *F*, and is fixed to the frame of the instrument.

As gear *H* rotates, it carries gears *E* and *F* with it, causing them to roll around gear *G*. Pins *M* and *N*, consequently, change their positions relative to gear *H;* that is, they move in a circle about the centers of gears *E* and *F*. This added motion to the pins applies a correction to the output links which compensates for the error caused by the angle between the horizontal center line and the links.

How It Works

In the uncompensated resolver, the horizontal displacement of the output link *S* contains no error when the link lies on the horizontal center line. However, as the angularity of link *S* is increased, the right end of the link is pulled too far to the left, causing an error in the output.

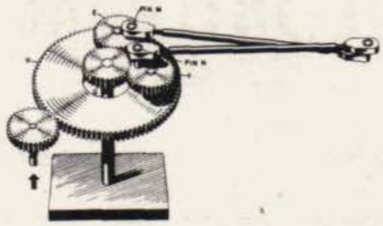


ZERO POSITION FOR COSINE OUTPUT

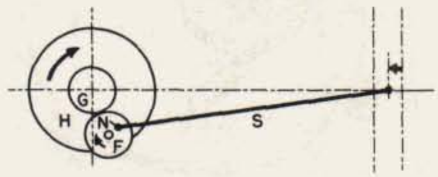
In the compensated resolver, the error at the output of link *S* is greatly reduced by shifting the location of pin *N* relative to gear *H*.

At the zero position, the center of gear *F* is on the horizontal center line and pin *N* is at the left of the center of gear *F*.

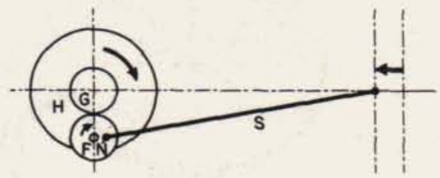
As gear *H* turns clockwise, rotation of gear *F*, due to its rolling around fixed gear *G*, moves pin *N* toward the right with respect to gear *H*. Movement of pin *N* toward the right introduces a correction that reduces the error caused by link *S* being too far to the left, as it was in the uncompensated resolver.



When gear *H* has turned clockwise through an angle of 90°, pin *N* has its greatest displacement to the right. Thus the correction increases as the angle between link *S* and the horizontal center line increases. The correction nearly balances the error, which increases with increasing angularity of link *S*.

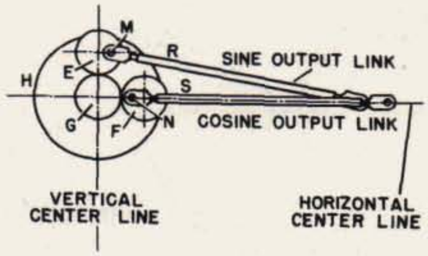


INPUT INCREASES



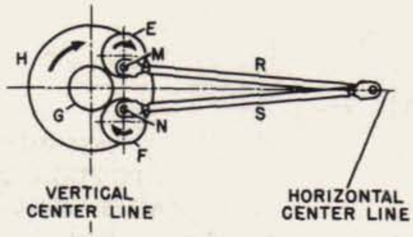
INPUT EQUALS 90°

Similarly, correction is made for the error at the output of link *R*, caused by the changing angle between link *R* and the horizontal center line. At the zero position, gear *E* is on the vertical center line and pin *M* is at the right of the center of gear *E*. The angle between link *R* and the horizontal center line tends to place the output end of link *R* too far to the left. This error, however, is compensated by right displacement of pin *M*.



ZERO POSITION

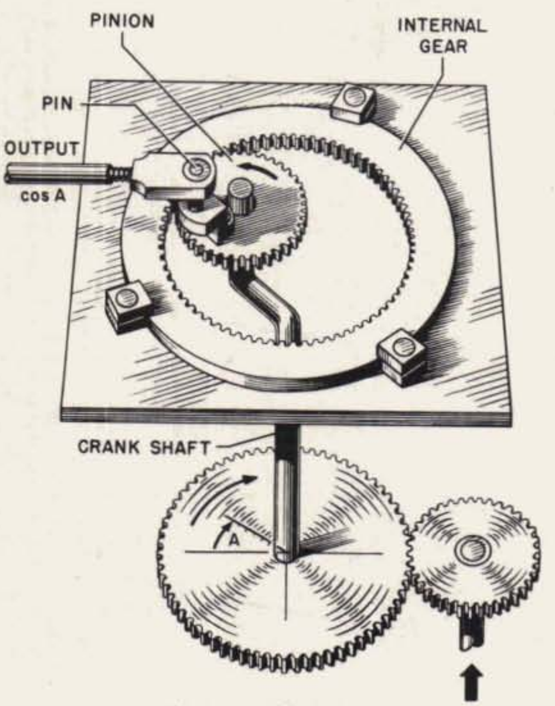
As angle *A* increases, pin *M* rotates about the axis of gear *E* and the correction decreases as the angle between link *R* and the horizontal center line decreases.



SINE AND COSINE OUTPUTS

The corrections introduced by shifting pins *M* and *N* do not completely compensate for the errors. These errors, however, are completely eliminated in the internal gear angle resolver.

THE INTERNAL GEAR ANGLE RESOLVER

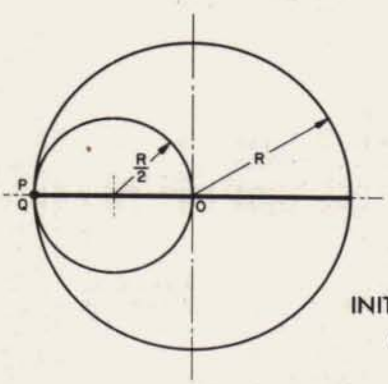


The internal gear angle resolver consists of two gears; an internal gear and a pinion. The pinion, which has half as many teeth as the internal gear, is bearing mounted on a crank shaft.

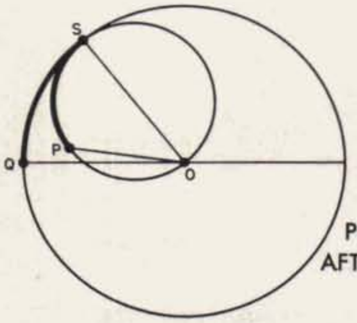
Rotation of the crank shaft causes the pinion to roll around the inside of the internal gear. A pin mounted near the periphery of the pinion moves the output link a distance proportional to the sine or the cosine of the angle through which the crank shaft is rotated.

This resolver computes true sine or cosine values. It is free from the distortion inherent in the other types.

How It Works



INITIAL POSITION OF POINT *P*



ASSUMED POSITION OF *P* AFTER SMALL CIRCLE HAS ROLLED TO POINT *S*

Assume that a point *P* is located on the circumference of a small circle rolling internally on another circle. As in the case of the pinion and internal gear, let the radius of the small circle be one-half the radius of the large circle. Under these two conditions, it can be shown that point *P* must move in a straight line along a diameter of the large circle.

The proof that point *P* moves along a diameter of the large circle is made by showing that no other path is possible. Initially, let point *P* of the small circle coincide with a point *Q* on the circumference of the large circle. A diameter drawn through the center *O* of the large circle to point *Q* establishes the stated path of point *P*. Secondly, let the small circle roll over an arc *QS* on the large circle. Then, as a matter of proof, assume that point *P* does not lie on line *QO*. However, it will be shown that this is a false assumption and that point *P* must lie on line *QO*.

To aid in this proof, a line is drawn from point O to the assumed position of point P and a second line is drawn from point O to the point of tangency, S, between the two circles. This construction forms angles POS and QOS. The proof is as follows:

1. Angle POS is an inscribed angle in the small circle. Therefore, according to a theorem of plane geometry, angle POS measured in degrees equals one-half arc PS measured in degrees.

2. Angle QOS is a central angle in the large circle. Therefore, angle QOS measured in degrees equals arc QS measured in degrees.

3. The linear lengths of arcs PS and QS are equal since they both are the distance traveled by the small circle. However, *measured in degrees,* arc PS is one-half arc QS because the radius of the small circle is one-half the radius of the large circle.

4. From the relations established in steps 1, 2, and 3, it follows that angle POS must equal angle QOS.

5. Since line OS is common to both angles and since the two equal angles overlay each other, lines PO and QO must coincide. Therefore, point P is on line QO, which extends to form a diameter of the large circle.

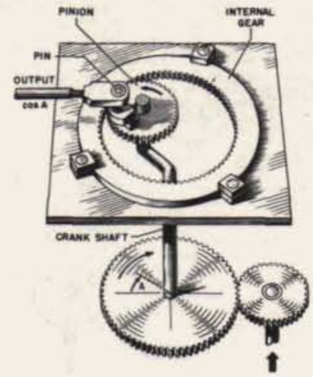
To analyze the displacement of point P along the horizontal diameter, assume that the small circle has rolled inside the large circle until the line joining the centers has moved through angle A. Point P lies at the intersection of the small circle and the horizontal center line. Lines joining points S, P, and O on the circumference of the small circle form a right triangle, with OS the hypotenuse; since an angle, like OPS, inscribed in a semi-circle is a right angle. Designate side OP, the distance of point P from the vertical center line, as x. Then:

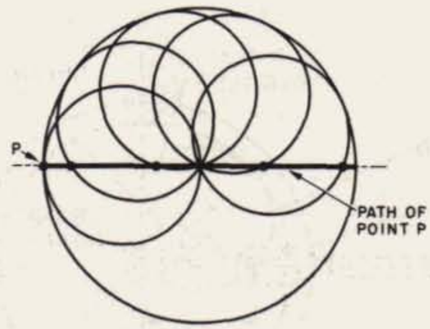
$$x = OS \cos A$$

Since OS is a constant, distance x is proportional to the cosine of angle A. Hence:

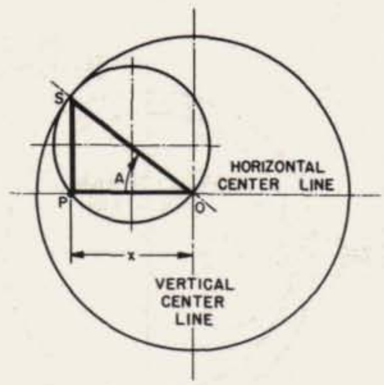
$$x = K \cos A$$

where K is a constant.

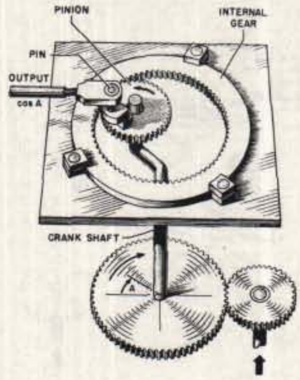




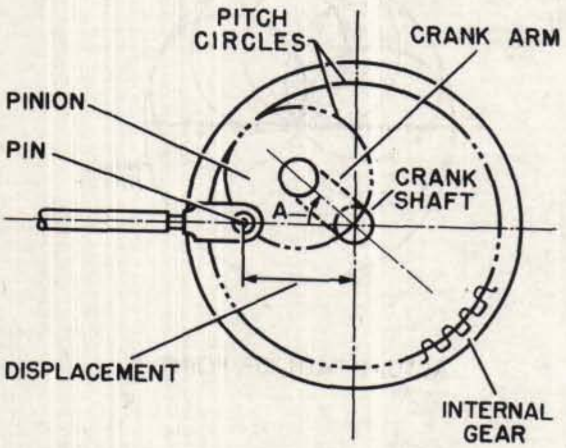
ACTUAL PATH OF POINT *P*



x IS PROPORTIONAL TO COSINE *A*

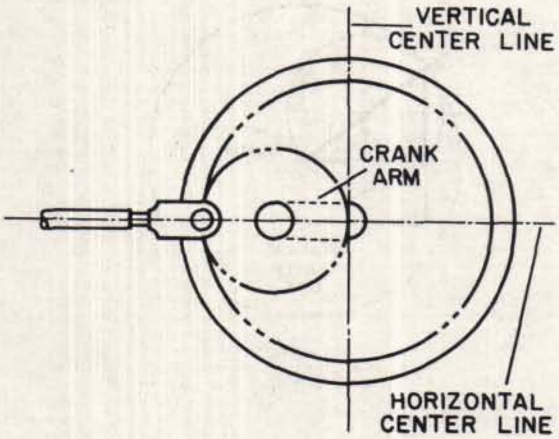


The circles of the illustrations that show how the internal gear angle resolver works meet the definition of pitch circles, which are circles that roll together with the same ratio as a pair of gears built upon them. The large circle represents the pitch circle of the internal gear of the actual mechanism, and the small circle represents the pitch circle of the pinion. Point *P* on the small circle represents the pin located over the pitch circle of the pinion.



ROLLING CIRCLES CORRESPOND TO PITCH CIRCLES

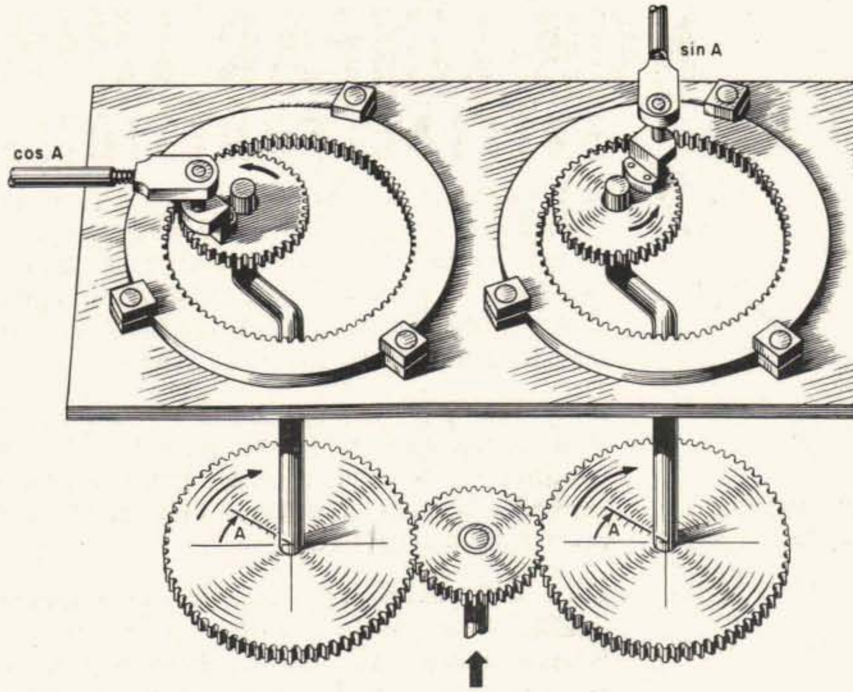
As the pinion, driven by the crank shaft, rolls around the inside of the internal gear, the pin moves in a straight line along a diameter of the internal gear, and the displacement of the output link is proportional to the cosine of the angle through which the crank shaft is rotated.



ZERO POSITION

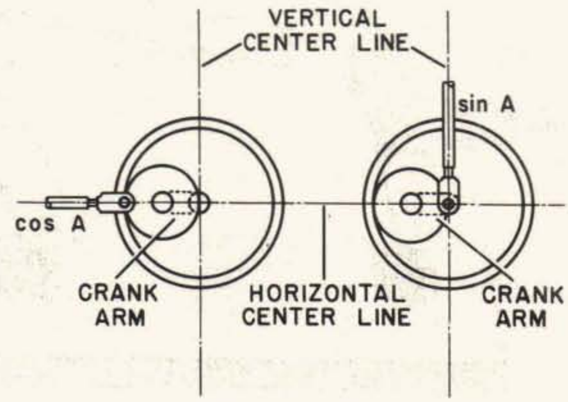
With angle *A* at zero, the crank shaft is so positioned that the crank arm is at the horizontal center line, and the output link is at its maximum displacement position to the left of the vertical center line. This corresponds to the value of unity for cos *A*.

As angle *A* increases, the output link moves to the right, its displacement from the vertical center line being proportional to the cosine of angle *A*.



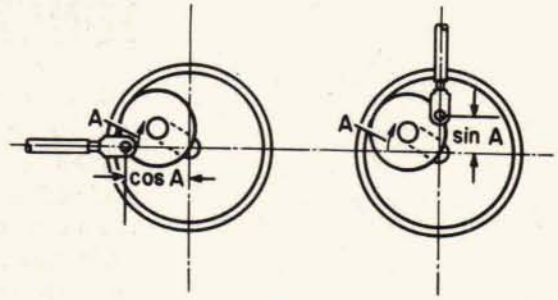
COMBINED SINE AND COSINE RESOLVERS

By gearing a second resolver to the first, and by positioning the output link and pinion differently in the second resolver, the sine of angle *A* can also be computed.



ZERO POSITION

With angle *A* at zero, the crank arm of the sine mechanism, like that of the cosine mechanism, is aligned with the horizontal center line; but the output link is at its zero displacement position. This zero position is the intersection of the horizontal and vertical center lines.

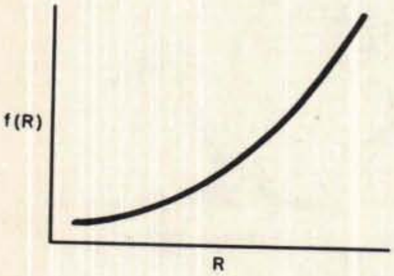


WITH ANGULAR INPUT

As angle *A* increases, the inside pinion turns counterclockwise. This causes the sine output link to move upward along the vertical center line, by an amount proportional to the sine of angle *A*.

NON-LINEAR LINKAGES

SPECIAL FUNCTIONS

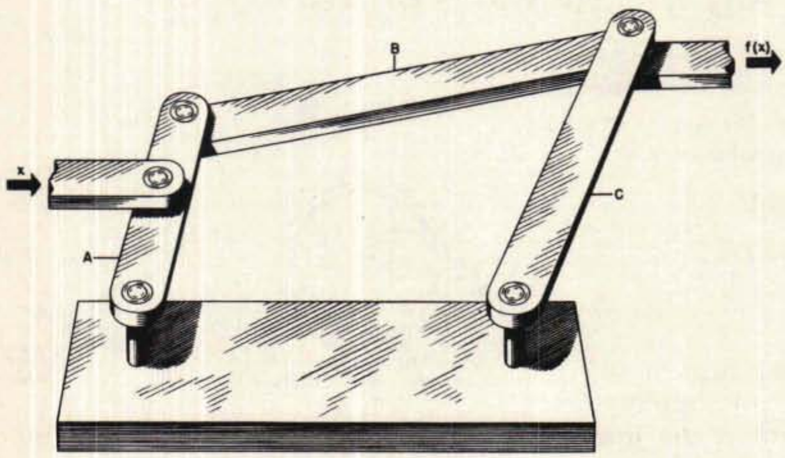


PLOT OF EMPIRICAL DATA

In the solution of a fire control problem, some data must be computed which cannot conveniently be expressed by a simple mathematical relationship. This data cannot, for example, be expressed as easily as the sine of an angle, or as the sum or product of several quantities. Empirical data of this type may be described by means of graphs.

As an example, assume a curve, plotted from empirical data, showing the variation of the quantity $f(R)$ with range. The variation is such that as range increases, the quantity $f(R)$ increases, but not according to any relation that can be expressed by a simple equation.

Since $f(R)$ cannot be expressed by a simple equation, the linkage mechanisms described previously cannot be used. Special linkage mechanisms, however, can be designed to generate complicated functions of an input variable.

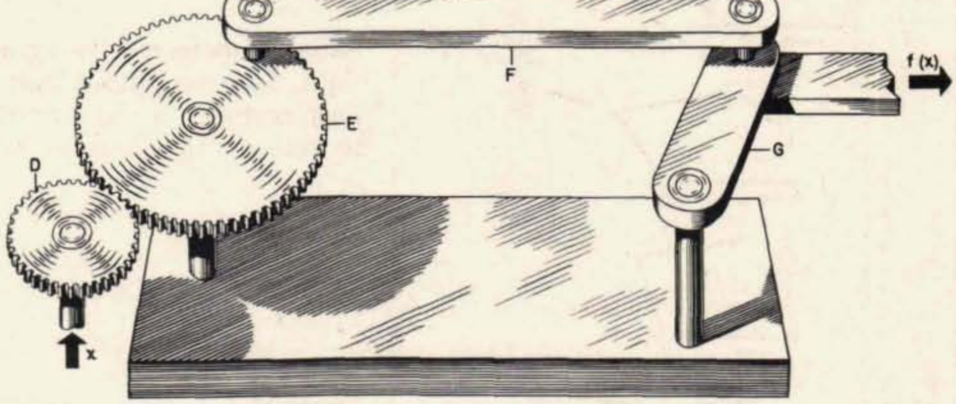


NON-LINEAR FUNCTION MECHANISM

FUNCTION MECHANISMS

One of the simplest function mechanisms consists of three links, *A*, *B*, and *C*. The lower ends of links *A* and *C* are connected to fixed pivots and the upper ends are connected to each other through link *B*. The input is carried through links *A* and *B* to the output link. The function computed depends upon the point on link *A* at which the input link is located and also upon the lengths and arrangement of the individual links.

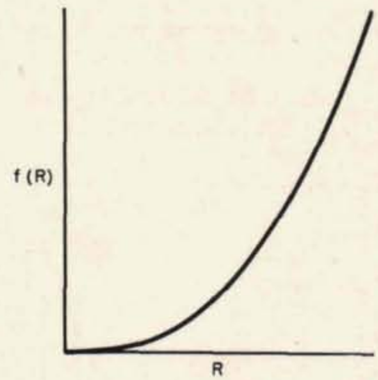
The same function can be computed for angular movements of the input by replacing the input link and link *A* with two gears, *D* and *E*. The input drives through gears *D* and *E* while a pin on gear *E* drives the output link through link *F*. The function computed depends upon the choice of gear and linkage dimensions.



ANOTHER ARRANGEMENT

How It Works

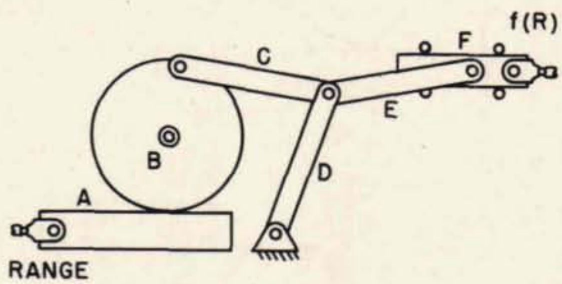
Assume that a curve has been plotted showing the variation, with range, of the quantity $f(R)$.



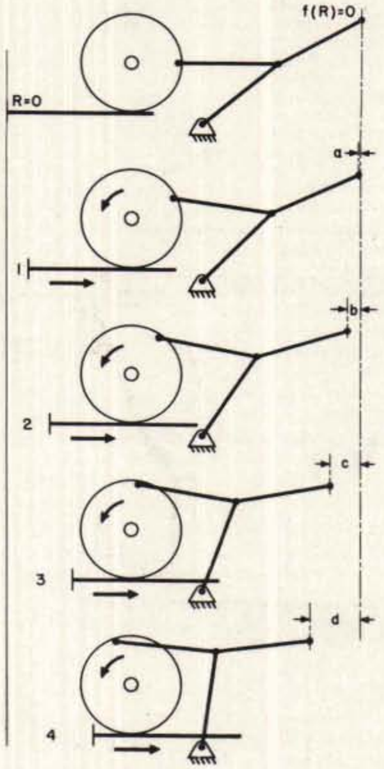
CURVE SHOWING *f(R)*

Also assume that a function mechanism has been selected to compute $f(R)$. Non-linear linkage mechanisms have been classified and the general types of functions computed by these mechanisms have been determined. From this classification, a particular mechanism can be selected to approximate $f(R)$. Then, by trial and error, the dimensions of links can be altered until the selected mechanism accurately computes $f(R)$.

In this case, the selected mechanism consists of rack *A*, gear *B*, links *C*, *D*, and *E*, and a guided bar *F*. The input is introduced at rack *A*, which, in turn, drives gear *B*. A pin on gear *B* drives links *D* and *E* through link *C*. Link *E* moves bar *F* between guide rollers. An input link connects with rack *A* and an output link connects with bar *F*.



MECHANISM FOR COMPUTING *f(R)*

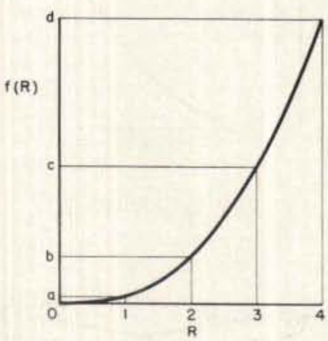


OUTPUT OF MECHANISM
FOR INCREMENTS OF *R*

With range at zero, the input link is at its extreme left position and the output link is at its extreme right position.

As range increases, the input link moves to the right, and the output link moves to the left. Displacement of the output link to the left of its zero position represents the quantity $f(R)$.

Successive increases in range cause further increases in $f(R)$.

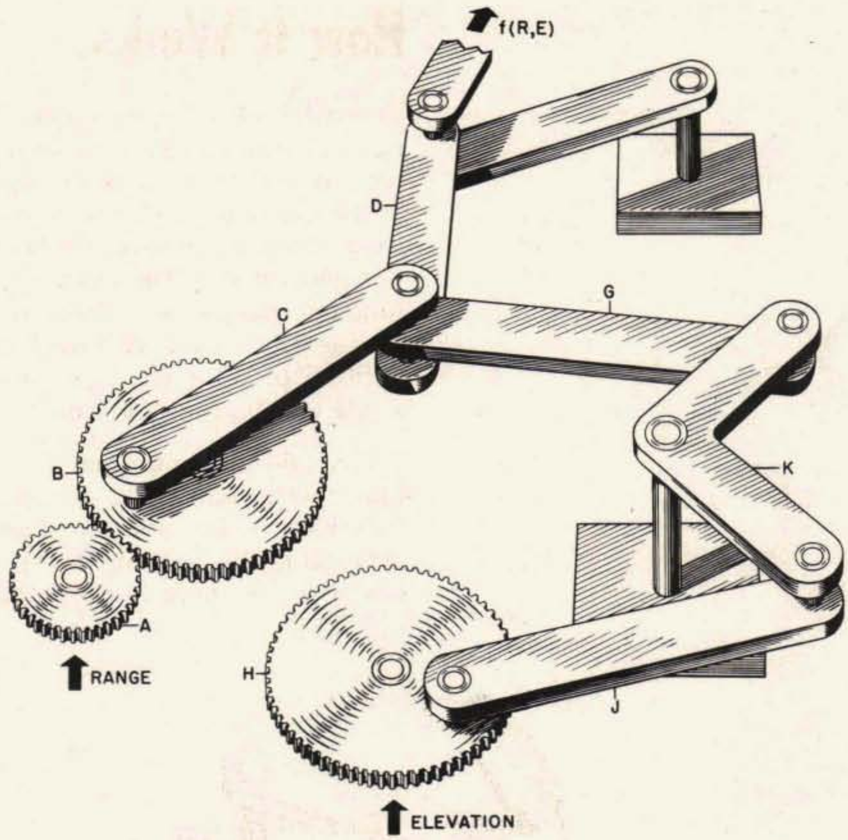


PLOT OF MECHANISM
OUTPUT

By plotting the values of the output link against the respective input values, a curve showing the increase in $f(R)$ with range is obtained. The shape of the curve is essentially the same as that of the original $f(R)$ curve. By the proper choice of gear and linkage dimensions, the mechanism can be designed so that its output is an extremely close approximation of the correct value of $f(R)$.

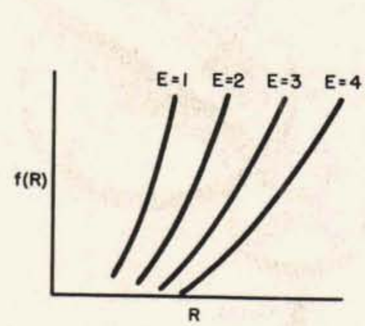
Schematic Symbol

The schematic symbol for the function mechanism consists of the lower case letter "f" enclosed in a circle.



COMPOUND FUNCTION MECHANISMS

Some of the special quantities used in the solution of a fire control problem are functions of two independent variables. For example, the quantity $f(R,E)$ is a function of both range and elevation. This function is shown graphically by means of a family of curves. Each individual curve represents values of $f(R)$ for all values of range at one elevation only. Separate curves are plotted for different elevations. The entire family of curves shows $f(R,E)$.



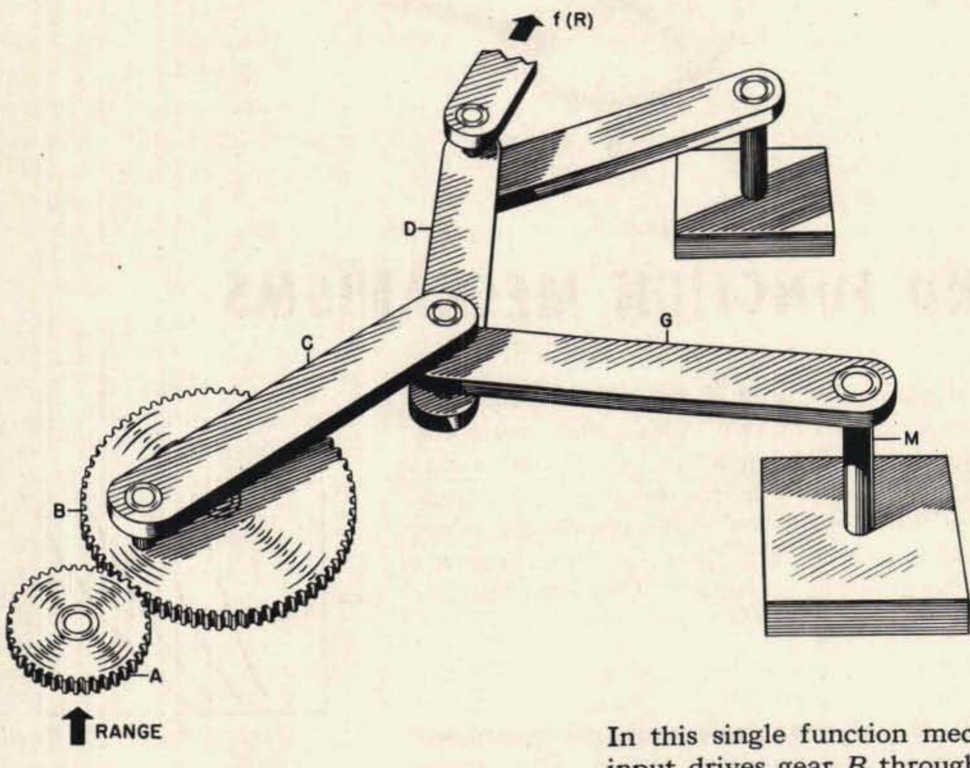
FAMILY OF CURVES SHOWING *f(R,E)*

Two single function mechanisms can be combined to compute the function represented by a family of curves. This combination is called a compound function mechanism. Such a mechanism has two inputs, each computing a part of the function. These two parts of the function are combined within the mechanism to produce the compound function.

How It Works

The value of a function generated by a single function mechanism depends not only upon the lengths of the links but also upon the locations of the fixed pivots. If the location of one of the fixed pivots is changed, the values of the function generated will also change. The compound function mechanism works on this principle; that is, the output is varied by changing the position of what would be a fixed pivot in a single function mechanism.

To explain the operation of a compound function mechanism, first assume that a single function mechanism has been designed for generating the quantity $f(R)$, values of which describe one curve in the family of curves for $f(R,E)$.

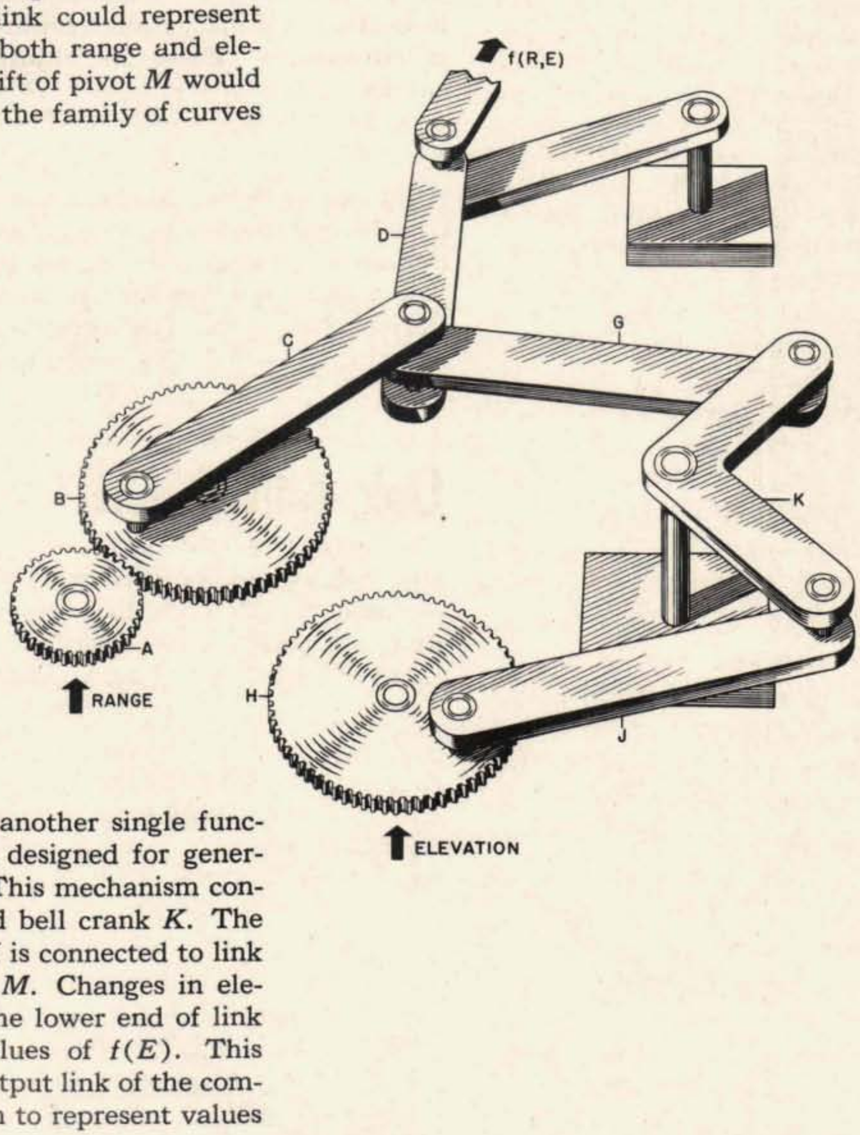


SINGLE FUNCTION MECHANISM

In this single function mechanism, the range input drives gear *B* through gear *A*, while an eccentric pin on gear *B* drives the output link through links *C* and *D*. Link *G*, pivoted at point *M*, guides the movement of links *C* and *D*, thereby determining the values of $f(R)$.

If fixed pivot M could be shifted to another point, a different set of values could be generated. These values could represent some other curve in the family of curves for $f(R,E)$.

Furthermore, if pivot M could be shifted to various positions representing values of $f(E)$, movement of the output link could represent the combined function of both range and elevation, or $f(R,E)$. Each shift of pivot M would represent another curve in the family of curves for $f(R,E)$.

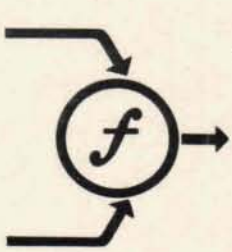


COMPOUND FUNCTION MECHANISM

Accordingly, assume that another single function mechanism has been designed for generating the quantity $f(E)$. This mechanism consists of gear H, link J, and bell crank K. The output end of bell crank K is connected to link G in place of fixed pivot M. Changes in elevation shift the pivot at the lower end of link G in accordance with values of $f(E)$. This causes movement of the output link of the compound function mechanism to represent values of the quantity $f(R,E)$.

Schematic Symbol

The schematic symbol is similar to that for the single function mechanism except that there are two inputs to the mechanism instead of only one.



LINKAGE ADJUSTMENTS

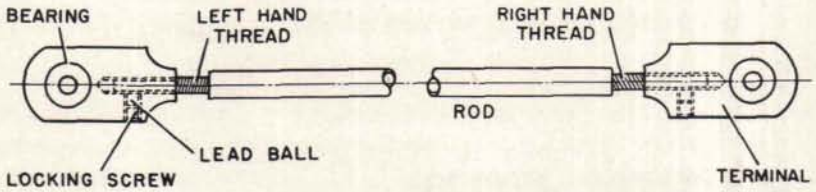
The use of clamps for adjustment of gearing was discussed in Section 1. Linkage mechanisms use two additional means of adjustment. These are adjustable links and adjustable cranks.

A measuring device to check the adjustment of a linkage is commonly provided by two accurately located holes having the same diameter. One hole is in the mechanism to be adjusted and the other hole is either in another link or in a stationary member. The correct adjustment is obtained when a setting rod can be inserted through both holes.

Link Adjustment

One linkage mechanism can be adjusted to another linkage mechanism by varying the distance between pivots in a link that connects the two mechanisms. This distance can be changed by the use of an adjustable link.

An adjustable link consists of a rod, threaded at both ends, and of terminals that screw on to the ends of this rod. Each terminal contains a bearing. One end of the rod has a right-hand thread and the other a left-hand thread.

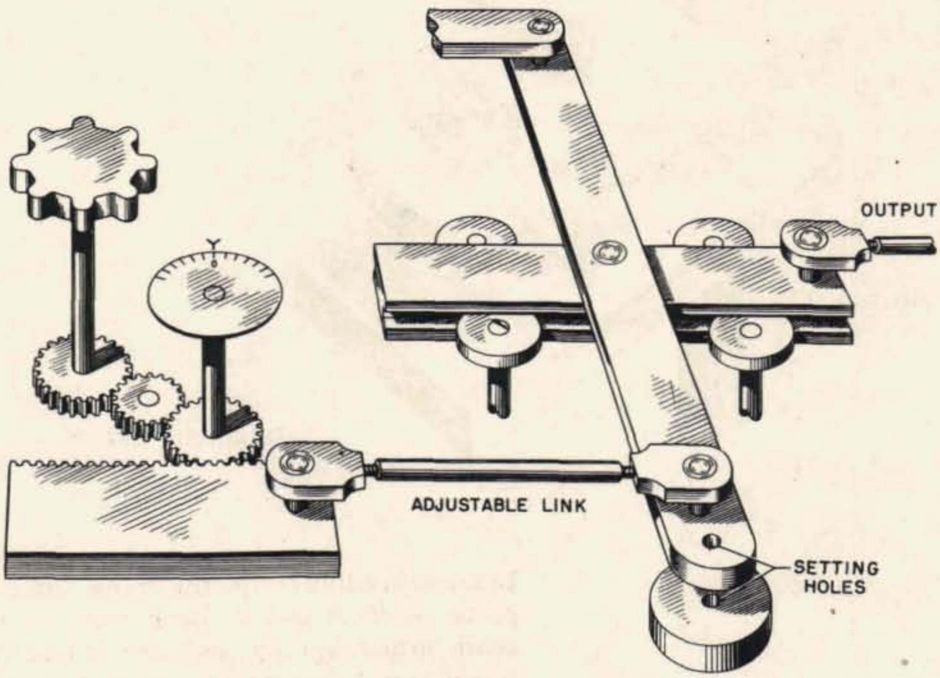


ADJUSTABLE LINK

The action of the adjustable link is similar to that of a turnbuckle. The right-hand and left-hand threads on the rod make it possible to vary the distance between terminals. Rotating the rod in one direction causes both threaded ends to screw into the terminals, so that the distance between the terminals decreases. Rotating the rod in the opposite direction causes the ends of the rod to back out of both terminals, thereby increasing the distance between terminals.

A locking screw threaded into the side of each terminal is used to secure the rod after an adjustment has been made. A lead ball between the locking screw and the rod prevents injury to the threads on the rod.

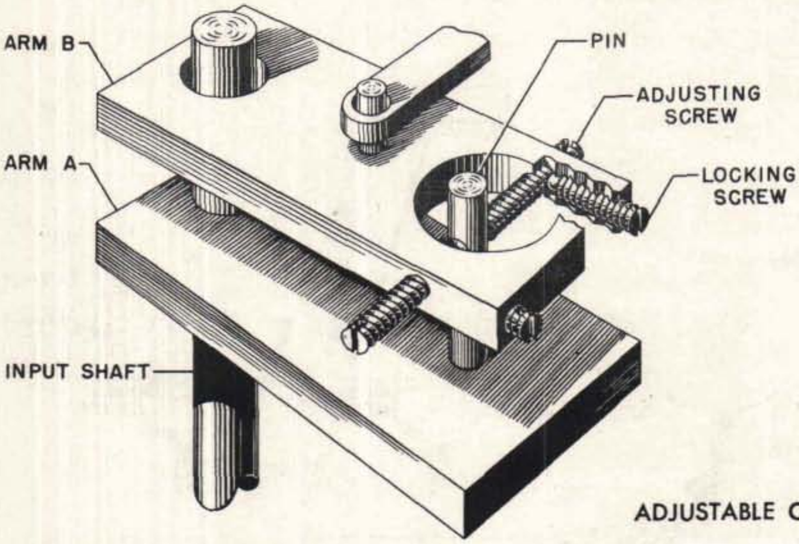
APPLICATION OF ADJUSTABLE LINK



As an example of how the adjustable link is used, assume that a linkage differential is to be adjusted to its zero position. One input to the differential is connected to another linkage mechanism by means of a non-adjustable link, and it is assumed that this mechanism is at its zero position. The second input is connected to a dial and a knob by means of gears, a rack, and an adjustable link. The adjustment consists of setting the adjustable link so that the differential is at its zero position when the dial is at zero. The dial merely indicates the zero position of the knob input. The zero position of the differential is obtained when a setting rod can be inserted through a hole in the differential and a hole in the instrument frame.

Crank Adjustment

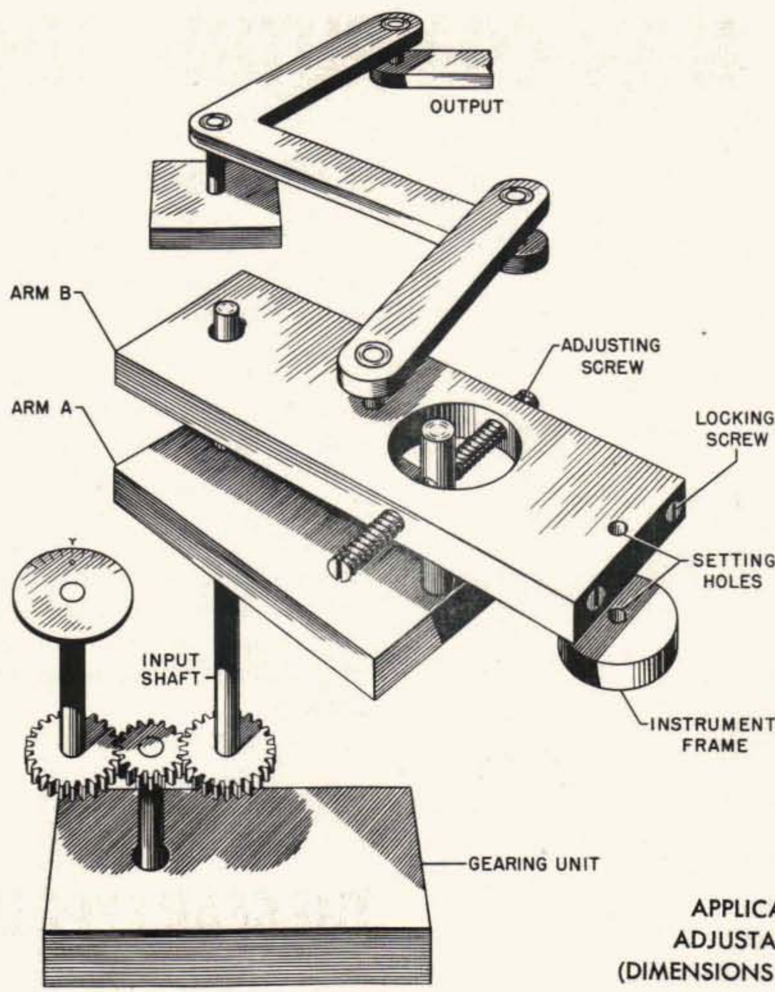
Transfer of shaft rotation to linkage movement can be accomplished by means of a crank. Adjustment between the shaft and the linkage can be made by an adjustable crank, which permits changing the angular relationship between the shaft and the crank.



ADJUSTABLE CRANK

In an adjustable crank, the crank arm consists of two separate parts: arms *A* and *B*. Both arms are mounted on the input shaft; arm *A* solidly, and arm *B* loosely. A pin, fixed to the top of arm *A*, is positioned in an opening in arm *B*. Adjusting screws on each side of the opening bear against the pin, causing the two arms to act as a single crank arm. Locking screws are used to secure the adjusting screws. A lead ball under each locking screw prevents injury to the thread of the adjusting screw. The output link is connected to the top of arm *B*.

The outer end of arm *B* can be moved relative to arm *A* by unscrewing one of the adjusting screws and by turning the other one in. Since arm *B* is mounted loosely on the input shaft, it can be rotated on the shaft relative to arm *A*. The amount of rotation is limited by the size of the opening in arm *B*.



APPLICATION OF
ADJUSTABLE CRANK
(DIMENSIONS EXAGGERATED)

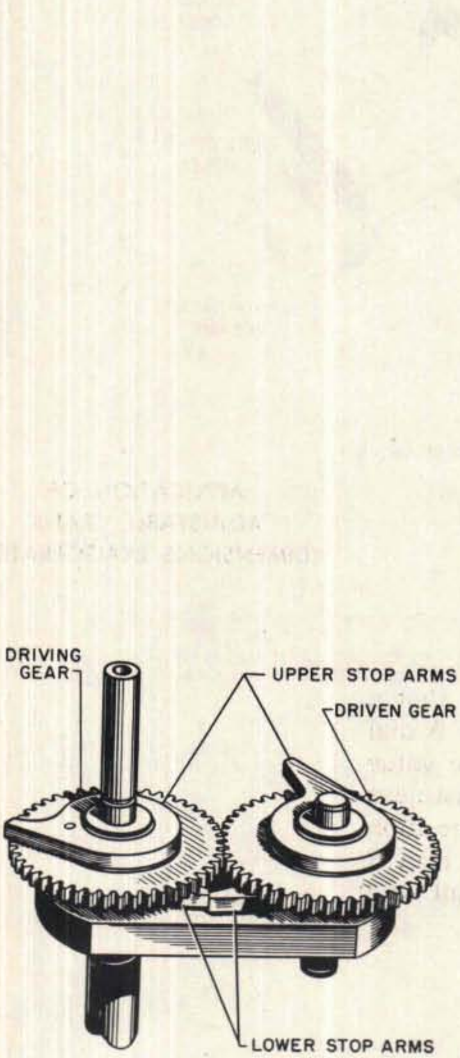
To illustrate the use of the adjustable crank, assume that a linkage mechanism is to be adjusted to a gearing unit. A dial which is geared to the crank input shaft indicates the value represented by the position of the shaft. The adjustment consists of bringing the linkage mechanism to the position where a setting rod can be inserted through a hole in arm *B* and a hole in the frame of the instrument, with the dial held at a prescribed value.

NON-COMPUTING LINKAGES

Linkage mechanisms, as in the case of other types of computing mechanisms, require safety devices to limit the movements of the various elements. These safety devices consist of limit stops, intermittent drives, and relief drives.

One type of limit stop and one type of intermittent drive are described in Section 2. An additional type of each is described here. A description of the relief drive, which may be considered as a linkage type of intermittent drive, is also included. The limit stop, intermittent drive, and relief drive described here are not only used with linkage mechanisms but may also be used with other types of computing devices.

THE GEAR TYPE LIMIT STOP



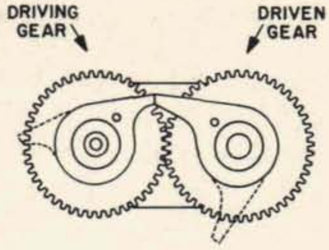
GEAR TYPE LIMIT STOP

Gear type limit stops serve the same general purpose as the screw type limit stops described on page 144.

The gear type limit stop consists of two pairs of stop arms and two meshing gears of unequal size. One stop arm is pinned separately to each face on each gear, forming a pair of stop arms above, and another below, the two gears. One arm of each pair is longer than the other. The longer arm is pinned to the larger gear. One pair of arms stops the rotation of the gears in one direction when the end faces of the two arms meet. Rotation of the gears in the opposite direction is stopped by the other pair of stop arms. The number of revolutions that the drive gear can turn before it is stopped depends upon the gear ratio and upon the relative positions of the stop arms on the gears.

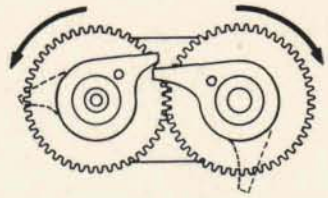
How the Stop Operates

Suppose, initially, that the ends of the upper stop arms are touching. At this stop position (limit *A*), engagement of the upper arms limits rotation in one direction. However, the gears can rotate in the opposite direction since the lower stop arms are apart.

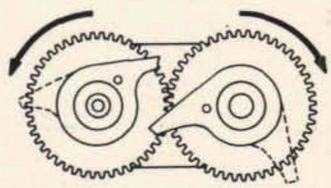


AT LIMIT A

Now, suppose that the small (driving) gear turns one revolution from limit *A*. The large (driven) gear turns with the small gear, but the large gear does not turn through a complete revolution because it has more teeth than the small gear. This gives the arm on the small gear a greater rotational movement than the arm on the large gear. Because of this difference in movement and because of the shape of the arms, the upper arms are sufficiently apart at the end of one revolution to clear each other.

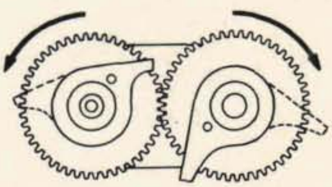


ONE TURN FROM LIMIT A



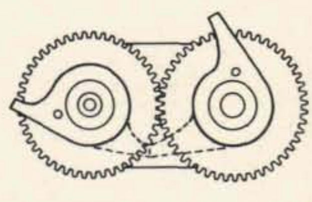
SIX TURNS FROM LIMIT A

As the two gears revolve, the difference in rotational movement increases. Consequently, the upper arms move farther apart. At the same time, the lower arms move closer together.



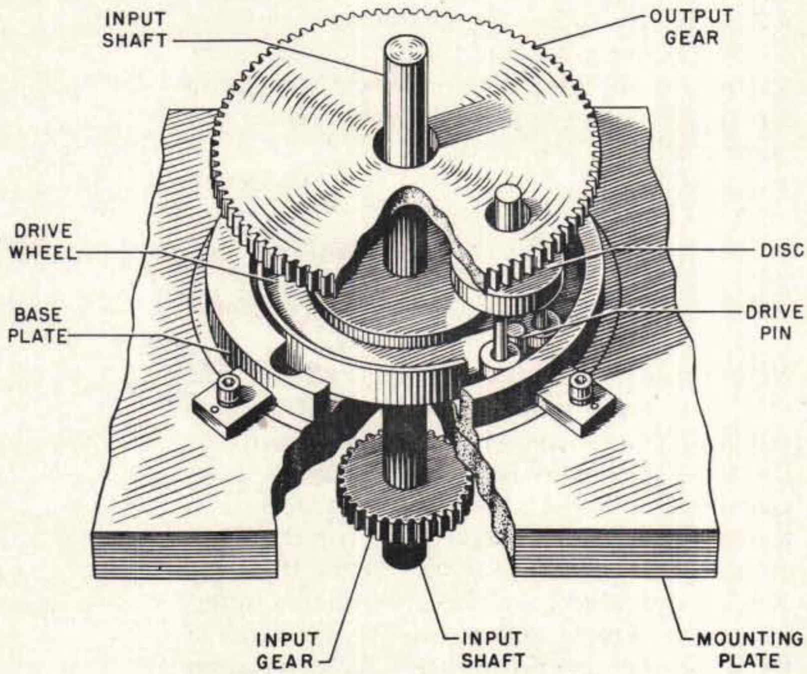
TWELVE TURNS FROM LIMIT A

After a number of revolutions, depending on the gear ratio and the angular spacing of the arms, the ends of the lower arms meet and stop rotation at limit *B*. However, the upper arms are now apart so that the gears can be rotated back to limit *A* again.



AT LIMIT *B*
($12\frac{1}{3}$ TURNS FROM LIMIT *A*)

THE INTERMITTENT DRIVE



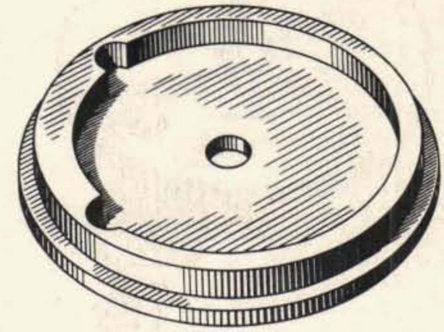
The function of an intermittent drive is described on page 146. A particular type of intermittent drive is also shown on that page. The description following page 146 shows that this type of drive permits many turns of its input and output shafts while the two shafts are connected, and also permits many additional turns of the input shaft while the output shaft is disconnected and locked. The type of drive shown there is particularly adaptable for use with a mechanism, such as a component solver, that requires many turns of the connecting shafts.

Another type of intermittent drive is illustrated here. This type differs from the type shown on page 146 in that its input and output shafts are connected for less than one revolution, and its input shaft is free to turn for less than one revolution after the output shaft has been disconnected and locked. It is adaptable for use in shaft lines which move through small angles, such as those which operate linkage mechanisms.

HOW THE DRIVE OPERATES

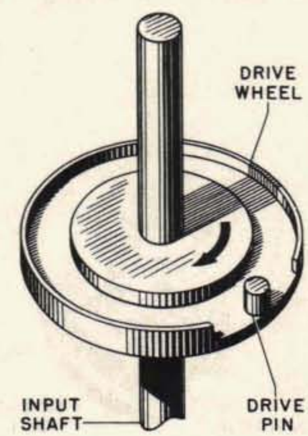
Driving the Output Gear

1. The input shaft moves a drive pin. The stationary base plate, which is clamped to a mounting plate, has a centrally located bearing. This bearing supports the input shaft so that the shaft can turn freely.



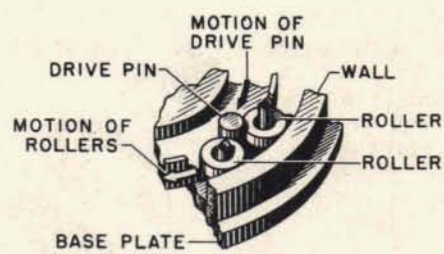
STATIONARY BASE PLATE

A drive wheel is secured to the input shaft. As the input shaft turns, it rotates the drive wheel. A drive pin is mounted near the edge of this wheel.

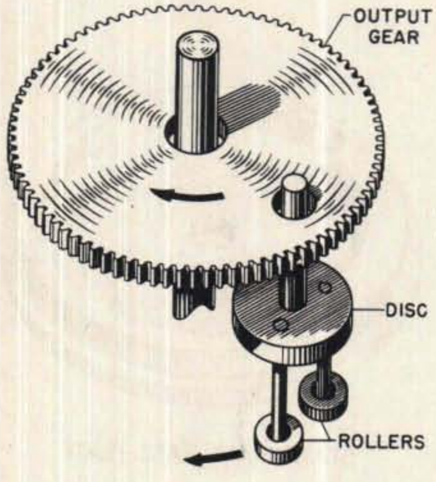


FROM INPUT SHAFT TO DRIVE PIN

2. The drive pin acts against two rollers. The drive pin is confined between two rollers. It pushes against them, causing the rollers to move around a cylindrical wall on the base plate.



FROM DRIVE PIN TO ROLLERS

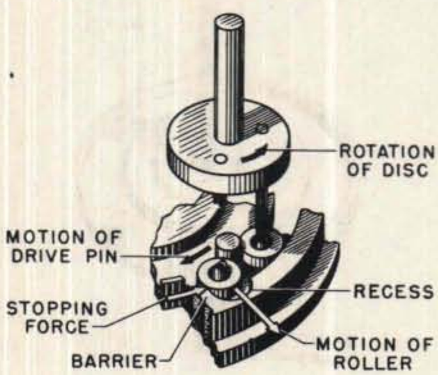


FROM ROLLERS TO OUTPUT GEAR

3. The rollers transfer motion of the drive pin to the output gear. The two rollers are attached to a disc which, in turn, is bearing-mounted on the output gear. Through this arrangement the drive pin, by pushing against the rollers, causes the output gear to turn with the input shaft.

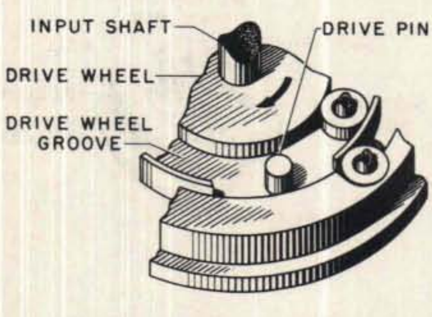
Cutout Action

1. The rollers are stopped by a barrier. The cylindrical wall on the base plate has a barrier formed by an increase in the wall thickness. When a roller strikes this barrier, it cannot travel farther around the wall. However, adjacent to this barrier is a recess into which the striking roller can enter.



TRANSFER FROM DRIVE POSITION TO CUTOUT

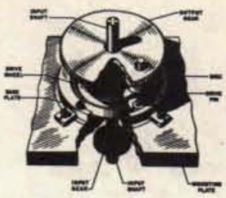
2. The disc rotates. The drive pin, pushing against the side of the roller which has struck the barrier, forces the roller into the recess in the base plate wall. This movement of the roller makes the disc rotate about its pivot. As the disc rotates, it carries the second roller around the drive pin to a position nearer the center of the drive wheel.

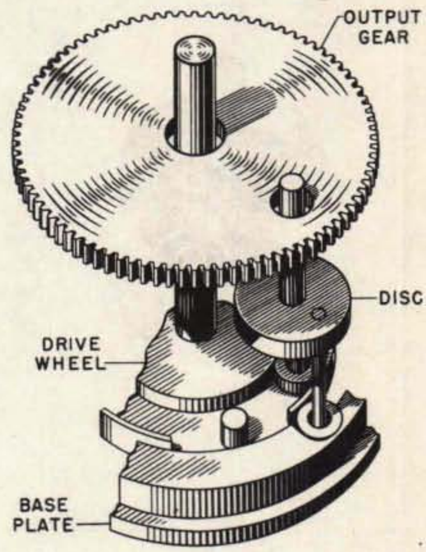


INPUT SHAFT CAN TURN

3. The input shaft is free to turn. With one roller in the recess and the other roller behind the drive pin, the drive pin is free of the rollers. Consequently, the drive wheel and input shaft can continue rotation without interference.

4. The output gear is locked. The roller confined in the recess in the stationary base plate cannot move. Consequently the output gear which supports this roller cannot turn. However, this locking action is positive only if the roller remains in the recess. To keep the roller there, the disc must be prevented from rotating back to its original position. This is accomplished by confining the inner roller within a groove cut into the drive wheel. On each side of the drive pin, the outer wall of this groove has been cut away. This cut-away portion allows the disc to carry the inner roller into alignment with the groove. Continued rotation of the drive wheel completely confines the roller within the groove so that the disc can no longer turn, thus the outer roller is locked within the recess in the base plate.

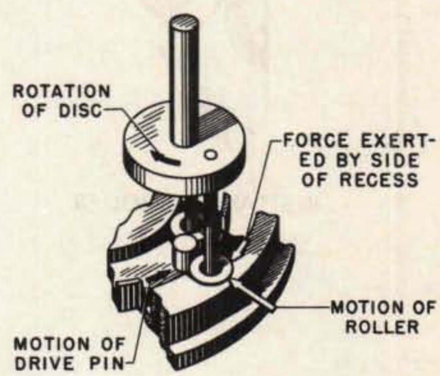




OUTPUT GEAR IS LOCKED TO BASE PLATE

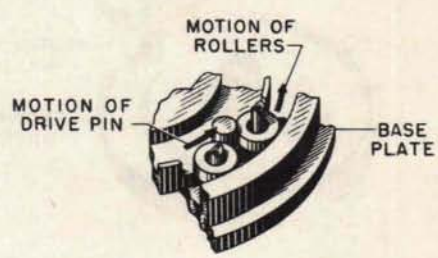
Returning to the Drive Position

1. The drive pin strikes a roller. If rotation of the input shaft is reversed, the drive pin moves back to the cutout position and strikes the roller that was confined in the drive wheel groove.



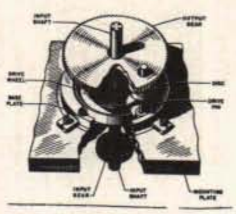
TRANSFER FROM CUTOUT TO DRIVE POSITION

2. The disc rotates. The force exerted by the drive pin and the force exerted by the side of the recess create a turning effort which rotates the disc back to the drive position. Rotation of the disc aligns the rollers with the wall of the base plate. The drive pin is now between the rollers.



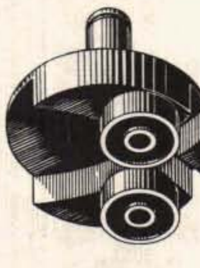
DRIVE POSITION

3. The output gear is driven by the input shaft. The drive pin pushes against the rollers, driving them around the base plate wall. This motion is transferred to the output gear, as described previously.



LIMITS OF OPERATION

The angle through which the output gear is able to rotate is limited to the angle between the two recesses, or cutout points, in the base plate. This angle, therefore, is dependent on the design of the base plate.



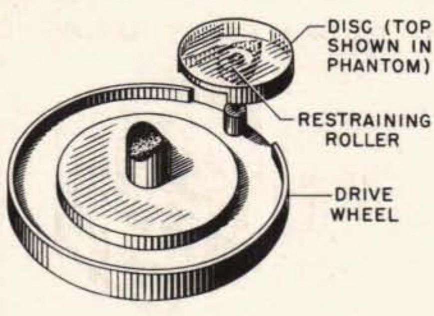
RECESS IN DISC

REFINEMENTS OF DESIGN

For simplicity of description, nothing has been said thus far about features of the intermittent drive that are not basically essential. These features serve primarily to improve the smoothness with which the drive cuts out, by preventing interference between the wall of the drive wheel groove and the roller that enters this groove.

RESTRAINING ROLLER

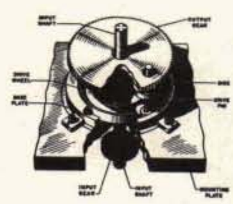
The features involved are (1) a recess in the face of the disc, and (2) a restrainer (a roller mounted on the drive pin). The recess is shaped like a V, except that the side walls are curved rather than straight. Slots extending from the bottom of the V are non-functional, being merely tool clearance incident to machining. The restrainer acts in the recess in the disc.

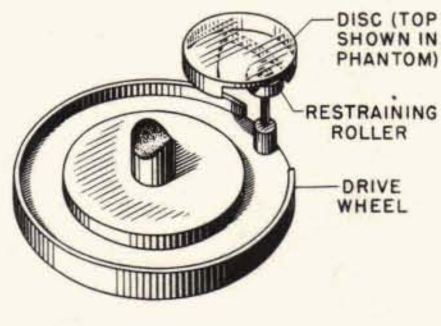


DRIVE POSITION

While the disc is in the drive position, the open end of the V-shaped recess faces the input shaft, and the restrainer bears against the sides of the V.

At cutout position, the disc is rotated so that one roller is in the recess of the base plate. After cutout, the drive pin moves beyond this roller, losing contact with it. This action starts before the outer wall of the drive wheel groove has confined the other roller. Therefore, at this time, the drive wheel groove cannot restrain the disc from turning back toward the drive position. The restrainer does, however, perform this function.





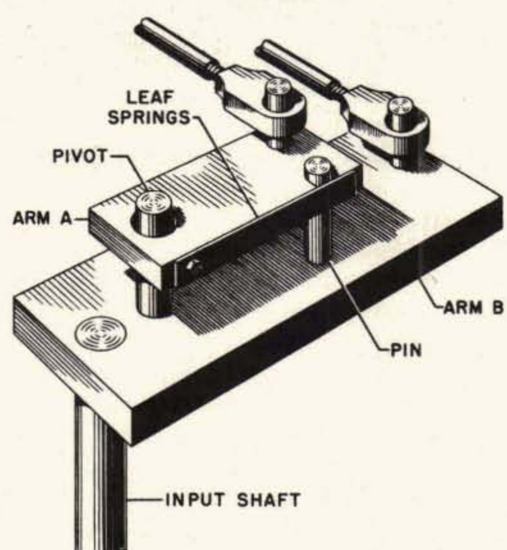
CUTOUT POSITION

With the disc at the cutout position, the V-shaped recess is oriented so that its open end faces in the direction of drive wheel motion, and so that one of its curved sides is parallel to the drive wheel rim. The restrainer maintains contact with the curved wall of the V and holds the disc at the cutout position. Hence, the roller is kept in alignment with the drive wheel groove, eliminating the possibility of the drive wheel wall striking the roller.

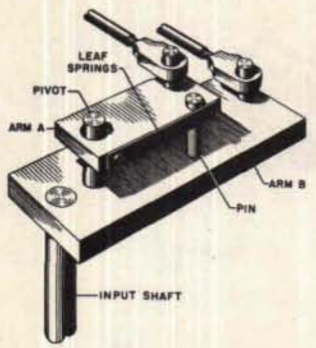
THE RELIEF DRIVE

The relief drive may be considered as a linkage type of intermittent drive which provides relief in one direction only.

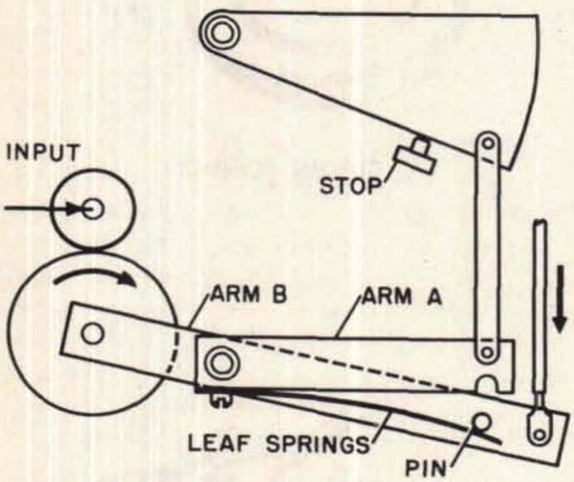
The relief drive consists of a spring-loaded crank arm, *A*, supported on an ordinary crank arm, *B*. Arm *B* is secured to the input shaft. The inner end of arm *A* pivots on arm *B*. A pin attached to arm *B* fits a recess in the side of arm *A*. Leaf springs, fixed at one end of the side of arm *A*, confine the pin in the recess. One output link connects with arm *A*. Another may connect with arm *B*.



RELIEF DRIVE



During normal operation the crank arms move as a single unit. Drive from arm *B* to arm *A* is positive in one direction, being transmitted directly from the pin on arm *B* to the side of arm *A*. In the other direction, the drive is from the pin on arm *B* through the leaf springs to the side of arm *A*. Normally, the stiffness of the springs is sufficient to prevent their bending, and arm *A* moves with arm *B*.



APPLICATION OF RELIEF DRIVE

If the mechanism driven by arm *A* reaches its limit of travel, arm *A* is unable to turn farther in the limited direction. However, the input shaft may continue turning past this point. The leaf springs, by bending, allow continued rotation of the shaft. Any mechanism connected to arm *B* will therefore continue to be driven by the input.

A LINKAGE NETWORK

The simplified network described here is not taken from any actual instrument. It is an imaginary network which illustrates how linkage mechanisms may be combined to solve the mathematical equations involved in a gun fire control problem.

This network computes an approximate solution of one part of a fire control problem. It computes sight angle (Vs), which is the angle that the guns must be elevated above the line of sight to allow for the drop of the projectile during its flight.

Sight angle in this problem is determined by advance range $(R2)$.

Advance range is here equal to the sum of present range and the predicted change in range caused by relative motion between own ship and target. The change in range is called range prediction (Rt).

In order to compute sight angle, two quantities are used as inputs to the network:

1. Present range (R), obtained from the radar equipment.
2. Range rate (dR), the rate at which present range is changing. It is assumed that dR is available from another network, not described here.

Computing Range Prediction (Rt)

Range prediction depends upon the time of flight of the projectile and upon the range rate. For simplicity, it is assumed that time of flight (Tf) is directly proportional to present range (R), or:

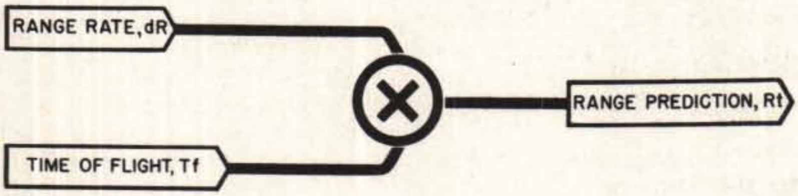
$$Tf = K \times R$$

A bell crank multiplier is used to multiply present range (R) by a constant (K) to compute Tf for any given value of R.

Range prediction (Rt) is the product of Tf and dR, or:

$$Rt = Tf \times dR$$

An XY multiplier is used to multiply Tf by dR, the quantity dR being supplied as an input to the network. The multiplier output is Rt.

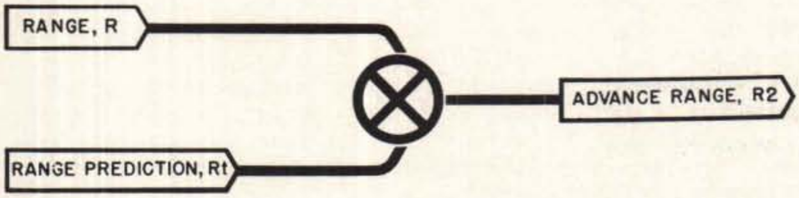


Computing Advance Range (R2)

Advance range is equal to the sum of present range and range prediction, or:

$$R2 = R + Rt$$

Both R and Rt become inputs to a linkage type differential, resulting in an output of advance range ($R2$).

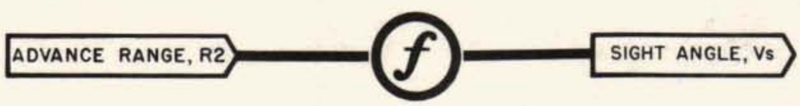


Computing Sight Angle (Vs)

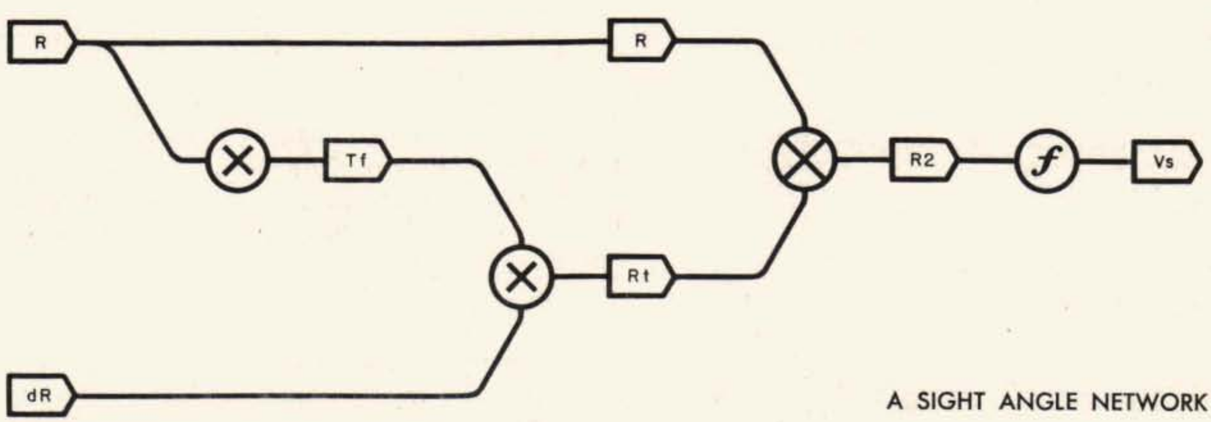
Advance range ($R2$) goes to a function mechanism which solves the equation:

$$Vs = f(R2)$$

This mechanism is designed so that its output is sight angle (Vs) corresponding to the value of advance range that is set into the mechanism.

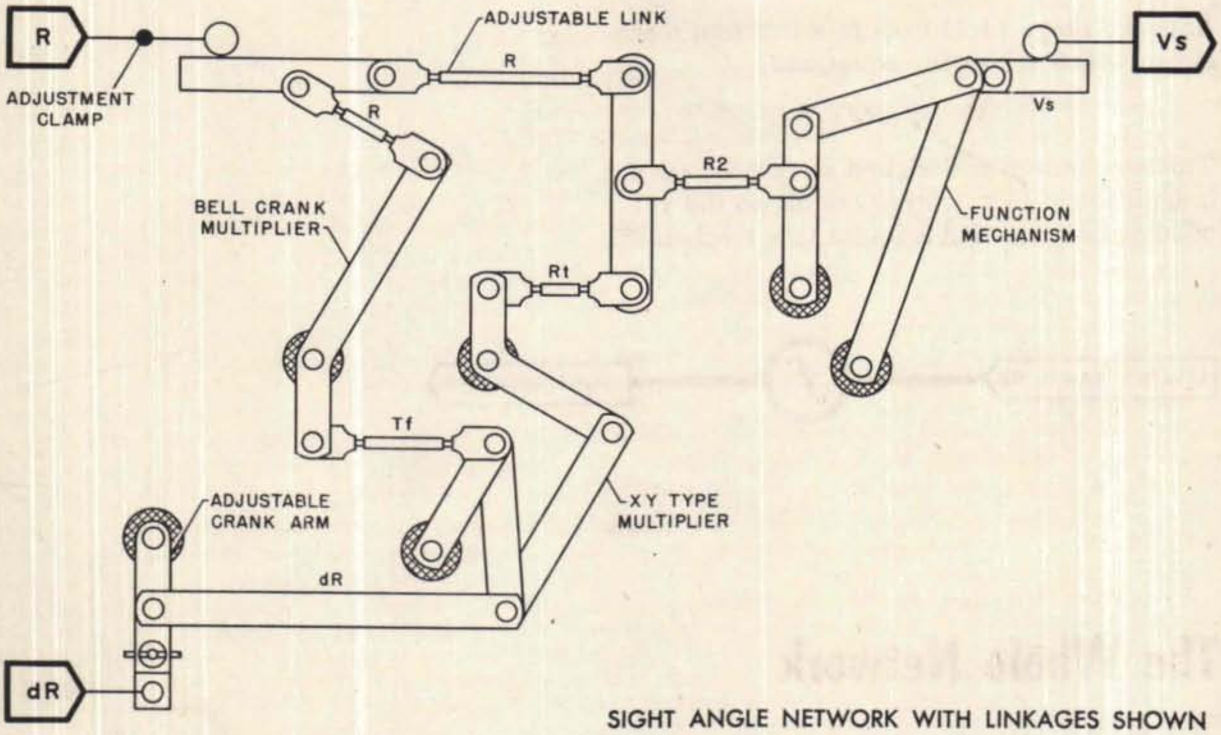


The Whole Network



A SIGHT ANGLE NETWORK

Here is what the entire network looks like, when shown schematically. Each change in the inputs to the network immediately changes the values all along the line, including the final output, sight angle.



SIGHT ANGLE NETWORK WITH LINKAGES SHOWN

Here is the complete network with the linkage mechanisms shown. The network consists of a bell crank multiplier, an *XY* multiplier, a linkage type differential, and a function mechanism.

Adjustments between the individual mechanisms are made at the points indicated.