CONSIDERATIONS IN HOT AND COLD
ALIGNMENT AND COUPLINGS

by

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ABSTRACT

Hot and cold alignment techniques are addressed, with graphical plotting of reverse dial indicator alignment maps. An alignment program with a two sets of data for cross checking, and suggested proper tooling and procedures are offered. One case history on steam turbine to barrel pump is presented where the turbine shaft was failed by reverse bending fatigue. The coupling portion of this study deals with the coupling's importance in the rotor effects and defects. Certain balance and assembly solutions and standards are suggested along with the proper fitting of a coupling. A procedure for re-positioning certain couplings to match the coupling's and rotor's residual imbalances. Two case histories are included.

ALIGNMENT

It seems amazing to me that we can pass a laser beam 200 feet, barring heat wave distortion, and hold a one to two mil accuracy. Further, we can attach a laser gun to one shaft and pass a beam (7 to 14 ft) to the other shaft, return it via mirror and have an accuracy within one mil and angularity within one minute. Yet, no one seems to know the true heat rise of a particular machine train. If a figure is tossed out, it seems to be plus or minus 10 mils. Machinery builders issue charts that are not clear whether the cold alignment is to be read in actual offsets or total indicator readings (TIR). That would only be off by 2:1. The data of estimated thermal movements, forwarded with a project, seems the same whether the machinery train is in a heated power plant building, offshore on a open platform, at the Equator, or at Prudhoe Bay.

I would like to define *Cold Alignment* as a "process of misaligning two or more casings in a train, so that the shafts connecting each pair of casings are collinear when a stabilized operating condition has been reached."

I would like to define *Hot Alignment* as a "process of measuring two or more casings in a train to accurately determine total casing movements in at least three planes." Normally, measurements will continue until measurements stop changing, i.e., less than one mil in one hour; or until the measurements begin to vacillate. The minimum timeframe for this stable equilibrium condition of both thermal and other forces, e.g., hydraulic, could be two hours (small/medium duty pumps w/o gears); the maximum timeframe has been 48 hours.

COLD ALIGNMENT DEVICES

There are several cold alignment techniques/devices available to perform alignment between two casing (excluding internal alignment within casings):

• Reverse Indicator Brackets: Two sets of bars extending from one shaft over to the opposite shaft, utilizing dial indicators, to indicate the total swept readings as one rotation, minimum, is completed. The bars should be sufficiently rigid and the reach should be practical, e.g., within three feet. The sag of the bar must be measured and that sag subtracted from the readings taken. From a realistic stand point, it is better to graph those readings and to cross check with a programmable calculator.

• Optical alignment using precision optical equipment in combination with internal targets, external targets, rotating mirrors, etc.

• Laser equipment with or without reflecting mirrors to translate the data points as either a full revolution is made or ¾, ½, or ¼ revolutions are made. These lasers and photoelectric receivers can be mounted to the shafts that are being rotated and require very small axial spacing on the shafts and can be used up to about 30 ft maximum spans.

• Mountings for cradling a standard accurately machined reference rod in a four way rectifier type of instrument readout. This allows shafts to be aligned without rotation of the shafts but requiring slightly over one inch, axial length, of clean diameter shafting.

• Micrometer readings taken between shaft ends on various quadrants. The face and rim method could be included with this type of measurement as they are both considered less accurate methods.

• Less accurate methods need not be discussed.

HOT ALIGNMENT DEVICES

There are several techniques/devices for measuring the thermal movements of machines while these machines reach equilibrium temperatures and forces:

• Eddy current probes mounted on water stands displaying the shaft position in three planes or the casing position in three planes. These stands are generally mounted to separate grouted baseplates or to stable sole plates of machinery. Water, taken from a stable temperature circulating systems, i.e., cooling towers, is used to keep the stands at a fixed reference position.

• Optical alignment equipment using precision levels and transits. This type of equipment is special and utilizes "split reflex bubble" leveling of the transit or level, on each reading, with the "gun" turning on jeweled bearings. Single or "combinations of" instruments-levels and "plunging" transits, bench marks and mirrors, plus "see through" targets allow different readings to be taken at proper time intervals.

• Benchmark gauges, which allow direct measurement in inches and mils to be physically measured between two reference tooling balls which are mounted in pairs at (1) the bearing housing (or casing location) and, (2) at the machine base, generally in the mounting platform, mezzanine, or grouted baseplate. An "inclinometer" plus calibrated distance measuring gauges are used in combination and can be graphed or programed on a calculator or computer.

• Mounted laser instruments attached to one machine and receiving or reflecting laser beams at an adjacent machine.

• Instrumented couplings able to record and transmit data as the articulation of the coupling is forced during transient movements.

• Eddy current probe instrumented reverse bars(cooled Invar) attached at one bearing housing and indicating to the adjacent bearing housing. These bars represent a dynamic version of a "reverse dial indicator bracket set."

• Strobe fired "freezing" of vernier measured variations of one shaft's movements relative to another sensed in the coupling area through special windows in the coupling guards [Murray, 1990].

ALIGNMENT GRAPHING (OR) CALCULATIONS

There are various types of graphs, plotting charts, ratio charts, etc., in order to present two or more shafts in some graphical form of understanding, to allow one to move equipment around correctly. Of equal importance, in my opinion, is conveying some understanding to the machinists or millwrights.

The advantage of a linear scaled graph showing the total machinery scale should be obvious, if not:

• It allows one to make judgments (adjustments) in total machinery move logic to best accommodate large changes, e.g., whereas, the motor was to be moved; it seems best to move the pump. Whereas, the turbine and the pump were to be aligned to the gear (normal), it is apparent that "dropping" the gear box straight down by 20 mils accomplishes the tolerances required.

• There is a normal logic to selecting the machine to be aligned. It is very simple-move the less complicated machine. Examples are:

· between a motor and pump, move the *motor*.

· between a steam turbine and pump, move the *pump*.

· between a steam turbine, gear, and pump; move the *turbine and pump*.

· between a steam turbine and compressor, move the compressor.

· between a gas turbine, gear box, and pump; move the *gear and pump*.

• Often the methodical process of moving down the train via two coupled shafts at a time would render the last body "bolt bound."

• *Regardless which type of hot alignment device is to be used, a cold position chart is necessary in the beginning, in the end, and normally throughout the process.*

• If one is to use a calculator to make the alignment calculations, the calculator program does not know *any* orientation, unless the programmer gives it one. The calculator is generally limited to two machines, although it does not need to be. Two machines will be shown here to illustrate the relationships of the "similar triangle relationship often used."

ALIGNMENT FIGURES AND EXPLANATION

Referring to Figure 1, a typical alignment plot of a pump(or compressor) is seen being aligned to a steam turbine "T." For first illustration, a simple cold alignment of the pump is required to the turbine, i.e., this is not a "normal field situation" that will be shown in Figure 4. Generally, the alignment programs will fix "orientation" looking *over* the stationary machine (turbine) looking *at the moveable* machine (pump/compressor). Further, this means that the "reverse" dial indicator readings for horizontal readings/moves will be defined as "left" or "right," since there seems to be little confusion on "top" and "bottom." In Figure 1, L3 signifies the "left field reading" taken with the dial indicator at the turbine while the "reach bar" is attached to the pump shaft. L4 is the corresponding "left" dial indicator reading with the dial indicator *at the pump shaft*, with the "reach bar" supported to the turbine shaft. The L1 and L2 values, in this example would be *desired readings* and are "zero" for L1 and L2 (refer to alignment program).
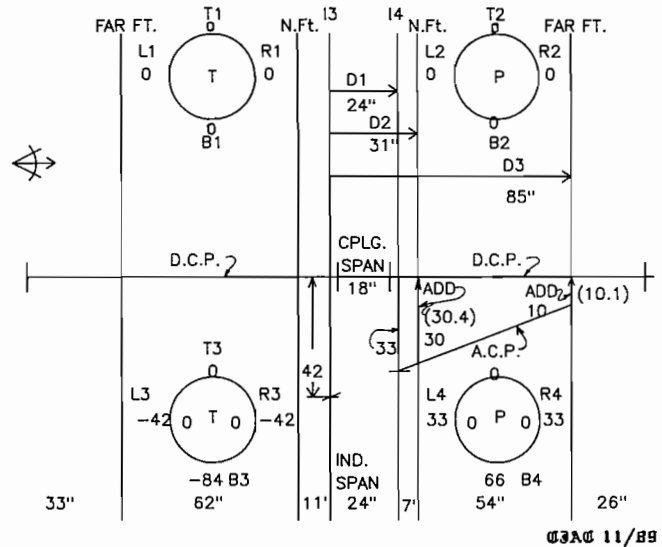

Figure 1. *Typical Turbine Pump Alignment Graphical Plot.*

In looking at Figure 1 and the "sag corrected" alignment data recorded, this condition can be corrected "vertically" by adding a 30 mil shim at the pump's *near* foot and a 10 mil shim under the pump's *far* foot. No horizontal misalignment is indicated as the "left" readings equal the "right" readings.

A large reverse dial indicator bar set is shown in Figure 5 being used on a turbine drive with seven inch diameter coupling hubs, a 16 in diameter coupling and an 18 in spacer. The "Sag Check" for 1 mil sag at 22 in span is shown in Figure 6.

However, to understand the technique and the relationship used, please refer to Figure 2. Now the vertical lines I3 and I4 indicate the reverse dial indicator spans, and the I3 position be-

comes the "common reference plane." The distance D1 is the indicator span(I3 to I4). D2 is the distance, I3 to the NF (near foot) , D3 is the distance I3 to FF (far foot). By enclosing the position of this shaft in the "triangles" shown, one can decide how an alignment program can be configured, i.e., D1/y1 is proportional to D2/y2..D3/y3..D4/y4.. Now, it can be seen that the near foot correction is (B3-B1)-(y2), and the far foot correction is (B3-B1)-(y3). Now, (B3-B1) is the algebraic difference in the "vertical" dial indicator readings divided by 2. And, y2 = (D2/D1)y1; and y3= (D3/D1) y1.
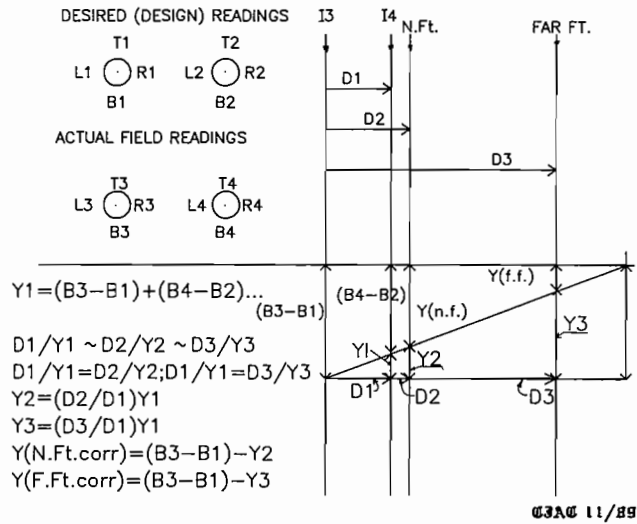

Figure 2. The Logic of Similar Triangles in Reverse Alignment Calculations.

The alignment program included in this paper (APPENDIX A):

• Allows one to enter the dial indicator readings desired, then enter the dial indicators readings from the field, then key in four keys for the correction necessary at the two feet.

• It asks for the distances D1, D2, and D3.

• It asks for the dial indicator readings is succession T1, L1, B1, R1,

• It assumes that the dial indicator readings in the field have been corrected for sag by subtracting out the minus readings based on a sag check of the bars to be used. Example, if the TIR sag of a bar were − 2 mils), then the horizontal readings would modified by adding + 1 mil = − (− 1 mil). The bottom vertical reading would be modified by adding + 2 mils = − (− 2 mils) [Jackson, 1985].

• It does not require a printer, but one can be used. It does require an HP-41 calculator.

• By entering [XEQ] ALPHA "I" ALPHA, a second set of readings can be entered for a "trim," without entering the distances or desired readings.

In Figure 3, one can see what would happen in graphical plotting, if the same vertical "reverse indicator readings" were used as in Figure 1, but the turbine is to be aligned to the pump. A slight shift in the horizontal values on the turbine readings is made to show the resulting swing of the "outboard" end of the turbine, horizontally, to the pump. This two mil difference on centerline for the one indicator, plots out as a 3.2 mil (near foot) and 8.3 mil (far foot) correction (right side move).
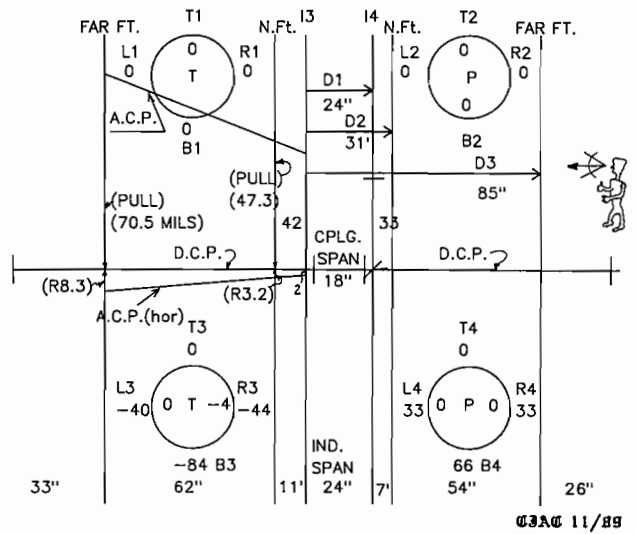

Figure 3. Alignment Plot Using Figure 1 Data but Moving the Turbine.

In Figure 4, the typical "required" (based on hot equilibrium conditions) plot can be seen. It simply says that both the turbine and the pump are going to heat up in service and move to some equilibrium position. That equilibrium condition should leave the turbine and pump as near collinear as can be determined through heat rise measurements. The upper set of numbers then become the "reverse indicator cold reading" to affect a successful hot running train. These readings become the desired cold position "readings." For turbomachinery trains (speeds greater than 3600 rpm) the offsets at each indicator location should be ½ mil (0.0005 in) per inch of axial span between the two indicators. For pumps 3600 rpm and less a 1 mil per inch tolerance can be accepted. The current actual cold position (ACP) would require the near foot of the pump to be raised 20-21 mils and the Far Foot to be raised 40-42 mils. No horizontal moves are required as the turbine is currently three mils to the right of the pump, a condition that could have been desired based on a movement of the turbine "left" three mils due to
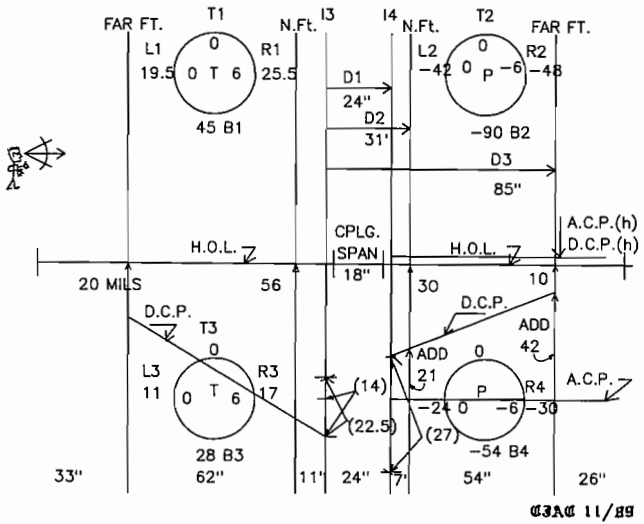

Figure 4. Typical Alignment Plot Based on Thermal Data-Turbine and Pump.

inlet/exhaust lines entering the turbine from the righthand side, for example.

The numbers in (parentheses) were outputted from the program shown in the APPENDIX and are generally more accurate, depending on the plotting accuracy. The field readings taken were already corrected for one mil bar sag (two mils TIR sag). The horizontal readings are corrected by subtracting a (− 1 mil) sag or − (− 1)= + 1. The bottom readings are corrected by subtracting a (− 2 mils) sag or − (− 2)= + 2. The pre-sag corrected readings for L3, B3, R3, L4, B4, R4 would have been + 10, + 26, + 16, − 25, − 56, − 31 respectively (Figure 6).
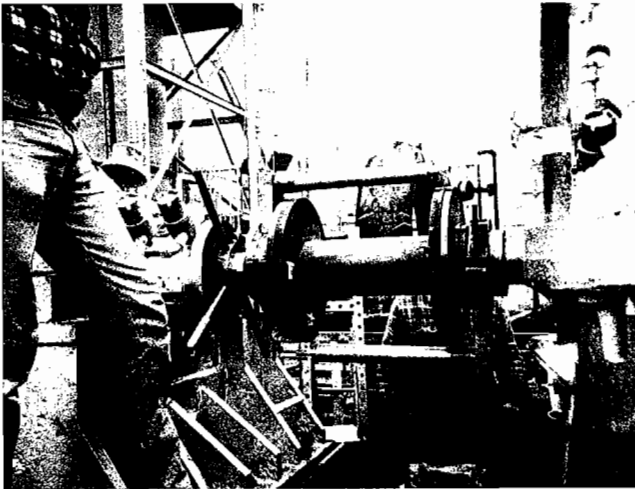

Figure 5. Photo of Reverse Indicator Bar Set for One Mil Sag in 22 Inch Span.
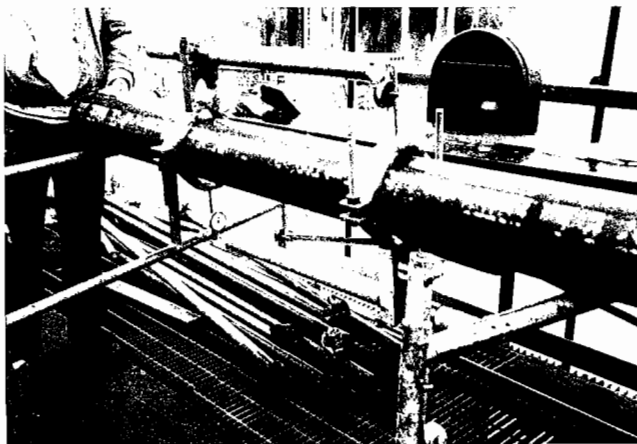

Figure 6. Photo of Sag Check for the Brackets in Figure 5 using Six Inch Pipe.

CASE HISTORY

The case history involves one of ten boiler feed pumps which have steam turbine drives in the 15,000 hp range with the barrel pumps delivering 4,000 psig water. The coupling is a flexible gear type at 18 in spacer length. The steam turbine can use either HP or LP steam. The exhaust bearing is overhung. The turbine has turning gear. The turbine shaft "broke through" near the turning gear. The overall cold misalignment was 87 mils, total. The offset plotted continued over 150 mils in the same,

i.e., "worst" direction. There was resistance to correct the situation since the inlet and outlet lines to the pump are all welded. Continued operation of the stubbed turbine shaft would have resulted in another failure. The vibration level was totally unacceptable, i.e., at certain speeds, over eight mils peak-to-peak.

The shaft orbit plotted by two orthogonal displacement probes represent the locus of the high spot of the shaft. Not only did this turbine drive exhibit strong "jump" phenomenon during the polar plots, but the circular orbits first showed the "deflated beach ball" pattern followed by the "figure eight" pattern, worst condition. This can be seen in Figures 7 (deflated beach ball orbit). The vibration spectrum showed the 2× at 200 percent of the 1× value.

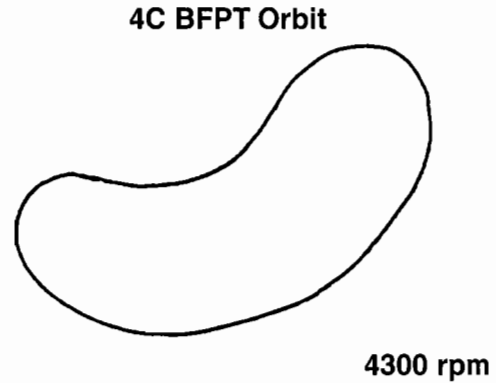**4C BFPT Orbit**

4300 rpm

Figure 7. Orbit of Boiler Feedwater Pump Turbine under Misalignment Preload.

The orbit is shown in Figure 8 of the (eight-bowtie pattern) as the preload increased with speed and load. The heavy shaft preload is in the direction 90 degrees to the long part of the figure eight. One way of looking at the vibration patterns shown in the orbit is to consider the shaft under preload. It cannot move in a circular direction, but must move where the high preload does not exist, i.e., in the direction of the long motion (or) the figure eight.

One must always consider that some fluid gyroscopics come into play, e.g., push down..go right..push up..go left. Further, depending on the direction of load/movement relative to the lo-
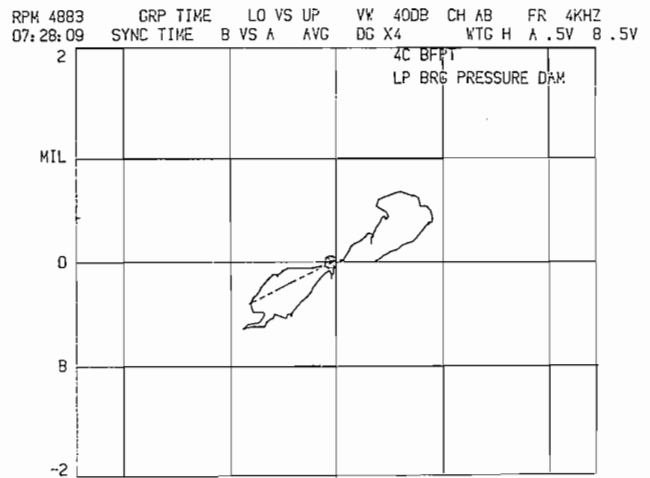

Figure 8. Orbit of Figure 7, but with Severe Misalignment 100-200 mils (Eight Inches).

cation of the vibration sensors, then one can probably deduce that both sensors can show $1 \times$ and $2 \times$ motion (in some cases), and one sensor can see strong $1 \times$ and the other sensor see strong $1 \times$ and even stronger $2 \times$. If one can carry that argument one step further, to where the flexible gear teeth of a coupling become "locked" then another strong component comes in from $1 \times$ unbalance and $1 \times$ and $2 \times$ axial contributions.

The motion of the turbine casing and bearing pedestal was measured with modal instrumentation at several points. This motion responding from the "preload" can be seen in Figure 9, where one would be looking at the turbine vertical motion looking back from the coupling position. The motion of the turbine case and separate overhung bearing pedestal is shown in Figure 10 as viewed from alongside the turbine/coupling looking parallel to the earth.
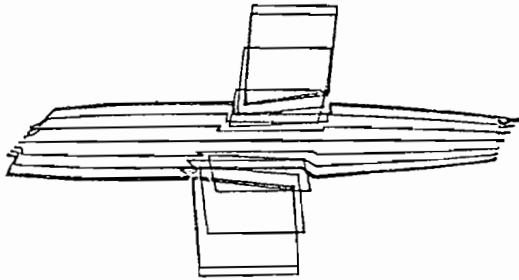

Figure 9. Modal Analysis Motion of Turbine Case/Bearing Pedestal (End View).
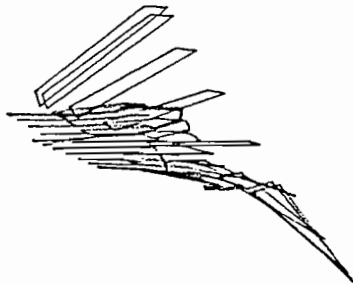

Figure 10. Same as Figure 9, but as Turbine Is Seen from Right-hand Side.

The cold alignment plot taken from laser bars is plotted in Figure 11. The Dodd bars (dyn-align) showed another 150+ mils in that same direction, before running out of Dodd bar sensor range.

NOTE There is a similarity of a "light rub" wherein a figure eight can be formed; BUT the shaft is "bouncing off" the rub contact point *in the direction of the figure eight*. However, in the *heavy rub* argument, first cousin to the strong *preload* cousin, the shaft motion is "restrained" against motion in the preload direction and is forced to move in the unconfined figure eight direction. In *both* cases, the shaft moved in the direction of the figure eight, but the direction of the light rub is near orthogonal to the direction of the heavy preload. One further suggestion is that "shaft relative" proximity probes *can* show less vibration under preloads of misalignment while extreme forces are being applied to the adjacent bearings.

Hint: (Rule 1) Do not move (realign) running machines around using improper vibration measurements as a "good"/ "bad" criteria. (Rule 2) Do not move running machines.
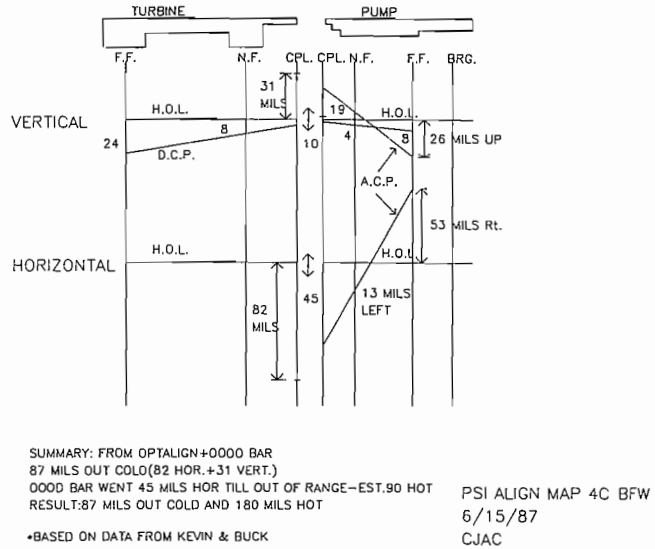

SUMMARY: FROM OPTALIGN+0000 BAR
87 MILS OUT COLD(82 HOR.+31 VERT.)
000D BAR WENT 45 MILS HOR TILL OUT OF RANGE—EST.90 HOT
RESULT:87 MILS OUT COLD AND 180 MILS HOT

•BASED ON DATA FROM KEVIN & BUCK

PSI ALIGN MAP 4C BFW
6/15/87
CJAC

Figure 11. Cold Alignment Graph Based on Laser Alignment Data (87 mil out).

ALIGNMENT GUIDELINES

• Use reverse indicator alignment. Plot and cross check with a calculator for agreement. It only takes a few minutes to plot and check. Align to ½ mil per inch of separation. Slower pumps could be 1 mil per inch of separation. *This tolerance is based on an adequate hot rise to equilibrium measure.*

• Install equipment with adequate spacer lengths for more allowable tolerance, plus, it gives more maintenance freedom.

• When installing in the field, put ¼ in (⅛ in + 1/16 + 1/16 in) full stainless steel shims under all feet of equipment to be aligned. For smaller equipment 3/16 in of total three piece shims may be adequate. Install a straight ground spacer shim under gear boxes—this is not a shim, it is a spacer.

• Provide x, y, z alignment bolts for equipment feet. Develop a desire for sole plates over fabricated baseplates. Make the sole plates an agreed dimension greater that the equipment feet on the three outer perimeters, e.g., six inches.

• Orient piping so that "pipe loops" and anchors give little force on the equipment and basically force pipe expansion into the pipe racks. (Pipe up a Pump—Don't Pump up a Pipe!) Pay attention to steam inlet and exhaust lines in their orientation to the turbine, Zeds (Zs) are worse.

• Make the vertical moves first to get the correct shim amounts under the feet. Keep two horizontal adjustment bolts (stops) *inplace* while installing vertical shims. Use four magnetic bases and long travel dials to keep equipment horizontally in position while vertical shims are adjusted. Remove any "soft feet" in the process of vertical shim adjustments.

• Use the reverse bars to confirm vertical moves while adjustments are in progress. Work out a ratio of lengths, when shimming, e.g., if D2 is ⅓ of D3 then it will take 12 mils move at the far foot to get four mils at the outer (far) indicator.

• Definitely use the reverse bars to "buck in" the horizontal readings, when the vertical shims are in place *before* a set of readings are taken. STEP 1: Zero the bars in the horizontal position. Reverse the bars 180 degrees to the opposite horizontal position; move the near foot to obtain ½ the near indicator reading (or) move the dial to ½ the distance to zero and move the near foot to obtain zero. Hint: The sign of the reading + or − will tell you which direction to move. STEP 2: Do the same for the

far foot to outer dial indicator reading. It is best to start with the one that has the worst (highest) reading but I guess (?) one would have figured that out. STEP 3: Recheck with a zero swing. STEP 4: Take a new complete set.

• Build low sag bars. Use dials with revolution counters, if possible. Indicate bar set to bar set and leave the coupling in place, if possible. Indicate on a flat surface which is perpendicular to the dial indicator stem.

• Get adequate tools before the job, not into the job.

• Use stainless shims. Confirm thicknesses with mikes. File off edges to be free from burrs, etc.

• Consider aligning the equipment on sole plates with greased adjustment bolts before grouting in the sole plates.

• Turn the equipment in the normal direction it turns. Particularly true with gears which must be backed out of mesh at each quadrant reading. Come up on the reading in the same direction. Take consistent readings while viewing from one position, e.g., top, right, bottom, left. Use levels (magnetic base — four way levels, for instance). Follow with mirror. Check sums of algebraic sum of horizontal readings vs algebraic sum of vertical readings. This should hopefully be within two to three mils. Don't use the indicator bars to turn. Turn from opposite end of shaft if possible. Release the torque when readings are taken. Keep the oil circulating in the bearings at constant temperature to assist and prevent damage.

• Attach the bars to give the longest reach for improved accuracy. It doesn't have to be coupling hub to coupling hub..it could be two to six inch longer than the coupling hub, if the shaft allows. Stencil "bar sag" on bars as checked.

• Give craftsmen, leadmen, foremen, and maintenance supervisors and engineers training in alignment. A trainer is shown in Figure 12 with a reverse indicator set that has ¾ mil sag on the eight inch span shown. This trainer allows the fast movement — one half, via dovetails and differential screws for initial training; and the slower method — opposite half, duplicating field conditions by using stainless shims, bolts, leveling bolts, etc.
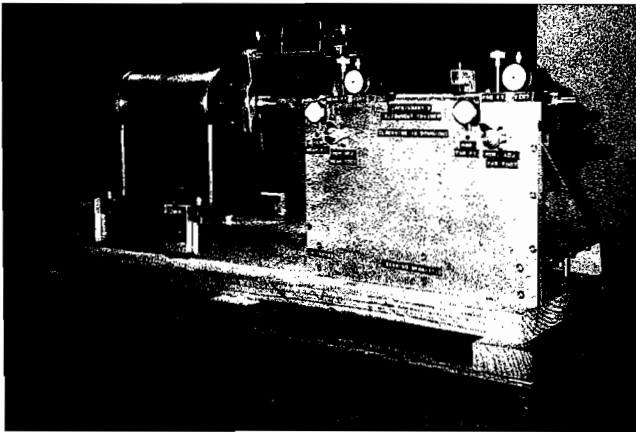

Figure 12. Smaller Reverse Alignment Bar on Alignment Trainer.

APPENDIX FOR ALIGNMENT

In the APPENDIX section, an alignment program is offered. The instructions are shown in Figure 13 for using the program which works on an HP 41 calculator with or without printer. The program itself is reflected in Figure 14. It does not have sag correction inputs nor heat rise inputs. It presumes that has been determined by hot measure, graphed, and sag corrected read-

ings applied. An example is shown in Figure 15 to verify the program and its use. A "trim" alignment or a second set of data can be reentered without changing the original "desired dial reading" or reentered the shaft and feet spans. Simply enter "I." The orientation must be "looking over the *fixed* equipment towards the *moveable* equipment. The horizontal corrected dial indicators (for graphing only) are shown within the circles. HOL stands for hot operating line. DCP (Desired) stands for desired cold position. ACT (actual) are field readings.

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PROCEDURE FOR USING "ALIGN" PROGRAM WRITTEN TO USE HP 41 IN FIELD WITHOUT THE
PRINTER:[NOTE: THE PRINTER CAN BE USED AND WOULD BE TURNED "ON",W/"NORMAL"]
1. TURN HP 41 C,CV,CX,ETC "ON", [XEQ][ALPHA][SIZE][ALPHA][034],..PUT IN [PRGM]
AND PUNCH IN PROGRAM (OR) LOAD CARDS (3) SIDES 1-5, PUT IN [USER] MODE.
2. [XEQ][ALPHA]"ALIGN"[ALPHA].
3. ENTER DATA AS CUED,E.G.D1,COUPLING SPAN(INCHES);D2,DIST. TO NEAR FOOT(NF);
D3, DISTANCE TO FAR FOOT(FF).
4. ENTER DIAL INDICATOR READINGS AS SEEN FROM THE FIXED MACHINE LOOKING
TOWARDS THE "MOVEABLE" MACHINE,E.G. L1(MILS),B1,AND R1,AND L2,B2,R2, FOR THE
"DESIRED" READING. THESE READING SHOULD BE THE DESIGN COLD POSITION WITH HEAT
RISES (OR FALLS) EXPECTED OR FIELD MEASURED.
5. ENTER THE "ACTUAL" DIAL READINGS TAKEN FROM REVERSE INDICATOR METHOD IN THE
SAME VIEWING LOGIC "LOOKING FROM" FIXED" MACHINE..L3,B3,R3; THEN L4,B4,R4.
6. THE PROGRAM WILL INSTRUCT YOU TO KEY A,B,C,D (KEYS) TO OBTAIN THE NEAR FOOT
HORIZONTAL AND VERTICAL MOVES [KEY A] & [KEY B] AND FAR FOOT [KEY C] & [KEY D]
AND THESE KEYS WILL DISPLAY AS LONG AS NEED FOR THE MOVES TO BE RECORDED (OR)
THEY CAN BE RE-KEYED IF ONE WISHES TO RE-CHECK...A PRINT OUT IS MADE BELOW FOR
THE TWO EXAMPLES GRAPHED ON THE PREVIOUS PAGE:(PLOT & ENTER SAG CORRECTED RDS)
DATA INPUT/OUTPUT          EXAMPLE ONE        EXAMPLE TWO
DISTANCES-?
D1<CPLG SPAN>=             10.0 (INCHES)      10.0
D2<NEAR FOOT>=             16.0               16.0
D3<FAR FOOT>=              66.0               66.0
DESIRED RDS.=
T1=0.0
L1=                        12.0               12.0
B1=                        24.0               24.0
R1=                        12.0               12.0
T2=0.0
L2=                        -14.0 (USE CHS FOR -) -14.0
B2=                        -28.0              -28.0
R2=                        -14.0              -14.0
ACTUAL RDS=
T3=0.0
L3=                        34.0               -4.0
B3=                        54.0               -4.0
R3=                        20.0               0.0
T4=0.0
L4=                        -34.5              -2.0
B4=                        -59.0              -8.0
R4=                        -24.5              -6.0
REALIGN:
KEY A, B, C, D
NF HORIZ.,                 <LEFT> 3.8(MILS)   <RIGHT>-2.0(MILS) *
NF VERTICAL                <PULL>-15.8 **     <ADD>7.6 ***
FF HORIZ.                  <RIGHT>-6.2        <RIGHT>-2.0
FF VERTICAL                <PULL>-18.3        <PULL>-12.4
```

NOTE: ON SAME MACHINE,DISTANCES & DESIRED RDNS;FOR NEW IND. RDNS., [KEY "I"].
*SIGNS(-) OR + ARE ONLY A CHECK ON THE SYSTEM-DISREGARD-ALL;(-)=<RIGHT>,I.E.
<RIGHT> -2.0 MEANS "MOVE RIGHT" 2.0 (MILS) IF (MILS) WERE USED IN DIAL RDS.
THE SAME IS TRUE FOR VERTICALS, (-) = <PULL>[A + <ADD>]
**<PULL> MEANS "PULL" SHIMS OR LOWER(DOWN) THE FOOT.
***<ADD> MEANS "ADD" SHIMS OR RAISE(UP) THE FOOT.   CJAC 8/86

Figure 13. Operating Instructions for HP 41 Alignment Program.

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ALIGNMENT PROGRAM- FOR TWO CASINGS- USING REVERSE INDICATOR TECHNIQUES
WRITTEN BY CHARLES JACKSON FOR USE WITH AN HP 41 CALCULATOR W/O PRINTER
Dial Indicator Readings are taken looking over fixed casing.(MILS)
DESIRED readings are inputed and ACTUAL field readings are inputed.(MILS)
Spans D1,D2,D3 are from the fixed machine indicator position.(INCHES)
D1=Coupling Span,D2=Near Foot Span, and D3=Far Foot Span.(NORMAL).
Indicator Readings can be any UNIT;corrections are in same UNITS (MILS).
LEFT and RIGHT movements from same viewing end;PULL means DOWN,ADD means UP.
Program Allows new indicator readings to be inputed by pressing [I].(SIZE 034)
Load cards,Press "USER",Initiate by , XEQ Alpha "ALIGN" Alpha. Program is:
```

01	LBL "ALIGN"	042	LBL I			163 PSE
02	CLA	043	"ACTUAL RDS."	083 -	123 RCL 20	164 RCL 19
03	CLRG	044	AVIEW	084 -	124 -	165 XEQ F
04	FIX 1	045	PSE	085 2	125 STO 22	166 ARCL 19
05	"DISTANCES=?"	046	"T3=0.0"	086 /	126 RCL 21	167 AVIEW
06	AVIEW	047	AVIEW	087 RCL 16	127 RCL 03	168 RTN
07	PSE	048	"L3="	088 +	128 *	169 LBL D
08	"D1<CPLG SPAN>="	049	XEQ E	089 STO 17	129 RCL 01	170 "FF VERTICAL"
09	XEQ E	050	STO 10	090 RCL 02	130 /	171 AVIEW
10	STO 01	051	"B3="	091 *	131 RCL 20	172 PSE
11	"D2<NEAR FOOT>="	052	XEQ E	092 RCL 01	132 -	173 RCL 23
12	XEQ E	053	STO 11	093 /	133 STO 23	174 XEQ G
13	STO 02	054	"R3="	094 RCL 16	134 "REALIGN"	175 ARCL 23
14	" D3<FAR FOOT>="	055	XEQ E	095 -	135 AVIEW	176 AVIEW
15	XEQ E	056	STO 12	096 STO 18	136 "KEY A,B,C,D"	177 ADV
16	STO 03	057	"T4=0.0"	097 RCL 17	137 AVIEW	178 ADV
17	"DESIRED RDS.=?"	058	AVIEW	098 RCL 03	138 RTN	179 ADV
18	AVIEW	059	"L4="	099 *	139 LBL E	180 ADV
19	PSE	060	XEQ E	100 RCL 01	140 PROMPT	181 RTN
20	"T1=0.0"	061	STO 13	101 /	141 RTN	182 LBL F
21	AVIEW	062	"B4="	102 RCL 16	142 LBL A	183 X>0?
22	"L1="	063	XEQ E	103 -	143 "NF HORIZ."	184 "<LEFT>"
23	XEQ E	064	STO 14	104 STO 19	144 AVIEW	185 X<=0?
24	STO 04	065	"R4="	105 RCL 11	145 PSE	186 "<RIGHT>"
25	"B1="	066	XEQ E	106 RCL 05	146 RCL 18	187 RTN
26	XEQ E	067	STO 15	107 -	147 XEQ F	188 LBL G
27	STO 05	068	RCL 12	108 2	148 ARCL 18	189 X>0?
28	"R1="	069	RCL 10	109 /	149 AVIEW	190 "<ADD>"
29	XEQ E	070	-	110 STO 20	150 RTN	191 X<=0?
30	STO 06	071	RCL 06	111 RCL 14	151 LBL B	192 "<PULL>"
31	"T2=0.0"	072	RCL 04	112 RCL 08	152 "NF VERTICAL"	193 RTN
32	AVIEW	073	-	113 -	153 AVIEW	194 END
33	"L2="	074	-	114 2	154 PSE	
34	XEQ E	075	2	115 /	155 RCL 22	
35	STO 07	076	/	116 RCL 20	156 XEQ G	
36	"B2="	077	STO 16	117 +	157 ARCL 22	
37	XEQ E	078	RCL 15	118 STO 21	158 AVIEW	
38	STO 08	079	RCL 13	119 RCL 02	159 RTN	
39	"R2="	080	-	120 *	160 LBL C	
40	XEQ E	081	RCL 09	121 RCL 01	161 "FF HORIZ."	
41	STO 09	082	RCL 07	122 /	162 AVIEW	CJAC 8/86

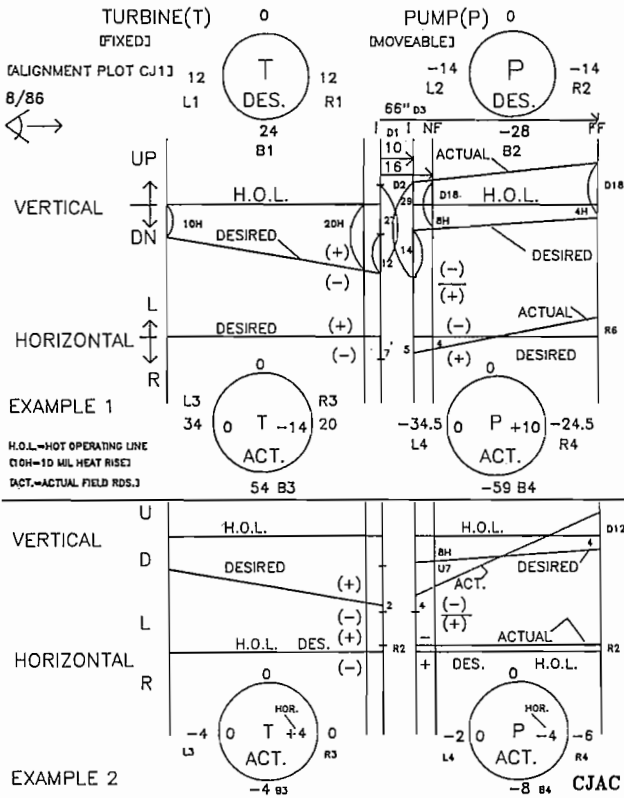Figure 14. HP 41 Alignment Program.

Figure 15. Two Illustrated Examples for Alignment Program.

COUPLINGS

This section is intended to highlight areas to give consideration which basically concentrate on factors other than specific coupling designs.

Flexible couplings are designed to transmit torque and to decouple motion. Some couplings are designed to allow a ½ degree misalignment. This generally means that if one takes the tangent of ½ degree (0.00873), the coupling may stand eight to nine mils of offset per inch of separation before the "gear involutes lock up," the shim packs fatigue, or whatever deleterious effect occurs that the coupling builder is concerned about.

In the above suggested alignment tolerances, it has been suggested that ½ mil per inch of separation be used as an alignment tolerance; thus providing 1/16 of the coupling builders concern level, based on ½ degree of misalignment capability. If the coupling can only stand ¼ degree then the tolerances get tighter, but one could still be safe at⅛ of the coupling builders parameter. Even taking the slower pump, the factor of ¼ to ⅛ is still present; and isn't it a simple tolerance?

Couplings must also provide axial growth capability. In several incidents where a condensing turbine drives through a topping (back pressure) turbine, it was calculated that the topping turbine did, by arrangements of dowels in the exhaust feet of each turbine, calculate to have 255 mils (0.255 in) of axial growth towards the condensing turbine. Fortunately, about ½ this growth occurs whilst the turbine is put on warmup/slow roll cycles. The "prestretch" dimension for this installation was set at 220 mils and that was met within ± 10 mils by actual measurement. This is to say that the cold space between shaft ends (BSE) should be almost a ¼ in longer than the actual free assembled coupling length, therefore affecting a prestretch.

In the present API 671 Coupling Standard, the purchaser has several balance selections. One specification calls for the residual unbalance (in oz-in) to be equal or less than 4W/N (where W is the coupling weight in pounds and N is the speed in rpm) for each individual component, balanced prior to coupling assembly, and the random assembled (no match marks) coupling "check balance" not to exceed 40W/N.

In high speed service, there is an alternate specification that requires the assembled (match marked) coupling to be balanced to 4W/N tolerances. The 4W/N requirement is very tight and equals a grade of 0.67 mm/sec on the ISO Balance Quality.

There is another stipulation that the units be assembled to within ⅓ mil TIR which is difficult to do. The significance of this can be shown in Figure 16, which was prepared by Drosjack for the API 671 Task Force Committee, when Essinger was Task Force Chairman [Drosjack, unpublished]. Hopefully, it will be in the appendix of the next revision.
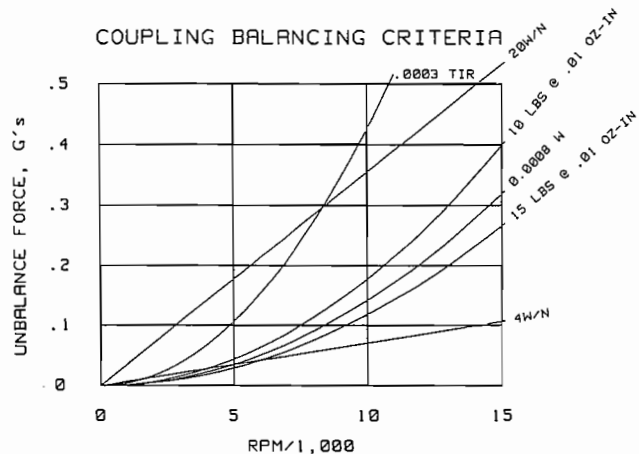

Figure 16. Unbalance Forces Based on Balancing Tolerances [After Drosjack].

The thrust of specifications affecting assembled parts on a rotor are significant. This chart shows that one can balance to 4W/N, but if the assembly is 0.0003 in nonconcentric, then a value greater than 20W/N is affected.

There is also a stipulation that rotors balanced without the contract coupling must use crowned half keys. This is shown in Figure 17, taken from the Schenck 84/85 Balancing News. It is a good practice. One that could be better addressed is the use of stepped keys, so that a good balanced rotor *and* a good balanced coupling do not get molested by an improperly fitted key. Further, a key that is too high in the slot will cause the mass of the coupling to be given a fixed eccentricity on assembly. Impellers and other assemblies have suffered this calamity in the past, and this one will defeat the shaft runout check on assemblies as well as the shaft face readings after fitting.

The significance of the overhung weight on a rotor can be noted on the rotordynamics effects [1] shown in Figure 18. Please note the effects on the second resonant modes. Not that this increased mass moment does not lower the first resonances, but it has strongest impact on the second mode. It is for this reason that clever machinery builders are sensitive about the weight of couplings and the amount of shaft overhang.

TWO CASE HISTORIES

In the first case history, approximately 24 pumps of the API 610 quality were installed at a petrochemical plant, and a "rubber tire type" coupling was used. Bearings were failing on all pumps in less than two weeks. In the final correction, the cou-
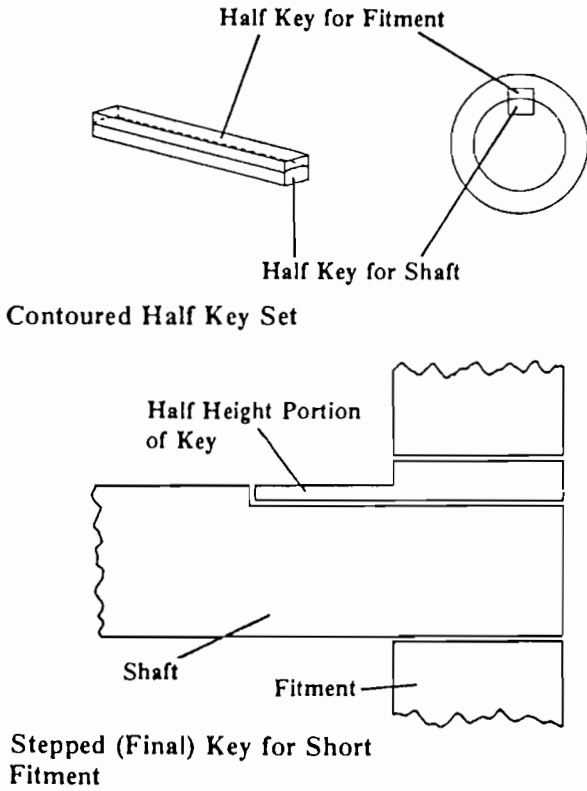
Half Key for Fitment

Half Key for Shaft

Contoured Half Key Set

Half Height Portion of Key

Shaft

Fitment

Stepped (Final) Key for Short Fitment

Figure 17. Illustrations of Contoured (Crowned) Half Keys and Stepped Keys.

FIG. 1 EFFECT OF MID SPAN WEIGHT LOCATION ON CRITICAL SPEED.

MID SPAN WEIGHT		CRITICAL SPEEDS		
		1ST RPM	2ND RPM	RATIO 2ND/1ST
A	72", 6" DIA.	5,516	22,058	4.00
B	280 LBS. 36" 36"	3,921	22,057	5.62
C	140 LBS. 140 LBS. 24" 24" 24"	4,194	16,670	3.98
D	140 LBS. 140 LBS. 12" 48" 12"	4,943	16,677	3.38

OVERHUNG WEIGHT		CRITICAL SPEEDS		
		1ST RPM	2ND RPM	RATIO 2nd/1st
A	72" 6"D.	5,516	22,058	4.00
B	40 LBS. 8"	5,443	21,000	3.86
C	40 LBS. 18"	4,977	13,050	2.62
D	40 LBS. 24"	4,404	10,028	2.27

REFERENCE:FIRM L. WEAVER,AIChE,9/1/71,ROTOR DESIGN & VIB. RESPONSE CJAC

Figure 18. Effect of Overhung Weight (Couplings) on Critical Speeds [1].

plings were replaced with lighter dry shim type couplings which were well engineered eliminating the problem. The first mode shape (4717 cpm) is reflected in Figures 19, 20, and 21 of the pump, with a rigid base and light coupling (Figure 19). Ninety-one percent of the strain energy is going into the shaft and nine percent is going into the bearings (Figure 20). Using the heavier coupling (actually heavier than the impeller) (Figure 21) on this $8 \times 6 \times 13$ pump not only lowers the resonance to running speed but 39 percent of the strain energy must be absorbed by the bearings (55 mm bore).

In the second case history, this drive train used a 500 lb weight, spigot fit, flexible shim pack type coupling at 18 in spacer length in this turbine-to-gear box location. The shop test on this steam turbine was made at ½ mil p/p max. On startup of this train, the vibration exceeded shutdown levels at six mils shaft relative and the turbine shutdown on supervisory instrumentation (refer to Figure 22, 1 of 4 points plotted).
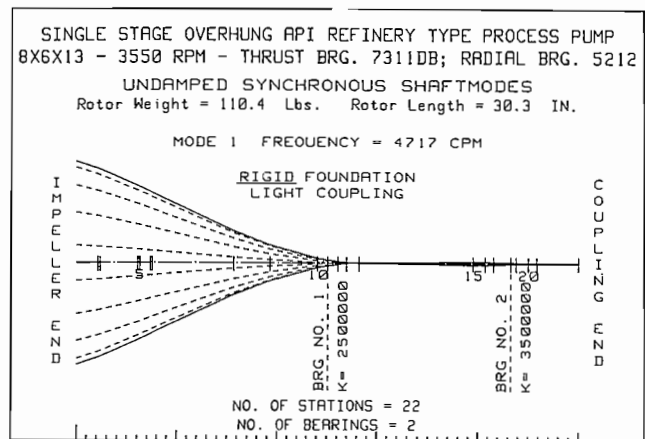

Figure 19. Mode Shape of 8 × 6 × 13 API Pump (Rigid Foundation/Light Coupling).

SYNCHRONOUS CRITICAL SPEED ANALYSIS

NO. (units)	CRITICAL SPEED RPM (HZ)		HMODE LB	IMODE LB/IN^2	KMODE LB/IN	USHAFT DIM. STRAIN ENERGY	UBEARING
1	4717	(79)	57.0	1.04E-01	3.60E+04	91	9
2	53482	(891)	6.0	3.59E-01	4.90E+05	36	64
3	67522	(1125)	9.8	6.42E-01	1.27E+06	42	58
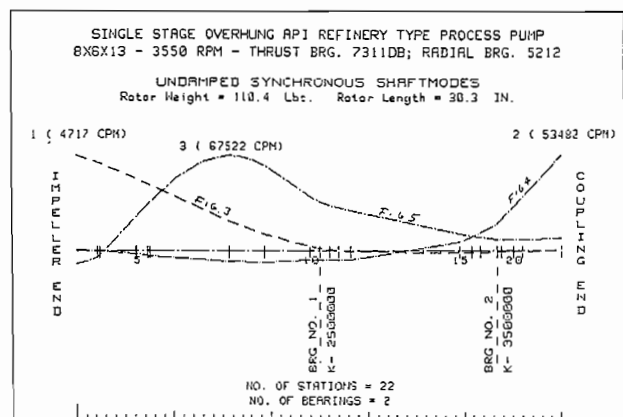
Figure 20. Critical Speed Modes and Strain Energy in Shaft and Bearings (Figure 19).

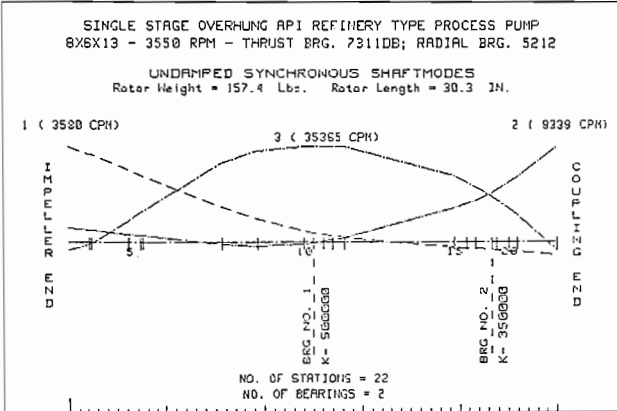| NO. units>	CRITICAL SPEED RPM (HZ)	WMODE LB	ITMODE LB/IN^2	KMODE LB/IN	USHAFT DIM.	UBEARING STRAIN ENERGY
1	3580 (60)	66.6	1.79E-01	2.43E+04	61	39
2	9339 (156)	59.6	3.47E-01	1.48E+05	35	65
3	35365 (589)	22.0	7.01E-01	7.82E+05	25	75

Figure 21. Criticals and Strain Energy with Heavy Coupling.
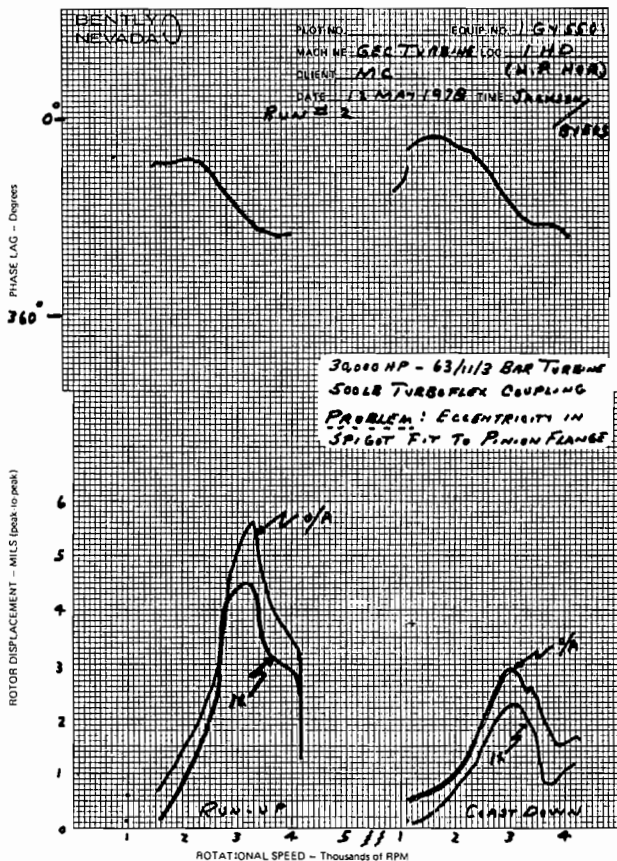
Figure 22. Response of 30,000 HP Drive Turbine to Coupling Eccentricity.

The gear pinion had been balanced and shop tested. The coupling had not only been well balanced, but had been indexed 180 degrees out at each component based on the high arbor center marks (highest eccentricity marks).

By placing dial indicators on each of the six primary components of this coupling assembly (shaft end attachments included), the dials agreed well in all positions except number five to number six, which was the coupling to pinion bolted spigot assembly (10 bolts). In Figure 23, the dial indications can be seen. The number five and number six position disagreed by 180 degrees, and about two to three mils eccentricity, which was found in the pinion spigot fit machining.
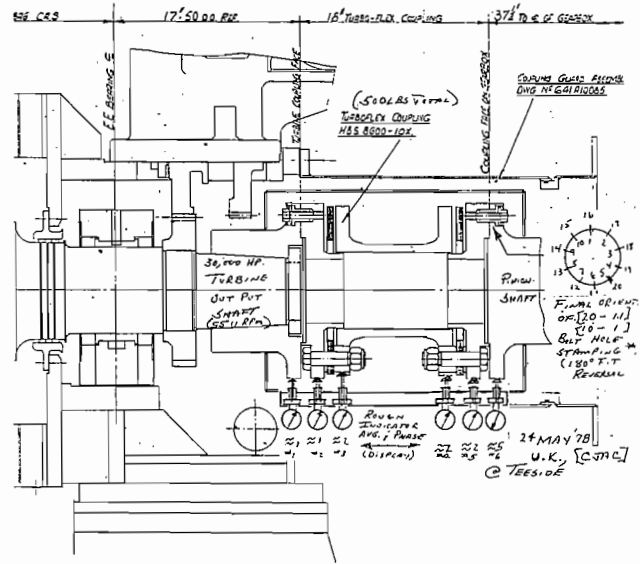
Figure 23. Dial Indicator Checks for Runout in Final Assembly to Pinion.

The bolts at this attachment were reversed by 180 degrees and the unit came up smoothly under full 30,000 hp load, as can be seen in Figure 24, with one mil unfiltered (¾ mil filtered) at the first resonance and running under ½ mil p/p filtered at full speed of 5511 rpm.

COMPENSATING A ROTOR'S RESIDUAL UNBALANCE USING THE RESIDUAL OF THE COUPLING

Given the ability to rotate a coupling's circumferential position on a rotor will allow one to match the coupling's residual unbalance location to compensate for the residual in the rotor. Couplings attached by hydraulic fit, marine type bolting, splines, etc., will lend themselves to this technique. Further, it can be used for any rotor assembled component; however, the scenario is more common with couplings. Take the example where say a turbine rotor assembly is balanced to one mil on test. The coupling is also balanced to one mil on test. Now, the coupling is assembled onto the rotor and the rotor now exhibits 2.25 mil vibration, which puts all the players into "Fingerpointing 104." By rotating the coupling exactly 180 degrees on the shaft, a second vector can be plotted to allow one to determine exactly how much to rotate the coupling for best solution; further, the new amount of vibration can be predicted.

This procedure was presented to the Vibration Institute at Nassau Bay [Winkler, 1983]. With permission, I presented a followup article on this procedure, with "proof plots," an HP 41 CPLGBAL program, with examples and a live demonstration [Jackson, 1984]. St. Elmo Smith successfully used the CPLGBAL program to obtain client (owner) acceptance on six
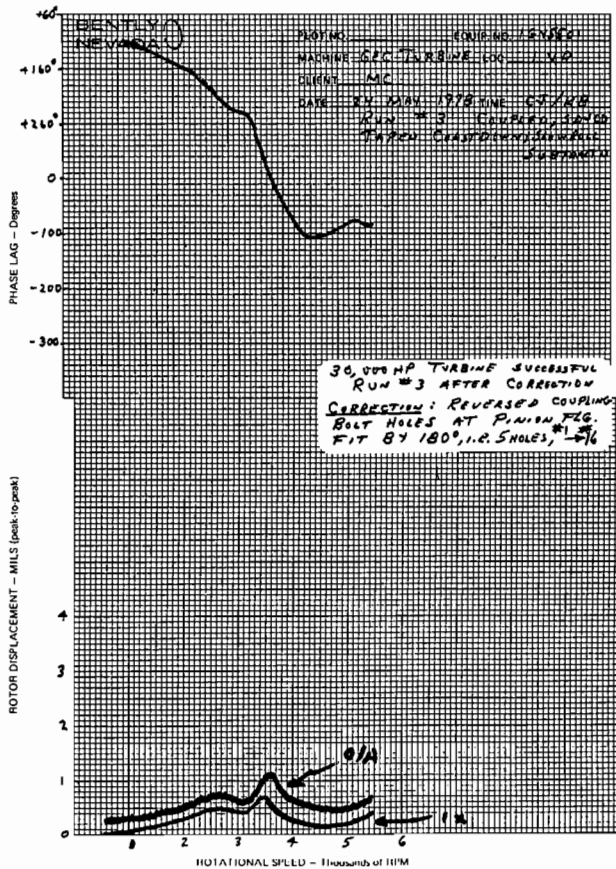
Figure 24. Vibration Response After Reversing 180 Degrees at Pinion Bolts.

Figure 25. CPLGBAL-Graphical Plot of Two Coupling Position 180 Degrees Apart with Shift Angle Determination for Best Balance.

gas turbine gear multistage injection pumps the following year. Shell has used the same program to rotate a different rotor assembly with the same predicted success. This procedure is outlined in Figures 25 and 26. The graphical plot is shown in Figure 25 of the two 180 degree opposite vectors. By connecting the heads of those two vectors with a common "couple" vector, the center of a circle representing the locus of *all* possible solutions is created. By rotating the couple vector from the last position (shift angle) so as to pass that vector through the center of the vector diagram (zero), a "best" solution can be achieved. Further, the difference (offset from zero) should be the resulting new vibration value.

The values via the HP calculator are shown in Figure 26. These data were taken from an actual 25 pound rotor model with a clamped release rotating element.

COUPLING GUIDELINES

• Review the coupling selected on all jobs. Determine if the horsepower and speed rating is satisfactory. Determine if the weight offers any adverse rotor problems. If the coupling is selected for proper torsional stiffness, assure that the lateral resonances have sufficient separation margin from the operating speeds and first orders of operating speeds.

• Determine that the operation of the elements will not be harmful to move say a drive motor off its magnetic center [Calistrat, 1989].

• Determine if an axial growth must be allowed for and how the coupling will accomplish this? Perhaps a "prestretch" should
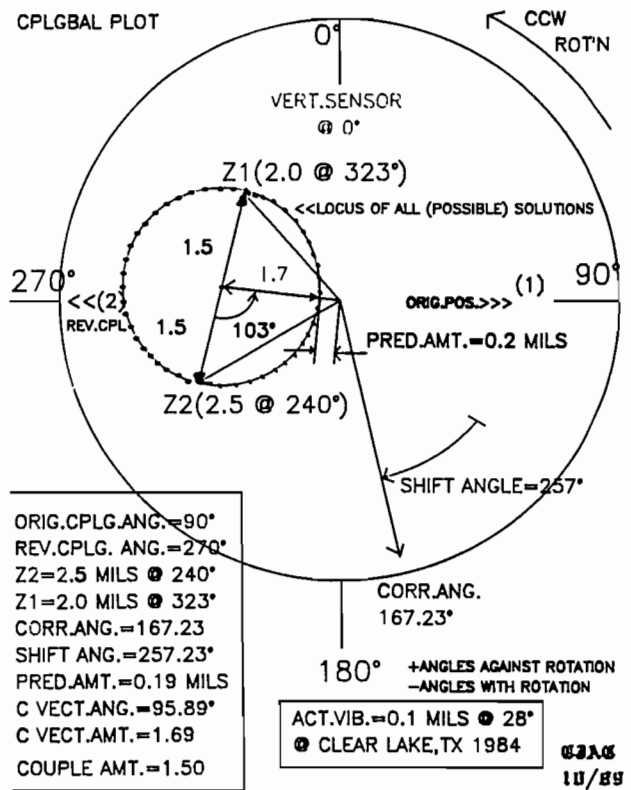
WORK/FILE SHEET
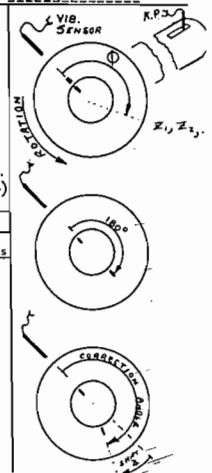## CPLGBAL —APPL. NOTE 84-01

Figure 26. CPLGBAL-HP 41 Calculated Values to Match the Graphical Vector Plot Shown in Figure 25.

have been called for in design by the agency with train responsibility.

• Determine which balance tolerance is acceptable, and how this is provided and maintained during normal maintenance.

• Determine how the assembly is to be registered (concentrically centered). How will this be checked after shaft mounting?

• Procedures for installing the coupling to assure proper interference fit must be developed. For example, I use one mil/in interference for keyed fits of couplings. That must be verified by the coupling manufacturer, as he may not have made allowance for that interference. If less than one mil/in is to be used, it would be good to determine if the interference is sufficient for any unseating of the fit at speed. Also, the keyway, by its pure nature under compression will relax the fit. The hydraulic fit may be two mils/in of diameter, or nominally 60,000 psi shrink pressure, as modified by the shaft and hub diameters. The shaft and hub each share this shrink pressure (total strain).

• Determine whether the coupling can be randomly assembled or do "match marks" certify that the proper assembly is obtained. Do the bolts need a specific strength and are they individually weight balanced? Can the bolts and nuts be reassembled repeatedly without replacement? For example, some bolt/nut assemblies can be made up for possibly ten cycles before the nuts lose their selflocking feature.

• Is the key to be moment weigh matched for the keyways provided? Would a stepped key be a best choice against unbalance in the coupling? Many couplings have the keyways cut after balancing the coupling. Is the keyway radiused to reduce stress raisers? Is the key fitted and does it have radiuses to not interfere with the keyway? Is the key proper width and height? There is an AGMA standard that can be applied here.

• Are the couplings elements suitable for the environment surrounding the coupling, e.g., is stress corrosion cracking, hydrogen embrittlement, corrosion fatigue cracking possible? HINT: Sulphides and chlorides are bad actors. Elastomeric elements are not free from this review.

• How will the coupling be aligned? What will be the heat rise allowed? Does the manufacturer know the heat rise (fall)? Have the piping stresses been reviewed? Using API 610, allowable forces and moments does not free one from concern on pumps, since those values have been proven to move a shaft 10 to 20 mil by actual tests using heavy pumps and baseplate designs.

• Can the coupling be removed in service? Can the seal be removed with the spacer provided? Is there any real problem in providing a ten inch spacer rather than a five inch spacer? Can the coupling be compressed in place to drop out the spacer without special tools? Hydraulic dilation requires one or two pumps; who is providing that? How much is the draw (advance)? What is the taper on the shaft? How is that taper being maintained? Are plug and ring gauges being supplied? Are they made to a standard? Does one use a gauge to lap a full taper? (NO!) Are lapping tools made separately and compared to a standard? (YES!) If I lap a shaft taper and a hub bore, do I change the required spacer length? (YES!) How do I maintain the proper spacer length without moving the equipment? (Special shims?) Do I have them? (Probably not). If I have to move the equipment, axially, will I be bolt bound? (Maybe.) If I move the pump or motor axially, do I need to recheck alignment? (YES!)

• Most motor drives require some form of "limited end float" during startup. How is that provided? Is it the correct amount? How will it be maintained?
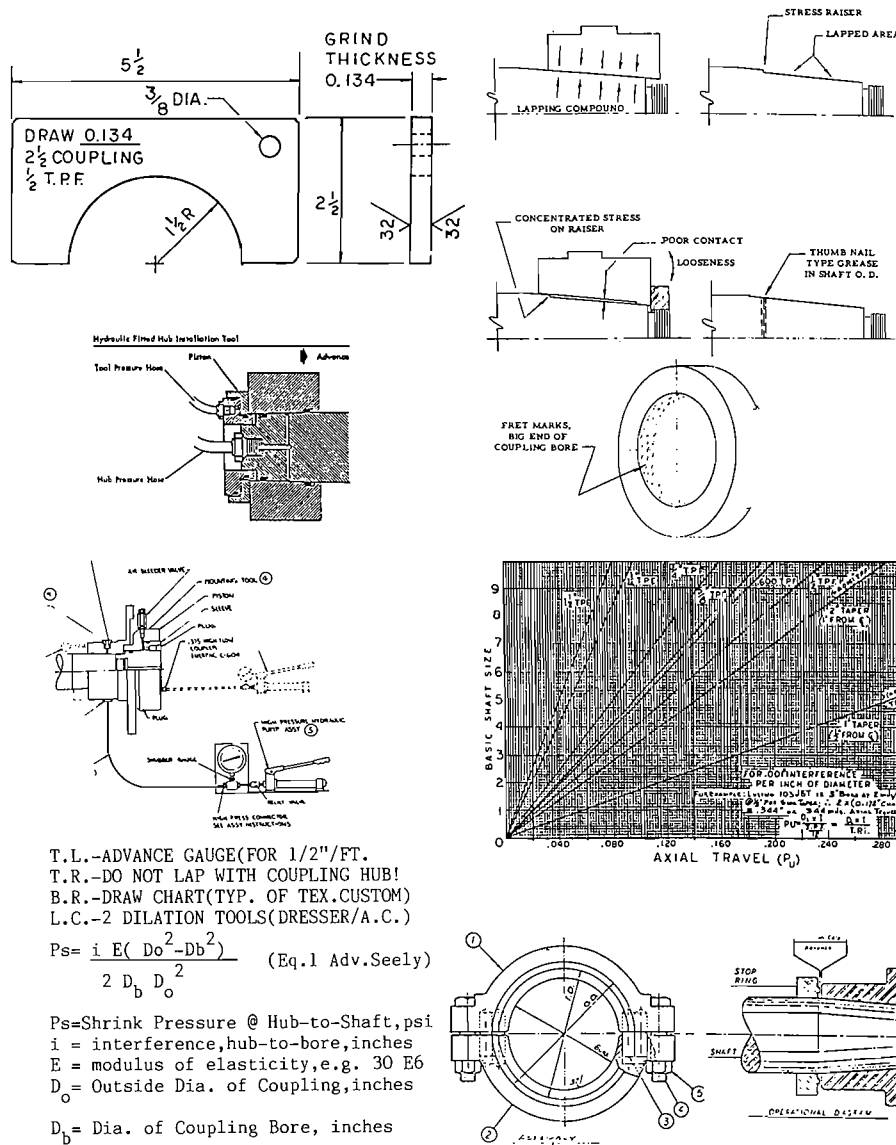
• Does the coupling require lubricant? Grease or oil? What type? How much? How is it provided? Packed or injected? How much oil must be injected?

APPENDIX 2

An APPENDIX has been added to suggest a method of using hydraulic dilation fitting of couplings to shaft. A specific interference is being used as an example (APPENDIX 2, Figure A). An interference chart is shown plus a formula for determining the fit given a set of conditions (APPENDIX 2, Figure B).

(1)

(2)

(3)

(4)
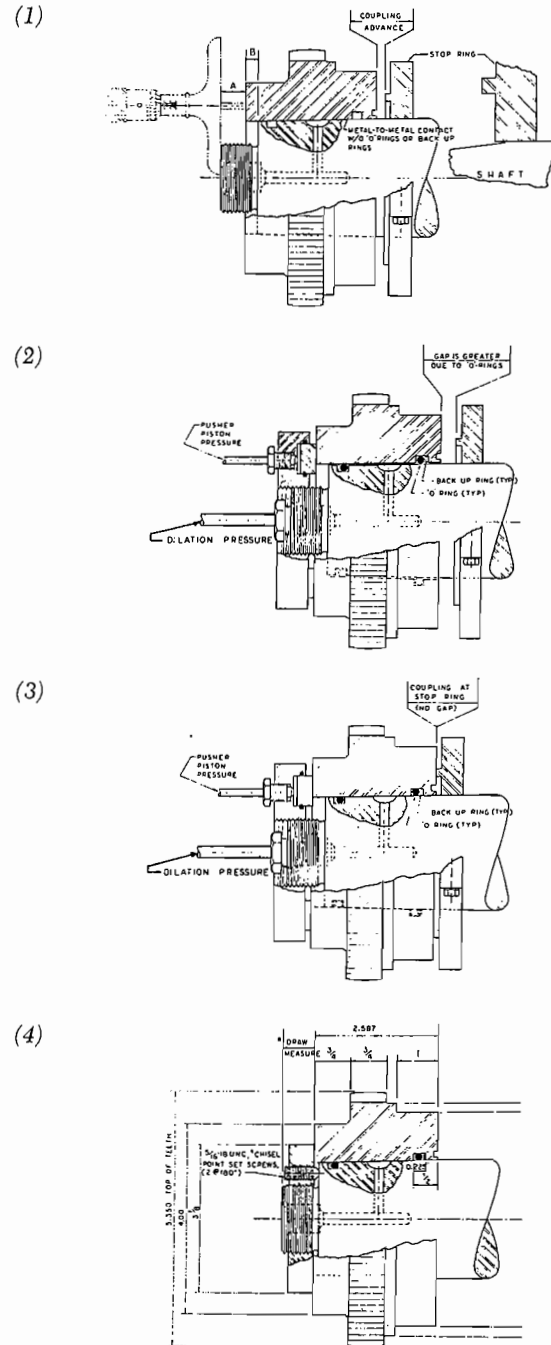
APPENDIX 2—Figure A. (1) Remove "O" Rings, Engage Hub to Shaft, Gage and Set Stop Ring for Advance; (2) Reinstall "O" Rings and Back-Up Rings, Re-engage Hub, Bleed and Connect Pumps; (3) Keep Pusher at 800-1000 PSI, Dilate With Bore Pressure, Hold @ Stop R; (4) Hold Pusher Pressure, Remove Dilation Pressure, e.g. 40,000 psi, Remove Pusher Pressure, Install Retention Nut (if any), Recheck/Remove Stop R.

T.L.-ADVANCE GAUGE(FOR 1/2"/FT.
T.R.-DO NOT LAP WITH COUPLING HUB!
B.R.-DRAW CHART(TYP. OF TEX.CUSTOM)
L.C.-2 DILATION TOOLS(DRESSER/A.C.)

$$Ps= \frac{i \; E(\; Do^2-Db^2)}{2 \; D_b \; D_o^2} \qquad \text{(Eq.1 Adv.Seely)}$$

Ps=Shrink Pressure @ Hub-to-Shaft,psi
i = interference,hub-to-bore,inches
E = modulus of elasticity,e.g. 30 E6
D_o= Outside Dia. of Coupling,inches

D_b= Dia. of Coupling Bore, inches

APPENDIX 2—Figure B. Interference Chart.

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