DEVELOPMENT OF A DIFFUSER PUMP FOR PIPELINE SERVICE

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ABSTRACT

This paper reviews Trans Mountain's decision to use diffuser pumps for pipeline service. This includes a review of the basic design parameters and the experience gained with wear rings, mechanical seals, bearings, lubrication and baseplate design during six years of operation.

NOMENCLATURE

BEP  Best Efficiency Point
CPM  Cycles Per Minute
CSt  Centistokes
Hz   Hertz (cycles per second)
IPS  Inches Per Second
NPSH\(_R\) Net Positive Suction Head (Required)
Nss  Suction Specific Speed
RPM  Revolutions Per Minute

INTRODUCTION

In the spring of 1988 Trans Mountain initiated engineering for a major expansion of the pipeline's capacity. The expansion plans included the construction of three new pump stations, which required the procurement of seven 1100-1500 kw (1500-2000 Hp) main line pumps.

This expansion would also require the modification of all existing pumps to ensure that their operating range complimented the new operating conditions. Such modifications have historically included removal of the pump from service, for modification of the casing volute tongue(s), and replacement or modification of the rotating element. These modifications are costly and require removal of the very equipment needed to increase capacity. Depending on the magnitude of the modification, the work can take months to complete on each pump.

As shown in Fig. 3, Trans Mountain's pipeline capacity and average throughput have varied significantly during forty years of operation. As of the spring of 1988, the pipeline was operating at 40% of its previous maximum capacity. To provide for future expansion and minimize the time required for pump conversion, it was decided to replace all existing main line pumps with a new variable flow design.

Trans Mountain has completed a second capacity expansion program and the ability to modify the pump internals will reduce power costs. This saving is expected to provide a seven year payback on the change-out costs. It would have cost an additional $250,000 to make these changes by modification of the pump volutes.

PUMP DESIGN

A specification was developed for a horizontally split case, double suction, between bearings design with horizontal suction and discharge nozzles. The specification required the pump to include removable volute inserts or diffuser section to allow the pump to operate at optimal efficiency at any capacity within the range specified. The specification was submitted to six pump vendors. Four vendors proposed a diffuser style design and two proposed double volute cases with removable volute tongue inserts. Sulzer Bingham's model HSD diffuser pump was selected based on price and technical merit.
The Diffuser Design

A centrifugal pump volute collects the liquid discharged by the impeller and converts velocity energy into pressure energy. In a diffuser pump, a series of curved vanes around the impeller serves the same function as shown in Fig. 1. The diffuser vanes also guide the fluid evenly through the case. Diffusers can simplify casing design, or as in the case of the HSD, permit the use of an oversized pump case to allow for future capacity increases or decreases.

Another advantage of the diffuser design is that the resulting flow distribution minimizes the hydraulic imbalance and the resultant radial loading on the pump shaft. As indicated by Nelson (1980), a double volute design does not necessarily ensure a balanced impeller load. "Since the volutes are cast into the case and no machine work is done on them they tend to have uneven surfaces and are irregular in shape and therefore, the hydraulic balancing does not occur. These irregularities can cause shaft deflection of a magnitude greater than the wearing ring clearances. This results in the rapid deterioration of wearing rings, bearing failures and shaft breakage".

Diffuser pumps are not commonly used in hydrocarbon applications, in spite of their capacity modification ability. As noted by Karassik et al (1976) and Steffeck (1981), the design is used primarily in multistage applications requiring high pressure and high flow, such as boiler-feed pumps. Hamreus and Deardorff (1985) suggest that "the higher initial cost is probably the one reason that the diffuser method has not been used by any major pipeline company, even though the future costs could possibly prove to be less than other methods due to the ability to maintain good efficiency over the complete range of capacities".

Efficiency and Suction Specific Speed (Nss)

Efficiency was another important design parameter. The desire for high efficiency was tempered by recognition of the relationship between pump efficiency and Suction Specific Speed (Nss) and their effect on range of operation. Hallam (1983) notes "Generally, pumps with higher Nss numbers must be operated closer to the BEP than pumps with lower Nss numbers". Fraser (1981) notes that "the closer the discharge recirculation capacity is to the rated capacity, the higher will be the pump efficiency", and "this unremitting drive for high efficiencies and high Nss has created designs that are very limited in their range of operation". Lobanoff (1985) also shows a dramatic decrease in stable operating range with increasing Nss.

A broad, stable operating range remains a prime consideration for Trans Mountain due to the variety of fluids transported. The pipeline transports materials as diverse as heavy crude oil with a viscosity range of 200 to 350 CSt to refined products and condensates with comparatively high vapour pressures. A large batch of heavy crude oil will reduce pipeline capacity 40% during its transit through the system. The pumps also have to be sized for a design or "peak" capacity at least 10% above the "sustainable" capacity to allow for make-up of lost throughput due to emergency outages and maintenance "shut-downs".

Predicting a pump's stable operating range is not straightforward. Nelson (1980), Lobanoff (1985) and Taylor (1977) state that standard "3% head-drop" NPSHr tests are poor indicators of operating stability for high head (200m/stage) or high Nss pumps. Nelson (1980) notes that narrow boiling point liquids such as gasoline and condensates can induce damaging vibration levels at flow rates well above that where recirculation cavitation has progressed sufficiently to cause impeller damage. Taylor (1977) adds that "excessive cavitation can occur at low flows on high energy pumps without significant performance impairment".

To ensure stable pump operation, most authorities recommend limiting the pump Nss. Based on a five year study of 480 refinery and petrochemical pumps, Hallam (1983) suggests a maximum Nss of 11,000 while Nelson (1980) indicates that for Nss between 10,000 and 12,000 the application requires "careful review".

The HSD pump selected has an Nss of 10,500 and has operated successfully within the vendor's recommended operating range of 50% to 120% of the BEP flow of 1250 m³/hr (5500 gpm). Pump performance curves are shown in Fig. 4.

OPERATING EXPERIENCE

One expression of the stable operating range of the different internals expressed as a function of unfiltered vibration is shown in Fig. 5. As the pump operates further away from the rotating assembly's BEP, the overall vibration level increases. At these elevated vibration levels, spectral analysis shows no discrete vibration frequencies. The spectrum shows only intermittent peaks occurring between 20 and 50 Hz similar to those shown in Fig.6.

Pump Internal Clearances

Careful design of the pump's internal clearances and impeller have helped to provide a robust pump in spite of the double suction design and the elevated Nss. As described by Makay (1978), "As an impeller vane passes by a stationary blade (diffuser tip, or volute tongue) a hydraulic shock occurs that can be observed in the liquid, on the structure, or can be noticed on the rotor vibration measured at the bearing or any part of the shaft". To minimize this "hydraulic shock" a staggered impeller vane design was specified and the internal clearances shown in Fig. 10 were kept within recommended tolerances.
Impeller/diffuser interaction can cause vibration in both the lateral and axial directions (with respect to the pump shaft). Axial vibrations result from the impact of pressure pulsations (created by hydraulic shock) on the outer surface of the impeller shroud. Powers (1995) indicates that these pulsations are effectively choked by Gap A clearances of between 0.9 mm (0.035") and 2.2 mm (0.085"). When an HSD impeller is trimmed, only the vane tip is modified. The impeller shroud is left at full trim and a radial clearance of 1.5 mm (0.060") is maintained between the shroud and the diffuser.

Gap B, the radial clearance between the diffuser and impeller vane tips, controls the strength of the pressure pulsations. Gap B clearances of 3\% to 5\% of the impeller diameter have been used. Tighter clearances were attempted during development testing of second generation impellers, but this resulted in an unstable performance curve with a steep head rise to shut-off, as predicted by Makay (1978) and the impeller trim had to be reduced. Recently some impellers were trimmed increasing Gap B from 3\% to 6\%. Although the original trim operated successfully, there was a marked reduction in vibration.

**Pump Case Stiffening**

After the pumps had been placed in service, a minor leak was observed at some of the casing studs on two of the pumps. The casing split line gasket had eroded and fluid had migrated along the casing studs. Subsequent examination revealed that the casing had distorted during hydrostatic testing and the positive "crown" on the casing split line flange had become negative. As shown in Fig. 11, crowning is the machining of the pump case to insure that the inner edge of the split line flange meets before or at the same time (neutral crown) as the outer edge. A negative crown causes the outer edges to meet first and can prevent mating of the inner edges of the split line.

The pump vendor conceded the design weakness and developed a modification to control the distortion. The modification included the installation of six (6) stiffening vanes as illustrated in Fig. 1. It was found that the vanes successfully controlled casing distortion, while improving pump efficiency. The modifications were undertaken at the pump vendor’s cost and took approximately one year to complete.

**Impeller Wear Rings**

Minimizing the deterioration of impeller wear ring clearances and the attendant drop in pump efficiency was another objective in the development of the pump. Trans Mountain had previously experimented with the concept of notching the wear rings, as shown in Fig. 2, with good success. The objective of the notches is to provide an alternate path for erosive particles to "slide" through. Therefore, the new pumps were ordered with two diametrically opposed, diagonal notches on the impeller wear rings.

Twelve pumps have been examined after four years of service and ten had clearances equal to or less than API 610 maximums. Two pumps had larger clearances which were attributed to high vibration caused by impeller imbalance. While the notches appear to be successful, low pump vibration levels and tight Gap A clearances may have also contributed to the preservation of wear ring clearances.

**Mechanical Seals**

The greatest problem experienced with mechanical seals was in the seal flush arrangement. The function of the seal flush is to dissipate the frictional heat developed at the seal faces. The flush arrangement was a modified API Plan 11 configuration. The flush originated at connections adjacent to the pump discharge nozzle and provided pumped fluid for both bearing cooling and seal flush.

The configuration proved unsatisfactory as it was difficult to balance the flow between the two demands. To correct the problem, new flush source connections were drilled as shown in Fig. 10. This new location appears to have provided a cleaner source of fluid due to the "centrifuge" like action of the impeller and has improved seal life significantly.

**Bearings and Lubrication**

The pump specification required hydrodynamic radial bearings on both the drive and non-drive ends and back to back mounted, angular contact, thrust bearings on the non-drive end of the pump. Anti-friction bearings were specified for the thrust bearing in spite of API 610’s (1989) requirement for hydrodynamic bearings at rated horsepower over 750 (@ 3600 rpm), because they had proven successful on the existing pumps.

The lubrication system has been kept as simple as possible, as 50\% of Trans Mountain’s pump stations are remote unmanned sites and only two are manned 24 hours a day. The lubricant is delivered to the bearings by "slinger" rings. The pumped product is the only cooling medium available at most stations, without resorting to forced air cooling, and it is circulated through small finned heat exchangers located in the bearing housing reservoirs to cool the lubricant.

Initially problems were encountered with high bearing temperatures. The pump vendor had provided bearings with internal clearances of 0.10 mm (0.004") on a 88.9 mm (3.5") shaft diameter. When these clearances were increased to 0.15 mm (0.006") the bearing temperatures dropped by 15°C. High temperatures were still experienced on some variable speed applications at shaft speeds greater than 3600 RPM so these pumps were modified to incorporate forced lubrication systems.
Baseplate Design

Upon installation of the new pumps, high vibration levels were experienced at some pump stations during pump start-up. These vibration levels ranged from 0.6 to 1.0 inches\(^2\)\(v\) for about 30 seconds, until the pump discharge valve was substantially open (fixed speed pumps are started against closed discharge valves). These vibrations were limited to a frequency range between 20 and 50 Hz, as shown by the cascade or "waterfall" plot in Fig. 6.

This vibration phenomena was associated with flow conditions well below the recommended minimum of the pump. This problem was alleviated by attenuating the vibration monitors for a fixed time during start-up and shutdown. Of greater concern was the tendency for the pumps to trip off on high vibration during upset conditions (e.g., loss of an upstream station), and then have this problem cascade down the pipeline.

Structural Dynamics Research Corporation (SDRC) was retained to evaluate the problem. The analysis found that a "system" resonance was being excited (system being a collective term for the pump, baseplate, foundation and piping). This resonance took the form of a torsional mode of vibration of the pump about its vertical axis, at 44 Hz (2640 cpm) as shown in Fig. 12.

The pump baseplate was modeled using finite element techniques and an in situ modification was developed. As shown by the frequency response function in Fig. 7 (this function is proportional to the reciprocal of stiffness), the modification increased the resonant frequency by approximately 50% to 69 Hz. This was sufficient to reduce the low flow vibration to acceptable levels as shown by start-up vibration data in Fig. 8 & 9.

In the latest expansion Trans Mountain worked with the pump vendor to develop a baseplate design that has a structural stiffness exceeding 5 x 10\(^6\) lb/in. in the lateral and axial directions and 2.5 x 10\(^6\) lb/in. in the vertical. This design has been used in five different locations and found to be quite effective in minimizing resonances.

CONCLUSIONS

Diffuser pumps are recommended for pipeline applications where significant changes in throughput are anticipated. Careful execution of the design and internal clearances can result in a robust pump with a broad stable range of operation. Gap A and B clearances should be specified in the purchase of large pumps. Optimizing these clearances can extend the stable operating range of a high head pump.

Although the horizontally split case pump presents an extended gasket joint, it is often preferred for ease of access to pump internals. Due to the complex nature of the joint it is recommended that a witnessed disassembly of the pump after hydro testing be included as part of acceptance testing. During this examination the crown on the split line flange should be measured to confirm its effectiveness.

Wear rings and mechanical seals merit special attention in services containing abrasives. The traditional approach is limited to specifying hard surfaces for both wear rings and mechanical seal faces. A recommended complementary strategy is to deal directly with the abrasives. The seal flush should be reviewed, to determine if a "cleaner" source is available. This source may be located at some point where centrifugal action tends to exclude particulate matter. Notching of the wear rings should also be considered to minimize the accumulation of abrasive matter at the wear rings.

Supply of lubricant by slinger rings is far simpler than by a forced lubrication system. The inherent simplicity is considered to be more reliable. In spite of this advantage, forced lubrication should be given serious consideration when rated horsepowers exceed 1100 kw (1500 Hp) at 3600 RPM and particularly when bearing peripheral speeds exceed 15 m/s (50 ft/s).

Baseplate designs for large pumps should be reviewed carefully. The original installations discussed in this paper had a lateral stiffness of 1 million lb/in and this has been increased to 5 x 10\(^6\) in later designs.

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**Figure 1:** Casing Discharge Volute

**Figure 2:** Impeller Wear Ring Notch
Figure 3: Average Annual Pipeline Throughput

Figure 4: Pump Performance Curves
Figure 5: Unfiltered Pump Vibration Vs Flow Rate

Figure 6: Pump Start-Up Spectra
Figure 7: Frequency Response Function, Hor, NDE, Bearing Housing

Figure 8: Overall Vibration @ Start-Up (Prior to Modification)

Figure 9: Overall Vibration @ Start-Up (After Modification)
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