# A NEW SHAFT ALIGNMENT TECHNIQUE

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## ABSTRACT

A new technique for shaft alignment is easy to apply for verification of alignment and can save users $10,000 to $20,000 in equipment costs. It is called the Reverse-Face Alignment technique. Examples are given with photographs for different styles of couplings such as gear, elastomeric and flex-pack types. Alignment acceptability is determined immediately without complex geometrical calculations as necessary with other methods. The alignment measurement equipment not only is inexpensive, but can be installed quickly for *quasi-hot* alignment checks. Remote readout and computer connection are possible, but these complications are not perceived as being a benefit in most cases. Pitfalls are discussed for this and other methods for comparison purposes.

## 1. INTRODUCTION

Shaft alignment can be measured and corrected in many ways. The goal is to cost-effectively and efficiently get a machine aligned and back running. The capital cost of the tools should be weighed against the total time required to do the alignment. Much of the time required to do an alignment is taken up by tasks not directly related to the measurement of misalignment. Therefore, acceleration of a portion of the alignment task through large capital expenditure may be marginally beneficial.

In some cases, a tool to quickly check the state of alignment is all that is required. What is the alignment and is the alignment within guideline? In other cases, the alignment may be expected (and found) to be good, but for trending purposes it is desirable to document the current alignment on an ongoing basis. Slow changes in alignment can indicate changes in foundations that require correction in the long term.

These days, the trend in some quarters seems to be to think that using a laser alignment system is the only way to do alignment. This paper is written by a ""laser alignment" iconoclast. There are many reasons why dial indicators are a viable option to measure alignment. The capital cost outlay for a laser system versus a set of dial indicators and magnetic bases is the obvious first reason for looking at efficient alignment methods that do not involve laser systems. In the author's experience, the Reverse-Face Method can allow faster attachment and faster data collection versus using a laser system. The format of the data from the Reverse-Face Method is immediately comparable with a guideline so that no complicated calculations are required to determine if the dial readings are indicative of acceptable alignment or not. Compared to the Reverse Dial Method, the installation of the dials is easier and faster for the Reverse-Face Method, and the interpretation of the results is easier.

The state of shaft alignment is traditionally described by including a measurement of parallel offset between the shafts. However, the angularity at flex-planes instead of parallel and angular offset of the shafts to define alignment quality is a fundamental point in this paper.

## 2. ALIGNMENT METHOD SELECTION ISSUES

The following list contains some of the issues that help to determine what will be the alignment method of choice:

- the time required to prepare to do the alignment check
- the time required to actually do the alignment check
- the time required to change the alignment
- the time to re-check the alignment after correction
- the cost of hardware and labour
- the resulting payout

## 3. METHODS DISCUSSED

The methods that will be referred to in this paper are:

- Reverse-Face
- Reverse-Rim (more familiarly called Reverse Dial Method)
- Rim-Face, and
- Laser

A good reference in the discussion of alignment is a book by John Piotrowski called "Shaft Alignment Handbook" (Ref. 1). Piotrowski's allowable misalignment guidelines will be discussed with this author" interpretation of how to apply the guidelines. Comments by Piotrowski, combined with a reference from another paper, about gear couplings are included. There is no mention of the Reverse-Face Method in Piotrowski's book.

## 4. THE GOAL OF ALIGNMENT

The goal of the alignment process is to make the angularity at each flex plane of the coupling sufficiently small with the machine in operation. This statement assumes a spool piece coupling. If the coupling has a single flex-plane, then the offset between the centerlines of the shafts at the flex-plane must be made sufficiently small, as well as the angularity. Note the use of "sufficiently small" as opposed to "minimized". In the case of gear couplings or u-joints, it is not desirable to eliminate angularity totally, as discussed below.

## 5. DESCRIPTION OF THE REVERSE-FACE METHOD

The dials indicators should be mounted in pairs, 180 degrees apart, at each power transmission point or flex-plane. The reason for this pairing is to compensate for axial float that can, and usually does, occur during the rotation of the shafts. (There are those who use one face dial and attempt to force the shafts into the same axial position for each reading. I do not recommend this approach.)

Refer to Photograph 1 for an example of how to mount a dial in the face direction across a flex-plane.

The dials should be labelled distinctively (eg: dial A1 and dial B1, dial A2 and dial B2). Start with dials A at 12:00 o'clock and B at 6:00. Record the readings in pairs for dials A at 12:00, 3:00, 6:00, 9:00 and 12:00. Convert the A&B readings into float-compensated A readings (eg: A*1 and A*2) by the formula [(A-B)/2].

There are two possible sign conventions for the dial readings. If the dial indicator base is mounted on one side of the flex-plane and the dial is pointing away from the magnetic base as it touches the other side of the flex-plane, then a positive change in the dial reading indicates the coupling halves are closing. This is the standard mounting convention. On the other hand, if the dial is turned around to point back toward the base as it touches the other side of the flex-plane, then the sign convention is reversed: a positive change in the dial reading indicates the coupling halves are opening. This is a potential source of confusion for the uninitiated. Photograph 1 shows a dial indicator in the standard orientation. (Dial indicators show a positive reading when the plunger is pushed in.)

The diameter of the circle described by the dials as the shaft is rotated is the basic dimension required for calculating flex-plane angularity. The lengths between flex-planes and the distances from flex-planes to feet on the machine-to-be-moved, are also required before alignment corrections can be calculated.

Turning the shaft to the usual 4 positions of the clock can be done by eye (often there are bolt patterns that help), or an inclinometer can be attached to the shaft for more precise guidance as to shaft angular position.

Data quality is checked by comparing the sum of the vertical dial readings with the sum of the horizontal readings. Ideally, the sums should be equal. In practice, small differences will be seen. Repeat the collection procedure if the differences are "large". Large is to be considered relative to the dial readings. If the misalignment is large, then a larger difference between horizontal and vertical dial sums can be accepted. The final sums for normal machines are usually no more than 1 or 2 thou different. Some alignment methods offer the option of not making a full turn on the dials or laser equivalent. This short-cut loses the data check discussed above.

For discussion purposes, a viewpoint for the machines is required. View the unit from the driven machine, looking toward the driver. Left and right sides of the unit are determined this way. The flex-planes can be referred to as the near and far, near being closest to the driven machine.

Usually, the driver will be the machine to be moved.
The allowable angularity should be determined at the start of the job if speed is of the essence. The graph in Figure 1, is recommended for this purpose. The maximum shaft speed is required as well as the type of coupling. Gear couplings should be left below the bottom line. Convert the alignment guideline expressed in mils per inch into mils TIR by multiplying the guideline by the diameter swept by the dials. The result is the largest difference that should be seen between the vertical or the horizontal dial readings [the axial float compensated readings of course.]
(Note: 1 mil = 1 thou = .001 inch)

The measured flex-plane angularities in the vertical and horizontal planes are calculated by dividing the difference between the dial readings by the dial swept diameter. A sample spreadsheet is included as an appendix. Alternatively, compare the dial reading differences with the number calculated from the guideline angularity.

The ultimate check of angularity is done by taking the square root of the sum of the squares of the horizontal and vertical angularities at each flex-plane. This is the correct way to determine angularity, but in most cases, will make only a small difference.

In all alignment work, the issue of hot versus cold alignment must be addressed. I have found that in most cases, the Reverse-Face Method allows me to mount the magnetic bases and collect the dial data within 5 minutes. In my experience, thermal changes of consequence occur after about 10 minutes. If hot alignment is more critical, [for example, if pipe loads or shaft torque influence are suspected to influence alignment], there are other methods such as Vernier Alignment or Essinger Bars that can be used. No method that requires the shutdown of the machine to check alignment is suitable, in the limit, for critical hot alignment checks.

The calculation of the horizontal moves and vertical shim changes is based on simple geometry. A spreadsheet is shown in the appendices that we use to do this calculation.

## 6. REVERSE-FACE METHOD VERSUS OTHER METHODS

Laser Systems cost significantly more than a set of dial indicators, lack a calculated angularity at the flex-planes, and are noted for problems due to misuse by users, in the author's experience. On the other hand, the speed of data collection and calculation of moves and shims is excellent. In some cases, the brackets are too bulky to swing a full circle. Some systems get around this problem by calculating the missing readings from several intermediate readings at known angles.

The Reverse Dial or Reverse-Rim Method requires complicated brackets plus correction for bracket sag. The installation of the brackets takes longer than mounting a set of face dials. The resulting dial readings must be converted via complex calculations to angularity numbers. The cost of the brackets is greater than the cost of face dial equipment. The speed of collection of data is certainly no faster than the Reverse-Face Method.

The Rim-Face Method has the same disadvantages as the Reverse-Rim Method for spool piece couplings. However, since it is insufficient for alignment determination in general to measure an angle across a single flex-plane coupling [in other words, the Reverse-Face Method does not work in this instance], one of the Rim-Face Method, or the other methods above, is required when there is no spool piece. Rim-Face is a logical method to complement the Reverse-Face Method since no special brackets are required to use Rim-Face for a single flex-plane coupling. All that is required is three magnetic bases and dials. In addition, for couplings like that shown in Photograph 3, the Rim-Face Method is the only logical method. Reverse-Rim or Laser can be used, but the brackets would be a sight to behold.

## 7. LIMITATIONS OF THE REVERSE-FACE METHOD

Care must be taken when using the Reverse-Face Method on gear couplings or elastomeric couplings that can develop radial clearance due to wear. Refer to Photograph 2 for an example of these two types of coupling on one machine. The radial clearance in the couplings must be measured. (This is not a hardship, since the clearance must be measured to monitor the wear in the coupling.) Then, the difference in the radial clearance between ends, must be used to correct the angularities calculated normally, A calculation of the radial clearance-induced angularity at the coupling flex planes can be done using simple geometry [angle = difference in the radial clearances/distance between flex-planes].

The other limitation, as discussed above, is the single flex-plane coupling, which requires the Rim-Face Method.

## 8. OPPORTUNITIES FOR ENHANCEMENTS OF THE REVERSE-FACED METHOD

If it is considered to improve the speed of data collection, or to automate the method, there are many things that could be done, as summarized below:

- special dial indicators with integral magnetic bases for specific coupling designs
- remote digital readout of dial indicators
- readings at 45 degree orientations, converted to horizontal and vertical
- calculation on a spreadsheet with manual data entry
- direct input to the computer using digital dial indicators
- addition of a digital inclinometer
- calculation of the angularities and corrections based on less than 360 degree rotation of shafts
- calculation of shims at many planes
- combination of cold and hot alignment data to calculate desired cold angularity
- error analysis calculation using tolerances on dial readings.

The author has generated a spreadsheet to calculate dial readings at orthogonal points [3:00, 6:00, 9:00] from dial readings at other angular locations. Again, this is a simple geometry exercise.

## 9. GUIDELINES FOR ALIGNMENT AND RELATED ISSUES

The original impetus to use the Reverse-Face Method was to make it easier to determine if a particular alignment was acceptable. Piotrowski's graph [see Figure 1] provides a strong argument for using angularity at each flex plane to determine acceptability.
Gear couplings are a special case. The relative tooth velocity should be calculated to determine if the oil film will break down due to misalignment (maximum of 5 in/sec pk (Ref. 2)). On the other hand, minimum angularity is also required to ensure that the oil will get between the teeth.

Many manufacturers and most Laser Alignment systems use guidelines based on the offset and angularity between the shaft centrelines at the middle of the coupling spool piece. However, consider alignment limits based on parallel offset alone with no shaft angularity, and then shaft angularity with no parallel offset. Either limit can be derived from flex-plane angularity considerations. In real alignments, the as-left alignment will have a combination of offset and angularity. If both shaft offset and angularity limits were reached at the same time, the resulting flex-plane angularity would be twice guideline.

In other words, the guideline based on shaft offset and angularity would have to be twice as strict to be equivalent in all cases to the flex-plane angularity guideline in Piotrowski's chart.

## 10. CONCLUSIONS

The author has found the Reverse-Face method to be a fast, accurate and inexpensive method of doing alignment measurements. It is hoped that the reader will find the method to be useful, too.

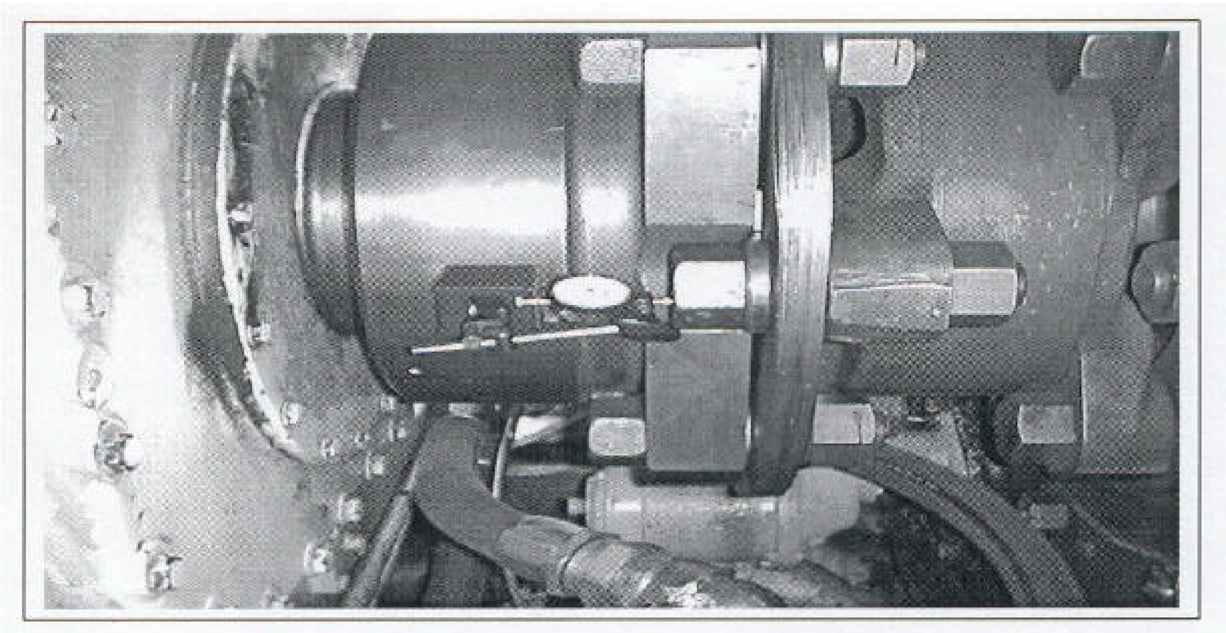
## 11. REFERENCES

1) Piotrowski, John; "Shaft Alignment Handbook"; Marcel Dekker, Inc., New York and Base)

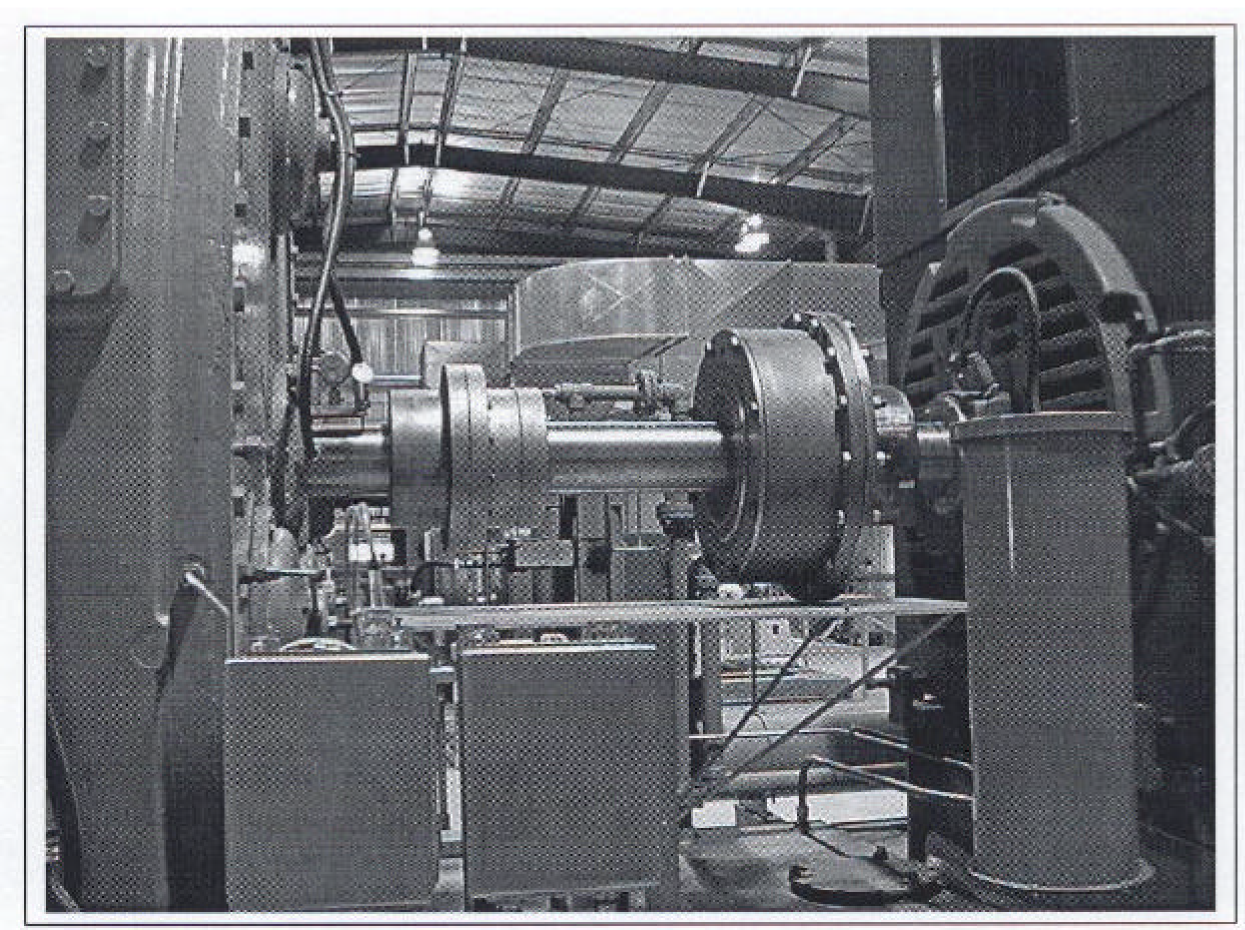
2) Crease, A.B.; "Design Principles and Lubrication of Gear Couplings"; Paper B1, International Conference on Flexible Couplings for High Powers and Speeds, June, 1977.

**BIOGRAPHY:**

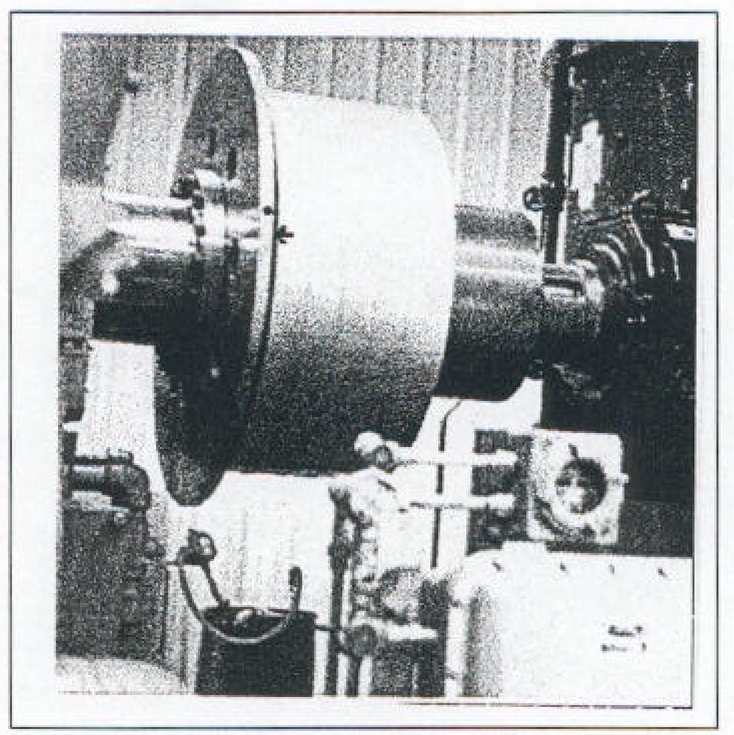
Brian Howes is Chief Engineer for Beta Machinery Analysis Ltd., Calgary. His previous experience includes: research and development in the area of pulsations and vibrations of reciprocating compressor piping systems, 28 years of troubleshooting problems, in many countries using a wide range of equipment including turbines, centrifugal and plunger pumps, centrifugal, screw and reciprocating compressors, pulp refiners, paper machines, ball mills, furnaces and piping systems. He has a Master of Science in Solid Mechanics from the University of Calgary, and is a member of the Board of Directors of the Canadian Machinery Vibration Association (CMVA).



Picture 1: Flex-pack coupling with a face dial mounted, showing interfering piping underneath coupling [102795]



Picture 2: Gear coupling on the left, and Elastomeric coupling on the right [102847]



Picture 3: A LoRez Coupling – single flex-plane – use Rim-Face Method

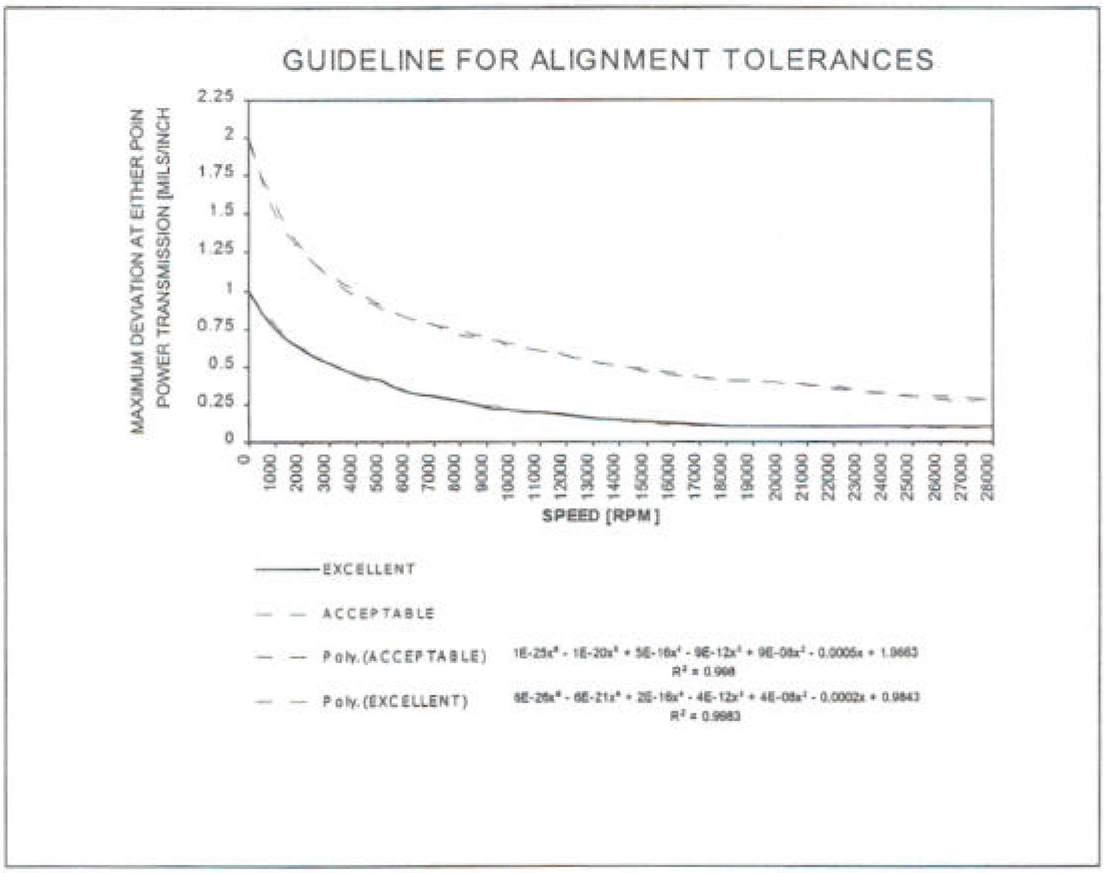



Figure 1: Alignment tolerance Chart (with polynomial curve fit equations) after Piotrowski

**APPENDICES**

1. Spreadsheet to document Reverse-Face dial readings and calculate angularities.
2. Spreadsheet to calculate hot alignment angularities given the cold alignment and Essinger Bar data.
3. Spreadsheet to calculate shims and moves for many feet from Reverse-Face data

**Appendix 1**

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | **Alignment Report** | | | | | | | |
| | | | **[Reverse Face Method]** | | | | | | | |
| **Owner:** | | | | | | | | | | |
| **Location:** | | | | | | | | | | |
| **Unit:** | **DeLaval HV-8** | | | | | | | | | |
| **Coupling:** | **Make:** | **?** | **Speed [rpm]** | | **600 max** | | | | | |
| | **Model:** | **?** | **Target angularity limit[thou/inch]** | | | | **1.5 to 0.8** | | **when hot** | |
| **Date:** | **August18/99** | | **Machine temperature during readings:** | | | | | **cold** | | |
| **Analyst:** | **B. Howes** | | **Target cold alignment:** | | | **compressor** | | **10** | **Thou** | |
| | | | | | | | **higher than the engine** | | | |
| **Dimensions:** | | **Diameter of dial circle:** | | **26** | **inches** | | | | | |
| | **Coupling shim pack spacing:** | | | **13** | **Inches** | | | | | |
| **As left angularities:** | | | **Vertical** | **Horizontal** | | **Total** | | | | |
| **@Fixed machine shim pack:** | | | **-0.83** | **-0.36** | | **0.90** | **thou/inch** | | | |
| **@Moved machine shim pack:** | | | **0.88** | **0.25** | | **0.92** | **thou/inch** | | | |
| **Guideline for angularity is a function of speed. Check book by Piotrowski for graph [page233].** | | | | | | | | | | |
| **Predicted hot angularities:** | | | **Vertical** | **Horizontal** | | **Total** | | | | |
| **@Fixed machine shim pack:** | | | **-0.06** | **-0.36** | | **0.36** | **thou/inch** | | | |
| **@Moved machine shim pack:** | | | **0.12** | **0.25** | | **0.28** | **thou/inch** | | | |
| Note: | {The predicted hot alignment angularities are based on the assumption that the engine will rise | | | | | | | | | |
| | 10 thou more than the compressor between cold stopped and hot running} | | | | | | | | | |
| | {The shaft motion within the bearing clearances is not included in these calculations.} | | | | | | | | | |
| | | | | | | | | | | |
| **Dials on fixed machine side** | | | | | | **Dials on "moved machine" side** | | | | |
| Dial "a" @ | Dial "a" | Dial "b" | (a-b)/2 | | | Dial "a" @ | Dial "a" | Dial "b" | (a-b)/2 | |
| **12:00** | 0 | 0 | **0** | | | **12:00** | 0 | 0 | **0** | |
| **3:00** | -3 | 7 | **-5** | 0 | | **3:00** | 3.5 | -14 | **8.75** | 0 |
| **6:00** | -25 | 18 | **-21.5** | | | **6:00** | 16 | -30 | **23** | |
| **9:00** | -21 | 7.5 | **-14.25** | -9.25 | | **9:00** | 12.5 | -18 | **15.25** | 6.5 |
| **12:00** | 0 | -1 | 0.5 | | | **12:00** | 0 | 0 | 0 | |
| | | | | | | | | | | |
| sum of vert | | -21.5 | | | | sum of vert | | 23 | | |
| sum of hor | | -19.25 | | | | sum of hor | | 24 | | |
| closure | | 0.5 | | | | closure | | 0 | | |
| **fmy** | **-21.5** | -ve = open at bottom | | | | **mmy** | **23** | +ve = closed at bottom | | |
| **fmx** | **-9.25** | -ve = open on left | | | | **mmx** | **6.5** | +ve = closed on left | | |
| | | | | | | | | | | |
| sum of vertical should be equal to sum of horizontal | | | | | | | | | | |
| closure [equivalent dial reading at second 12:00] should be zero | | | | | | | | | | |
| equivalent dial reading [compensated for float] is (a-b)/2 | | | | | | | | | | |
| fmy, fmx, mmy, mmx are the face results at 6:00 and 9:00 for the 2 planes | | | | | | | | | | |
| | | | | | | | | | | |

**Appendix 2**

Essinger Bar readings to angularities at the power transmission points of a spool piece coupling
Refer to "Shaft Alignment Handbook" by Piotrowski for more details and
A graph on allowable alignment angularity Brian Howes, December, 1999

Unit: Boiler Feedwater #1 Location:
Date: Nov29/99 to Dec 2/99 Condition: As found

Dimensions [inches]

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| | | | | | | | up/right |
| Motor end | **L1** | **L2** | **L3** | **L4** | **L5** | Pump end | |
| | 77 | 10 | 7 | 9 | 81 | | down/left |

Essinger Bar CHANGES FROM COLD TO HOT [thou]

| | | | | | | |
|---|---|---|---|---|---|---|
| | **E1** | **E2** | | **E3** | **E4** | |
| Vertical | 0.4 | 2.1 | y1, y2 ,y3, y4 | 1.8 | -0.7 | +ve = up |
| Horizontal | 0.3 | 0.6 | X1, x2, x3, x4 | -5.7 | -0.7 | +ve = right |
| | MOB | MIB | | PIB | POB | |

Calculated position of shaft centerline at the flex planes after cold to hot change

| | Motor shaft | | PUMP SHAFT | | |
|---|---|---|---|---|---|
| | V1 @ F1 | 2.32 | V2 @ F2 | 2.08 | |
| Vertical | V3 @ F2 | 2.48 | V4 @ F1 | 2.29 | +ve = up |
| Horizontal | H1 @ F1 | 0.64 | H2 @ F2 | -6.26 | +ve = right |
| | H3 @ F2 | 0.67 | H4 @ F1 | -6.69 | |

Calculated angles at the shim packs [flex points, power transmission p oints]

| | Motor end | | Pump end | | | | |
|---|---|---|---|---|---|---|---|
| Vertical | Alpha1 | -0.06 | Alpha2 | 0.00 | angle units are thou/inch | | |
| Horizontal | Beta1 | -0.99 | Beta2 | 1.05 | 1 thou/inch = 1 milliradian | | |
| Total | Gamma1 | 0.99 | Gamma2 | 1.05 | | | |
| Guideline | 0 | 0.5 | | | Speed | 3600 | [rpm] |
| | minimum | maximum | | | | | |

**Now assume the motor is set 6 thou lower than the pump, Cold**

Calculated position of shaft centerline after the change
From cold to hot at the flex planes

| | Motor shaft | | PUMP SHAFT | | |
|---|---|---|---|---|---|
| | V1 @ F1 | -3.68 | V2 @ F2 | 2.08 | |
| Vertical | V3 @ F2 | -3.52 | V4 @ F1 | 2.29 | +ve = up |
| Horizontal | H1 @ F1 | 0.64 | H2 @ F2 | -6.26 | +ve = right |
| | H3 @ F2 | 0.67 | H4 @ F1 | -6.69 | |

Calculated angles at the shim packs [flex points, power transmission points]

| | Motor end | | Pump end | | | | |
|---|---|---|---|---|---|---|---|
| Vertical | Alpha1 | 0.80 | Alpha2 | -0.85 | angle units are thou/inch | | |
| Horizontal | Beta1 | -0.99 | Beta2 | 1.05 | 1 thou/inch = 1 milliradian | | |
| Total | Gamma1 | 1.27 | Gamma2 | 1.35 | | | |
| Guideline | 0 | 0.5 | Thou/inch at | | Speed = | 3600 | [rpm] |
| | minimum | Maximum | | | | | |

**Appendix 3**

**Reverse-Face Alignment Calculation Sheet**
[For unit where driver is the fixed machine]

**Date:** **Time:** **Customer:** **Unit:**

Reverse-Face Dial Readings

| fixed machine=driver | | | | | machine to be moved | | | |
|---|---|---|---|---|---|---|---|---|
| a@ | a | b | (a-b)/2 | | A@ | a | b | (a-b)/ |
| 12 | 0 | 0 | 0 | | 12 | 0 | 0 | 0 |
| 9 | | | | | 9 | | | |
| 6 | | | | | 6 | | | |
| 3 | | | | | 3 | | | |
| 12 | | | | | 12 | | | |

Dial Data Quality Check

| | | | | |
|---|---|---|---|---|
| sum of vert | 0 | | sum of vert | 0 |
| sum of hor | 0 | | sum of hor | 0 |
| closure | 0 | [from 12 to 12] | closure | 0 |

[The sums in the vertical and the horizontal should be equal]
[Closure of the dial readings at 12:00 should be zero]

| | | | |
|---|---|---|---|
| Fmy | 0 | fmy | 0 |
| fmx | 0 | fmx | 0 |

| | **fmy** | **fmx** |
|---|---|---|
| Fixed fm* | 0 | 0 |
| Move fm* | 0 | 0 |

| | | SHIMS [thou] | MOVES [thou] |
|---|---|---|---|
| DIA | 26 | | |
| CPLG | 13 | | |
| L1 | 38 | 0 | 0 |
| L2 | 73 | 0 | 0 |
| L3 | 53 | 0 | 0 |
| L4 | 88 | 0 | 0 |
| L5 | 33 | 0 | 0 |
| L6 | 49 | 0 | 0 |
| L7 | 74 | 0 | 0 |
| L8 | 98 | 0 | 0 |

[fmy is the bottom less the top
Dial reading after axial float compensation]
[fmx is the right less the left
Dial reading after axial float compensation]

Lengths Ln are from the flex-plane closest to the fixed machine
"Fixed fm*" and "Move fm*" are the values of fmy or fmx at the flex-planes
on the fixed machine and the machine to be moved sides respectively.
Lengths Ln and Diameter DIA in inches CPLG, inches between flex-planes

Sign Conventions
looking from driven machine to driver
positive y is up
positive x is right
reference vertical readings to 0 @ 12 o'clock
reference horizontal readings to 0 @ 9 o'clock
Positive shim value means add shim