COMPONENT BALANCE FOR CENTRIFUGAL PUMPS— EFFECTS OF PRODUCT DESIGN, BALANCE MACHINE TOOLING, AND THE BALANCING PROCESS ON BALANCING REPEATABILITY

by

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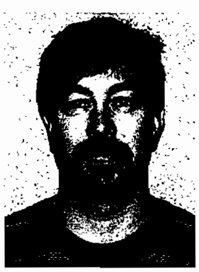

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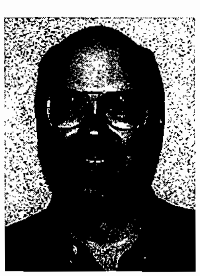

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ABSTRACT

Balancing of pump rotating components, especially impellers, is common practice. The purpose of this effort was to determine the proper design of pump components and balance arbors, create appropriate procedures, and establish their validity through repeatability and reproducibility studies. Examples of tests and recommendations for users are included.

BACKGROUND

Pump component and rotor balance have been scrutinized in recent years by both users and manufacturers. While it may appear that more restrictive (i.e., lower) unbalance levels will automatically result in lower vibrations and longer machine life, studies by Nelik and Jackson (1995) indicated such was not the case. Their research showed that hydraulic unbalance contributes much more significantly to vibration than mechanical unbalance.

ISO 1940/1 (1986) specifies balance grades ("G" levels) for various types of rotating equipment. For pumps, the general recommendation is G6.3. Some users and manufacturers specify G2.5 for certain services, such as fan and boiler feed pumps.

Typically, balance processes (tooling, balance machine capabilities, compensation cycles and methods) are taken for granted (not even considered) when balancing rotating components of centrifugal machines. Because of the above, attempts to verify proper balance may make the part appear to be grossly out of tolerance, especially as compared with the minimal tolerances allowed by the lower balance grades. While API 610 (1995) requires ISO 1940/1 grade G0.7, it recognizes the inability to reliably reproduce any levels below G2.5.

INTRODUCTION

The purpose of this effort was to determine the proper design of pump components and balance arbors, create appropriate procedures, and establish their validity through repeatability and reproducibility studies.

Test programs evaluated standard balance machines with a variety of impeller and tooling (mandrel) designs. Newly developed compensation procedures minimized or eliminated residual unbalance created by the mandrel design.

Traditional gauge repeatability and reproducibility studies measure only one-dimensional elements. Unbalance, by its very nature, is two-dimensional. Presented is a two-dimensional analysis developed to overcome the inadequacies of traditional (standard) analysis.

Before embarking on the presentation of the research, a review of unbalance types is in order.

TYPES OF UNBALANCE

The term "unbalance" will be used throughout, as that is the entity detected, located, and measured by a balance machine. Balance is a zero quantity, determined by measuring an absence of unbalance.

The following paragraphs explain and illustrate the two main types of unbalance as defined by international standards.

• *Static*—Static unbalance exists when the principal inertia axis is displaced parallel to the shaft axis/axis of rotation, as illustrated in Figure 1.

This type of unbalance exists in narrow disk-type rotating components, such as turbine wheels, flywheels, and low specific speed pump impellers. It also exists if the disk's center of mass is located eccentric to its center of rotation, due to nonuniformity in density of the disk, for example.

Static unbalance, if of sufficient magnitude, may be detected by mounting the rotor on knife-edges (ways) or rollers. If the ways are level, as shown in Figure 2, the rotor will turn until the heavy portion is down. While this may be adequate for low speed equipment, it is not a sufficient balancing method for higher speed units. Centrifugal means, as obtained on balancing machines, measure static unbalance more accurately, and must be employed for high-speed equipment.

Correction of static unbalance may be accomplished by adding weight opposite the unbalance or by material removal at the unbalance location.

• *Dynamic*—Dynamic unbalance exists when the principal inertia axis and the shaft axis/axis of rotation neither intersect nor are parallel to one another, as illustrated in Figure 3.

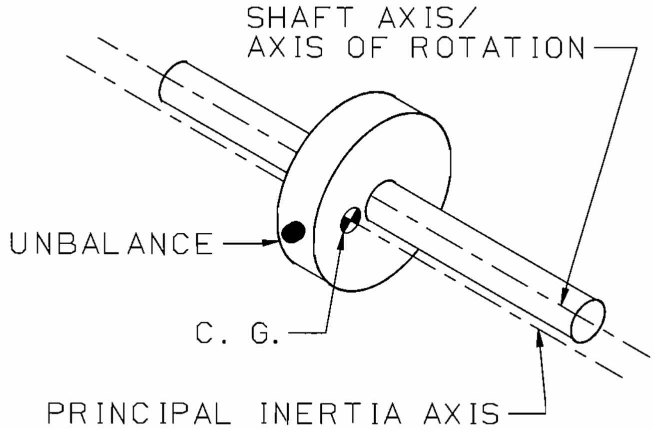

Figure 1. Static Unbalance–Principal Inertia Axis Displaced from Axis of Rotation.

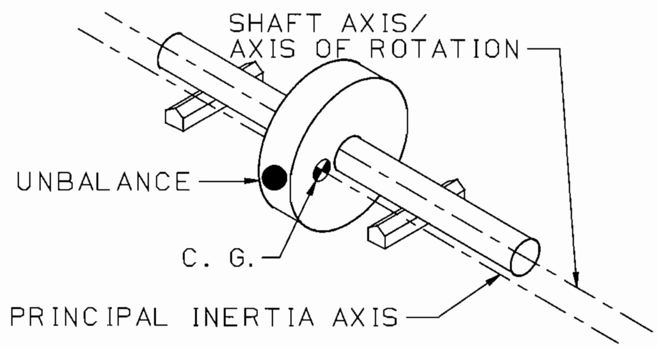

Figure 2. Detecting Static Unbalance Using Gravity Method (Ways).

Dynamic unbalance is the most common type encountered. Dynamic unbalance requires dynamic methods for detection. Correction employs adding weights to create a couple equal in magnitude but 180 degrees opposed to the original couple, or by removal of material equating to the original couple.

Two special cases of dynamic unbalance, known as couple and quasi-static unbalance, are also defined in literature. *Couple* unbalance occurs when the principal inertia axis intersects the shaft axis/axis of rotation at the center of gravity. *Quasi-static* unbalance occurs when the principal inertia axis intersects the shaft axis/axis of rotation at a point other than the center of gravity. Both these special cases require dynamic detection methods and addition or removal of material appropriately located to negate the original couple.

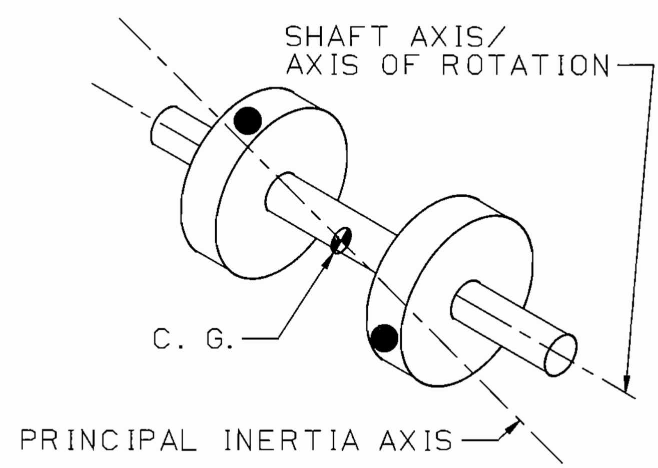

Figure 3. Dynamic Unbalance–Principal Inertia Axis and Axis of Rotation Neither Intersect Nor Are Parallel to One Another.

CONTRIBUTORS TO UNBALANCE

Unbalance occurs in a component, in this case an impeller, due to several factors.

The vast majority of impellers are made from castings. Even precision casting processes may have nonhomogenities in material due to gas holes, shrinks, or variations in material density related to the relative vertical position in the mold. Less precise processes (e.g., sand-casting) allow some amount of variation (shift) from part to part.

Tolerances must be allowed in alignment of a casting during the initial machining process. Alignment in the machine will coincide with the part geometry, which may not be identical to the mass distribution.

Design considerations related to unbalance include keyway design, shaft/impeller fit, and unfilled voids. These considerations relate to unbalance of the entire rotating assembly as well as the impeller.

An incorrect or incapable process, which includes the tooling, balance machine, method, and operator, can unbalance an impeller to a level greater than its initial unbalance.

Unbalance caused by variations in the casting, machining, and design are typically small in comparison with the cost associated with making each of those aspects "perfect." A capable balance process allows for economical correction of these inherent unbalances to reduce the unbalance to an acceptable level.

DETERMINATION OF STATIC OR DYNAMIC BALANCE

As stated in a previous section, static unbalance commonly exists in narrow disk-type rotors, and dynamic unbalance exists in wider rotors. Because balance machines have a practical limit on their ability to sense unbalance in planes that are relatively close to one another, it is necessary to establish criteria for determining whether a component should be statically (single-plane) or dynamically (two-plane) balanced. API 610, Eighth Edition (1995), and ANSI B73.1 (1991) specify a dynamic balance for components that have a ratio of diameter to peripheral width of six or less ($D/b \leq 6$). Ratios greater than six should be statically balanced. Examples of pump components and determining dimensions are shown in Figures 4 through 7.

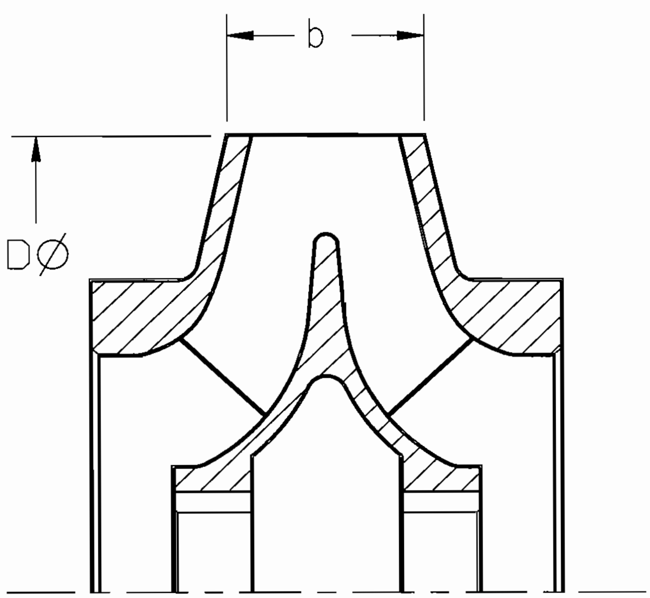

Figure 4. Determination of Balance Type–Double Suction Impeller.

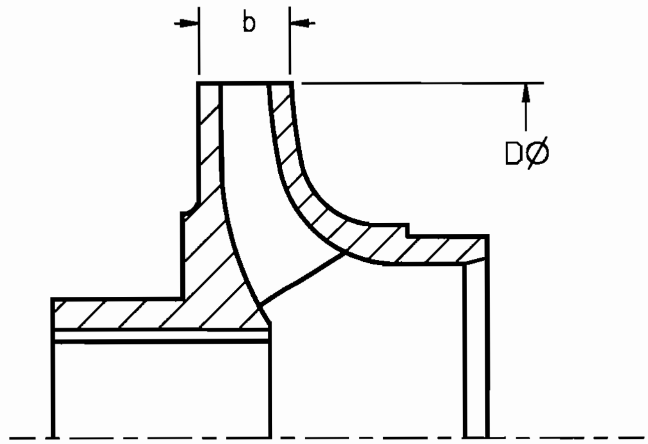

Figure 5. Determination of Balance Type–End Suction Enclosed Impeller

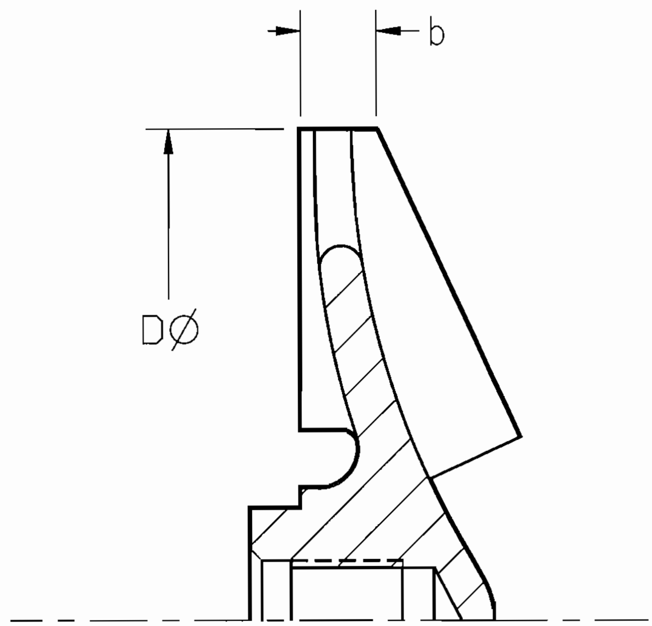

Figure 6. Determination of Balance Type–End Suction Open Impeller.

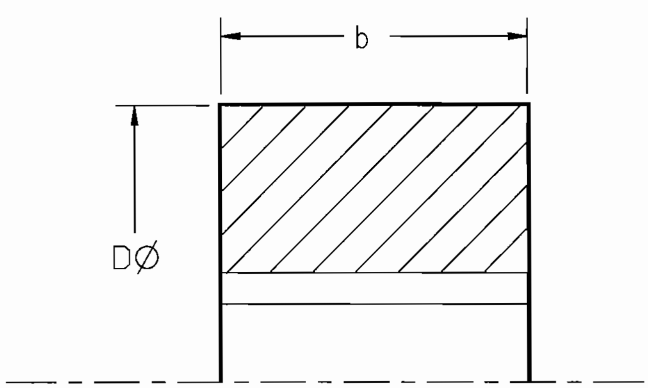

Figure 7. Determination of Balance Type–Disk.

TOLERANCE SELECTION

ISO 1940/1, "Balance Quality Requirements of Rigid Rotors" is a widely accepted standard used to reference rotor balance quality grades. The standard gives recommendations for determining permissible unbalance and offers methods of allocating these tolerances.

Various levels (balance quality grades) are specified within the standard, from very low grades used for extremely precise instruments (G0.4), to very high levels used for crankshafts of slow marine diesel engines (G4000).

Independent research conducted by Nelik and Jackson (1995) indicated that a balance level of G14 or lower had no effect on vibration levels of the pump tested. Hydraulic loading contributed more to the vibration than did the mechanical loading caused by unbalance.

While the above-cited research indicates no improvement in pump vibration below a balance quality grade of G14, ISO 1940/1 recommends G6.3 for pump impellers, based on the practical experience of manufacturers and users of various pump types. The authors concur with this recommendation.

Unbalance is measured in units of mass and length, and is typically expressed as ounce-inches (oz-in), gram-inches (g-in), or gram-millimeters (g-mm). The permissible residual unbalance, U_{per}, varies directly with the mass of the rotor, m, and may be expressed as:

$$U_{per} = e_{per} \times m \qquad (1)$$

where e_{per} = permissible residual specific unbalance value = eccentricity of the center of mass of the components. e_{per} varies inversely as the speed of the rotor. For a given grade:

$$e_{per} \times \omega = \text{constant} \qquad (2)$$

where $\omega = 2\pi N/60$ = angular velocity of the rotor at maximum service speed (N). Rearranging the above and solving for e_{per} gives:

$$e_{per} = \frac{U_{per}}{m} = \frac{\text{constant}}{\omega} \qquad (3)$$

For a given balance grade and rotating speed, the permissible residual specific unbalance value, e_{per}, is constant.

For convenience, approximate formulas relating impeller weight and rotating speed to various balance grades are tabulated in Table 1.

Table 1. Formulas for Maximum Permissible Unbalance, English Units.

ISO Balance Grade	Maximum Permissible Unbalance, U_{per} (oz – in)
G0.7 (API 610)	4 W/N
G2.5	15 W/N
G6.3	40 W/N

Where: W = impeller weight, lb
N = rotating speed, rpm.

DESIGN CONSIDERATIONS

When speaking of balancing for single stage pumps, we often concentrate on the main component—the impeller. Depending on the type of pump, the unbalance attributable to the impeller may account for as little as 30 percent of the total rotor unbalance. Figures 8 and 9 illustrate the impeller and coupling ends of the shaft of a typical heavy-duty end suction overhung pump. Note the voids left by standard machining practices and square-end keys. Clearance fits of the impeller to the shaft and the sleeve to the shaft also contribute to rotor unbalance. The unbalance magnitude of each portion is summarized in Table 2.

A redesign of the impeller, shaft, keyway, keys, and shaft sleeve, and the impeller end, as shown in Figure 10, yields a reduction in unbalance of 42 percent, as shown in Table 3.

MANUFACTURING CONSIDERATIONS AND BALANCE PROCESS

Once an acceptable balance level has been selected, it becomes necessary to manufacture impellers that are, in fact, reliably balanced to the stated level. A process capable of achieving this goal has many facets that must be evaluated. This study evaluated the items that follow.

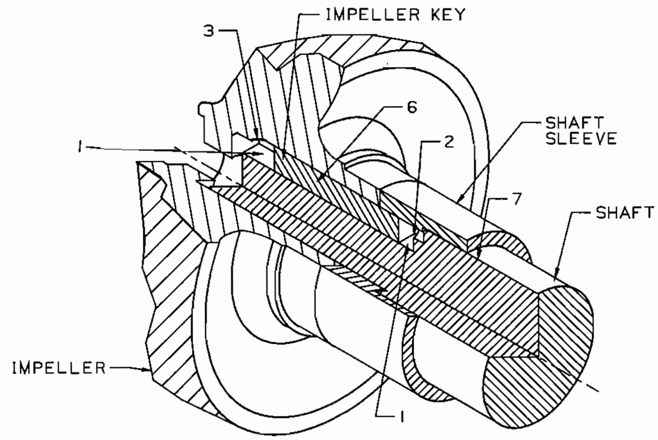

Figure 8. Typical End Suction, Heavy Duty Pump–Impeller End, Original Design.

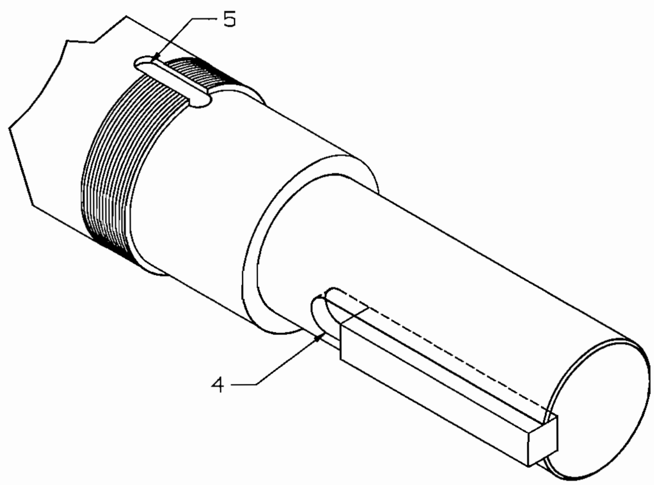

Figure 9. Typical End Suction, Heavy Duty Pump–Coupling End.

Table 2. Typical End Suction, Heavy Duty Pump Rotor Unbalance Summary, Original Design.

Locator	Description	Unbalance (oz-in)
1	Shaft Impeller Keyway	0.94
2	Sleeve Keyway	0.72
3	Impeller Keyway	0.33
4	Coupling Keyway	0.25
5	Bearing Locknut Keyway	0.38
6	Impeller/Shaft Fit	1.0
7	Sleeve/Shaft Fit	0.13
	Rotor subtotal [1]	2.9
	Impeller @ ISO G6.3	1.05
Total Rotor Unbalance		3.95

[1]Added vectorially

Part Orientation

Most single-stage and some multistage pumps have a clearance between the impeller and the shaft. This clearance allows for ready assembly and disassembly without the need for thermal expansion of the impeller bore. At the speeds normally encountered (≤ 3600 rpm), the design is more than adequate. Torque is often transmitted through a key and keyways in both the shaft and impeller.

It was postulated that, due to the clearance between the shaft and impeller, orientation of the part on the balance tooling (mandrels) would affect impeller balance. Numerous balance readings were taken, half with the key facing up (12 o'clock) and half with the key facing down (six o'clock). Figure 11 shows the plotted data.

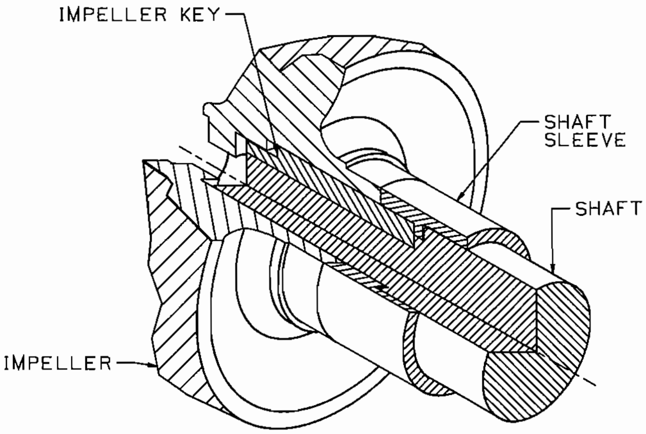

Figure 10. Typical End Suction, Heavy Duty Pump–Impeller End, after Redesign.

Table 3. Typical End Suction, Heavy Duty Pump Rotor Unbalance Summary, after Redesign.

Locator	Description	Unbalance (oz-in)
1	Shaft Impeller Keyway	0.0
2	Sleeve Keyway	0.0
3	Impeller Keyway	0.0
4	Coupling Keyway	0.25
5	Bearing Locknut Keyway	0.38
6	Impeller Shaft Fit	1.0
7	Sleeve Fit	0.13
	Rotor subtotal (1)	1.26
	Impeller @ ISO G6.3	1.05
Total Rotor Unbalance		2.31
% Reduction		42%

(1) Added vectorially

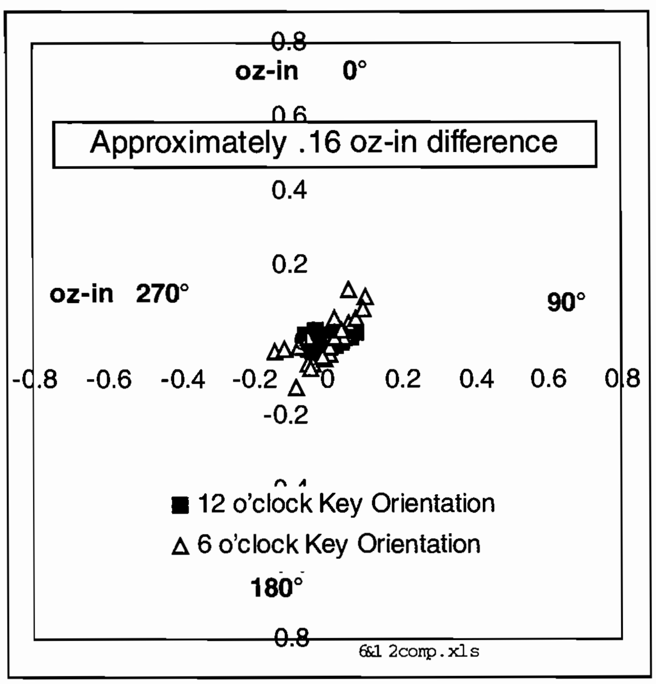

Figure 11. Effect of Part Orientation on Unbalance.

At 1800 rpm operating speed and balance grade G6.3, the allowable unbalance of the impeller is 0.69 oz-in. Analysis of these data show the orientation consumes 0.16 oz-in, or 22 percent, of the total tolerance. Balancing about the impeller's axis of rotation reduces the total amount by 50 percent, however, or 11 percent of the total tolerance band. (Note: This may be equated to runout (TIR) as compared with eccentricity, which is one-half the TIR.) Part orientation, therefore, should not be considered a significant factor.

Balance Machine Validation

Balance machines are typically calibrated by the machine manufacturer at regular intervals. In order to assure proper unbalance readings, the calibration process must include the production tooling and must reflect the production balancing process. Our studies showed variations of ± 33 percent, based on known unbalance values, between the balance machine's calibration configuration and its production configuration.

It is necessary to proceed beyond the standard balance machine calibration process if reliable and repeatable readings are desired. A balance machine "validation" process must:

- Include the balance machine/tooling configuration.
- Determine the balance machine response to a known unbalance.
- Ensure machine repeats within the balance tolerance level selected.
- Ensure electronic compensation cycle is operating properly.

Impeller unbalance values will be questionable if any of the above are found to be erroneous.

Runout of Tooling and Machine

At the outset of this study, measurements were taken of the tooling (mandrel) and machine spindle runouts, and maximum values were found to be 0.0035 inch and 0.005 inch TIR (total indicator reading), respectively. Since eccentricity is one-half the TIR, total eccentricity of the spindle/mandrel combination is 0.00425 inch. Allowable eccentricity, e_{per}, for balance grade G6.3 at 1800 rpm is 0.00131 inch. Eccentricity of the spindle/mandrel combination was, therefore, 420 percent of the maximum allowable eccentricity for the desired grade!

As a result of the findings, balance machine and mandrels were repaired. It is recommended that machine spindle runout not exceed 0.003 inch TIR, and that mandrel runout not exceed 0.0005 inch TIR. Balance machine spindle and mandrel runouts in the ranges specified do not tax balance machines' compensation cycles.

Balance Machine Tooling (Mandrels)

A variety of different mandrel designs may be used for impeller balancing operations. Basic designs include a cut off shaft, a solid mandrel with fixed key, and expansion types. Expansion types are either mechanically or hydraulically expandable.

A proprietary solid mandrel that incorporates indexing capabilities has been developed. The indexing capability optimizes a balance machine's compensation cycle.

Figure 12 compares results of testing an impeller with an index-capable solid arbor and a mechanical expansion arbor. The solid arbor repeated within nine percent of the total allowable G6.3 tolerance, while the mechanical expansion arbor consumed 20 percent of the tolerance.

Figure 13 shows results of testing the same impeller on two different mandrels. For this 25 lb impeller, allowable unbalance is 0.53 oz-in. Difference in unbalance between centers of the two patterns shown is 0.57 oz-in. Therefore, even if an impeller were corrected to "zero" unbalance on one mandrel, it would appear to exceed the allowable unbalance if checked on the other mandrel.

Figure 13 also shows the difference in results between a fixed key solid mandrel and the proprietary design with indexing capability. The smaller pattern of data indicates better repeatability. For production purposes, an index-capable solid mandrel is recommended.

Axis Definition

While it would not appear inherently obvious, it is necessary to define the axis about which the impeller must be balanced.

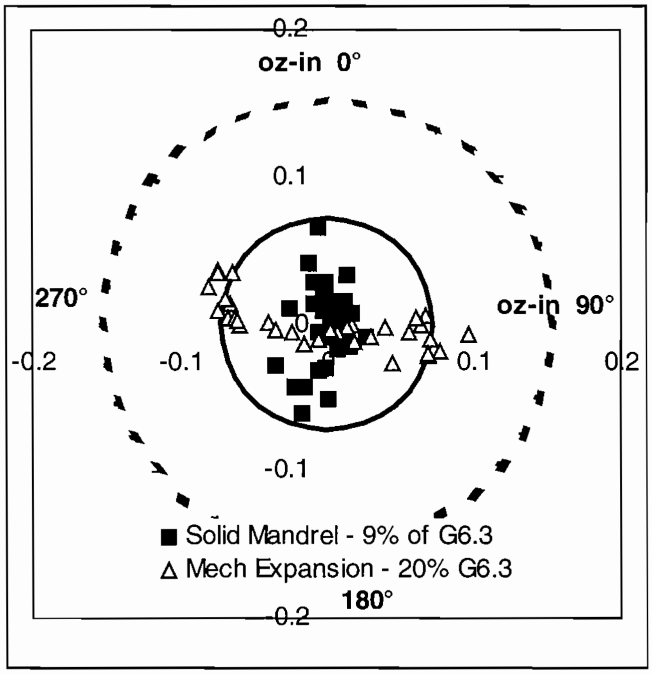

Figure 12. Solid Versus Expansion Type Mandrel.

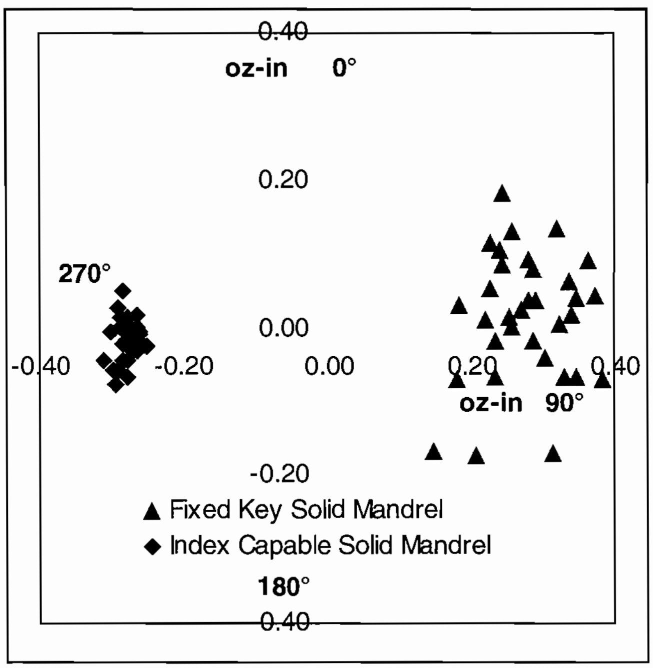

Figure 13. Solid, Fixed-Key Versus Indexable Mandrel.

Balance machines, through the compensation cycle, have the capability to electronically isolate various rotational axes. Figure 14 illustrates the variation encountered when the unbalance of an impeller balanced about its geometric axis is measured about a different machine or tooling axis.

In the figure, the circle about each point is the allowable total unbalance. If, for example, the machine spindle axis were used for verification, the impeller would appear to be approximately 180 percent over the maximum allowable unbalance. Conversely, an impeller balanced about the spindle axis would be unbalanced about its own axis. The authors' recommendation is to use the impeller axis rather than the spindle, mandrel, or shaft axis, as the balancing axis.

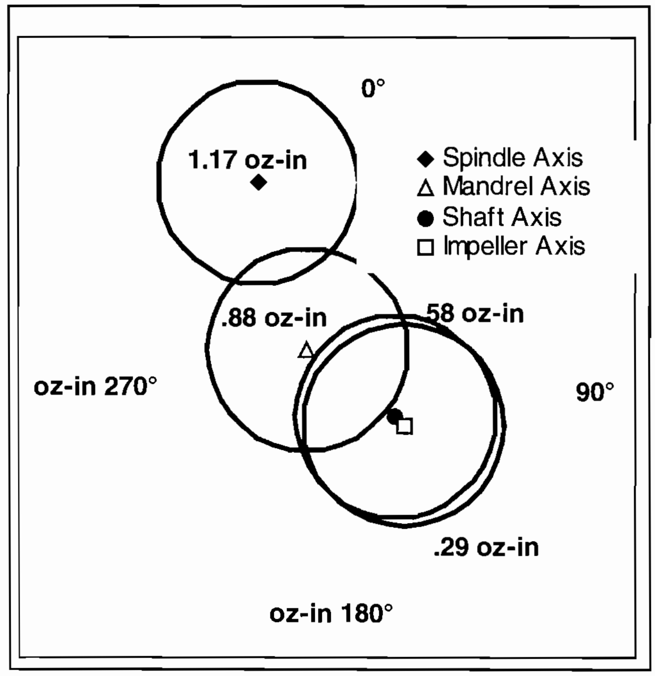

Figure 14. Examples of Unbalance Based on Four Separate Axes of Rotation.

Initial Unbalance Effect

An impeller that is grossly out of balance, this study found, will require a process modification to assure balance specification requirements.

Impellers with initial unbalances ranging from six to 40 times the maximum allowable unbalance (i.e., tolerance) were balanced to one-third the tolerance value. A compensation cycle was performed, and the results of the initial balance procedure were checked.

Instead of the data points falling along the horizontal line in Figure 15, they fell well above it in most cases. The balance machine's measurement scale is most likely affected by large amounts of initial unbalance.

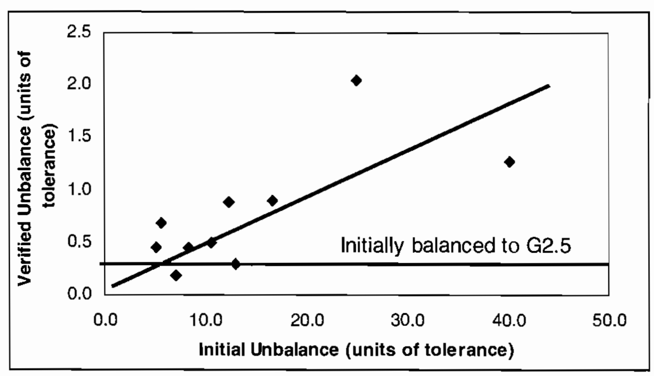

Figure 15. Verification of "Final" Balance Based on Initial Unbalance.

The authors recommend that, if initial unbalance exceeds 10 times the allowable unbalance, a "rough" balance operation be performed, followed by a compensation cycle and a final balance/verification.

CALCULATING REPEATABILITY AND REPRODUCIBILITY IN UNBALANCE R&R STUDIES

In this section, we will discuss how gauge R&R studies should be executed when unbalance is being measured. For simplicity, only the case of static (one-plane) unbalance will be considered. Although the design of such studies is briefly discussed, the major emphasis will be on their analysis. We show that the analysis using the standard one-dimensional paradigm is inconsistent and therefore incorrect; show a consistent way to extend this paradigm to the two-dimensional case, which includes unbalance measurements; remark on how we recommend such studies be designed; and provide two examples.

The Standard Gauge R&R Analysis

Unbalance, by its very nature, is two-dimensional. However, historically, gauge R&R studies have been analyzed using techniques that are designed for one-dimensional measurements. The reduction from two dimensions to one is accomplished by analyzing the magnitude of the unbalance, but not its direction. This one-dimensional analysis will be referred to as a *standard analysis*. In all cases, we assume the reader is familiar with the standard analysis, including terms such as repeatability (or EV, equipment variation) and reproducibility (or AV, appraiser variation). The words "appraiser" and "operator" are used interchangeably in most of this section.

There are clear inconsistencies in using the standard analysis for unbalance data. This is illustrated for the repeatability case, but the same problem exists for reproducibility as well. In Figure 16, the open-circled data represent 20 repeat readings of unbalance. The second set of readings (solid-square data) is the first set of readings, but offset by +1 unit in each of the "X" and "Y" directions. The third set of readings (asterisk data) are the same as the second set of readings, but rotated 90 degrees around their mean. The variability of the measurements is the same in all three cases, so any summary measures of them should be algebraically identical. However, when the standard analyses are done, the repeatability values for the three sets of data are 2.17, 1.40, and 3.67, respectively.

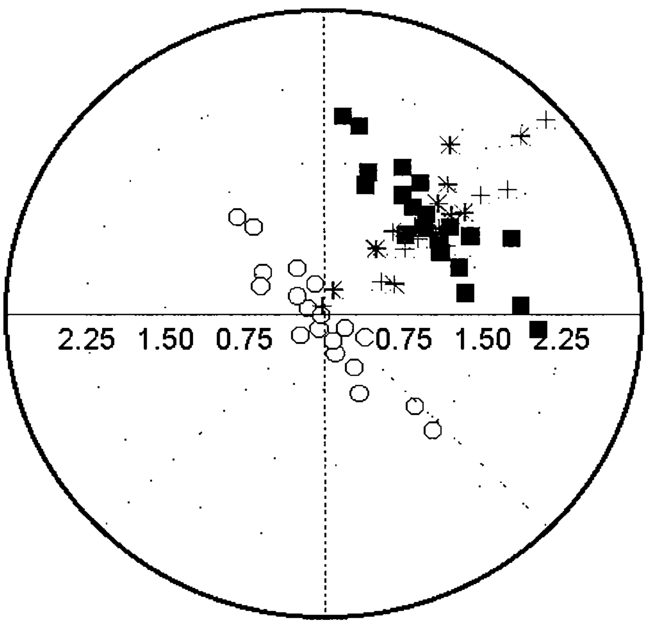

Figure 16. Examples of Three Sets of Equivalent Data That Let to Different EV Using the Standard Analysis.

In spite of these inconsistencies, we have observed that the standard analysis has been, to our knowledge, the only method used to analyze R&R in unbalance studies.

The 2D Gauge R&R Analysis

The analysis proposed can be used on a wide variety of data that are measured in two dimensions. For this reason, this analysis is referred to as a *2D analysis*, even though only its use for (static) unbalance readings is considered here.

Throughout this section, the emphasis is on an explanation of how to extend R&R studies from one to two dimensions, not on the mathematical details. For mathematical details, refer to Voelkel (1998). The subscript "EV" will be used to denote repeatability, while "AV" will be used to denote reproducibility. The variation due to both EV and AV is called "R&R." In this subsection, a distinction will not be made between a particular value, such as σ_{EV}, and an estimate of it, such as s_{EV}.

2D Repeatability and 2D Reproducibility

First, just consider repeatability. We will make the reasonable assumption that the variation in unbalance readings, when the r, ϕ readings are translated into x, y dimensions, can be modelled with a bivariate normal distribution. This is the natural statistical way to extend from one to two dimensions, and the data we have observed are usually consistent with this model.

In the one-dimensional case, the repeatability, or *EV*, is the length of an interval that captures 99 percent of repeat readings, by the same appraiser using the same gauge. This idea is extended to the two-dimensional case by defining *2D repeatability* as the diameter of a circle that captures 99 percent of repeat readings. This value will be called *EV*, as well. It turns out that *EV* is a function of the eigenvalues of Σ_{EV}, the variance-covariance matrix of the repeat readings. In particular, this means that *EV* depends on neither the average nor the rotation of the readings. For this reason, each of the three data sets in Figure 16 lead to the same *EV* value, which turns out to be 3.73. Figure 17 shows the first data set, centered to the origin, along with the circle corresponding to *EV*. The definition of reproducibility can be extended to *2D reproducibility*, or *AV*, in the same way.

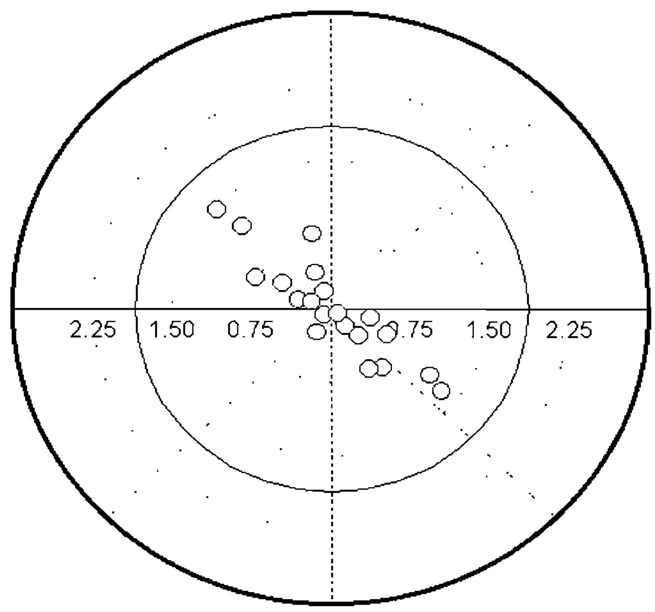

Figure 17. Example of Two-Dimensional Summary by EV.

Three other methods were considered for the definition of 2D repeatability. These were:

- The area of the circle,
- The area of an ellipse, and
- The length of the major axis of an ellipse,

each of which are constructed to capture 99 percent of repeat readings. However, all these measures turned out to have inconsistencies and so were not acceptable—refer to Voelkel (1998) for details.

2D R&R

In the one-dimensional case, *R&R* is essentially defined as the length of an interval that captures 99 percent of readings made by many appraisers, each of whom make many readings of the same part. Under the usual reasonable assumptions in such studies, it is well known that:

$$\mathrm{R\&R} = \sqrt{\mathrm{EV}^2 + \mathrm{AV}^2} \tag{4}$$

However, the two-dimensional case turns out to be more complex. In fact, it turns out that:

$$\mathrm{R\&R} \leq \sqrt{\mathrm{EV}^2 + \mathrm{AV}^2} \tag{5}$$

for this case.

Some Recommendations on R&R Studies

The typical so-called "long form" in a standard R&R study often uses two repeat readings, or trials, on each of 10 impellers by each of two operators. In practice, all these readings are usually made over a very short period of time. In addition, the statistical techniques upon which *EV* and *AV* are calculated make several assumptions, one of which is that the operators chosen in the study are a random sample from a large number of operators.

It has been the authors' experience that typically only a few operators are trained to make unbalance measurements, so the "large number of operators" really does not exist. Also, when operators make all the unbalance reading over a short period of time, there is no information in the gauge R&R study regarding the consistency of measurements over time. For these two reasons, the authors make the following recommendations. If only two operators usually made the readings, have them collect readings for at least three time periods. If only three or four operators usually made the readings, have them collect readings for at least two time periods. If five or more operators usually made the readings, have them collect readings for at least one time period, but preferably two time periods.

Also, the authors note that while variation among impellers can, technically, be estimated from such studies, in general the number of impellers tested (10 at most, frequently fewer) is not enough to derive a precise measure of impeller variation. There are reasons, however, to use a number of impellers that are thought to represent those being produced—some may be more easily measured than others, for example. The summary measures of *EV*, *AV*, and *R&R* would take this into account. For such reasons, it is not good practice to use only especially good or only especially poor impellers for the study.

If these recommendations are followed, two trials for each impeller/operator/time-period combination are sufficient.

Two Examples

The standard gauge R&R study is often intended to be used after process improvements have been made, to provide a summary of what the measurement process is. However, most of our studies to date have been devoted to measurement-process understanding and improvement rather than summarization. For this reason, both examples are somewhat nonstandard, but they do show the power of the methods proposed.

The first example consists of a study in which two operators made readings on each of two machines at each of two rpm's. These eight combinations were grouped under the factor "appraiser" to provide an overall measure of how these three factors affected the results. To the extent that their effects were small, it indicates that the operators, machines, and rpm settings are interchangeable. (A more sophisticated analysis was performed that separated out these effects, but it is not reported here.) Each "appraiser" measured each of two impellers twice, for a total of $8 \times 2 \times 2 = 32$ readings. To graph these results to emphasize the R&R variation, but not the impeller variation, we took the 16 readings for each impeller and centered these readings at the origin. The results are shown in Figure 18. For these data, we estimate that $EV = 0.26$ oz-in, $AV = 0.78$ oz-in, and $R\&R = 0.82$ oz-in, while the tolerance is ±0.50 oz-in = 1.00 oz-in. So an estimated 82 percent of the tolerance is being used by the measurement process. It turns out that much of the *AV* was due to differences between the two operators. Each operator individually (across the machines, rpm's, and impellers) had an *R&R* of approximately 0.40 in-oz. This is still a fairly large fraction of the tolerance.

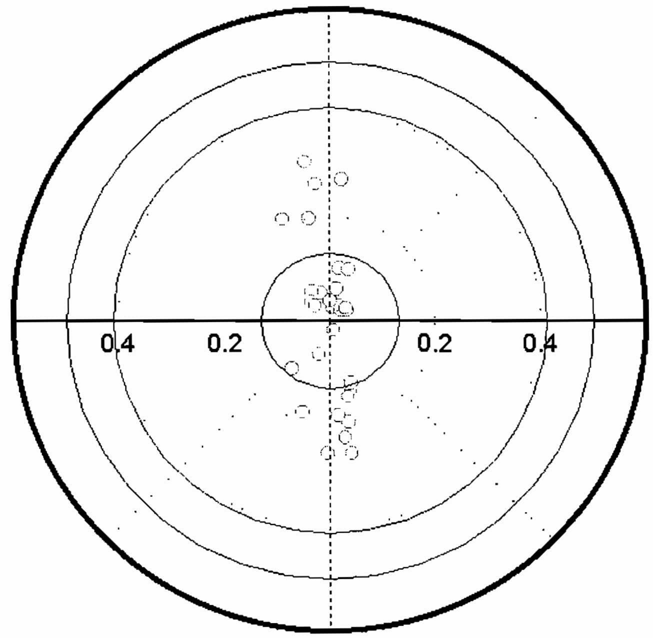

Figure 18. Example 1—Data Adjusted for Impeller. Increasing Diameter Circles Represent EV, R&R, and Tolerance.

Another feature of these data that can be seen in Figure 18 is that the *R&R* variation is predominantly in the "Y" direction. We have not yet been able to explain this phenomenon, but we believe that uncovering it could lead to a substantial reduction in variation.

The second example is based on a study whose objective, in part, was to see whether repeatability or reproducibility depend on impellers' weights. For this study, 10 impellers were selected. They consisted of two impellers at five weights, which were roughly 15, 20, 24, 25, and 36 lb. The same three operators measured each impeller's unbalance in two trials, and they did this on each of two days. There are, thus, 10 impellers × 3 operators × 2 time periods × 2 trials = 120 unbalance readings.

To see whether repeatability and reproducibility were affected by the impeller weights, a separate analysis was performed for each of the five weight groups. This lead to *EV* and to *AV* unbalance values, measured in in-oz, for each weight group. The values were also divided by the average weight of the impellers in that group, leading to *EV* and *AV* specific unbalance, or "e per," or eccentricity, values.

The results for repeatability are shown in Figure 19. These data do suggest that the *EV* depends on impeller weight. In fact, a statistical analysis indicates that this relationship is statistically significant ($P = 0.04$) and, furthermore, that this relationship is consistent with the model "*EV* is directly proportional to impeller weight." The few data points used, and the unusual value at 20 lb, suggest that more data would be desirable. Similar results hold for the AV measurements shown in Figure 20.

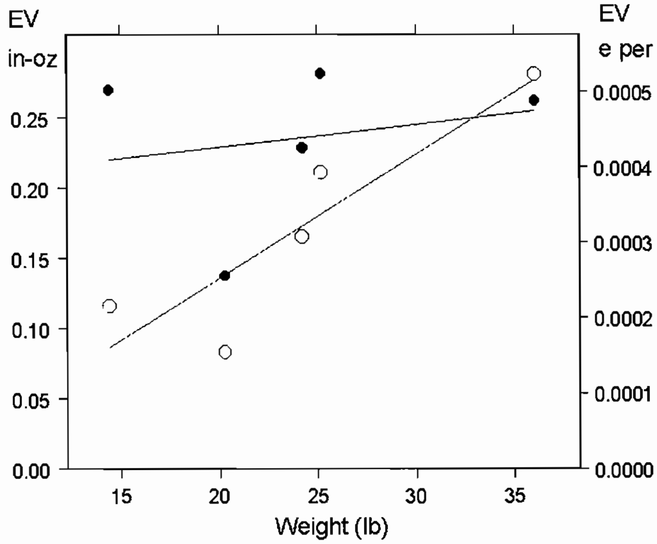

Figure 19. Example 2—EV, Measured in Both Oz-In (Open Circle) and Eccentricity (Solid Circle) Scales.

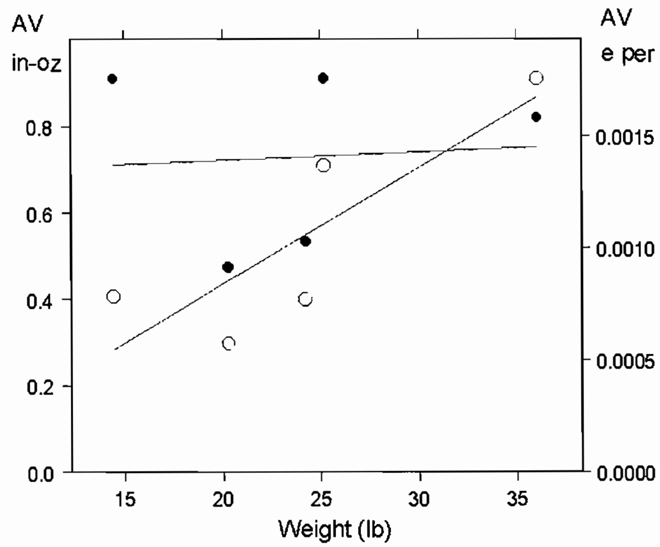

Figure 20. Example 2—AV, Measured in Both Oz-In (Open Circle) and Eccentricity (Solid Circle) Scales.

CONCLUSIONS

Despite attempts to achieve superficially low amounts of unbalance in the impeller, an overall unbalance effect from the rotor design is more significant than the unbalance contribution of the impeller. Previous research has shown that hydraulic loading contributed more to pump vibration than unbalance of the impeller.

The process used to balance the impeller, including tooling and tooling compensation, is critical in obtaining accurate and repeatable balancing. Without a well-defined and accurate process, significantly greater amounts of unbalance can be added to the rotor.

Traditional R&R studies used to quantify variation are incorrect when applied in two dimensions. A method was developed and presented that allows accurate quantification of data in a two-dimensional format.

R&R studies indicate that, with a well controlled balancing process, repeatability is approximately 58 percent of a G6.3 tolerance.

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