Benefits of Conducting Periodic Critical Pump Hydraulic and Mechanical Performance Audits

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Abstract

The operability and efficiency of critical pumping equipment is essential. A cost-effective tool that has proven valuable in predictive and preventive maintenance and in the avoidance of unscheduled pumping equipment outages is a comprehensive program of periodic hydraulic and mechanical performance audits.

This paper explains how hydraulic and mechanical field testing can be conducted without interrupting plant operations using non-intrusive measurement equipment including: ultrasonic flow measurements, vibration signature analysis, and for electric motor driven pumps, power and dielectric condition analysis. Methods and requirements to conduct the audits will be discussed and a case study with cost benefit analysis is presented.

Pumps are basically energy transfer devices. The energy of the pump driver, e.g. electric motor, steam or gas turbine, reciprocating engine, is transferred into the pumpage to move it through the system at sufficient pressure to overcome system losses and meet process requirements. It is the efficient transference of the applied energy that is a critical consideration in pump design and operation.

Efficiency of operation is of interest to everyone. The pump designer is motivated to maximize pump efficiency to remain competitive given the increasing cost of energy. However, the designer must not get overly zealous in the quest for pumping efficiency sacrificing reliability and durability. The end-user wants the most efficient pump with longest mean time between repair (MTBR) intervals to both reduce operating cost and minimizing production losses.

With the reasons for maximizing pumping efficiencies and longer MTBR intervals established, the value of periodic in situ performance testing or auditing becomes an increasingly valuable and cost effective predictive and preventative maintenance tool.

Centrifugal Pumps Terminology

Centrifugal pumps (Figure 1) are the most common pump type and are the primary focus of this paper; however, many of the techniques, procedures and benefits described herein equally apply to displacement type pumps including: reciprocating, diaphragm, and rotary.

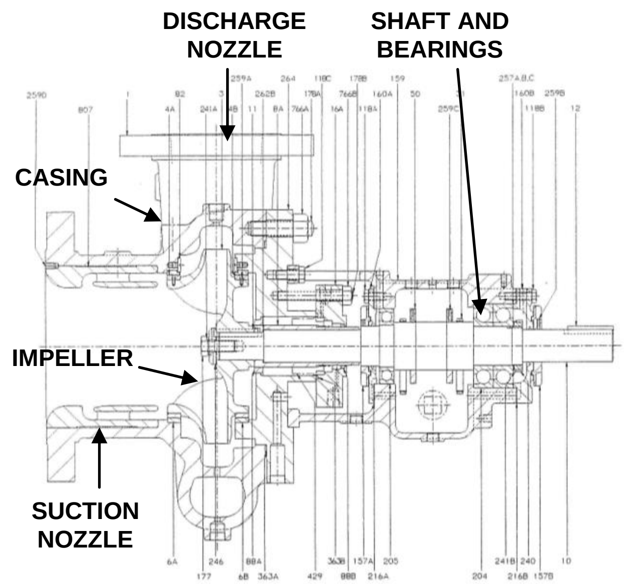

Figure 1 – Typical Centrifugal Pump

Centrifugal pumps are typically designed based on the application and mechanical requirements. Despite the design differences the basics of operation are the same. Every centrifugal pump is designed to impart energy to the liquid and subsequently increase the pressure of the pumpage to higher levels and overcome system resistance to move the liquid. The pumps consist of three principal components, an impeller, shaft and casing. The impeller mounted on a shaft accelerates the liquid (imparts velocity head) and the pump casing both guides and decelerates the liquid (converting the velocity head to pressure head).

The pump casing also provides support for the bearings and houses the entire rotating assembly. The shaft is sealed where it protrudes from the casing to prevent external leakage. Methods of sealing include both mechanical seals and packing.

Renewable stationary rings (Figure 2), also referred to as casing wear-rings, are mounted and fitted in the casing to control leakage of high pressure developed by the impeller back to the impeller inlet. Excessive leakage through the wear-rings is typically referred to as recirculation.

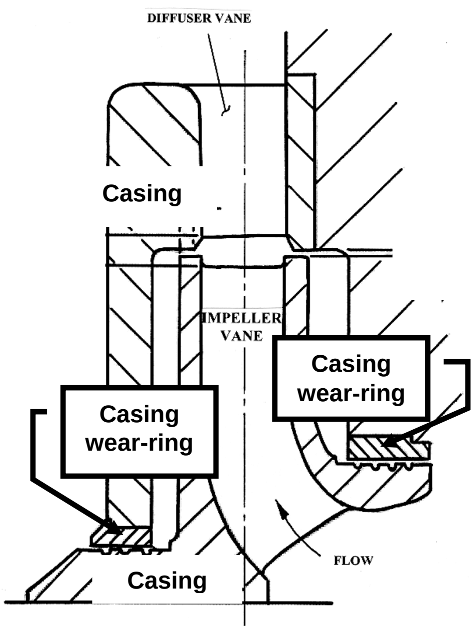

Figure 2 – Impeller Wear-Rings

In multistage pumps recirculation affects the efficient transference of stage pressure from one impeller to the next decreasing the total pumping efficiency of the unit.

Pump Characteristics

Typically pump manufacturers provide with the pump a set of performance or characteristic curves (Figure 3). The curves are developed from empirical data acquired during the testing at the pump manufacturer's facility and provide the end-user with a guide on how the pump will operate over its capacity range.

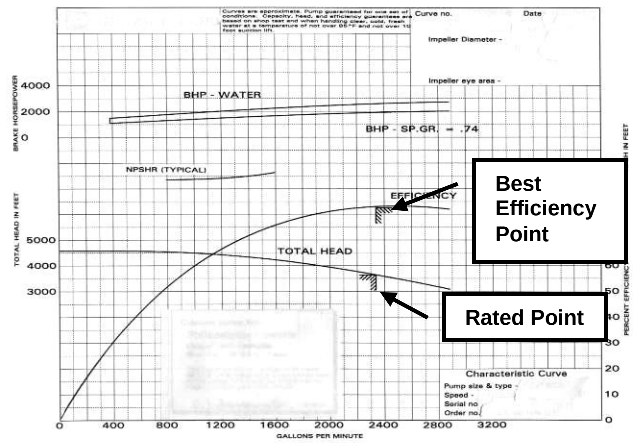

Figure 3 – Typical Pump Characteristic Curves

Pump performance curves plot the variations of the total dynamic head (TDH) representing the net work done on a unit weight of liquid in passing from the pump suction to the discharge developed by the pump at a constant speed over the capacity of the pump.

The pump capacity (Q) is the volume of liquid per unit of time delivered by the pump. In addition to the TDH-Q curve, a complete set of curves includes the brake horsepower (BHP) and pump efficiency (η). BHP is the total power delivered to the pump shaft (water horsepower less the mechanical and hydraulic losses) and the pump efficiency is calculated from the liquid horsepower divided by the brake horsepower (Refer to Figure 6).

Unless the pump is specifically designed for the system it is intended to operate in, variation in a pump's installed performance is not uncommon. Typically, an installed baseline performance audit is conducted and the manufacturer's provided pump curves are adjusted accordingly. However, if a baseline audit was not conducted, comparison to the original manufacturer provided curves is still of significant value for trending a pump's performance.

Measurement of Field Parameters

Capacity

Flow measurement devices typically require sensors of the device to be in contact with the flow or "wetted". The most common are the use of differential pressure devices requiring the installation an orifice or venturi into the system and, unless already installed, are impractical as portable field measurement instruments. Ultrasonic flow measurement provides a nonintrusive method and has become the dominant portable technology. Several manufacturers offer instrumentation employing ultrasonic technology and its variants which has become increasingly accurate and easier to use.

The ultrasonic transceivers are mounted external to the piping (Figure 4) and, utilizing transit time algorithms, are able to determine of the flow rate. Typical accuracy is 1% to 2% depending on the installation and configuration of the transceivers.

Figure 4 - Ultrasonic Flow Cells on the Discharge Piping

Suction flow is the most desired providing the total flow into the pump; however, recirculation and interstage take-offs should be taken into account. If the pump has an internal hydraulic balancing device, typically found in multistage pumps, the flow through the balancing line leak-off should also be measured to provide additional insight into the condition of the hydraulic clearances within the pump.

Pressure and Temperature Measurements

To be able to accurately determine the TDH an accurate, properly ranged and calibrated set of test pressure gauges and temperature measurement devices are essential and should have a full scale accuracy of at least 0.25%. The gauges should be installed just prior to the testing and removed after for recalibration and storage.

There are several electronic based gauges available that offer exceptional accuracy with temperature compensation. Their size and weight makes for ease of portability. Regardless of the type, the test gauges should be treated with care.

Pressure measurements must include the suction and discharge and should be measured as close to the pump nozzle flange as practical. Additional pressure measurements such as balancing line leak-off and any interstage take-offs may also be required.

To facilitate temperature measurements, portable infrared temperature measurement instruments provide sufficient accuracy and should be part of the standard audit kit inventory.

Brake Horsepower

If the pump is driven by an electric motor the brake horsepower can be reasonably calculated by measurement of the motor current and voltage. If the motor efficiency and system power factor are known the calculated brake horsepower accuracy will be improved.

For pumps not driven by electric motors other methods of estimating brake horsepower can be used.

Field-testing is never as accurate as those conducted in a controlled environment such as a manufacturer's test facility. The expected accuracies for field-testing are presented in Table 1.

Table 1 – Typical Field Testing Accuracy

Variable	Laboratory Test [1]	Best Field Test or Retest of Same Pump [2]	Average Field Test [3]
TDH	0.5	1.0	2.0
Capacity	0.75	2.0	4.0
RPM	0.1	0.1	0.2
BHP	1.0	2.0	3.0
Efficiency	1.4	3.0	5.4

Vibration Signatures

Vibration signature analysis will compliment the audit providing further insight into hydraulic anomalies such as recirculation and the mechanical condition of the rotating components.

Pump Driver and Ancillary Devices

A comprehensive audit should also include the pump driver and any ancillary devices such as an external lubrication and seal flush systems.

Performance testing of turbines is beyond the scope of this paper, but as a minimum vibration signatures should be obtained as high vibration due to turbine anomalies can influence the vibration signatures measured from the pump.

Electric motor drivers can be comprehensively tested using the available portable diagnostic tools. Including the motors in the audit is becoming of greater importance as variable frequency drives (VFD) become more prevalent. Motor voltage and current analysis provides significant insight into how the motor is operating under load and dielectric testing can alert maintenance to pending insulation faults. Additionally, vibration signature analysis will provide insight in to both electrical and mechanical anomalies.

Case Study

The case study included in this paper was an actual hydraulic and mechanical performance audit for a major petroleum pipe line.

Historical Background

The subject pump (Figure 5), tagged MLPU 1 is an axial split case 8X10X13 - 3 stage pump in pipeline service. The line transfers crude oil of various viscosities. The pump is driven by a constant speed 2000 Hp induction motor.

Figure 5 – Main Line Pumping Unit (MLPU) 8X10X13-3 Stage Pump

The crude is from sources that have known high silica content. It is suspected that the silica is eroding the hydraulic fits within the pump and the seal faces. The current seal flush method is an API Plan 11 which provides for the raw pumpage to circulate from the pump into the seals and back to the pump.

The following hydraulic, mechanical and electric operating parameters were measured and recorded during the course of the audit:

- Suction flow (GPM)
- Balance line leak-off flow (GPM)
- Suction and discharge pressures and temperatures (psig, °F)
- Pump speed (RPM)
- Motor current and voltage (VAC, AMPS)
- Pump and motor vibration signatures

Hydraulic data and vibration signatures were collected under three operating conditions. The flow rates through the pipeline were increased, stabilized and held constant at each operating point to facilitate the collection of the data.

Discussion of Data

The measured hydraulic data was summarized in Table 2 and compared to the original test hydraulic performance curves in Figure 6. As indicated, the pump's total dynamic head (TDH) at 1744 GPM flow rate was approximately 3% below expected and at 3121 GPM the TDH was approximately 20% below the expected value of 1260 ft. Concurrently, the brake horsepower was 20% and 19% above the expected values. The calculated efficiency was approximated 13 to 26 points below the values indicated on the original test curve in Figure 6. The measured balance line leakoff was an average of 131 GPM, which is approximately twice the recommended upper limit of 60 GPM typical for pumps of this design.

Figure 7 shows the overall vibratory amplitudes of the pump and motor fell within acceptable limits. The higher amplitudes of the pump at the lower flow rate were expected when the pump was operating at approximately 51% of the best efficiency point (BEP). As the flow rate increased to approximately 67% of BEP the overall vibratory amplitudes attenuated primarily due to the reduction in the amplitudes of the vane-pass components as illustrated in Figure 8.

The spectra at the two different flow rates, compared Figure 8, illustrate that the reduction in vane-pass energies (6X and 7X) at the higher flow rate of 1744 GPM) were the most influential in reducing the overall amplitudes. The reduction in the 1X running speed amplitude was only approximately 2%, compared to the combined reduction of the vane-pass (6X and 7X) amplitudes of approximately 33%.

Additionally, the spectra of Figure 8 provide insight into the loss of the pump's hydraulic performance. Note the significant amount and amplitudes of random frequencies between the 1X and vane-pass (6X and 7X) components. Commonly referred to as the "noise floor" these amplitudes are typically comparatively low; however, as the flow through the pump becomes increasingly unstable due to hydraulic anomalies such as recirculation the amplitudes of these random frequencies will increase accordingly.

Conclusions and Recommendations

The data indicates significant hydraulic performance degradation. The drop-off of the TDH (Figure 6) at the higher flow rates is consistent with excessive casing-ring clearances. Historical records indicate that the pumps have been in service approximately two years (16,000 hrs) and the measured losses are atypical for the time in service, but the reported levels of silica contamination within the crude oil have been known to accelerate the erosion of critical internal hydraulic clearances.

It is recommended that the pump be subjected to a detailed internal inspection of both the rotating and stationary components. It is estimated continued operation of the pump in its current condition will increase annual operating costs by approximately $111,699.00 (Table 2). Consideration should be given to reviewing the type of materials used for the pump internal components and their compatibly with the pumpage.

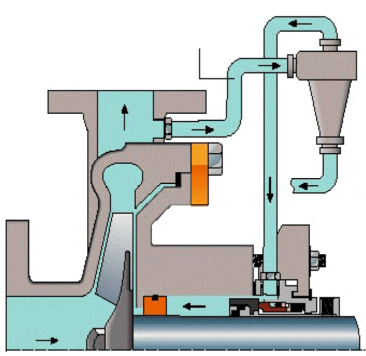

Figure 9 – API Seal Flush Plan 31 with Cyclone Separator

Additionally, the current seal flush is an API Plan 11; given the level of silica contamination API seal flush Plan 31 should be considered (Figure 9). The cyclone separator will reduce the amount of abrasive particulates suspended in the seal flush.

Summary

The loss of hydraulic performance in the case study was extreme and the incremental additional operating cost is higher than typical, but it is a good illustration that the potential penalties of poor hydraulic and mechanical performance can be significant. Typical results of pump audits reveal:

- 67% of pumps were operating at flow rates 65% or less below the best efficiency point (BEP) flow, additionally, 30% of audited pumps were operating below 50% of BEP
- 46% of pumps had control valve outputs of less than 50%
- 37% of the pumps had significant performances losses due to excessive internal clearances

System requirements can change over time. For example the black liquor systems in many paper mills have been optimized and operated with greater efficiency than the original system design reducing the flow requirements of the system. However, the pumps were never resized for the lower flow rates and TDH requirements and are typically throttled to operate significantly below the optimum BEP point resulting in additional pump operating cost diluting the system efficiency gains.

The two most significant benefits of scheduled hydraulic and mechanical pump and driver performance audits are:

1. Determining the energy losses due to inefficiencies of operation and once identified and corrected, continue to monitor and adjust as required to match system requirements
2. Anticipate and schedule repairs reducing unscheduled outages and consequential production losses. Additionally, provides greater repair budgeting accuracy by identifying failing components before they become critical

Pumps are significant components in any fluid handling system and as the cost of energy continues to increase period the pump hydraulic and mechanical performance audit has proven to be a valuable maintenance tool.

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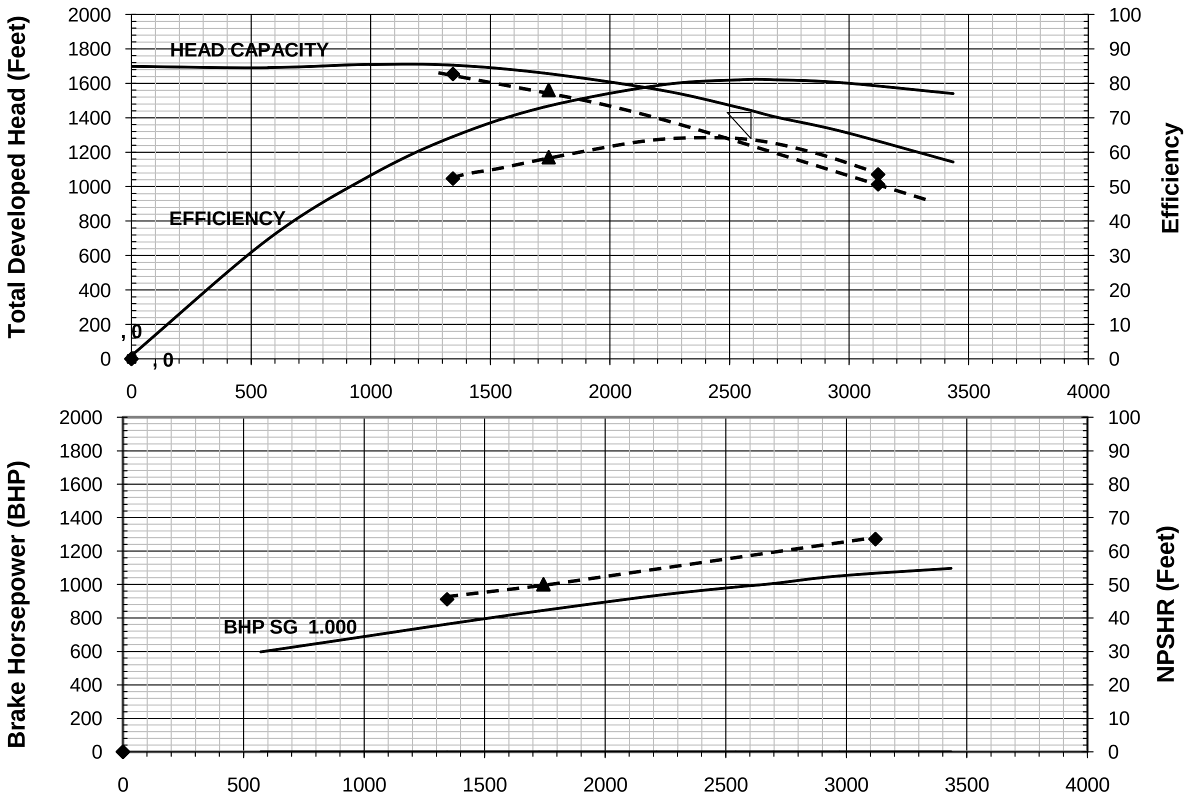
Table 2 -Main Line Pumping Unit 1 Hydraulic Condition Summary

FLOW RATE USGPM	TOTAL DEVELOPED HEAD			EFFICIENCY @ MEASURED FLOW			BRAKE HORSEPOWER			BALANCE-LINE LEAKOFF FLOW (GPM)			INCREMENTAL ANNUAL IMPACT OF OPERATION	
	DESIGN	ACTUAL	% VARIANCE	DESIGN%	ACTUAL%	Pts VARIANCE	DESIGN	ACTUAL	% VARIANCE	EST FLOW RANGE 45-60 GPM	ACTUAL	% VARIANCE	ESTIMATED OPERATING HOURS	EST IMPACT AT $0.07 KW/Hr
1,344	1706	1654	-3	65	52	-13	763	912	20	60	131	219	8000	($111,699)
1,744	1,744	1560	-11	73.8	59	-15	842	1000	19					
3,121	1260	1014	-20	79.8	54	-26	1070	1271	19					

Figure 6 - Measured Hydraulic Performance of MLPU 1

DATE	02/25/08	TAG No.	MLPU 1	Design Cap	2590	GPM
CUSTOMER	0	Mfg	0	Design Head	1430	Feet
LOCATION	0	TYPE	Axial Split, Multi-stage	BEP Cap	2590	GPM
UNIT	Pump Station	MODEL	0	BEP	81	%
PROCESS	Hydrocarbon	SN	0	Suc Imp Dia	11.43	Inches
OEM Test	3-Aug-05	Curve No.	62997	No.Stages 3	Inter Stg Dia 11.43	Inches



All data adjusted to 3560 RPM

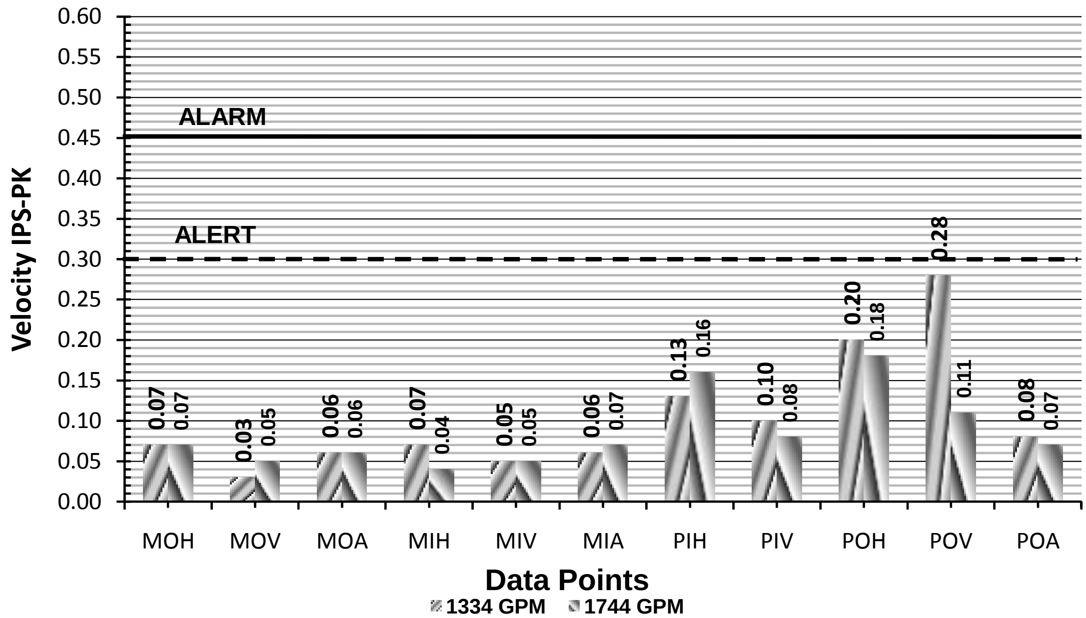
Gallons Per Minute

Brake Horsepower Adjusted to Specific Wt 1.00 g/cm³

Measured Avg Speed 3593 RPM

Figure 7 - Measured Overall Vibration Amplitudes MLPU 1

DATE	02/25/08		TAG No.	MLPU 1		Design Cap	2,590	GPM
CUSTOMER			Mfg			Design Head	1430	Feet
LOCATION			TYPE	Axial Split, Multi-stage		BEP Cap	2590	GPM
UNIT	MLPU 1		MODEL			BEP	81	%
PROCESS	Hydrocarbon		SN			Suc Imp Dia	11.43	Inches
OEM Test	3-Aug-05	Curve No.	62997	No.Stages	3	Inter Stg Dia	11.43	Inches



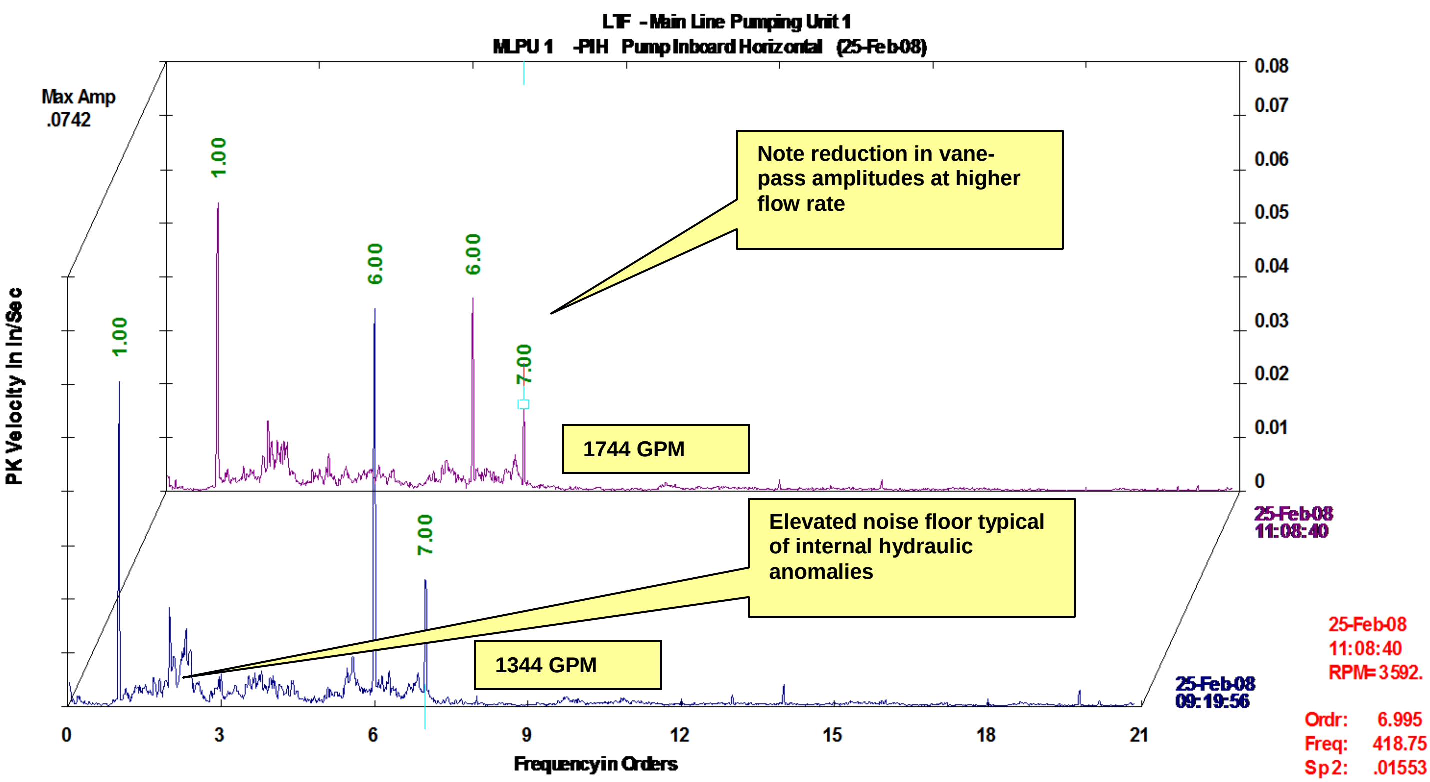


Figure 8 – PIH spectra at different flow rates exemplifying the reduction of vane-pass amplitudes at higher flow rate