Application of PAN^{TM} Filters for Avoidance of Pulsation Induced Surge in Centrifugal Compressors

Pulsation control with minimal pressure drop preserves low flow operating range.

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THE PROBLEM

Gas pipeline, storage and process systems increasingly employ both reciprocating and centrifugal compressors connected in fairly close proximity to each other. The mixed operation of centrifugal and reciprocating compressors in a compressor station has become a common practice as a result of capacity expansion and horsepower replacement programs. Operation of reciprocating compressors in the vicinity of centrifugal compressors often pose questions and concerns about the effect of pulsations on the centrifugal compressor surge margin and the ability of the surge control system to maintain the compressor operation safely away from surge. Traditionally, it was assumed that good reciprocating compressor pulsation bottle and manifold designs would result in minimal impact on the operational stability of the centrifugal compressor. This assumption was generally reliable for small horsepower and slow-speed applications and with older, high pressure drop and often significantly oversized pulsation control systems. However, more recent experience with compressor stations utilizing large power, high-speed reciprocating compressors with modern, efficient pulsation control systems shows that centrifugal compressors can encounter pulsation induced operational instability when operated in either parallel or in series with the reciprocating units.

The performance of a pipeline centrifugal compressor is commonly defined by a map of adiabatic head versus volumetric flow bounded by the surge and stall regions. Aerodynamic instability, known as surge, sets the limit for the centrifugal compressor's lowest flow and highest head/pressure ratio operation at a given operating speed. When surge occurs, the cyclic flow reversal within the compressor is sufficient to cause serious damage to the seals and bearings in most cases. And in extreme cases, it can fail bearings and seals and even damage the rotor shaft, impellers and other components. It is therefore imperative to avoid operating a centrifugal compressor in the surge region.

The exact location of the surge line on the map has a measure of uncertainty, as it can vary depending on the operating conditions. As a result, the typical surge margin for the minimum operating flow is 10 to 20% higher than the flow at the presumed surge line[10]. A recycle line with a control valve is installed around the compressor, and the valve opens when the pipeline flow drops to less than this predetermined margin from the theoretical or factory test measured surge line.

This map is critical for assessing the safe and useful operating range of a compressor, however the map is generic to any piping arrangement and does not provide a complete picture of how the compressor will respond to rapid transient inputs and how its surge behavior is affected by these events[4,11]. Specifically, the response of the compressor to rapid transient events such as single or multiple (periodic) pressure pulses is also a function of the compressor's upstream and downstream piping system acoustic response and impedance characteristics. Notably, the centrifugal compressor's behavior, and the resulting pressure ratio or head required, is affected by pressure pulsations in the suction or discharge lines. The piping system connected to the compressor either attenuates or amplifies these pulsations as demonstrated by its reaction to any fluctuation in flow with a fluctuation in head[12] as shown in Figure 1. The compressor head-flow characteristic will differ widely from the steady-state characteristic for higher fluctuation frequencies. Sparks[12] discusses piping system design approaches to reduce the pulsation levels using acoustic elements such as bottles, nozzles, choke-tubes and resonators.

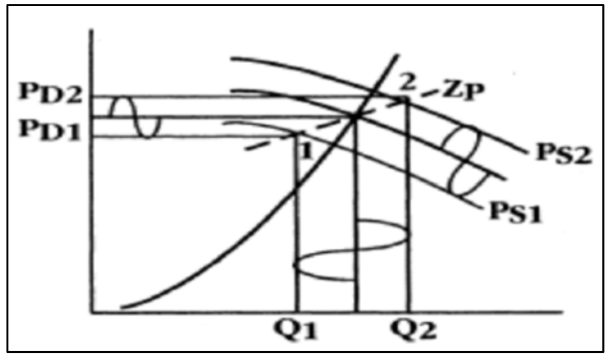

Figure 1: Effect of line pressure fluctuations on the surge margin of a centrifugal compressor[12].

Later investigations by Brun et al.[4] found that the onset of a surge event can occur when short pressure pulses interact with the piping system impedance line to cause a large flow fluctuation that crosses the surge line momentarily. Although this flow reversal event may be very short-lived (depending on the frequency of the flow fluctuation), it is usually sufficient to drive the compressor into a full surge cycle. Thus, even if a compressor is operating with an adequate surge margin based on the mean-flow, high inlet or discharge side pulsations have the potential to cause the compressor to operate in periodic unsteady surge cycles. This becomes a significant operational (and safety) concern in compressor stations where one or more centrifugal compressors operate in series or in parallel with reciprocating compressors.

More recently, Brun, et. al.[5] characterized the instability and explained the issue of transient surge, concluding that the surge margin reduction caused by the flow pulsation is less than the flow fluctuation magnitude. While not yet fully understood, it confirms that analytical prediction of the effect of pulsations on surge still has a measure of uncertainty. Adding this uncertainty to uncertainty in the prediction of pulsation amplitudes means that there can be significant risk associated with operating a centrifugal compressor in the low flow region of its operating map. One strategy for reducing the risk of pulsation induced surge is to operate with larger (mean-flow) surge control margins. For example, a 10 or 20% mean-flow surge control margin may need to be 25 or 35% to ensure that the compressor completely avoids surge when pulsations are present. But this reduces the useful flow range of the compressor, and it may result in increased levels of recycled flow when operation at low pipeline flows is required.

A more dependable strategy of efficiently reducing the magnitude of line pulsations is often required. Because of the high flow rates that are typical of centrifugal compressors, the use of orifice plates and traditional pulsation dampening bottles with choke tubes, unless exceptionally large and therefore very costly, can cause high pressure drops. The added resistance of the pulsation control elements must be overcome by the compressor at all operating conditions. This reduces the overall system efficiency and increases the power required for all operating conditions across the map. Even more of a concern is the fact that since large pipeline centrifugal compressors are most often single-stage units with relatively flat head-flow characteristics, the added resistance moves the compressor operating points closer to surge in all cases. This requires moving the surge control line to higher flow rates on the map, which can significantly reduce the low flow operating range of the compressor, countering some or all of the benefit that might be expected with the pulsation control system.

Because of many uncertainties in modeling and boundary condition assumptions, the best pulsation analyses generally predict pulsation frequencies within 2 to 3% accuracy, but pulsation amplitudes may be off by as much as 25%[9]. The state of the art often predicts problems near centrifugal compressors that never materialize, probably because the system impedance is more than assumed in the study or because the problematic operating conditions were never encountered in practice. In other cases, problems occur that were never predicted. Simplifying assumptions must be made in pulsation studies, and it is typically not practical to model and simulate all the potential combinations of units running, the ranges of speeds and the cross-combinations of load steps with multiple reciprocating units operating. Accordingly, an efficient pulsation control alternative should be an important consideration in the design of compressor stations having a mixture of centrifugal and reciprocating compressors.

THE SOLUTION

If the pulsation near the centrifugal compressor is of a single dominant frequency, a pulsation cancelling technology such as a side branch absorber or a single PAN™ Filter loop provides a solution to the pulsation problem without causing significant pressure drop. If a number of frequencies or a range of frequencies must be dealt with, then a series of PAN Filters or, in some cases, a tunable side branch absorber[3] (TSBA™) is required.

PAN Filters, a relatively new field proven technology, can be used to solve pulsation problems with little or no pressure drop or compromise in system efficiency[6]. PAN Filters can be used to attenuate pulsation in virtually any piping system, including reciprocating compressor suction and discharge headers[7,8], upstream of flow metering stations[2], and upstream or downstream of centrifugal compressors. They can be designed to replace most pipeline pulsation control bottles in service today, and they are worthy of consideration as an alternative to bottles for new compressor applications.

The application of PAN Filters to centrifugal compressor suction and/or discharge headers is the focus of this particular paper. PAN Filters can be applied to reduce harmful pulsations and associated shaking forces and pipe stresses in reciprocating compressor suction and discharge headers, without causing any significant pressure losses. Similarly, they can be applied to any other header or pipeline, including the suction or discharge lines of centrifugal compressors.

In a PAN Filter, the flow stream, as well as the pulsation energy, is carefully and equally split into two paths. As explained in detail in a previous paper[6], a carefully engineered TST-collector directs half of the pulsating flow through a loop of pipe and the other half of the pulsating flow passes straight through the TST-collector. The PAN Filter's primary cancellation frequency is the frequency that has a wavelength of twice the difference in the two flow path lengths, i.e., the path that includes the loop of pipe and the path straight through the TST-collector. When the two waves of equal amplitude, at this primary frequency, travel through the two flow paths and then are carefully rejoined 180˚out of phase by the TST-collector, one wave completely cancels the other, eliminating the pulsation at that frequency in the downstream flow.

Additional loops can be added in series to produce broad bands of pulsation attenuation that fill in the cancellation gaps between the odd and even orders of the primary frequency. Two or more loops may be necessary for applications with variable speed compressors, varying gas compositions and temperatures, varying flow velocities and flow streams with complex pulsation signatures. A 2-loop PAN Filter is shown in Figure 2. Note that, when necessary, the footprint can be reduced by folding loop pipes over themselves. Although the TST-collectors normally have flanged connections, which simplifies the hydrostatic testing and individual certification, where necessary, the TST-collectors can also be designed with welded pipe connections.

Figure 3 shows the extensive band of pulsation frequency cancellation that can be achieved with a 4-Loop PAN Filter. In this example, the loops are designed to cancel primary frequencies of 4, 8, 16, and 32 Hz. Note that 4 Hz and all 14 harmonics of 4 Hz up to 60 Hz are completely canceled. In addition, all the frequencies from 11 Hz to 53 Hz are attenuated by at least 90% or more.

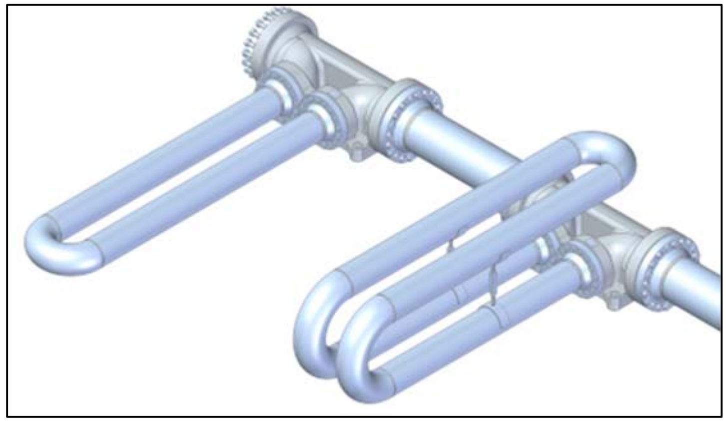

Figure 2: 2-Loop PAN Filter – Loops in series

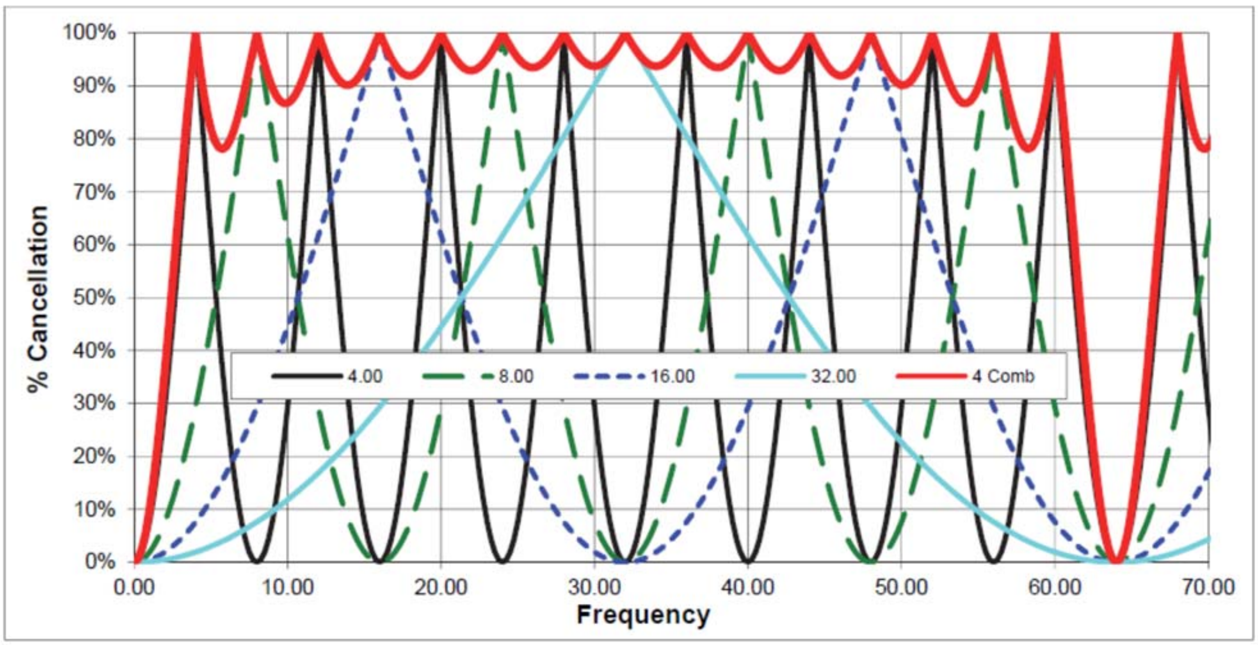

Figure 3: Example of 4-Loop PAN Filter pulsation cancellation

Pressure losses in PAN Filters and PAN Hi-Performance Compressor Manifolds are nearly zero[2,6,7,8], even in the highest transmission line flow cases. As a result, PAN Filters will increase operating efficiency, so that more flow can be produced for a given horsepower input, or less horsepower is required for a given flow rate. Either way, the use of a PAN Filter reduces fuel cost and greenhouse gas (GHG) emissions.

EXAMPLE OF PAN FILTER APPLICATION TO A CENTRIFUGAL COMPRESSOR

Figure 4 shows the adiabatic head vs. inlet volume flow operating map for a Solar C651 single-stage centrifugal compressor used on a natural gas pipeline. The compressor has 30 in. (762 mm) 900# flanges for direct coupling into the transmission pipeline. The rated operating speed is 9000 rpm, with a 4000 to

9500 rpm continuous operating speed range. As is typical of pipeline booster compressors, it is designed to have a wide operating range. Figure 5 is an expanded view of the low flow side of the same map with a surge control line that is 20% away from the actual surge line. The head, flow and minimum suction pressure that can be achieved at speeds of 9500, 9000, 8000 and 7000 rpm when operating on the 20% surge control line with a 1200 psig discharge pressure is shown in Table 1. A 20% surge margin is generally accepted as being conservative and therefore safe for operation of the compressor. Operating at flows below the control line at any given speed, requires recycling flow from the discharge back to suction, wasting power.

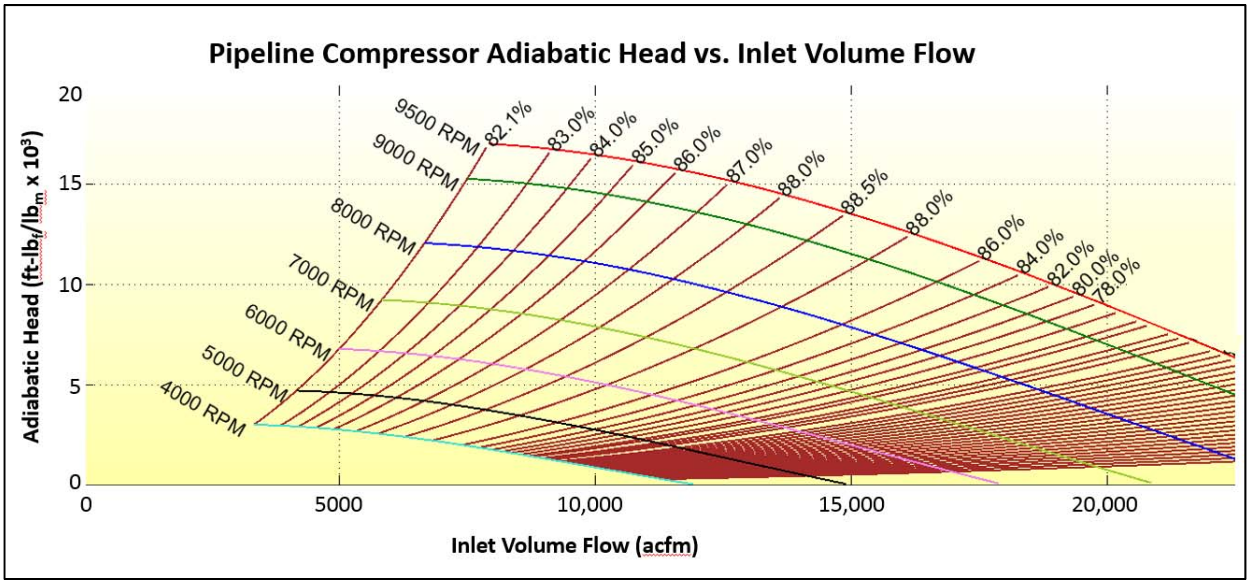

Figure 4: Head-Flow map for a Solar C651 single-stage pipeline centrifugal compressor

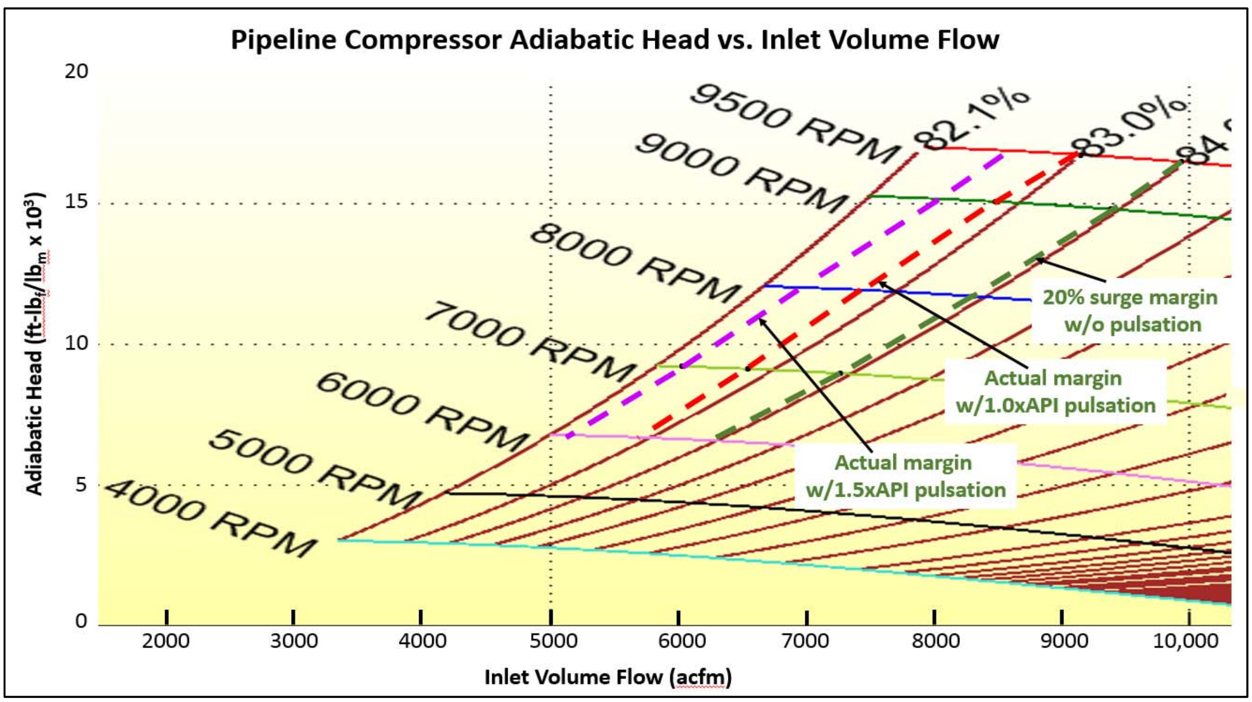

Figure 5: Low-flow end of Solar C651 single-stage pipeline centrifugal compressor head-flow map showing effective surge margin with suction line pulsations present.

But the picture changes significantly when pulsations are present in the suction line. The common industry guideline, API-618 5th Edition[1], specifies (paragraph 7.9.4.2.5.2.2.1) that the maximum allowable peak-to-peak pulsation level at any discrete frequency = $10/(P_L^{1/3})$% of the absolute line pressure.

Table 1: Head, flow and minimum suction pressure that can be achieved when operating on the 20% surge control line with a 1200 psig discharge pressure.

Pulsation Level	Pulsation Control	Condition for P_{disch} = 1200 psig	Speed (rpm)			
			9500	9000	8000	7000
None	None	Head (ft-lb_f/lb_m)	16,458	14,791	11,701	8,871
None	None	Min. P_{suct} (psig)	842.6	872.3	930.7	988.5
None	None	Q (acfm)	9942	9409	8348	7250
None	None	Surge margin	20.0%	20.0%	20.0%	20.0%

For a suction line pressure of 872.3 psig (886.8 psia in this case) at the 9000 rpm rated speed, the allowable peak-to-peak pulsation is 1.04% and the half-peak pulsation is 0.52% or 4.6 psi. This pulsation causes the suction pressure to oscillate from 867.6 psig to 876.9 psig at the pulsation frequency. When the cycle is at the minimum suction pressure, the head required from the compressor at 9000 rpm momentarily increases from 14,971 ft-lb_f/lb_m, at the 20% control line, to 15,046 ft-lb_f/lb_m. This moves the operating point to the left, so that the actual surge control line becomes the red line shown to the left of the 20% surge margin line in Figure 5. In most control schemes, the compressor driver would attempt to speed up in response to the demand for a higher head and pressure ratio. However the momentum of the train is such that it cannot respond as fast as the pulsation frequency. And if it did, it would immediately have to slow down as the suction line pressure swung to the maximum peak. So, in effect, the compressor continues to run at 9000 rpm and the operating flow momentarily swings closer to surge than intended.

At 9000 rpm, the surge margin is reduced to 11.2% instead of the intended 20.0%. Similar effects occur at the other speeds across the map as shown in Table 2 and Figure 5. The actual operating margin is only 10.0% at 8000 rpm. A surge margin of 10% is marginal for safe operation of the compressor. Therefore, this case would require that the intended surge control line be moved farther to the right, perhaps to 30% above the surge line, to ensure safe operation.

Table 2: Head, flow and minimum suction pressure that can be achieved when operating on the 20% surge control line with a 1200 psig discharge pressure and 1.0 x API-618 suction line pulsation.

Pulsation Level	Pulsation Control	Condition for P_{disch} = 1200 psig	Speed (rpm)			
			9500	9000	8000	7000
1.0xAPI618	None	Head (ft-lb_f/lb_m)	16,718	15,046	11,947	9,108
1.0xAPI618	None	Min. P_{suct} (psig)	838.1	867.6	925.9	983.4
1.0xAPI618	None	Q (acfm)	9113	8477	7420	6545
1.0xAPI618	None	Surge margin	12.7%	11.2%	10.0%	11.4%

In practice, acoustic modeling of compressor stations and systems rarely covers all the potential operating cases, especially when a number of reciprocating compressors are included in the system. There is therefore a significant risk that pulsations may be above the API-618 limit. To show the sensitivity of such a case, in this example, if the line pulsations are 150% of the API-618 limit, the operation of the compressor clearly approaches a dangerous operating zone as shown in by the purple, left-most line in Figure 5.

Table 3 shows the implications of pulsation levels of 150% of the API-618 limit. The surge margin at 9000 rpm is reduced to 9.6% and the margin at 7000 rpm is reduced to only 5.3%. Given the overall accuracy of surge prediction, coupled with control dead-bands, accuracy and response times, this is clearly an unsafe scenario that risks damaging the compressor. This can be a serious dilemma. Although moving the surge control line to higher flow rates - to reduce the risk of pulsation induced surge - is the generally most common solution attempted, such an approach reduces the operating range of the compressor. For a given head or pressure ratio, if the pipeline requires less flow than the minimum flow at the surge control line, all the excess flow must be recycled from discharge back to suction. For example, 10% recycle wastes 10% of the power used to compress the gas, which can equate to \$500 to \$1500/day of additional fuel cost on a typical pipeline centrifugal compressor. Further, the approach does not guarantee protection from surge if the pulsation amplitude is higher than predicted or expected.

Table 3: Head, flow and minimum suction pressure that can be achieved when operating on the 20% surge control line with a 1200 psig discharge pressure and 1.5 x API-618 suction line pulsation.

Pulsation Level	Pulsation Control	Condition for P_{disch} = 1200 psig	Speed (rpm)			
			9500	9000	8000	7000
1.5xAPI618	None	Head (ft-lb_f/lb_m)	16,848	15,174	12,070	9,228
1.5xAPI618	None	Min. P_{suct} (psig)	835.8	865.3	923.5	980.9
1.5xAPI618	None	Q (acfm)	8850	8325	7100	6125
1.5xAPI618	None	Surge margin	10.1%	9.6%	5.9%	5.3%

Attempting to reduce the line pulsation at the centrifugal compressor suction and/or discharge with traditional pulsation control methods typically becomes counter-productive. The line sizes are very large, 30 in. (762 mm) in this example, so that a pulsation bottle would be much larger in diameter, making it extremely large and costly. The use of orifice plates and choke tubes would cause pressure drop, which even if contained to one or two percent, would essentially have the same effect on increasing the effective head and pressure ratio as the pulsation does. The increased pressure drop also increases the required head and power at all operating conditions, not just near surge.

Methods of pulsation control that can cancel, rather than dampen, pulsation are therefore desirable for solving the problem of pulsations at the inlet or discharge of a centrifugal compressor. As discussed in previous papers[2,6,7,8], PAN Filters have been shown to control pulsations with minimal pressure drop. In order to explore the potential of PAN Filters for solving the pulsation problem in this example, the effects of adding one to three PAN Filters in series was analyzed. The number and amplitude of primary frequencies that must be controlled dictates whether one, two, three or in extreme cases, four, PAN Filters are required.

A single PAN Filter, i.e., one delay loop, is designed to cancel a primary frequency, and it will also cancel all the odd harmonics of that frequency as shown in Figure 6. This particular PAN Filter is designed to cancel 100% of the pulsation amplitude at 4.5 Hz (the primary frequency for 270 rpm in single-acting cylinder mode). It also cancels at least 90% over the range from 4.0 to 5.0 Hz (240 to 300 rpm), which is a typical operating range for a slow-speed integral engine-compressor. Although this example focuses on pulsation generated by a slow-speed compressor, a PAN Filter can be designed for pulsations generated by medium-speed or high-speed compressors just as effectively.

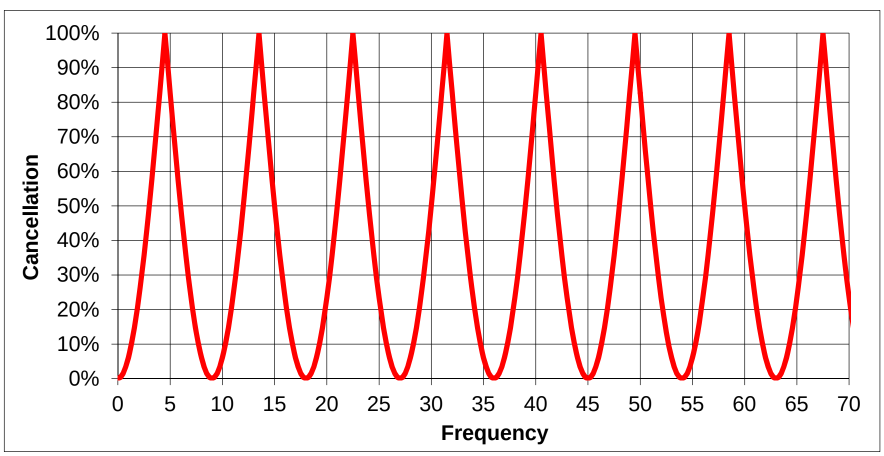

Figure 6: Pulsation cancellation for a single PAN Filter (i.e., one delay loop) designed for a primary frequency of 4.5 Hz.

If there are other pulsation frequencies that must be controlled, two PAN Filters may be placed in series. Figure 7 shows the result of adding a second PAN Filter designed to cancel a primary frequency of 9.0 Hz (the primary frequency for 270 rpm in double-acting cylinder mode). This combination achieves pulsation cancellation over wider bands of frequencies.

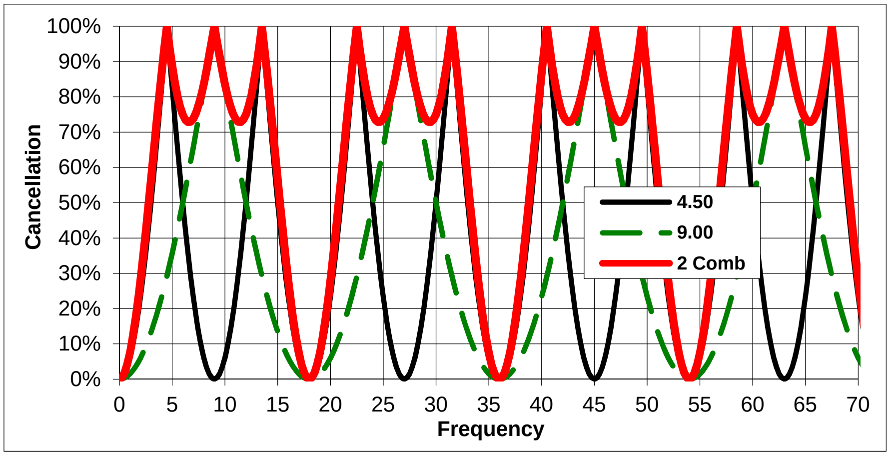

Figure 7: Pulsation cancellation for two PAN Filters (i.e., two delay loops in series) designed for primary frequencies of 4.5 and 9.0 Hz.

Figure 8 shows the effect of adding a third PAN Filter, designed to cancel a primary frequency of 18.0 Hz (the second order for 270 rpm in double-acting cylinder mode) in series. With this arrangement, the bands of pulsation cancellation become much broader. In extreme cases where there are a large number of pulsation frequencies to be canceled, four PAN Filters can be used.

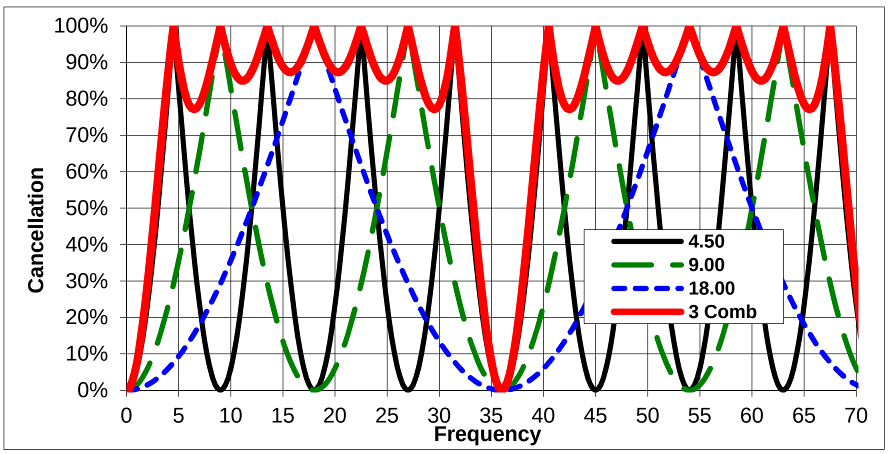

Figure 8: Pulsation cancellation for three PAN Filters (i.e., three delay loops in series) designed for primary frequencies of 4.5, 9.0 and 18.0 Hz.

Figure 9 shows the effect of adding a fourth PAN filter added, designed for a primary frequency of 36.0 Hz. This approach provides a high level of cancellation over a broad range of frequencies.

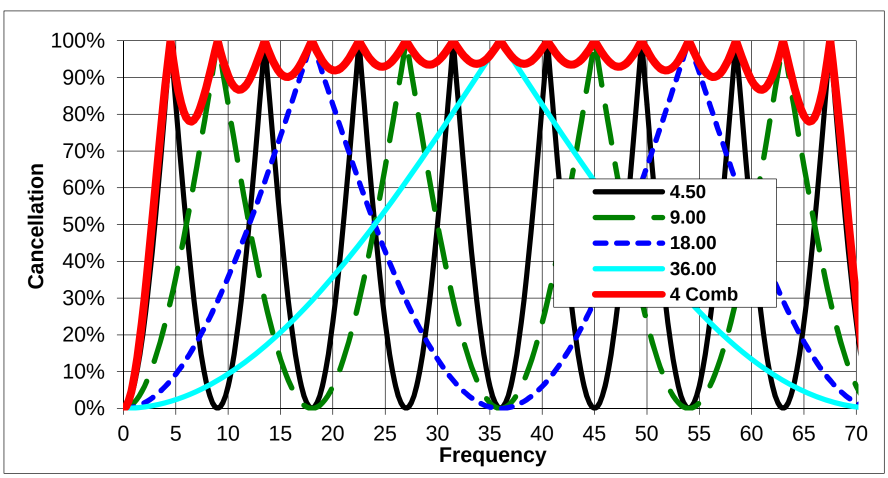

Figure 9: Pulsation cancellation for four PAN Filters (i.e., four delay loops in series) designed for primary frequencies of 4.5, 9.0, 18.0 and 36.0 Hz.

For purposes of examining the effects of using PAN Filters to reduce the risk of premature surge in the compressor discussed in this paper, it is assumed that the PAN Filters are designed to be effective in cancelling 90% of the pulsation. For a 30 in. (762 mm) diameter pipeline, each PAN Filter would consist of a Double-T TST-collector with 20 in. (508 mm) diameter loops of appropriate lengths to provide the intended frequency cancellation. Table 4 shows the results of modeling the effect of one, two and three PAN Filters to preserve the surge margin of the compressor. Although 90% of the pulsation is eliminated, a small pressure drop is added, conservatively calculated to be 0.95 psi per PAN Filter. The net effect of adding the PAN filter(s) is beneficial, even with three PAN Filters. In all cases the surge control margin should keep the machining operating safely, as the worst case is more than 13% away from surge.

Figure 10 shows the resulting operating lines compared with the uncontrolled operating line with 150% of the API-618 pulsation level. At 9000 rpm, the uncontrolled surge margin is only 9.6%. If the frequency content is such that one PAN Filter is all that is required for controlling pulsations, the surge range is 17.2% (i.e., only 2.8% less than the 20% control line). If two PAN Filters are required, the cumulative pressure drop uses up a little more of the surge range, but it is still 15.5%, much better than with no control. If three PAN Filters are required, the surge range is 14.2%. At 8000 and 7000 rpm, the improvement is even more striking, so that with any of these PAN Filter arrangements (i.e., one, two or three loops) the compressor would operate safely away of surge using the original intended 20% margin control line, even when the line pulsation coming into the PAN Filters is 150% of the API-618 limit. The higher the pulsation level, the more effective PAN Filters are in recovering surge margin.

Table 4: Head, flow and minimum suction pressure that can be achieved when operating on the 20% surge control line with a 1200 psig discharge pressure, with 1.5 x API-618 suction line pulsation controlled by one, two or three PAN Filters.

Level	Control	P_{disch} = 1200 psig	9500	9000	8000	7000
1.5xAPI618	1 PAN Loop	Head (ft-lb_f/lb_m)	16,552	14,882	11,786	8,951
1.5xAPI618	1 PAN Loop	Min. P_{suct} (psig)	840.9	870.6	929.0	986.7
1.5xAPI618	1 PAN Loop	Q (acfm)	9727	9091	8091	7091
1.5xAPI618	1 PAN Loop	Surge margin	18.2%	17.2%	17.5%	18.2%
1.5xAPI618	2 PAN Loops	Head (ft-lb_f/lb_m)	16,606	14,934	11,835	8,996
1.5xAPI618	2 PAN Loops	Min. P_{suct} (psig)	840.0	869.7	928.1	985.8
1.5xAPI618	2 PAN Loops	Q (acfm)	9545	8909	7909	6932
1.5xAPI618	2 PAN Loops	Surge margin	16.7%	15.5%	15.6%	16.3%
1.5xAPI618	3 PAN Loops	Head (ft-lb_f/lb_m)	16,661	14,987	11,883	9,041
1.5xAPI618	3 PAN Loops	Min. P_{suct} (psig)	839.0	868.7	927.1	984.8
1.5xAPI618	3 PAN Loops	Q (acfm)	9318	8772	7727	6704
1.5xAPI618	3 PAN Loops	Surge margin	14.6%	14.2%	13.6%	13.5%

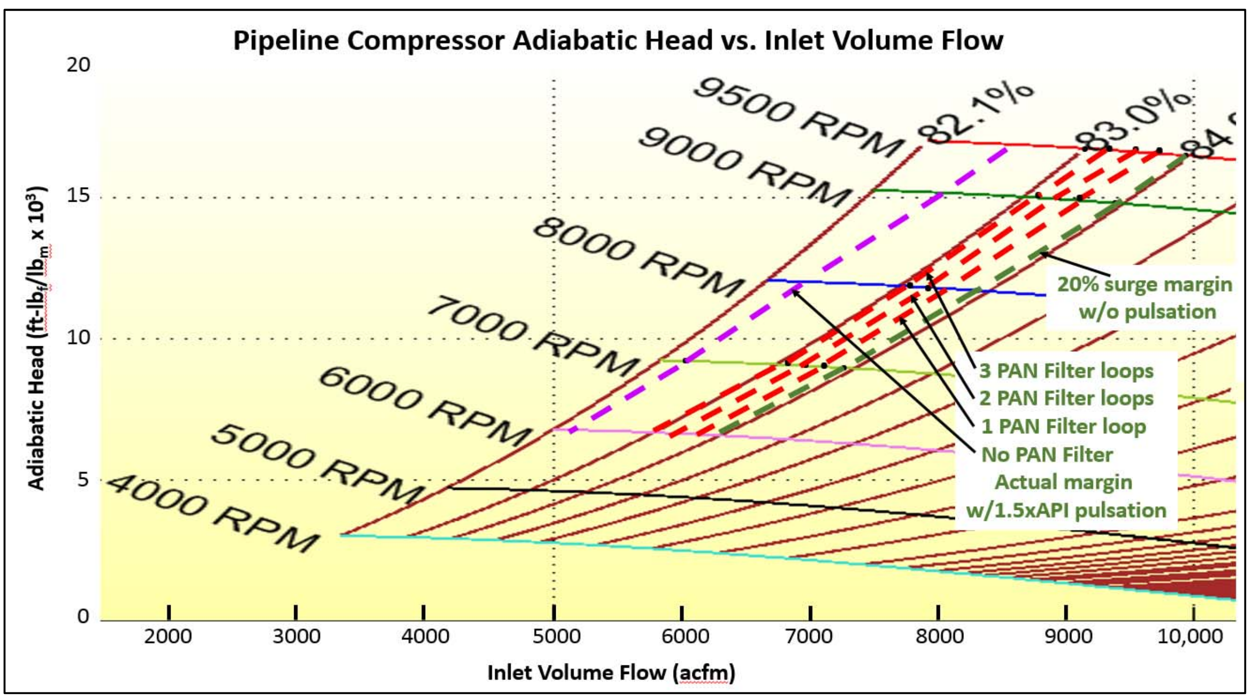

Figure 10: Low-flow end of Solar C651 single-stage pipeline centrifugal compressor head-flow map showing effective surge margin with PAN Filters used to control suction line pulsations.

Although this example considered the use of PAN Filters to successfully control pulsations in the compressor suction line to maintain a safe surge control margin, PAN Filters can also be used to control pulsations in the compressor discharge line. Further, in addition to pulsations from slow-speed compressors, as considered in this example, PAN Filter technology applies equally well to pulsations resulting from high-speed compressors or mixtures of slow- and high-speed units. In fact, the PAN Filter loop lengths required for high-speed compressors are shorter than those required in the aforementioned slow-speed compressor example, so that the pressure drop through each PAN Filter loop is also less, increasing the benefit even more.

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