MULTISTAGE PUMP EFFICIENCY GAINS THROUGH THE ELIMINATION OF THE SEAL CHAMBER PRESSURE EQUALIZATION LINE

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ABSTRACT

The majority of horizontally split multistage pumps were designed when packing was the only cost effective method of sealing the pumpage from atmosphere. As a result, even modern pumps employ a stuffing box pressure equalization line (balance line) to lower stuffing box pressures. This technique of pressure equalization has a major drawback: reduced pump efficiency. The work that is done on the fluid that is throttled back to suction through the balance line is lost and acts to reduce the useful work done by the pump. While this was an acceptable compromise in the 1950s, it is now a true waste of very expensive electrical power. The modern solution to this problem is to eliminate the balance line and apply a correctly specified mechanical seal to the high pressure side of the pump. This paper serves to show that, with careful consideration of the effects of balance line removal, pump life can be maintained and efficiency gains achieved.

INTRODUCTION

A single suction, horizontally split, eight-stage centrifugal pump was chosen to show that significant efficiency gains can be achieved by closing the seal chamber balance line. The test pump had a back-to-back impeller arrangement, meaning the first four stages face the opposite direction of the last four stages. A crossover cast in the pump casing transfers pumpage from stages one through four to stages five through eight (Figure 1). The back-to-back design was developed by pump manufacturers primarily to create an axially balanced rotor. Unfortunately, there is a trade-off associated with high pressure pumps of this type.

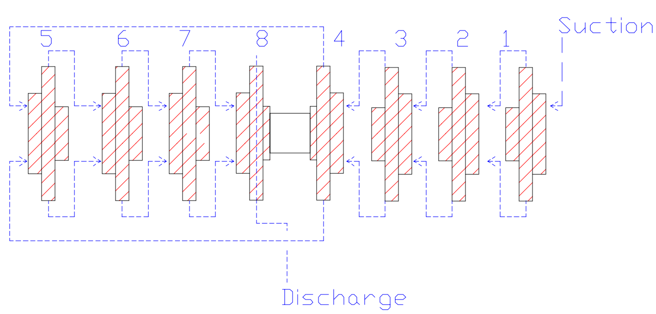

Figure 1. Common Multistage Centrifugal Pump Internal Flow Paths.

The majority of modern horizontally split multistage pumps were designed when packing was the only cost effective method of sealing the pump from atmosphere. One stuffing box (seal chamber) of the pump in a multistage design will, inevitably, be subject to high pressure. The high pressure had to be reduced to allow packing to "seal" this side of the pump. The solution was not to allow high pressure to build at one side of the pump. To relieve the pressure in the stuffing box (inboard or outboard depending on design), a throttle bushing was used to separate the casing and stuffing box on the high pressure side. The bushing accelerates a portion of the crossover flow into the stuffing box and an internal or external line is used to route this fluid back to suction (Figure 2). This passage is often referred to as the balance or stuffing box pressure equalization line. The trade-off is lost efficiency due to bypassed fluid that returns to suction. With today's modern mechanical seals, it is no longer necessary to compromise the pump design with a balance line. The following report outlines a test procedure and the results that show that it is feasible to increase pump efficiency by eliminating the balance line.

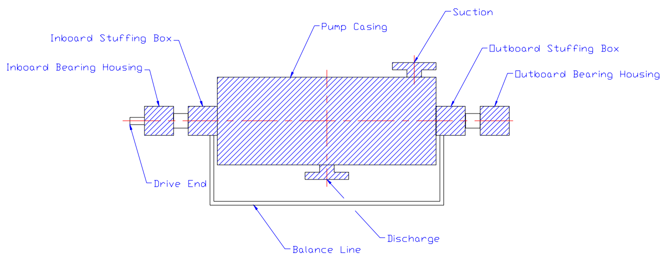

Figure 2. Common Multistage Centrifugal Pump.

PROCEDURE AND PREPARATION

The most important consideration in the pretest planning was the increased axial thrust load induced by higher pressures in the inboard stuffing box area. To quantify this, the thrust load was calculated at each operating point for both a balance line and no balance line case (refer to *Thrust Calculations* in RESULTS AND DISCUSSION section) It became obvious that modifications to the pump would be necessary to avoid premature thrust-bearing failure.

To reduce the thrust load caused by increased inboard stuffing box pressures, the last four stages of the pump were modified. Stages five through eight were fitted with larger impeller-eye and case wear rings. The factory suction eye ring diameter was 6.110 inches; it was increased to 6.781 inches on the last four impellers. Corresponding case rings were made for proper clearance (Figures 3 and 4). The larger eye rings reduced the amount of impeller shroud area subjected to impeller discharge pressure and increased the area exposed to stage suction pressure (the suction pressure of each impeller after #1 is approximately the discharge pressure of the previous stage). The modification was relatively simple and required no permanent changes to the pump (some shaft modifications might be necessary in different pump designs [refer to *Effects of Throttle Bushing Flow Reversal* section]). The new rings were constructed of bronze to reduce costs, but any material can be used to achieve similar results. Figure 5 shows the factory rotor installed in the pump casing while Figure 6 shows the modified rotor.

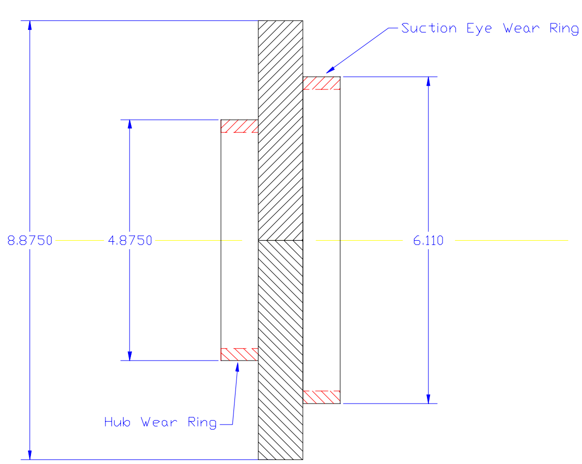

Figure 3. Representative Impeller Dimensions before Modification.

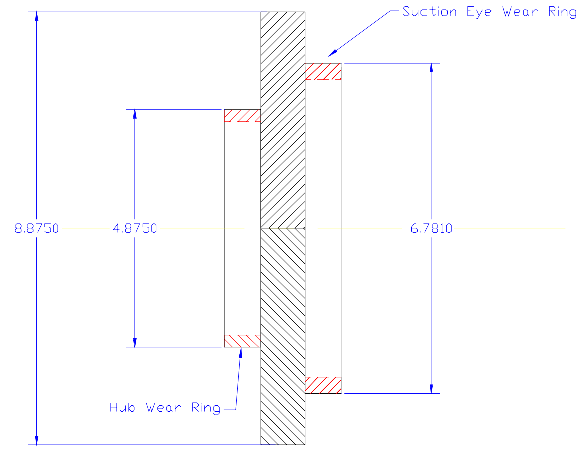

Figure 4. Representative Impeller Dimensions after Modification.

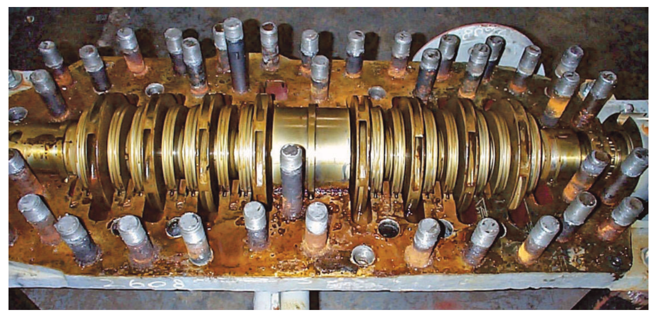

Figure 5. Pump in Factory Condition.

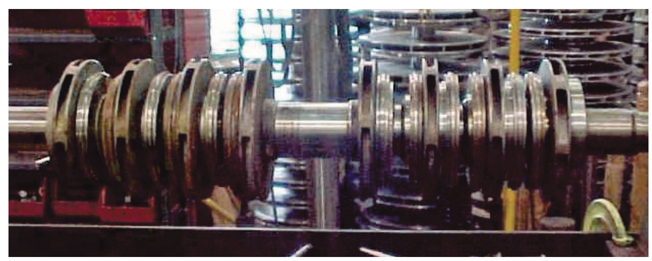

Figure 6. Modified Rotor.

The pump was tested at seven operating points from shut-off to run-out. Suction pressure, flow, discharge pressure, bearing temperature, seal gland temperature, stuffing box pressure, and efficiency were measured at each test point. Efficiency is calculated by computing the hydraulic power (Equation (1)) and dividing it by the horsepower draw of a calibrated test motor (Equation (3)).

$$Hydraulic\ Power = \frac{TDH\left[ft, H_2O\right] \times Flow\left[GPM\right]}{3960} \qquad (1)$$

The modified pump was run for 20 minutes to come to steady-state. The performance test was completed for both the balance line and no balance line case. Data were taken at roughly the same points for each test to ensure a larger degree of comparability.

RESULTS AND DISCUSSION

Thrust Calculations

Table 1 shows the results of the thrust load analysis for several cases. In the factory condition, it is clear that the pump will not operate reliably with the balance line closed. This is especially true as total dynamic head (TDH) approaches "shut-off." Columns three and four show two different seal flush conditions on the line closed modified rotor where flush pressure was altered. The test pump was run with the flush valves in the 80 percent open position in the balance line closed case as shown in Column 3 in Table 1. Note that the thrust load is increased dramatically in the balance line *open* case for the modified rotor over the factory condition (Column 2, Table 1). What is important to understand about the modified condition results is that the thrust can be reduced to acceptable levels with the balance line closed. There are several methods of thrust load analysis and the details of how axial thrust was calculated for the test pump are beyond the scope of this paper. The relationships published by Miyashiro, et al. (1980), were used to calculate axial thrust for all conditions. Lobanoff and Ross's (1986) volute/impeller velocity ratios were also utilized in the thrust calculation.

Table 1. Thrust Calculation Results.

TDH [ft-H2O]	Mod Cond: Open Line Thrust [lbs]	Mod Cond: Closed Line Thrust, 80% Open Flush Valve [lbs]	Mod Cond: Closed Line Thrust, Flush 20% Over 4th Stage Discharge [lbs]	Factory Cond: Balance Line Open Thrust [lbs]	Factory Cond: Balance Line Closed Thrust [lbs]
2560.1	2763.93	-20.70	578.81	-71.89	-3848.78
2530.1	2725.66	-23.84	568.64	-72.11	-3793.03
2360.4	2509.19	-41.61	511.13	-73.30	-3477.67
2029.7	2087.36	-76.24	399.06	-75.64	-2863.12
1725.9	1699.84	-108.05	296.11	-77.78	-2298.56
1420.7	1310.53	-140.01	192.68	-79.94	-1731.40
1131.4	941.51	-170.31	94.64	-81.98	-1193.79

Sign Convention:	(+): Tension-Thrust Toward Inboard (-): Compression-Thrust Toward Outboard

Efficiency

The results of this test are encouraging; Figure 7 shows the impact the balance line had on efficiency. When the balance line was closed, efficiency increased, on average, 3.1 points or 5.8 percent. At best efficiency point (BEP), efficiency was increased by 3.4 points. (BEP is considered to be at 490 gpm [balance line open] and 510 gpm [balance line closed], Figure 8.) Tables 2 and 3 show abbreviated results of all parameters for both tests. Note that BEP shifted from 490 gpm in the balance line open case to 510 gpm in the balance line closed case (Figures 7 and 8).

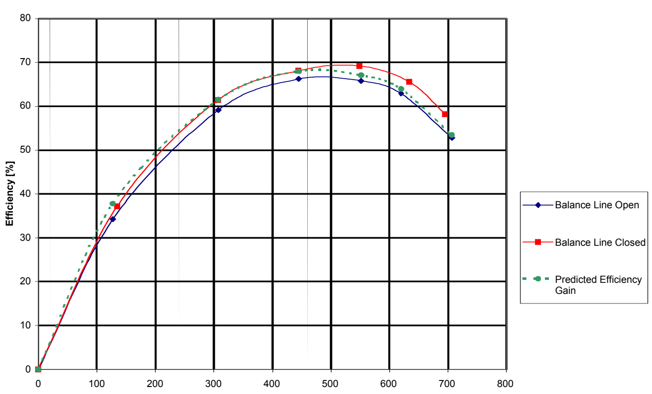

Figure 7. Balance Line Effects on Efficiency—Observed Versus Predicted.

The balance line is wasting a portion of the energy that is being drawn from the motor. The energy is lost in two ways. First, high pressure fluid leaving the fourth impeller discharge (crossover) is throttled across the inboard bushing to the lower stuffing box pressure. Potential energy (pressure head) is converted to kinetic (velocity head) and thermal (friction) energy through this process. The second way energy is wasted is by the actual mass transfer (flow) through the balance line. The pump is forced to transfer energy to the bypassed fluid twice before *useful* work is done.

Table 2. Pump Performance with Balance Line Open.

GPM	TDH	HP	EFF	RPM
0	2560.1	185.8	0	3575
127.1	2525.1	237	34.2	3575
307.3	2316.6	304.4	59.1	3575
444.9	1985.6	336.9	66.2	3575
551.2	1638.3	346.7	65.8	3575
619.7	1399.8	348.2	62.9	3575
706.5	1008.8	341	52.8	3575

Table 3. Pump Performance with Balance Line Closed.

GPM	TDH	HP	EFF [%]	Delta	RPM
0	2560.1	185.8	0	0	3575
127.1	2533.2	228.7	35.5	+1.3	3575
307.3	2359.5	298.1	61.4	+2.3	3575
444.9	2029.2	333.5	68.1	+1.9	3575
551.2	1716.7	345.4	69.2	+3.4	3575
619.7	1475.88	346.8	66.7	+3.8	3575
706.5	1136.7	340.8	58.2	+5.4	3575

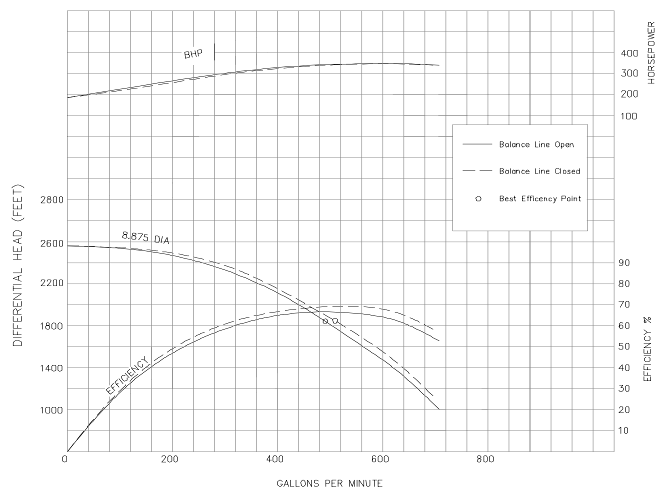

Figure 8. Test Pump Performance Curve with and without Balance Line.

To understand the shift in BEP, it is important to understand the components that determine pump efficiency. Total pump efficiency is defined by the product of the pump's hydraulic, volumetric, and mechanical efficiencies. When the balance line is closed, the pump's volumetric efficiency is increased.

The volumetric efficiency of a multistage pump is defined as the capacity of the first stage impeller versus the discharge flow rate of the pump.

$$e_v = \frac{Q}{Q + Q_L} \quad (2)$$

where:

e_v = Volumetric efficiency
Q = Discharge capacity of the multistage pump
QL = Capacity lost to leakage
$Q+Q_L$ = Capacity of the first stage impeller

Equation (2) demonstrates how closing the balance line increases volumetric efficiency. The capacity lost to leakage (Q_L) is reduced because there is no longer a balance line flow and the discharge flow of the pump is increased (Q). (*Note that there are still several sources of leakage in the multistage pumps that are not eliminated by closing the balance line.*) The denominator of the equation, $Q + Q_L$ will remain constant whether the balance line is open or closed as this is equivalent to the capacity of the first stage of the pump (Karassik, et al., 2001).

The actual shift in BEP as shown in Table 3 is due to an increase in overall pump flow similar to increasing pump speed or impeller diameter. Again, the augmented flow at BEP seen in the balance line closed test data is caused by the reduction in Q_L.

Vibration

Table 4 shows the biaxial vibration measurements taken at BEP for each test. Vibration data were taken to establish a baseline for the first (line open) test and to evaluate performance on the second (line closed) test. At BEP, overall (outboard) vibration levels decreased 28 percent in the vertical plane and 35 percent in the horizontal when the balance line was closed. The reduction in vibration in the line closed test is due to a reduction in axial thrust loading. The modifications to stages five through eight caused a much higher thrust load than normal to be applied in the line open test, resulting in relatively high vibration energy (refer to Table 1). Peak velocity at BEP in both cases did not violate API specifications.

Table 4. Summary Vibration Data—Filtered Overall Velocity Levels for Both Tests at BEP.

Test	IB Vertical [in/s-pk]	IB Horizontal [in/s-pk]	OB Vertical [in/s-pk]	OB Horizontal [in/s-pk]
Balance Line Open	0.13	0.11	0.11	0.20
Balance Line Closed	0.13	0.13	0.08	0.13

Temperature

Table 5 shows the end-of-test temperature at different locations for each test case on the modified pump. *It should be noted that the test was not run long enough to verify that these temperatures represent a maximum value*. The higher bearing housing temperatures in the balance line closed case are not considered a result of the closure of the balance line, rather, the normal warming trend of the bearings and oil reservoir with time. The total test time was less than one hour with the balance line closed test taking place last. It is expected that the balance line *open* case, with the modified rotor, would ultimately result in higher bearing temperatures than the balance line closed case.

Table 5. End-of-Test Temperature for Both Tests.

Location	Balance Line OPEN	Balance Line CLOSED
OB Bearing Housing	100 °F	110 °F
IB Bearing Housing	98 °F	103 °F
IB Seal Gland	97 °F	103 °F
OB Seal Gland	99 °F	111 °F

Axial Thrust Considerations

Besides efficiency gains, there were two main concerns in conducting this test and applying the technology to future applications. It would be pointless to explore options to improve pump efficiency if reliability or stability were to decrease.

The first concern was increased axial thrust load induced by the higher pressures in the inboard stuffing box. Figure 9 displays the dramatic increase in thrust when the balance line is closed. In the balance line open position (factory condition), thrust load is low and relatively flat across the differential head range. The inboard (high pressure) stuffing box pressure remains roughly constant due to the increasing flow through the balance line as differential head increases. Stuffing box pressure on the inboard side causes the majority of the thrust load in the balance line closed condition. As differential head increases in the line closed condition, there is no way for the stuffing box to relieve the building pressure from the fourth stage and the thrust is increased to a maximum just before shut-off (left side of Figure 9).

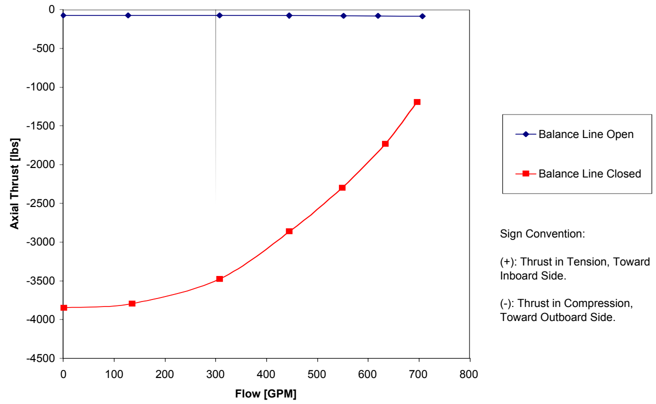

Figure 9. Factory Condition Pump Thrust Calculation Results.

It is clear that closing the balance line without modifications to the pump would significantly reduce its life. The modifications discussed previously in the PROCEDURES AND PREPARATION section were completed and the resultant thrust load is shown in Figure 10 and Table 1. In the modified condition, higher thrust loads were experienced in the balance line *open* case. This was expected and vibration data support this calculation. The test was conducted with the flush control valves positioned at 80 percent open. Axial thrust under these conditions was very low and in compression. At BEP, theoretical thrust load is as low as 108 lb. L10 bearing life calculated at BEP yields a predicted service life of over 100 years (Figure 11). Clearly, the bearings will not be the source of failure in a pump running in this condition.

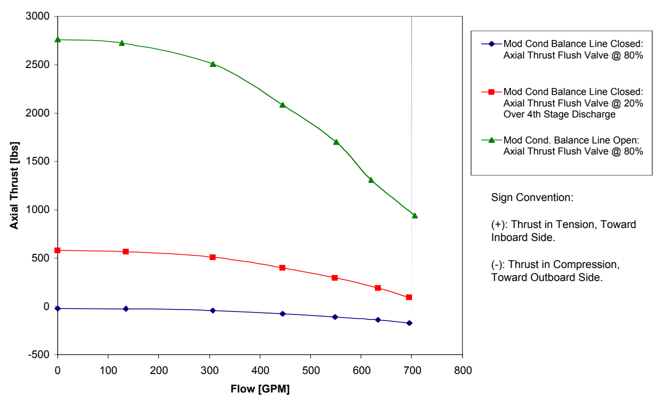

Figure 10. Modified Pump Thrust Calculation Results.

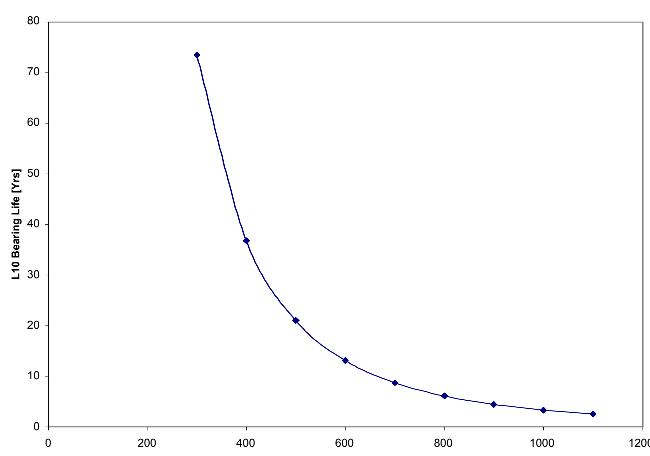

Figure 11. L10 Life of Double Row, Deep Groove Thrust Bearing (with Constant Radial Loading).

Figure 10 also shows a third thrust load curve. This curve was calculated based on a reduction in inboard stuffing box pressure by reducing the pressure of the flush line. The theoretical inboard stuffing box pressure required to maintain an adequate flush to the seal is 20 percent over the fourth stage discharge (crossover pressure). With the lower stuffing box pressure, the rotor is placed in tension and loaded to approximately 300 lb at BEP. The inboard flush pressure can be altered by changing the position of a flush line valve or through the installation of a properly sized orifice. It is typically preferred that the rotor in a multistage pump have some axial thrust to keep it from shuttling from inboard to outboard during a process variation. When the flush pressure valve is at 80 percent open, the thrust on the rotor is very low and it is conceivable that secondary flows may cause the rotor to shuttle axially and prematurely wear the thrust bearing.

Mechanical Seals

Sealing a high pressure multistage pump with no balance line is a challenge that, until recently, could not be met cost effectively. This is the second major concern of closing the balance line in a multistage pump. The outboard (suction) stuffing box pressure will not change significantly when the line is closed. However, the inboard stuffing box pressure will be raised to nearly discharge pressure, depending on the source of the seal flush. A cartridge-mounted single mechanical seal was selected for the tests both with and without a balance line. Double and tandem arrangements will be more common in field installations. The seal has a balanced design with antimony-impregnated carbon versus tungsten carbide faces. Because the inboard stuffing box pressure was expected to rise to at least fourth stage discharge, the Plan 11 flush was sourced from the discharge of the pump. This allowed the inboard seal to receive an appropriate amount of cooling flush even when the stuffing box pressure increased on the balance line-closed test. Needle valves were installed in the flush lines to each seal to regulate the flow. During the test, the seal temperature remained consistently below normal operating temperatures of 115 to 120°F (Table 5).

There are inherent safety and design issues that need to be addressed when dealing with higher stuffing box pressures. Obviously, a tremendous number of high pressure mechanical seals of different configurations have successfully been applied in the last 10 years. Many of these designs operate with the same reliability as lower pressure seals with the natural difference being cost.

While no high pressure mechanical seals have been tested extensively in this particular application, there are many hundreds of single and double/tandem mechanical seals running reliably in high suction pressure applications. The removal of the balance line, from a sealing perspective, is very similar to a pump operating at high suction pressure as seen in some pipeline and carbon dioxide (CO_2) applications. The high suction pressure causes a high stuffing box pressure in both the inboard and the outboard side of the pump. High stuffing box pressures can also be seen in vertical multistage pumps even with low suction pressures. Therefore, mechanical seals have been, and will continue to be, applied in high pressure stuffing box environments with reasonable reliability.

Balance Line Flow and Efficiency Gain Prediction

The experimental efficiency gains do not follow the predicted trend of higher gains at higher differential heads and lower gains at lower heads. Figure 7 shows the efficiency versus flow for the experimental line open, predicted line closed, and experimental line closed cases. The prediction was based on the amount of leakage across the inboard throttle bushing. Bernoulli's principles can be applied to roughly calculate the bushing leakage (Equation (4) (Stephanof, 1957). Since the flow across the bushing represents the bulk of the balance line flow, it is an elemental calculation to determine the effect it has on efficiency. Equation (3) was used along with line open data to predict the efficiency increase when the line was closed.

$$Pump\ Efficiency = \frac{Hydraulic\ Power}{Electric\ Power} \tag{3}$$

The theoretical (predicted) efficiency curve (Figure 7) shows larger efficiency gains toward shut-off. This is because the pressure differential across the throttle bushing is much higher at shut-off than run-out. Flow is dependent on the pressure at each end of the bushing and it follows that higher differential pressures result in greater bushing leakage. However, as stated above, the test results did not support this prediction. The results show higher efficiency gains at higher flows (lower pressures). This trend had not been fully explored at the time this report was written. A computational fluid dynamics (CFD) analysis and further testing will be necessary to understand what is causing the observed phenomena.

$$H_L = \left(f\frac{L}{d} + 1.5 \right) \frac{v^2}{2g} \tag{4}$$

where:
H_L = Head loss across the inboard throttle bushing in ft
f = Friction factor of the annular clearance based on Stepanoff's (1957) experimental results
v = Velocity of the fluid passing through the throttle bushing (this is the unknown and what is used to determine bushing flow or leakage)
L = Length of the throttle bushing in ft
d = "Hydraulic diameter of the throttle bushing" (equal to bushing radial clearance in ft)
g = 32.3 ft/sec (constant)

While there are many factors that affect the efficiency gains that were realized in this test, the prediction method that was used should serve as a conservative forecast to the "real world" performance.

Specific Speed and Number of Stages

As a general rule, the efficiency gains associated with closing the balance line will be maximized on lower specific speed pumps when compared to similar higher specific speed units. Low specific speed impellers generate a relatively large amount of head versus flow (Equation (5)). For the removal of the balance line in a multistage pump to make sense, the balance line flow in the unmodified pump should be a relatively large percentage of total pump flow. Since balance line flow (throttle bushing leakage) is primarily a function of crossover and suction pressure, lower specific speed pumps will have higher flow rates through the balance line. In the test unit, the estimated balance line flow in the factory condition case is between 1.6 percent and 12 percent of total pump flow (3.2 percent at BEP).

$$Ns = \frac{RPMx\sqrt{GPM}}{TDH^{\frac{3}{4}}} \tag{5}$$

where:
GPM = Evaluated at BEP and maximum impeller diameter
TDH = Evaluated at BEP, maximum impeller diameter, and per stage

Efficiency gains are also directly proportional to the number of stages. Multistage pumps with more stages will have a higher crossover pressure and, therefore, higher balance line flow when compared to similar pumps of fewer stages. The test pump had eight stages and it follows that the efficiency gains on a similar 10- or 12-stage unit would be larger.

Effects of Throttle Bushing Flow Reversal

When the balance line is closed, the flow through the throttle bushing will reverse direction. Flow through the throttle bushing is from outboard to inboard (toward high pressure stuffing box) in the unmodified, balance line open case (Figure 12). Conversely, when the line is closed, fluid will travel from the high pressure stuffing box back into the pump (inboard to outboard) (Figure 13). This is due to the seal flush pressure that is now higher than the crossover pressure by at least 20 percent. Since all multistage pumps with a back-to-back impeller arrangement were designed to have fluid moving toward the stuffing box through the throttle bushing, some manufacturers only trap the throttle sleeve in one direction. This can pose a problem when the balance line is closed and the throttle bushing flow reverses.

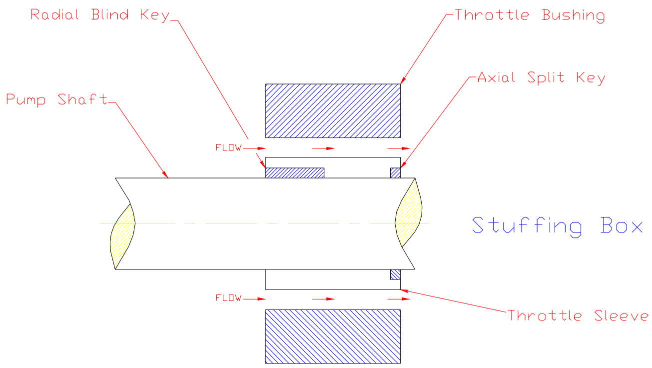

Figure 12. Typical Throttle Bushing Design and Leakage Flow Path on Pumps with a Balance Line.

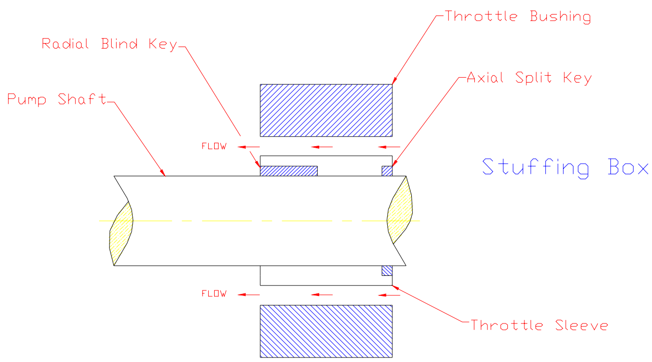

Figure 13. Throttle Bushing Leakage Flow Path on Pumps without a Balance Line (Bushing Shown Before Required Modification for Flow Reversal).

The throttle sleeve (Figures 12 and 13) is the dynamic member of the throttle annulus and rides inside the throttle bushing. Some designs use a split axial key to keep the throttle sleeve from moving. The sleeve is installed on the shaft and has an axial keyway that mates with the split key. In the balance line closed case, the flow reversal through the throttle annulus might cause the throttle sleeve to back off the axial key and severely damage the pump. Most all designs of this type also incorporate a radial key in the throttle sleeve to ensure it does not rotate relative to the shaft during operation. This radial key is typically blind so that high pressure fluid cannot bypass the throttle annulus through the keyway. The blind radial key would indeed trap the sleeve in the opposite direction of the axial key and prevent the throttle sleeve from moving when the balance line is closed. However, this is not an acceptable method of axially trapping and locating a sleeve on a high energy centrifugal pump.

Therefore, if the balance line is to be closed on multistage pumps of this type, the throttle sleeve must be trapped utilizing an approved method. This may or may not involve changing the shaft design of the pump.

There are also manufacturers that use countersunk setscrews or a threaded sleeve/shaft to secure the throttle sleeve to the shaft. These designs are inherently bidirectionally trapped and will not need to be altered for closed balance line operation. The test pump utilized a set-screwed throttle sleeve and performed acceptably for the duration of the testing.

Performance Matching

Closing the balance line does have a positive effect on both flow and developed head. This might present a problem to an end-user who simply wants to save money on electricity while remaining at the same rated flow and head as their factory pump. The modified pump will produce more flow at a given system head and the customer would not see any noticeable reduction in energy costs. The modified pump would draw marginally less if not the same amount of power from the driver as the factory pump in this condition. To ensure that the modified pump will perform acceptably in this case, a slight impeller trim will be required. It is expected that this reduction in impeller diameter of some or all the impellers will reduce the efficiency gains from closing the balance line by 25 to 30 percent, depending upon impeller hydraulic design and magnitude of impeller trim. However, the impeller trim will enable the end-user to realize some energy savings without system modification or increased pump throughput.

If the end-user has driver speed variability through a variable frequency driven motor, steam, or gas turbine, the negative effects of performance matching can be completely eliminated. Instead of trimming the impeller to match previous performance of the pump, the speed of the driver can be reduced slightly until the pump performs as it did before the balance line was removed. The full benefit of removing the balance line can then be realized as reduced electricity, steam, or fuel consumption.

Economic Impacts

Ultimately, cost savings will drive the development and implementation of this technology. Table 6 is an estimate of the annual savings versus the cost of modification for pumps similar to the test unit. Over an estimated 10 year life of the pump, the savings are considerable. All calculations are based on an energy cost of eight cents per kilowatt-hour. It is easy to see that a plant with 10 to 20 high-energy pumps would see a significant impact on their energy bill even on a monthly scale. These are savings that are literally being thrown away with old technology.

Table 6. Annual Savings and Cost of Modification.

HP	Annual Gross Savings	Cost of Modification (less Mech. Seals)	Total Savings Before Overhaul (Based on 10 Yr Life)
250	$4,834	$8,000	**$40,340**
500	$9,668	$12,000	**$84,680**
750	$14,502	$20,000	**$125,020**
1000	$19,336	$25,000	**$168,360**

The removal of the balance line in multistage pumps is most ideally suited to end-users who would like more capacity and/or discharge pressure out of their existing units. Tables 2 and 3 show that for a given flow or TDH the modified pump outperforms the factory condition unit and often at a lower horsepower requirement. This would be useful for pipeline operators that are often paid by the barrel of transported product.

CONCLUSIONS

The multistage pump balance line is truly an antiquated feature on some of today’s pumps. High-energy pump efficiency can be effectively augmented through the elimination of the balance line. The effects of higher stuffing box pressure, throttle bushing flow reversal, and axial thrust load can be dealt with in the modification process. With these issues adequately addressed, there is no reason pumps without balance lines cannot run as reliably as similar units with balance lines. Truly, this modification is not ideally suited for *all* multistage pumps and care must be taken to properly select candidate pumps to justify costs. Pending successful extended field trials and analysis, balance lines can be eliminated from many multistage centrifugal pumps to increase efficiency and reduce operating costs.

Precautionary Note

The fluid patterns that generate the axial hydraulic thrust in multistage pumps are highly complex. The pressure distributions acting on the front shroud and the back shroud of each stage impeller are heavily influenced by the leakage (amount and direction, i.e., radially inward or outward). Therefore the prediction of the hydraulic axial impeller forces requires a sufficiently accurate computer model for the leakage influence. Moreover, the leakage effect is not yet fully understood. This means that even more sophisticated theoretical calculations of axial thrust are determined under several assumptions that require experimental validation. It is highly recommended that theoretical predictions of axial hydraulic thrust in multistage pumps, including configurations with opposite impellers (apparently autobalanced), following design changes such as impellers and bushings, are backed up with actual measurements.

The end-user should always take care, with any modification, to adequately account for factors that affect pump reliability and performance. Considerations such as axial thrust, internal flow reversals, mechanical seal selection, and mechanical seal flush selection are all very important in performing the modifications discussed in this paper. If these factors are not specifically addressed, it is likely that pump reliability will be decreased.

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