COLD AND HOT ALIGNMENT TECHNIQUES OF TURBOMACHINERY

by

Charles Jackson
Engineering Fellow
Monsanto Polymers and Petrochemical Co.
Texas City, Texas

Charles Jackson is an engineering fellow in a Mechanical Technology Section working on mechanical and method-oriented problems at the Monsanto Polmers and Petrochemical Company, Texas City, Texas. He has been with the company for 21 years working in Division Engineering, Production, Maintenance and Plant Design, before setting up the present department in 1960 later being appointed engineering specialist, then senior engineering specialist. Mr. Jackson holds a B.S. degree in mechanical engineering from Texas A&M University, plus an AAS degree in electronics technology from College of the Mainland, Texas City, Texas. He is a registered professional engineer, a member of ASME, API and SESA.

The first five years of vibration analysis work on turbomachinery indicated that 60% of the problems encountered were a result of casing-to-casing misalignment.

In all cases, the indicators were twice rotating speed (2 × synchronous) frequencies plus significant axial viabration at single and double rotating speed frequencies. More normally, the axial vibration will be equal or higher in magnitude to the radial vibrations.

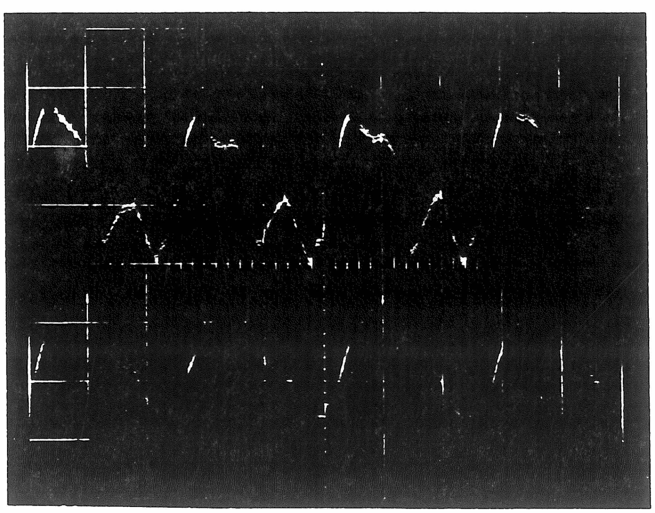


Figure 1. ***Typical vibration signal with significant twice*** *running speed frequency from external shaft misalignment. Speed is $\approx$ 5000 RPM (from once-per-turn marker on photo). Calibrations 1 cm = 0.5 mils p/p and 5 m sec sweep.*

Severe misalignment will ultimately cause coupling and or shaft failures. Several other exterior indications are worth noting should one be limited in vibration measurements:

1. Heat generation at the coupling faces.

2. Fretting of coupling hubs.

3. Shaft cracks at the large end of the coupling hub taper and keyway.

4. Abrasive failure due to extreme wear at the coupling teeth.

5. Circular flexure fatigue of diaphragm type coupling.

6. Shredding of shims or discs from multiple membrane type coupling.

Because a flexible gear type coupling is attemping to correct for all the shaft misalignment, the surface pattern may take the form of a metal "pimple" that is worked up in the center of the contact face and often works outward across the face leaving a gouge or scoring mark.

Several shaft cracks have been detected and fortunately were detected before failure from increased synchronous vibration pointing to coupling unbalance which

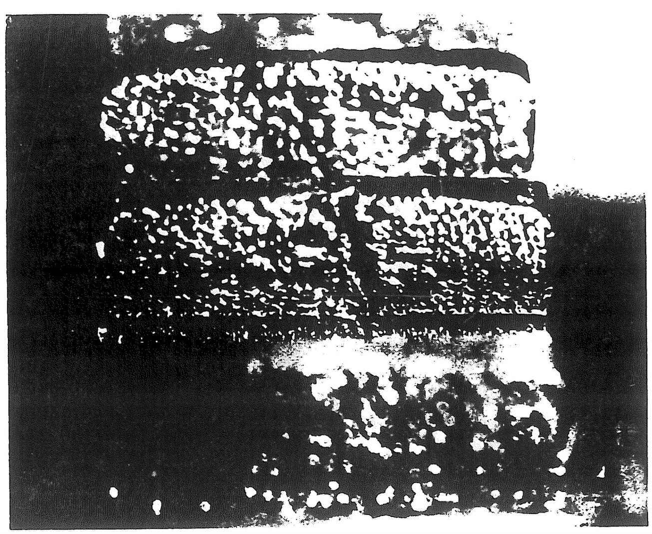


Figure 2. Flexible coupling tooth face photographed at 1:2 (110 mm with 55 mm lens). This damage occurred in three weeks on turbine-compressor coupling operating at 8800 RPM and 9000 HP with turbine rise at 100 mils over a given 60 mils rise. Turbine exhaust is 450 psig steam.

is affected by a transverse fatigue crack. The failure illustrated in Figure 3 had progressed 60% through the shaft in a transverse plane passing through the keyway at the shaft to hub change of section.

Having suffered through the problems from inaccurate measure of hot alignment, a study was instigated in 1965 to better our abilities. Often, three to four days had been required in getting questionable accuracy on a multicase compressor train with turbine drive. Extrapolated values had been based on indicator readings taken on time intervals immediately following shutdown with the set-up time lag varying from thirty minutes to two hours.

After air gauges used on thrust deflection turned up short on range, proximity probes mounted on water cooled reference stands were put into practice late in 1965. These probes taking the form of non-contacting electronic micrometers, were placed 90° apart to affect

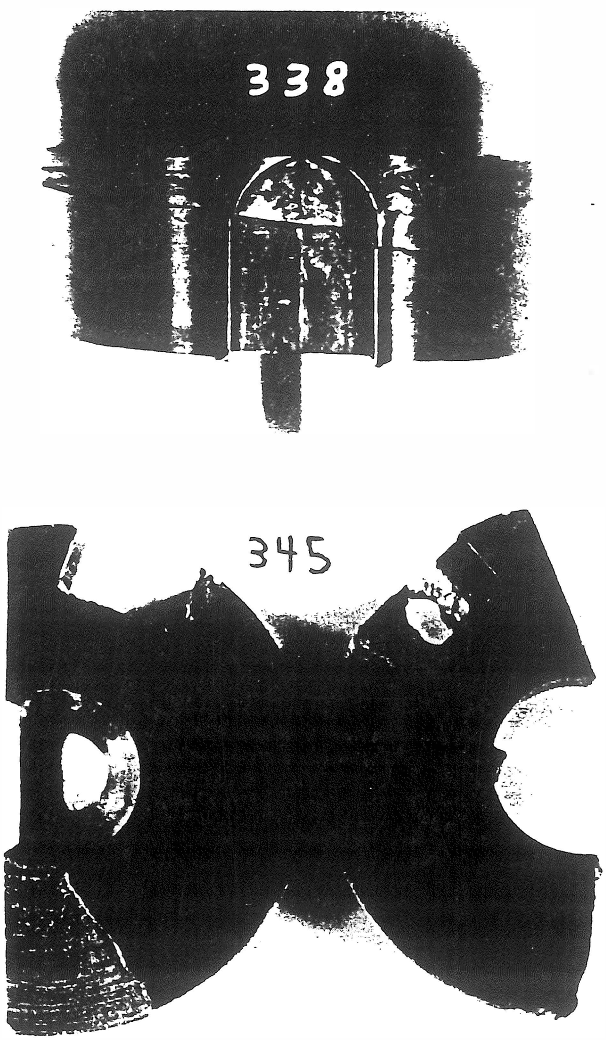


Figure 3. Transverse crack in a steam turbine shaft due to reverse bending fatigue in a plane through the keyway at the shaft-to-coupling hub fit. 40% of shaft remained at shutdown. Prior to shutdown vibration was mostly synchronous only 20% at 2 × f_r. Shaft relative vibration was 5 mils p/p and bearing caps were 0.5 ips velocity, peak.

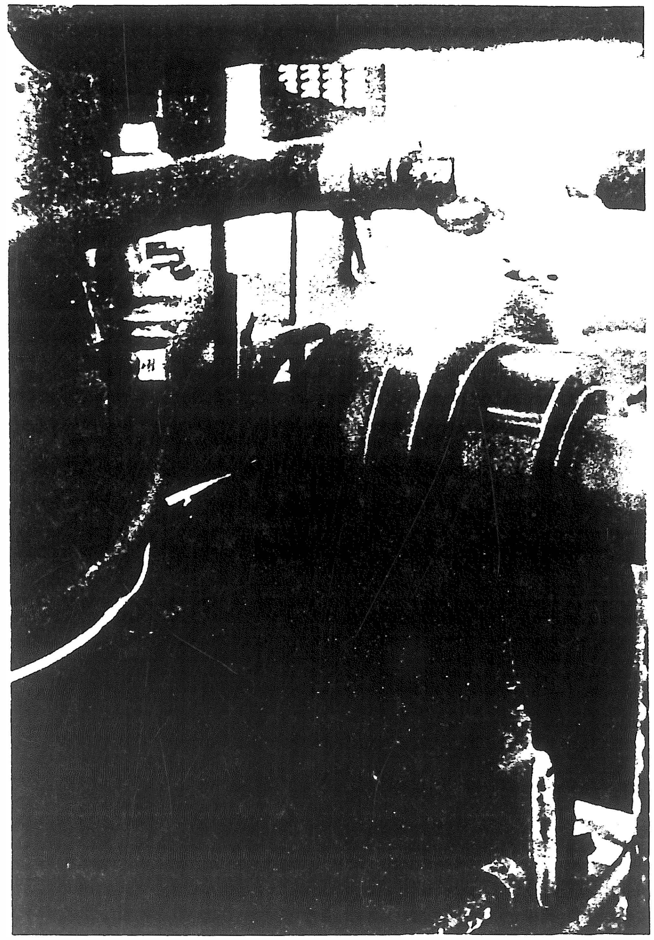


Figure 4. Probes are held 90° apart, reading directly to the shaft.

Cartesian coordinates in the horizontal and vertical transverse measurements directly to the shaft axis in each position.

In 1968, this system was modified to provide a procedure for measuring high speed barrel type centrifugal compressors with covered continuous lube couplings. Eight measurements per casing were required to do this correctly.

Several interesting facts have been derived from several years of precision hot alignment measure.

1. Many equipment manufacturers have limited to no useful data on expected heat rises.

2. The pipe strains transmitted to equipment of many installations have been drastic, i.e., over 100 mils in extreme cases.

3. The highest measured misalignment in our records has been 150 mils total centerline offset. Forty (40) mils vertical rise over supplied data is common.

4. The steam end "wobble" or flex plate designs on steam turbines will often rise seven or more mils with a change of wind direction or velocity. Supports well under bearing housing or water jackets reduce this.

5. Two cycles of heat up and cool down are required as a minimum before final locking or doweling is performed.

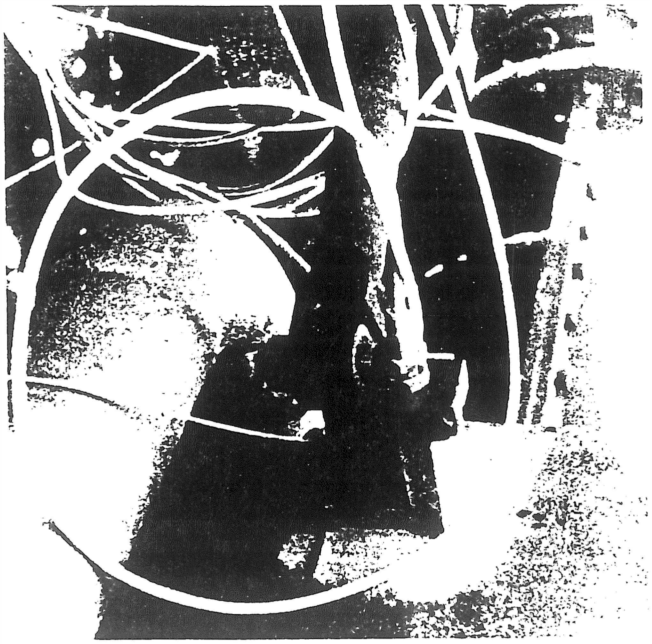


Figure 5. Barrel-type compressors require probes mounted to read vertical and horizontal shift, casing growth, and torsional roll.

6. Two dowels in the same transverse plane is the maximum in doweling. Criss-cross doweling is poor practice.

7. Measures directly to the shaft are best. The second choice would be the bearing housings; third choice, is the outer casing.

8. Horizontal lateral measures taken from *one* side of a casing only is poor. One must provide good alignment screw in the initial installation to allow shifts in alignment to be made. Refer to API 617, 1973.

9. Refrigeration compressor to condenser expansion joints are often too stiff.

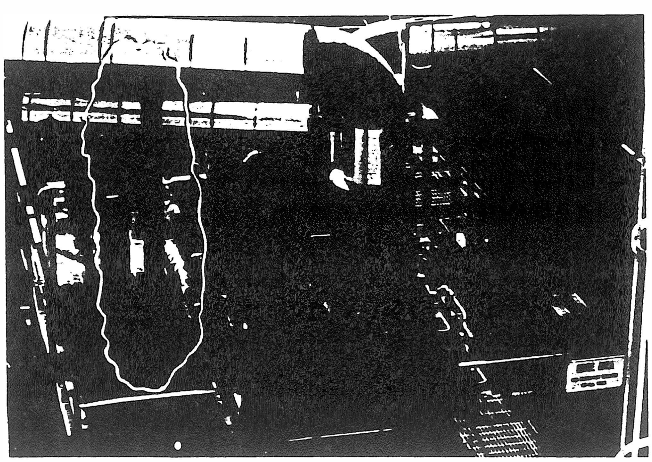


Figure 6. Three pipe columns were quickly constructed on five trains in this unit to remove strains from poor suction pipe supports. 20 mils of roll across the compressor occurred here.

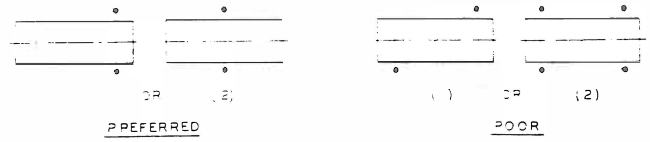


Figure 7. Poor *and* preferred *doweling is illustrated.*

10. A steam turbine should *first* be confirmed to have near zero shift when the steam line is *pressured* and *hot* to the trip throttle valve.

11. Concrete mezzanine, grouting, base plates, do not cure, shrink, and creep evenly. A recheck in 3-5 years is advised. A ten foot high concrete compressor base will cure shrink 60 mils (.060") in the first six years.

12. Where practical, a compromise heat rise for hottest day and coldest day of the year is best.

13. Measurement target designs on casings should consider all types of measurement, e.g., proximity, optics, laser, feelers, and indicators. Targets should be clean, square, magnetic, firm, non-corrosive, and attached without damage to the casing. Target designs should consider minimum personnel on the alignment team—preferably one person. Remember alignment must be covered on a 24 hour per day basis, if necessary.

14. The minimum distance to measure will improve accuracy linearly, i.e., twice the error at 40 feet over 20 feet; or 4 feet vs 4 inches, may increase the error 12:1.

15. Alignment must be simple. Procedures calling for several measures from different stations or use of complicated trig functions and ratios will lead to errors.

16. Maintenance, operating people, and curious visitors, can easily destroy all your data. Areas under study should be roped off. Strong consideration has been given to a shotgun loaded with salt pellets.

17. Corrections should always be made to give zero misalignment. There are hidden errors in any system plus the alignment varies throughout the day and year. It requires the same work to correct 20 mils as 10 mils provided a shim pack still remains.

18. Our present policy calls for 18 inches as a minimum new installation coupling spacer length for compressor drives. Several installations up to 24 inches have been in operation for three years or better. Excessive lengths will affect other considerations such as coupling weight, torque wrap-up and available space.

19. Permanent facilities for continual or routine examination on alignment on large single train compressors need not require any more outlay of money. Recording additions can be added at about $125 per point hardware costs from the proximity probe system. A cost estimate of proximity systems versus optic systems is made in bibliography, Reference No. 2.

20. When an alignment check is scheduled, one or all of the following conditions will occur:

a. Severe rain storms lasting throughout normal equilibrium temperature rise time;

b. Winds up to 15 mph with strong gusts during readout intervals;

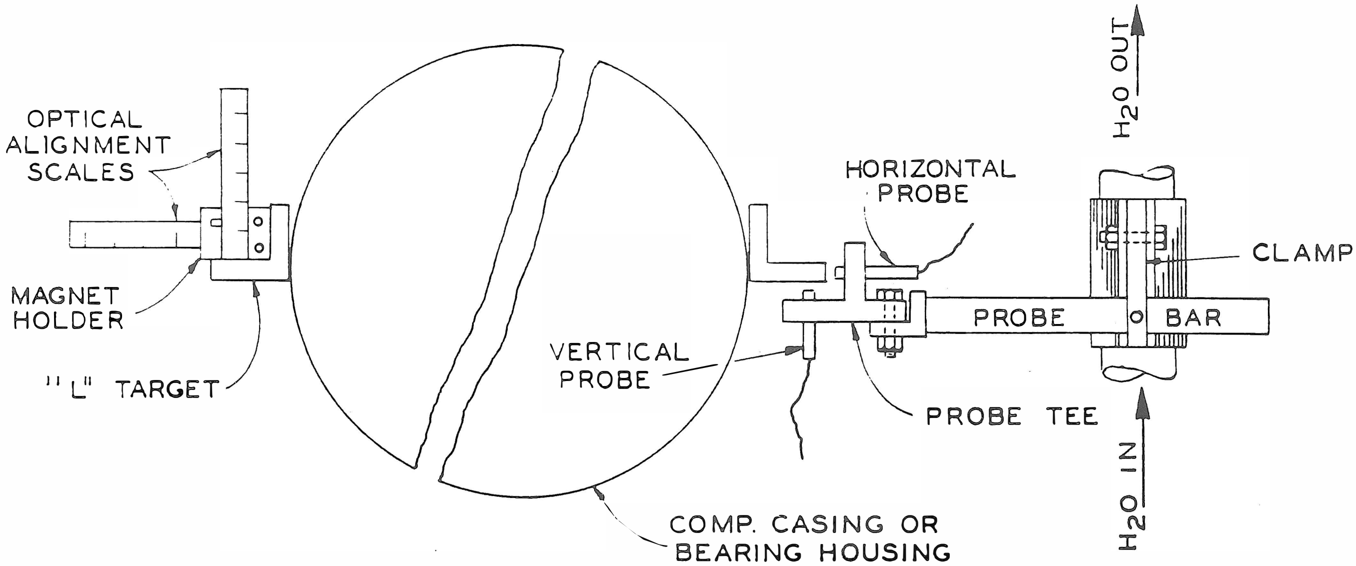


Figure 8. Targets can also be used for optical alignment.

c. Peak movements within the plant of trains, trucks and other mobile equipment. accompanied by jack-hammer operation, pile driving, insulation, overhead welding, flares and steam line blowdowns.

Absolute limits of misalignment are difficult to establish. I have been well impressed with efforts made by the Koppers Coupling Company and several compressors manufacturers, e.g., Dresser-Clark in taking the ever criticized position of establishing limits. These limits for sure should commit a correction, but I have a deep admiration for one trying to sincerely establish good guidelines. Our position cannot be simplified in a unified statement, e.g., asymmetric angular misalignment, in my opinion, is more damaging than parallel offset at the same magnitude.

I feel safe in stating that we attempt to obtain a measured 1/4 mil misalignment per one inch of coupling spacer length, e.g., gear-to-gear, and would probably not make a correction if 1/2 mil per inch were obtained. This means that one would wish to obtain about 4 1/2-5 mils on an 18″ spacer length coupling.

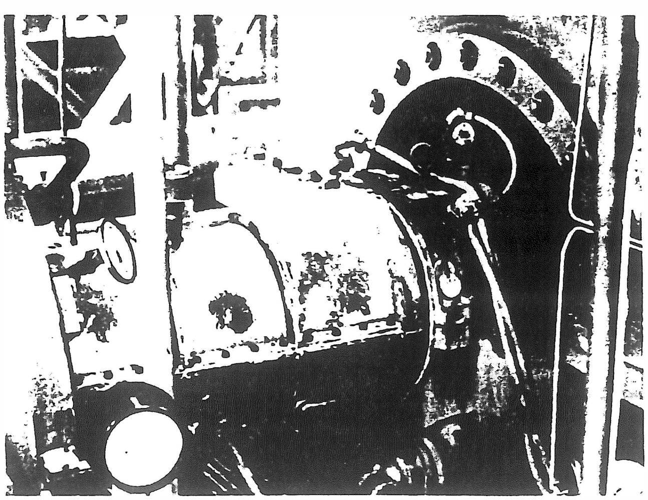


Figure 9. A 24″ spacer length is shown on this 14,000 h.p. drive.

COLD ALIGNMENT

After years of emphasis on good hot alignment, it became apparent that good cold alignment or base reference alignment practice was not consistently performed.

There are many good techniques in this area, but our group prefers "reverse indicator graphical plotting." Since this procedure has been published, a step-by-step example of an alignment will be outlined in this paper.

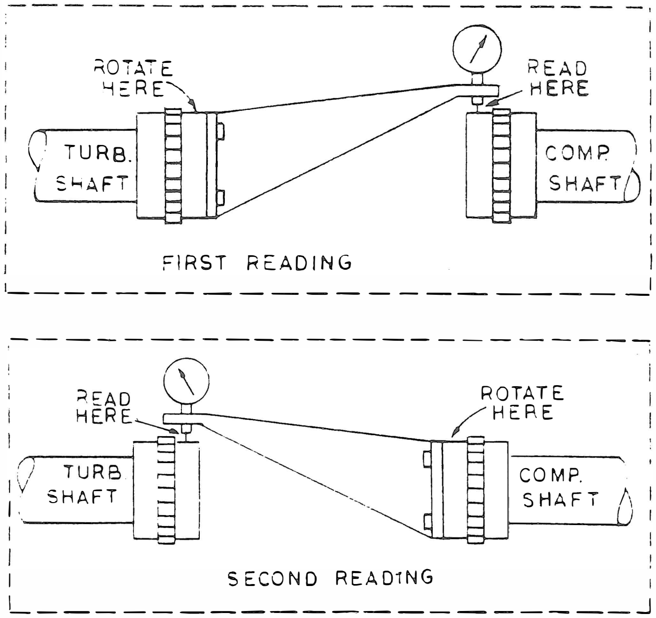
This measure was made in February 1972 as part of 3 trains of steam turbine driven rotary screw compressors. The turbine operated on saturated 200 psig steam exhausting at 75 psig. The compressors were 25″ screw compressors rated at 1200 hp on low mol wt. gas.

Since the thrust bearings on the driver and driven are in the outboard positions, two axial probes were included to determine shaft growths and resulting coupling compression on this installation. It was quickly determined that this torsional damped design flexible disc coupling was compressed to its maximum safe guaranteed limit in axial deflection at 75°F below compressor design discharge temperature. As a result, the turbine was moved outward to affect a near zero axial deflection of the coupling elements at rated load conditions.

Only vertical alignment will be discussed to conserve length, although the horizontal alignment would be performed in a similar manner. One driver and one driven machine will be used in this example. Though multiple cases merely extend the graphics.

The procedures will normally go in this order:

1. Lay out a scaled graph with the desired *hot* operating line in correct position for the shafts involved. (L&N chart paper, Cat. No. 691180 is convenient). The hot operating line is the desired final position of design operating equilibrium conditions.



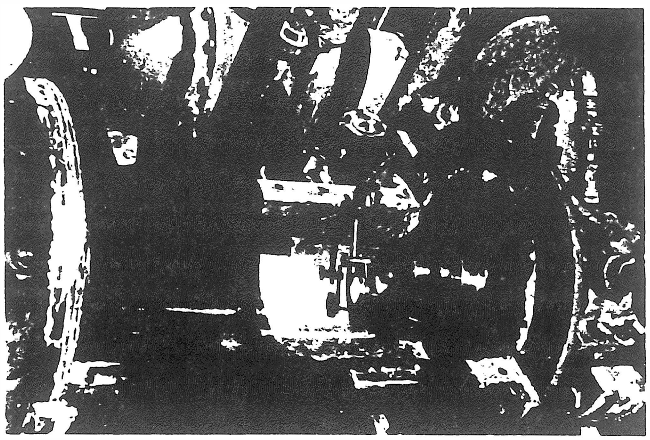


Figure 10. Reverse indicator set up.

The graph is laid out on a 500:1 scale, i.e.. each small division reyresents 0.5 inches in length but 0.001″ (1 mil) in the vertical (transverse) direction. The machines are measured with a tape and laid out on the target (T) end-of-the- machine (EOM) to its first support centerline (Cl) to the other support centerline (Cl) to the probe location (P) to the coupling spacer (C), across the spacer (C), to the probe position (P), support (Cl), support (Cl), and end-of-machine and target (T & EOM). Target or probe positions will vary from job to job. This particular job was a hybrid; i.e., half optics and half proximity technique.

The available information was:

Steam End	Exhaust End	Comp Suction Inboard	Compressor Discharge Outboard
2 mils rise (Monsanto)	11 mils rise (Turbine Vendor)	4 mils rise	6 mils rise

2. The "desired cold position" of the shafts is then plotted based on information calculated, given by the manufacturer, or per past measured rises measured on similar machines. Hot (rise) machines are plotted below the desired hot operating line. (Cold (fall) machines, e.g., refrigeration compressors, are plotted above the hot operating line. Plot position is opposite expected movement.

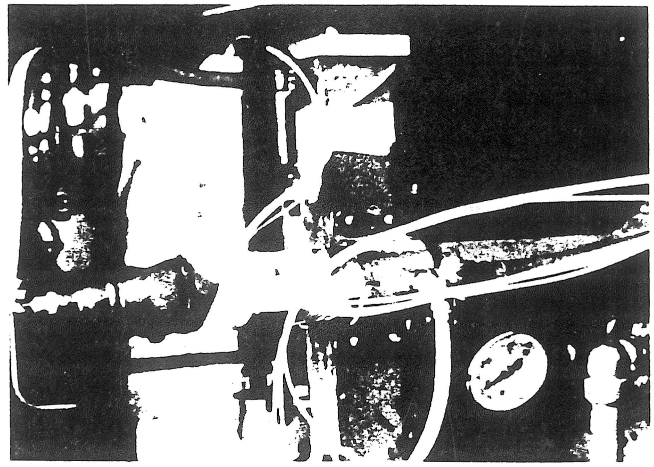


Figure 11. Alignment probes at shafts of turbine driven screw compressor. Water flows through stand and across support.

3. Reverse Indicators Readings required for cold position are given to the field machinists or mill wrights as taken from the map. Recall that a total indicator reading (T.I.R.) will be double centerline offset.

4. The machinist or mill wright will then report back the reverse indicator redings (T.I.R.) taken. This information is plotted on the map as *"Cold Actual"* position.

5. Shim corrections are established from the map and applied to the machine. Vertical corrections with zero horizontal shift must be established before horizontal corrections. Horizontals growth predictions are generally predicted to be zero.

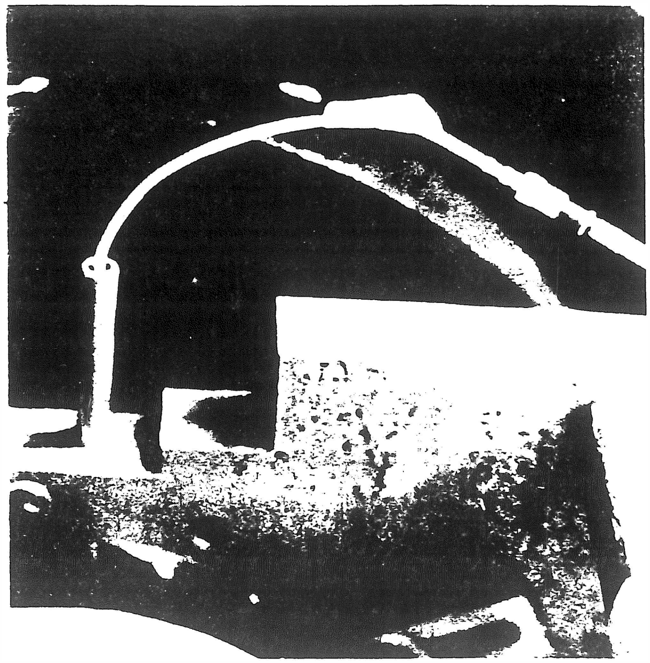
6. Confirmation of reverse indicator reading are made and used for cold (base) actual measured alignment runs. Agreement to cold desired position within 4 mils are often accepted at this point.

7. Turbine and compressor are brought to design conditions (load and temperatures). If this is impossible in pre-start-up, then bring machine to near design temperatures. Equilibrium conditions must be allowed; i.e., a heat sink is initially in progress. Machinery may take 4-24 hours to stabilize. Vacillations in readings or less than 1 mil changes in 1-2 hours is a good termination point provided the original heat direction has been arrested.

8. New value heat rise data is plotted reciprocally from the desired hot operating line.

9. Re-evaluate shim corrections from the chart. Republish shim corrections and reverse indicator readings to the field.

10. Confirm to the field that reverse indicator readings are obtained with all machines bolted down (normal) in a final position. Log information in Step 9 in the file for reference and supervision.



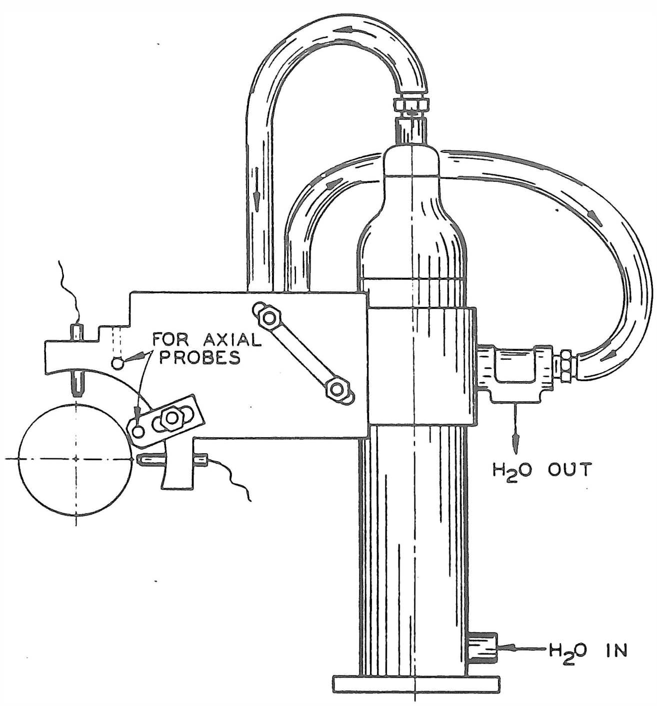


Figure 12. Close up of probe holders. Axial probe was added to read each coupling hub for shaft growth.

This paper has reviewed a method of hot alignment measure and a method of cold alignment measure. You will note that no face readings were required since the angular position is established by graphing. There are many tricks of the trade that cannot be covered in such a paper. Only a few are listed here:

1. Clean areas for shimming from grout chips and other interferences.

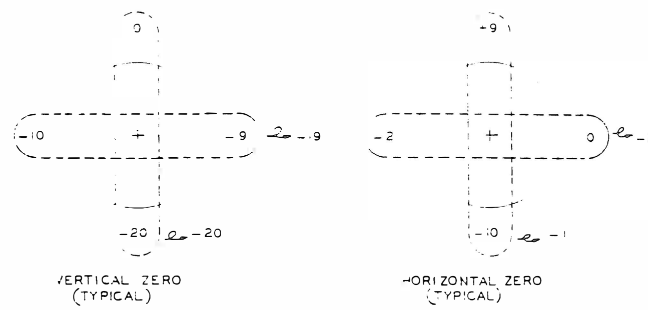


Figure 14. Indicator reading should near sum in each diametrical pair as illustrated.

2. Use full shims. We prefer stainless shims under turbines. Laminated peel-off shims on other equipment will speed up this work.

3. Start with *all* shims at 125 mils or greater.

4. Use long range dial indicators with plus and minus directions confirmed. 200 mil minimum in each direction are suggested. Place indicators at true and smooth surfaces.

5. Follow indicator at all times with a mirror.

6. Turn shafts with indicators in the same direction on all readings, e.g., top, north, bottom, south, top. Clockwise or counter clockwise readings taken facing each machine will only cause mistakes.

7. Indicator holders should be minimum and equal sag types.

8. Indicator readings that do not closely sum arithmatically in opposite pair positions should be taken over. Record a horizontal zero set as a recheck. It will assist the horizontal plots.

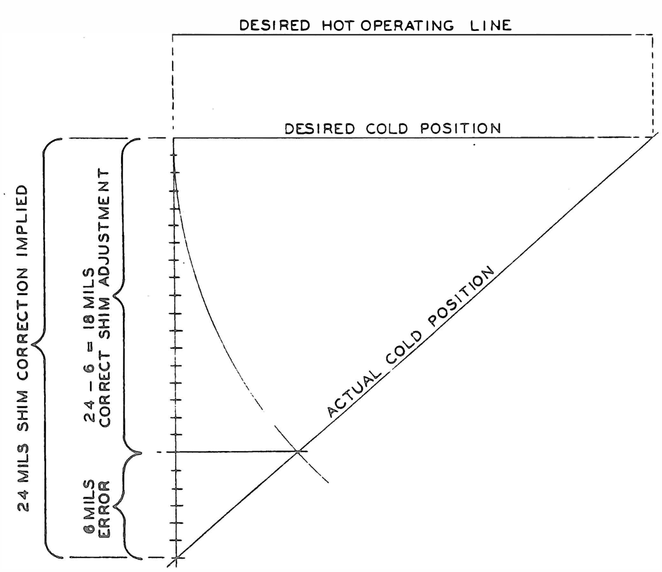
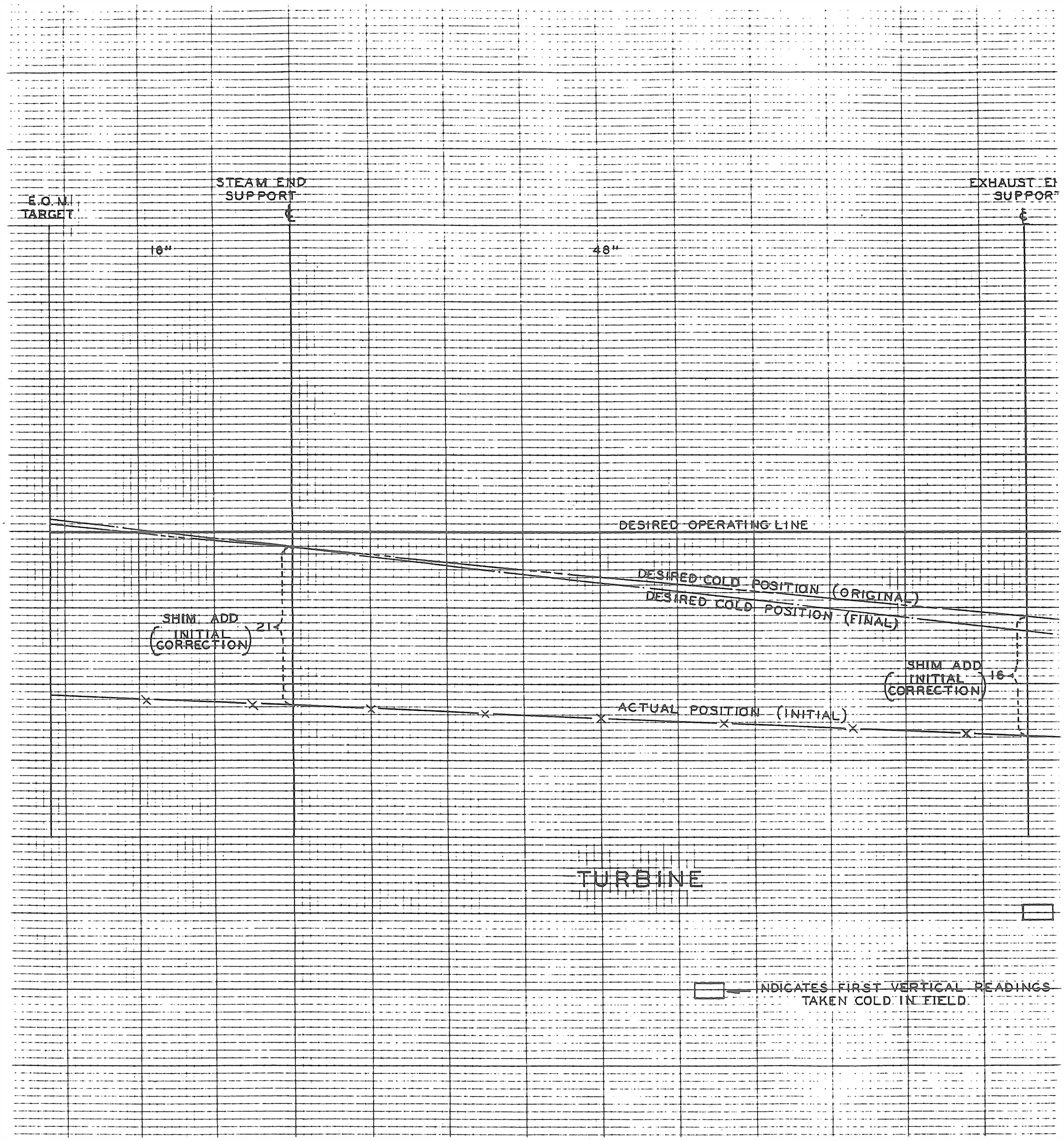


Figure 15. Extremely large shim correction will cause an error which can be compensated as shown.



E.O.M.
TARGET
STEAM END
SUPPORT
EXHAUST E
SUPPOR
16"
48"
DESIRED OPERATING LINE
DESIRED COLD POSITION (ORIGINAL)
DESIRED COLD POSITION (FINAL)
SHIM ADD (INITIAL CORRECTION) 21
SHIM ADD (INITIAL CORRECTION) 16
ACTUAL POSITION (INITIAL)
TURBINE
INDICATES FIRST VERTICAL READINGS TAKEN COLD IN FIELD

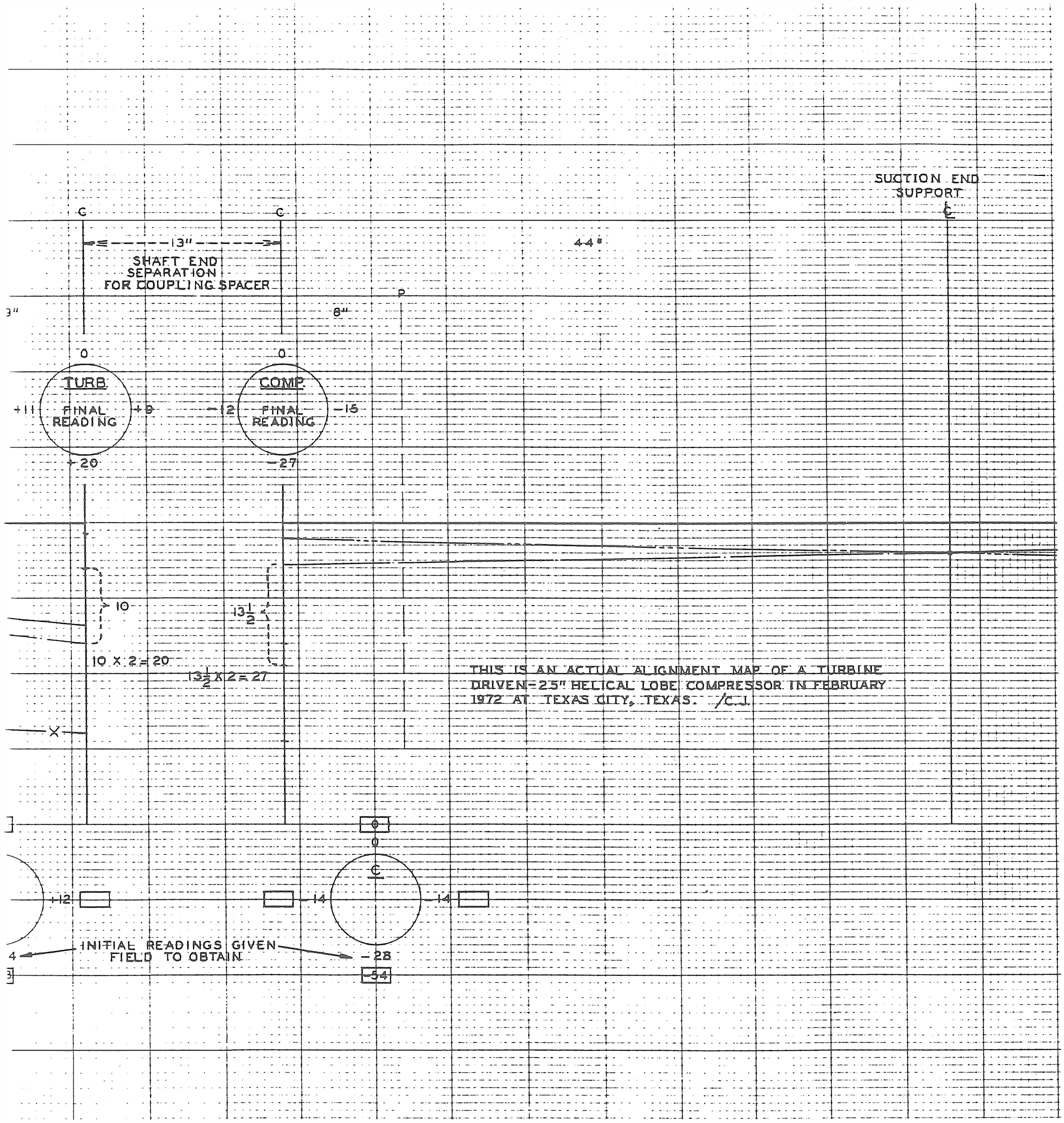
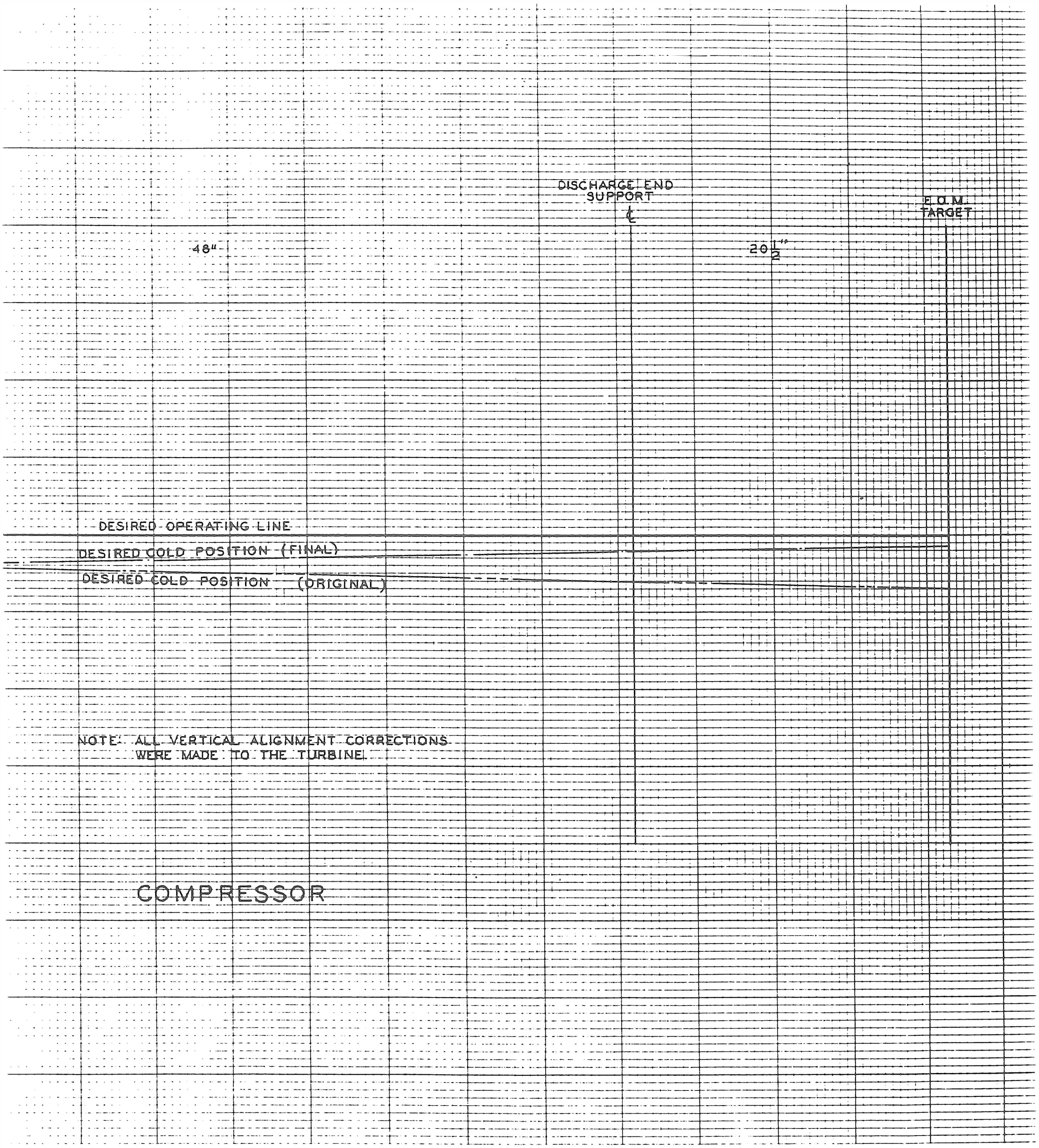


Figure 13. Graphical plot of alignment steps shown in text.


DISCHARGE END
SUPPORT
℄
F.O.M.
TARGET
48"
20 1/2"
DESIRED OPERATING LINE
DESIRED COLD POSITION (FINAL)
DESIRED COLD POSITION (ORIGINAL)
NOTE: ALL VERTICAL ALIGNMENT CORRECTIONS
WERE MADE TO THE TURBINE.
COMPRESSOR

9. Extremely large shimming changes should be corrected for a graphical error as shown. Mr. Howard Blackburn, Monsanto, is credited with this simple correction. The correction recognizes that on paper, only, can one stretch a shaft in length over such a long span. It is for this reason that a 500:1 plot is made by our group rather than a previous 1000:1 scale plot. Multi case trains may justify reversal to the 1000:1 (1″ = 1 Div in length only).

10. Good indicators should be used. They will be special for longer range measure and should, therefore, be boxed and stored for only special (not every day) use.

Other methods are in use and this procedure merely outlines one that our group uses and finds comfortable. Our craftsmen have indicated to us on several occasions that they understand this technique. The graphical plot to them easily reasons with known indicator movements, e.g., if the shaft to be read is lower, then the dial indicator plunger will be pushed *in* and machinists generally know that to be a plus (+) or "right" (CW) reading.

Aside from hot alignment measure, one should consider the time and money saved by accurately choosing shims over "trial and error" methods. Working men will promote a system that saves time and effort, particularly, if it is simple. Engineers often over complicate simple solutions with meticulous and copious details.

The misalignment effects outside of vibration signals, coupling, and shafts are not reviewed in this paper.

REFERENCES

1. Jackson, Chas., "Successful Shaft Hot-Alignment," Hydrocarbon Processing, January 1969, Vol. 48, No. 1 [Ref: ASME Paper 68-PET-25, "Shaft Alignment Using Proximity Transducers," by Chas. Jackson, Sept. 1968.]

2. Jackson, Chas., "How to Align Barrel-Type Centrifugal Compressors," Hydrocarbon Processing, Sept. 1971, Vol. 50, No. 9, Pgs. 189-194.

3. Jackson, Chas., "Vibration Measurement on Turbomachinery," AIChE, CEP Technical Manual, 1972, Vol. 14, Pg. 113.

4. "Optical Alignment—A Maintenance Service to Reduce Your Machinery Downtime," Dresser Machinery Group, Dresser Industries, Bulletin 371-5000 GP 1971.

5. Sohre, J. S., ASME Paper 62-WA-250, "Foundations for High Speed Machinery," Nov. 1962.

6. Dreymala, James, "How To Align Machine Shafts," Power Transmission Design, March, 1972, Pg. 66-74, 88.

ACKNOWLEDGMENT

The author wishes to acknowledge the contributions of Messrs. Carl H. Duhon, Howard Blackburn, and James H. Ingram of the Mechanical Technology Department of Monsanto Polymers & Petrochemicals Company located at Texas City, Texas.