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MITIGATION OF WIND-INDUCED VIBRATION OF ARCTIC PIPELINE SYSTEMS

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Abstract

Due to arctic permafrost conditions, pipelines in Alaska's North Slope oil fields are located on supports approximately 5 feet above the tundra, with spans ranging from 40 to 60 feet. Steady arctic winds blowing across the flat topography of the region often induce vertical pipeline vibration. The vibration is the result of cyclic lift forces associated with the vortex shedding phenomenon. Although the amplitudes of vibration are typically quite small, the accumulation of vibration cycles can induce fatigue damage, especially in pipeline field welds.

A series of pipeline wind-induced oscillation studies undertaken at the Kuparuk oil field are summarized in this paper. A general discussion of the wind-induced pipeline vibration phenomenon and observations from a field measurement program are presented. The results from full profile fatigue tests on typical pipeline field welds are presented. A brief review of conventional wind-induced vibration mitigation systems is included, followed by a discussion of field experiments in which the performance of promising mitigative systems was evaluated. The design of pipeline vibration dampers (PVD's) suitable for field-wide application at Kuparuk is then summarized.

1. Introduction and Objectives

Due to arctic permafrost conditions, pipelines in Alaska's North Slope oil fields are located on supports approximately 5 feet above the tundra, with spans ranging from 40 to 60-feet. Steady arctic winds (see the North Slope wind rosette in Figure 1) blowing across the flat topography of the region often induce vertical pipeline vibration. These oscillations are vortex-induced cross-flow (vertical) vibration caused by (horizontal) wind flow across the pipelines. Although the amplitudes of vibration are typically quite small, the accumulation of vibration cycles can induce fatigue damage, especially in pipeline field welds. Several analytical and experimental studies were undertaken to better understand and control the pipeline vibration phenomenon.

The pipeline wind-induced vibration (WIV) studies undertaken at the Kuparuk oil field are summarized in this paper. In Section 2, the results of a field measurement program are presented. In Section 3, observations and implications from full profile fatigue tests on typical pipeline field welds are presented. A brief review of conventional WIV mitigation systems is included in Section 4, followed by Sections 5 and 6 which discuss the development of promising mitigative systems. Section 7 presents field experiments in which the performance of these mitigative systems was evaluated. The design of a system of pipeline vibration dampers suitable for field-wide application at Kuparuk is summarized in Section 8. Important conclusions and observations from the Kuparuk pipeline vibration studies are presented in Section 9.

2. Field Investigation

The objective of the field investigation was to measure the displacement and stress responses of a pipeline during WIV, and correlate these displacements and stresses with those predicted by analytical models. This was accomplished by placing instruments on several spans of pipeline and measuring the response during wind-induced vibration and during several controlled vibration tests. Instruments were placed on a single pipeline in the Kuparuk system. The pipeline selected for instrumentation was an 8-inch gas injection (GI) line with an 8.625-inch outside diameter, a 0.5 inch wall thickness, a 3-inch layer of polyurethane insulation, and a 0.0276-inch thick corrugated metal jacket. The pipeline is supported on 55-foot spans and has a projected (aerodynamic) diameter of 14.625 inches.

The instrumentation consisted of an array of sensors located within or near three straight-run pipeline spans, beginning at a straight-run anchor. Twelve accelerometers measured vertical accelerations of the pipeline. Three horizontally oriented accelerometers at midspan locations measured the "in-line" vibration of the pipeline. Five displacement transducers measured vertical displacements at various locations. Eight pairs of strain gauges (one at the top and one at the bottom of the pipe

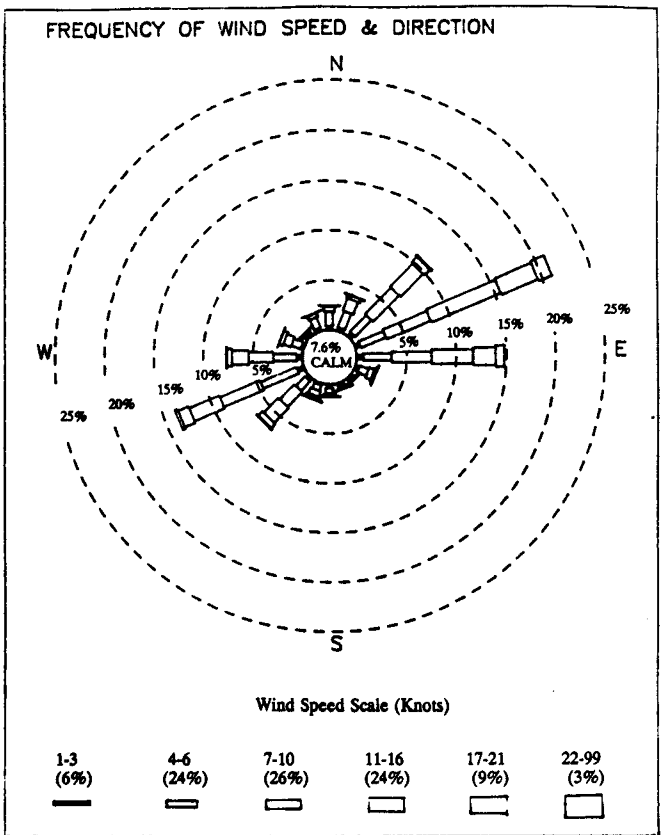

Figure 1 Prudhoe Bay Wind Rosette - 56,121 Observations (1978-1987)
(note - wind direction is the direction wind is blowing from)

cross section) were applied to the pipeline to measure longitudinal strains due to bending. The strain stations were located at the middle and near the supports of the pipeline spans, which are regions of maximum bending moment during WIV. In addition, three sensors measured the temperature of the pipe surface, the surface of the insulation jacket, and the air. Three meteorological stations measured wind velocity and direction. The meteorological stations were seven feet above the height of the snow, roughly two feet above the centerline height of the pipeline, and at least 50 feet from the pipeline to reduce disturbance of the wind when the sensors were leeward of the pipeline.

2.1 Free-Vibration Testing

The instrumentation and data acquisition system were used to measure static and dynamic response to free-vibration testing. The tests involved loading the pipeline into a deformed configuration, measuring the static deflections and strains, then instantaneously releasing the pipeline, and recording the free-vibration. The static responses were used to identify the stiffness properties of the pipeline, while the dynamic free-vibration responses were used to determine the vertical vibration mode shapes, frequencies and damping ratios.

The static and dynamic data from free-vibration testing was studied in detail using modal analysis techniques presented in (Ewins, 1984). An anchor-to-anchor finite element model of the pipeline was developed and used to correlate with the measured data. Nodes of the finite element model coincided with the locations of instruments so that predicted and measured results could be directly compared. Correlating the measured data with the finite element model provided insight into the important modeling parameters, which is essential for accurately modeling other pipelines. The most important observations are summarized as follows:

(1) The correlation with the static results effectively "tuned" the finite element model such that its dynamic properties were very close to the measured properties. Eleven vertical vibration frequencies and mode shapes were extracted from the dynamic portions of the free-vibration test data. Because the correlation coefficients between the eleven measured and the first eleven predicted mode shapes were very high (see Table 1), it is believed that these vibration modes were accurately captured.

(2) The correlation of the measured and finite element mode shapes was excellent. This is significant, since the finite element mode shapes represent the entire anchor-to-anchor portion of the pipeline, while the measured modes only contain information within the instrumented spans.

(3) The damping ratios obtained with these methods range from 0.10% to 0.39%. The best estimate of the damping ratio for use in analytical studies is 0.4%.

Table 1
Measured and Predicted Frequencies and Mode Shape Correlation Coefficients

Mode Number	Measured Frequency (Hz)	Predicted Frequency (Hz)	MSCC
1	1.98	1.98	.9989
2	2.02	2.04	.9989
3	2.12	2.14	.9987
4	2.25	2.27	.9990
5	2.39	2.43	.9974
6	2.54	2.61	.9712
7	2.71	2.80	.9959
8	2.83	2.99	.9936
9	2.91	3.19	.9924
10	3.05	3.37	.9008
11	3.49	3.54	.9699

2.2 WIV Data

The data acquisition system recorded data after initiation by either a displacement trigger or a wind trigger. A displacement trigger occurs when the root-mean-square (RMS) of the pipeline displacement at a midspan displacement transducer falls within a specified range. A wind trigger occurs when the average wind speed at one of the three meteorological stations falls within a specified range. This section discusses the data obtained during WIV events. The data characterizes the pipeline response under severe vibration conditions, as well as under more frequent, but less severe conditions. The data from two WIV events are examined in detail below. These events were selected because of the frequency content of their vibration response. The response recorded during Event 010101_T2 consisted of vibration in one mode only (i.e., uni-modal vibration), while the response in Event 060402_T7 was a combination of many vibration modes (i.e., multi-modal vibration). The typical pipeline frequency response to wind-induced dynamic forces is bounded by these two cases.

Narrow Banded, Uni-Modal Case (Event 010101_T2)
For this event, the average wind speed resolved perpendicular to the pipeline alignment (i.e., perpendicular wind speed) was 9.1 mph with a standard deviation of 0.9 mph. Plots of displacement history and its Fourier Amplitude Spectrum (FAS) are shown in Figure 2. It is clear from these plots that the displacement response is dominated by vibration at a frequency of 2.25 Hz, which corresponds to vibration mode 4 of the pipeline. All of the displacement, acceleration and stress response signals were observed to be near-harmonic wave forms at a frequency of 2.25 Hz.

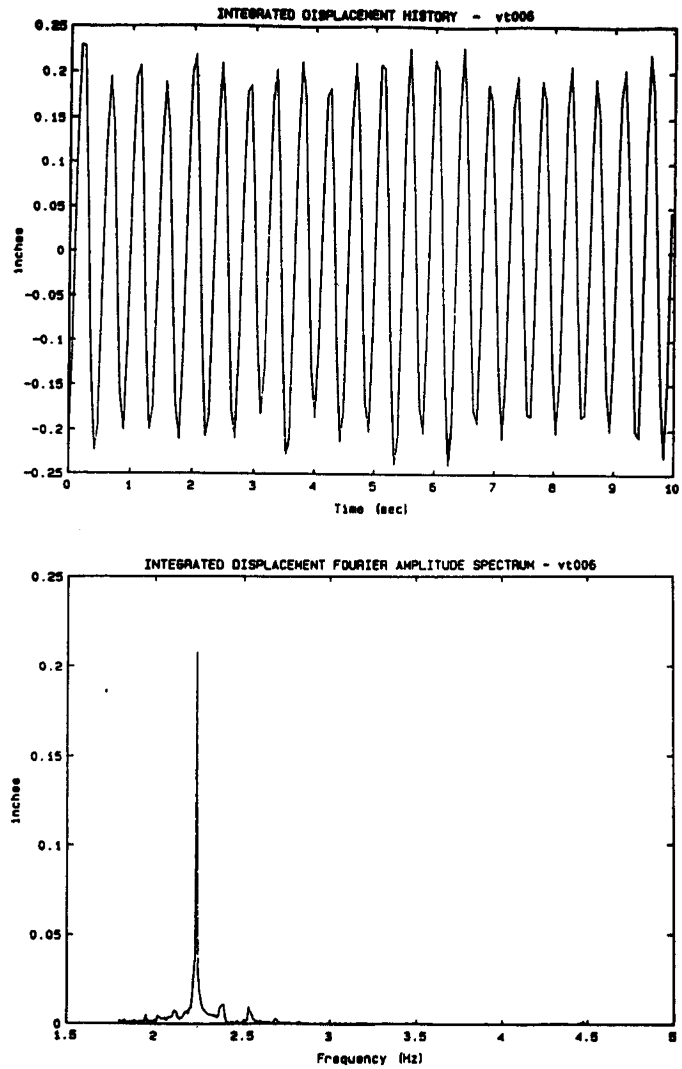

Figure 2 Narrow-Banded, Uni-Modal Response

Broad Banded, Multi-Modal Case (Event 060402_T7)
For this event, the average perpendicular wind speed was 13.2 mph with a standard deviation of 1.3 mph. Plots of integrated displacement history and its Fourier Amplitude Spectrum (FAS) are shown in Figure 3. Plots of displacement FAS indicate the presence of significant vibration at several separate frequencies ranging from 2 to 3.5 Hz. The frequencies indicate that modes 4, 5, 7, 8 and 9 have an appreciable participation in the response, while mode 6 is dominant at a frequency of 2.54 Hz. The plots of portions of the displacement histories reflect a rather complicated wave form resulting from vibration at several closely spaced frequencies.

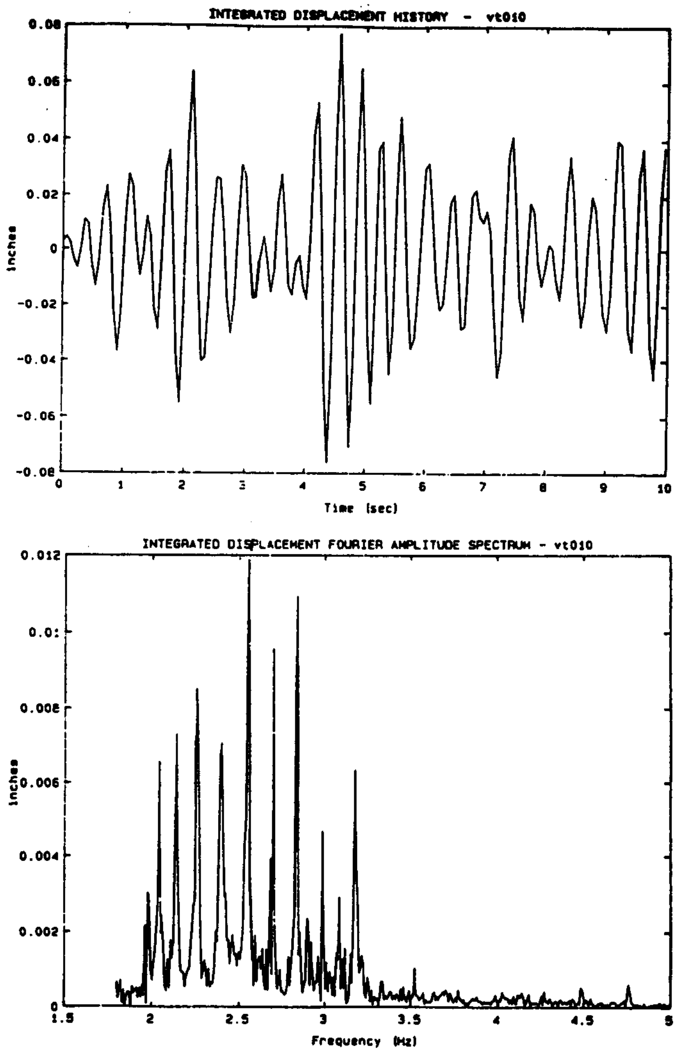

Figure 3 Broad-Banded, Multi-Modal Response

2.3 Observations

The data from the wind events characterize the pipeline vibration under various conditions. The most important findings are summarized in this section:

(1) The pipeline vibration modes and frequencies that are involved in WIV are bounded by the fundamental mode vibration frequencies for a single span pipeline with pinned- and fixed-end boundary conditions, respectively. These modes are illustrated for a seven span system in Figure 4.

(2) The largest measured displacements and stresses occur when the pipeline is undergoing narrow-banded vibration events. The displacements and stresses are notably smaller during broad-banded vibration. The maximum observed midspan displacement was roughly 0.6 inches peak-to-peak, and the maximum observed stress was roughly 1.8 ksi peak-to-peak.

(3) The most severe WIV events resulted from winds with normal wind speeds in the 9 to 14 mph range. The narrow banded events were caused by winds with normal wind speeds in the 9 to 10 mph range with relatively small standard deviations, indicating a relatively steady wind.

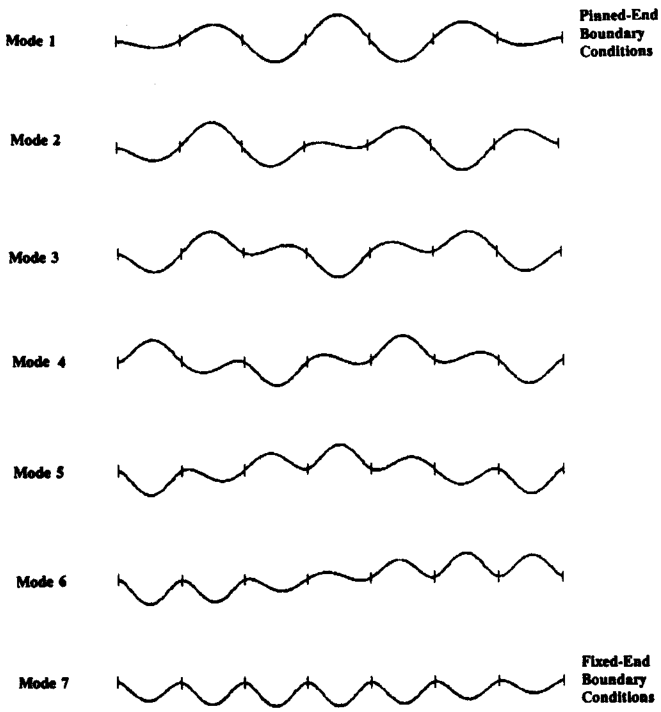

Figure 4 Mode Shapes for Seven Span System

Lower wind speeds tend to excite a narrow band of the lower natural frequencies of the pipeline, while higher wind speeds tend to excite a broader band of higher natural frequencies (see Figure 5).

(4) It was observed that vibration at lower frequencies produced maximum stresses at the midspan locations, while vibration at higher frequencies produced maximum stresses at the support and anchor locations. This trend is consistent with results obtained from the finite element model.

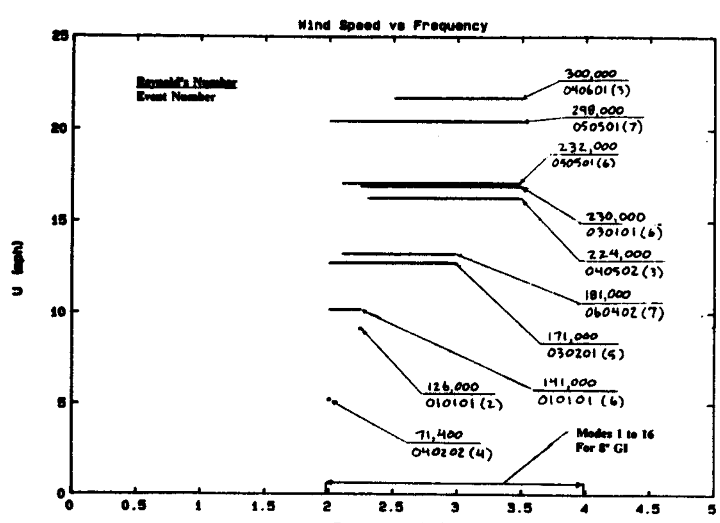

Figure 5 Observed Wind Speed - Vibration Frequency Relationships

2.4 Correlation with Analytical Models

In order to apply the results of the experimental investigation to other pipelines in the Kuparuk system, analytical models were correlated with field data. A standard model was used to estimate the vortex shedding frequency (and thus the pipeline vibration frequency) from the wind speed (Blevins, 1977). Another model, termed the SSD aerodynamic model, was developed to estimate the vibration amplitude; this model is similar to those presented in (Blevins, 1977, and ESDU, 1985). The models are dependant on the Reynold's number of the air flow past the pipeline, which depends on the wind speed, air temperature, and pipe diameter. In addition, the SSD aerodynamic model depends on the stiffness, mass, and damping of the multi-span pipeline system; the air density; and a Reynold's number dependant lift coefficient established from experimental data (Cheung, 1983). At a certain Reynold's number, a transition in flow past the pipeline occurs, which causes narrow banded vortex shedding to breakdown (Schewe, 1983). This leads to smaller lift forces spread over a broad band of frequencies (Cheung, 1983 and Schewe, 1983). The Reynold's number at transition depends on several parameters, including the turbulence in the wind. Wind turbulence was apparently an important factor in the measured response of the pipeline, causing a broad-banded frequency response of the pipeline at relatively low Reynold's numbers, and causing reduced lift forces which reduced the amplitude of response. Without the apparent turbulence, the response of the pipeline could have been narrow-banded over a larger range of wind speeds, and the response amplitudes could have been larger than observed due to increased lift force amplitudes. The lift coefficient in the SSD aerodynamic model depends on the degree of wind turbulence; coefficients for both laminar and turbulent wind conditions are included in the model. Good agreement between amplitudes measured in the field were obtained when turbulent wind conditions were assumed. The analytical models and correlation studies are presented in more detail in (Sause, 1992).

3. Evaluation of Pipeline Field Weld Fatigue Threat

Concern with pipeline WIV predated any documented failures related to the phenomenon. Prior to the studies described in this paper, rough field amplitude measurements and engineering analysis indicated that the cyclic stress ranges associated with WIV were below the commonly accepted endurance limit stress range of 9 to 12 ksi. The first documented fatigue failure of a pipeline weld raised the question whether the field weld endurance limit is less than 9 to 12 ksi. For this reason, experimental fatigue assessment work was initiated and an inspection of field welds in high risk locations was performed.

3.1 Weld Fatigue Strength Assessment

A fatigue strength assessment was initiated with the Edison Welding Institute and their U.K. affiliate The Welding Institute. Five typical field girth welds were obtained from a new pipeline construction project. Full profile weld segments were cut from these samples and subjected to fatigue testing to failure. The use of full weld profile testing is significant in that it fully incorporates the in-service geometry as opposed to the more common machined coupon testing which does not. The experimental data was used to generate S/N curves for typical Kuparuk field welds. While a detailed discussion of the fatigue work is outside the scope of this paper, the results are summarized in the attached Figure 6. Figure 6 establishes 6.8 ksi as the endurance limit, or

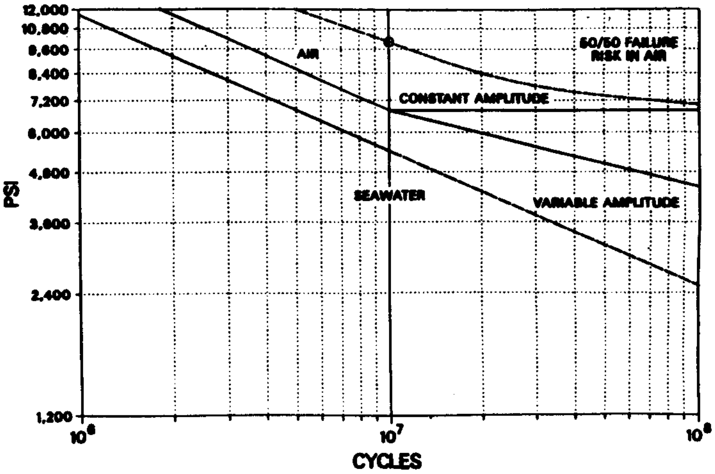

Figure 6 S-N Curves for Pipeline Girth Welds (Edison Welding Institute)

expressed alternatively if all stress ranges are less than 6.8 ksi, then the design life is infinite. This experimentally derived endurance limit is reduced by the presence of code allowable root defects. A fracture mechanics analysis was used to assess the detrimental effect of root defect size and shape. Figure 7 shows the results for two different shaped surface breaking root defects. The results indicate that for a 0.2-inch deep, 1.0-inch long linear defect (typical for lack of fusion defects) the nominal 6.8 ksi endurance limit is reduced by 50% to 3.4 ksi. This defect size was selected as a reasonable worst case defect.

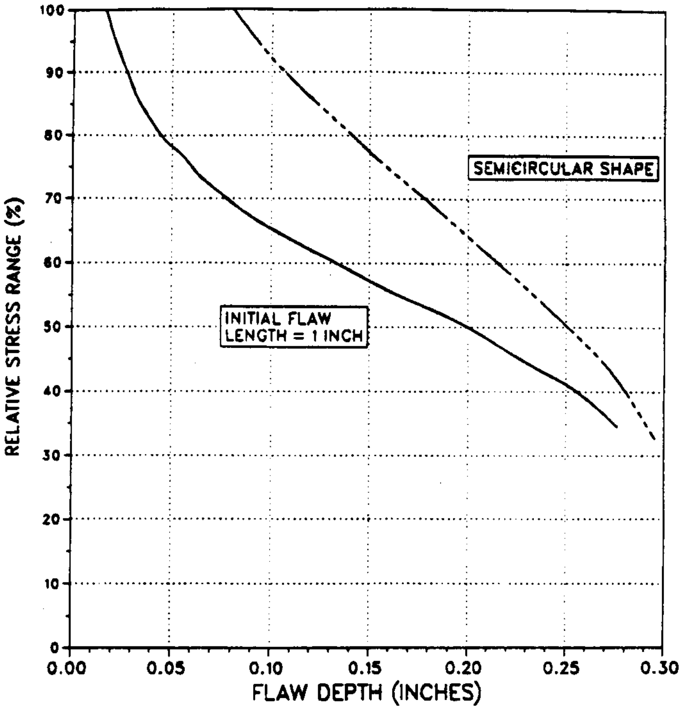

Figure 7 Effect of Root Flaws on Stress Range (Edison Welding Institute)

3.2 Implications for Field Wide Analyses

In order to assess the fatigue risk for systems at Kuparuk, analyses of an inventory of pipeline configurations was undertaken. The evaluation, which focused on small diameter pipeline configurations (≤ 12 inches) with various contents, involved finite element modeling of the systems and application of the SSD aerodynamic model to estimate the wind-induced stresses. Using the fatigue criteria and the analytical results, the following observations and conclusions were made:

(1) In turbulent wind conditions, the maximum stress ranges during vibration are between 3.4 and 6.8 ksi for the 6-inch through 10-inch gas pipelines. Thus, if one of these systems has a weld located near a location of maximum stress along the pipeline, and if this weld contains a defect positioned near a point of maximum stress within the pipe cross-section, then this weld may accumulate fatigue damage during vibration in turbulent wind conditions. This type of vibration occurs regularly for pipelines aligned perpendicular to the prevailing wind directions (Figure 1).

(2) In laminar wind conditions, the maximum stress ranges during vibration exceed 6.8 ksi for 6-inch and 8-inch water and oil pipelines, and for 6-inch through 10-inch gas pipelines. Thus, if one of these systems has a weld located near a location of maximum stress along the pipeline, then this weld may accumulate fatigue damage during vibration in laminar wind conditions even if it has no defects. It is uncertain how often this type of response occurs.

(3) In laminar wind conditions, the maximum stress ranges for the 10-inch water and 12-inch gas lift pipelines are between 3.4 and 6.8 ksi. Thus, such a system may accumulate weld fatigue damage during vibration in laminar wind conditions.

(4) Based on these observations it was concluded that fatigue damage of welds in the Kuparuk pipeline system appear to be occurring on a regular basis, especially in smaller diameter (6 and 8-inch) pipelines which contain defects. Thus, WIV in the pipeline system poses the risk of fatigue failure.

3.3 Field Wide Survey

With the confirmation of fatigue risk to pipeline field welds, an inspection of welds in high risk locations was undertaken at Kuparuk. All pipeline field welds which were located within three feet of a support, in systems with nominal diameters of 12 inches or less, and with alignments between N-S and NNW-SSE were surveyed. Support locations were critical since they are high stress locations, the diameter criterion identifies pipelines with a high potential to vibrate, and the alignment criterion identifies systems which are oriented perpendicular to the prevailing wind directions, shown in Figure 1.

3.4 Results of Field Wide Survey

Using the criteria outlined above, 918 field welds were inspected using an automatic ultrasonic shear wave technique. Since all of the inspected welds were located at supports, only the top portion of each weld (from the 10 to 2 o'clock position) was scanned. It is this portion of the cross-section that has the maximum tensile bending stress. This inspection identified six welds with strong indications of planar discontinuity. These six welds were removed from service and five of the six were examined in detail. Two welds had a severe lack of penetration defect, two welds had fatigue cracks that initiated at the inner wall and extended halfway through the pipe thickness, and one weld had excessive porosity and slag. Based on the results of the field wide analysis, it was concluded that WIV causes fatigue damage in pipeline field welds at Kuparuk on a regular basis. The field wide survey of welds in high risk pipelines confirmed this conclusion by identifying the existence of fatigue cracks in high risk pipeline systems.

4. Review of WIV Mitigation Systems

In the interest of avoiding additional fatigue damage and fatigue failures, an investigation of methods of mitigating the pipeline vibration was undertaken. A literature study of mitigation systems was conducted, and two promising systems for the existing pipeline system were identified.

4.1 General Methods of WIV Mitigation

The literature contains two broad categories of mitigation methods, namely structural methods and aerodynamic methods. Structural methods modify the dynamic properties of a vibration susceptible system, by increasing the natural frequencies of the system above the frequency range of the lift forces, or by decreasing the amplitude of the vibration to an acceptable level. Structural methods may increase the stiffness, mass, or damping of the system. Increasing the stiffness increases its natural frequencies. Stiffness modifications (e.g., adding intermediate supports to the pipeline), can be prohibitively costly. Increasing the mass reduces the vibration amplitude, but also decreases the system's natural frequencies. Increasing the damping of the system reduces only the amplitude of vibration. Aerodynamic methods for suppressing vortex-induced vibration typically aim to suppress the formation of vortices in the wake near the vibration susceptible body. Common aerodynamic methods can be categorized as follows (Zdravkovich, 1981):

(1) Surface protrusions (strakes, wires, fins, studs, etc.), which disrupt regular vortex shedding along the cylinder;

(2) Shrouds (perforated, gauze, axial rods, slats, etc.), which affect the entrainment layers;

(3) Nearwake stabilizers (splitter and sawtooth plates, etc.), which control the confluence point in the nearwake.

Other aerodynamic approaches include increasing the effective diameter and increasing the surface roughness. Increasing the effective diameter increases the critical wind speed required to cause vibration. The increased diameter and wind speed are associated with higher Reynold's numbers, and these higher Reynold's numbers are associated with a transition of the boundary layer, which leads to smaller lift forces spread over a broad band of frequencies (Cheung, 1983, Schewe, 1983). The smaller lift forces produce a decreased vibration amplitude. Increasing the surface roughness promotes the boundary layer transition at lower Reynold's numbers, which also reduces the lift forces.

Helical Strakes

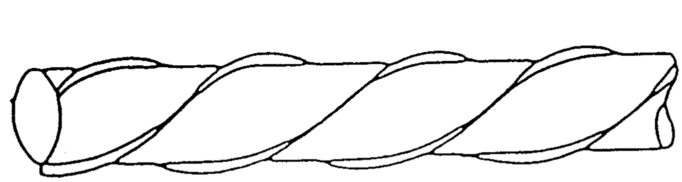

Figure 8 Illustration of Helical Strakes

4.2 Recommended WIV Mitigation Systems

The recommendation of WIV mitigation systems is based on three main factors, namely (1) whether the method has been the subject of favorable reports in the literature, (2) whether the method appears economical and practical for existing pipelines in arctic conditions, and (3) whether the method is associated with significant uncertainties when implemented on several in-line cylinders (as the Kuparuk system where there are usually several parallel pipelines on a series of supports). Based on these factors, there is no ideal method of pipeline vibration mitigation however, helical strakes and tuned dampers were judged to be worthy of further investigation.

Helical strakes (Figure 8) have successfully mitigated vortex-induced vibration of stacks and towers. Laboratory and field investigations of helical strakes have been so widespread that optimal design parameters have been identified. However, strakes may not be effective in preventing vibration of systems composed of several in-line bodies, such as parallel pipelines, especially if the in-line bodies are closely spaced (Walshe, 1970). In some cases, the buffeting effect of adjacent structures is reduced by helical strakes, while in other cases it may be increased (Every, 1982).

An increase in the damping of a vibration susceptible system will directly reduce the vibration amplitude, since the amplitude is inversely proportional to the damping ratio (Clough, 1975). An auxiliary mass damper (Harris, 1988) is a mass-spring-damper device fitted to a vibration prone structure. The damper is tuned such that the characteristic motion of the primary system results in resonant (or near-resonant) response in the damper. It has been found that a damper having a mass equal to one percent of the mass of the structure is sufficient to control vibration, although in practice, installed dampers have had a mass of two percent of the structure mass (Wardlaw, 1988).

5. Development of Helical Strakes for Kuparuk Pipelines

Despite potential problems associated with several in-line cylinders, it is believed that helical strakes are the best aerodynamic method for mitigating pipeline WIV.

5.1 Optimum Design

Investigations of helical strake configurations, both in wind tunnels and field installations, indicate that the important configuration parameters are: (1) the number of strakes, (2) the height of the strake relative to the diameter of the cylinder (h/D ratio), (3) pitch angle of the helix, and (4) the strake cross-section. The optimum helical strake configuration is as follows:

(1) three (or more) strakes,

(2) h/D ratio greater than 7.5%, with optimum h/D ratio between 10% and 12%,

(3) a pitch of five times the cylinder diameter (5D),

(4) slender, sharp edged rectangular strakes, with a maximum thickness (t) of one fifth of the strake height.

A design parameter which is not established in the literature is the portion of the vibration-susceptible body that must be covered by strakes. Coverage of the portion of the body which undergoes significant motion is recommended. For vertical smoke stacks, coverage ranging from 20% to 67% of the top portion of the stack has been implemented (Wardlaw, 1979). For horizontal pipelines, coverage should be provided in the middle portion of the span where the vibration amplitude is the largest; as a minimum, 50% of each span should be covered.

5.2 Design for Field Evaluation

The rectangular cross section of a strake is controlled by the ratio of the height of the strake to the diameter of the cylinder (h/D ratio), and the ratio of the height of the strake to the thickness of the strake (h/t ratio). For an 8-inch nominal diameter pipeline (insulated diameter D=14.625 inches), h/D ratios of 10% and 12% are obtained for h=1.46 inches and h=1.76 inches, respectively. Based on these values, a strake height of h=1.5 inches was selected to provide h/D ratios of 10% for 8-inch nominal diameter pipelines. In order to keep the strakes relatively slender, the minimum h/t ratio of 5 is considered. The maximum thickness of the strake was, therefore, selected to be 0.25 inches, corresponding to an h/t ratio of 6. The final aspect of the helical strake design is the length of strake required for the pipelines in the Kuparuk system. For pipelines with a nominal diameter of 8 inches, the minimum length of an individual strake required to cover 50% of the span is 390 inches. The selection of a strake material which can easily (and economically) be installed on pipelines in arctic oil fields, while remaining aerodynamically effective over long time periods is a difficult, uncertain problem. For the field evaluation, a sheet metal strake system was constructed which met the geometric requirements specified above. The sheet metal strakes are attached to the pipeline via a system of metal bands.

6. Development of PVD's for Kuparuk Pipelines

The pipeline vibration damper (PVD) concept consists of one damper in each span of the pipeline (see Figure 9) with each damper tuned to one of three frequencies. The damper consists of a weight (mass) and a damped spring. The requirements of the PVD systems are:

(1) The system must mitigate vibration over a broad temperature range and be durable under arctic conditions for extended periods.

(2) The system need reduce only vertical pipeline vibrations which dominate the response to vortex-shedding excitation.

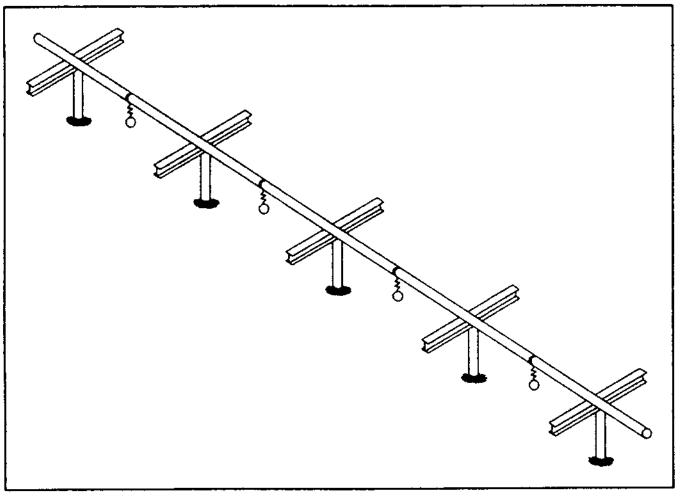

Figure 9 Multi-Span Configuration with Dampers

(3) The system must provide sufficient damping in the frequency range of 1.5 Hz to 6.0 Hz, corresponding to vibration-susceptible modes of pipelines in the Kuparuk oil field.

(4) The weight of damper must not excessively stress or deflect the pipeline system.

(5) The individual dampers should be compact and should not undergo excessive static deformations.

6.1 Elastomeric Springs

Metal, plastic or elastomeric spring elements are often used to isolate structures and equipment from vibration (Harris, 1988). Elastomeric springs are widely used, because they can sustain large deformations and return to their original state with virtually no damage. For PVD systems, elastomeric springs are superior to other types of springs because for a given amount of elasticity, deflection capacity and energy dissipation they require less weight and less space. Also, different elastomer compounds may be molded into different configurations, generally at low cost. Because of these favorable characteristics, elastomeric springs were considered most appropriate for arctic PVD systems.

6.2 Cold Weather Properties of Elastomers

When exposed to low temperatures, elastomers become harder, stiffer and less resilient. In arctic applications, the primary concern is that the damper maintain effective dynamic tuning and provide sufficient damping over the range of expected temperatures. Special elastomeric compounding can provide favorable dynamic properties over a broad temperature range, but all compounds will exhibit undesirable stiffening trends at low temperatures (Harris, 1988). Hence, low temperature testing of the elastomeric springs is mandatory for the development of arctic PVD's. The temperature dependent dynamic properties of the elastomeric springs are used to evaluate the effectiveness of the damper system over a range of temperatures.

6.3 Prototype Design and Testing

Shear deformation is known to provide the maximum energy dissipation per unit volume of elastomeric material and excellent resistance to creep (Harris, 1988). Thus the PVD is composed of a mass suspended by elastomeric shear springs. The mass is supported by a "ladder" suspension configuration of springs arranged in series with each spring serving as a tier of the ladder (see Figure 10). The arrangement of the springs and the mass such that the line of action passes through the damper centerline minimizes the bending stress in the elastomeric springs, enhancing the damping and creep performance. The series arrangement provides the flexibility needed to create a low frequency (1 to 10 Hz) damper using light-weight masses. The modular arrangement of springs provides numerous device configurations including 1, 2, 3, 4, 5, 6, 7, or 8 tiers (see Figure 10). Combined with a mass that weighs between 10 and 90 lbs, this arrangement allows the damper to be tuned to a specific target vibration frequency. Each spring is a small cylinder of elastomeric material bonded to metal connectors which have protruding threaded studs. The PVD illustrated in Figure 10 incorporates two different commercially available "sandwich mounts", each made of a compound of natural rubber and a low temperature man-made polymer. Metal plates link the springs together and attach the springs to the suspended mass and a pipe collar at midspan.

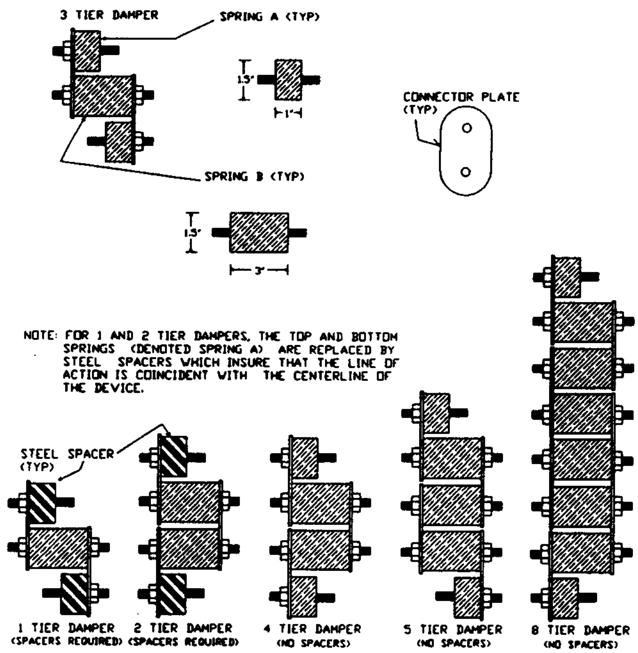

Figure 10 Illustration of PVD Configuration

Several damper configurations were laboratory tested at ARCO's research facility in Plano, Texas. The objective was to establish the vibration properties of different configurations for a range of temperatures, displacement amplitudes, and frequencies. The testing involved controlled displacement cycling in a dynamic testing machine equipped with a cold box. The tests considered a temperature range of +50° F to -50° F, a displacement range of +/- 0.1 to +/- 0.5 inch, and a frequency range of 1.5 to 4 Hz. The force and displacement histories provided the effective stiffness and effective damping ratio of the tested configuration. The tests were conducted on 3 and 4 tier systems, with the results for systems with more or less tiers obtained by modeling the stiffness of the system as a series of springs. Additional tests were conducted on a 3 tier configuration after it had been exposed to ultraviolet (UV) bombardment to simulate the effects of aging and weathering and to establish long term performance and replacement criteria.

A summary of the test results is presented in Figure 11 which illustrates the dramatic increase in effective stiffness and damping ratio with decreasing temperatures. The low temperature results indicate that the increase in stiffness is associated with a dramatic increase in damping ratio. Thus the potential loss of tuning due to stiffening may be compensated by extremely high damping.

7. Field Evaluation of Promising Mitigation Systems

The helical strake and pipeline vibration dampers were field evaluated on a section of an 8-inch gas lift (GL) pipeline with an 8.625-inch outside diameter, a 0.322-inch wall thickness, a 3-inch layer of polyurethane insulation and a 0.0276-inch thick corrugated metal jacket. The pipeline is supported on 55-foot spans and has a projected (aerodynamic) diameter of 14.625 inches. Three anchor-to-anchor sections (nominally 20 straight run spans each) were used in the evaluation. The helical strake system was installed on one 20-span section, the damper system

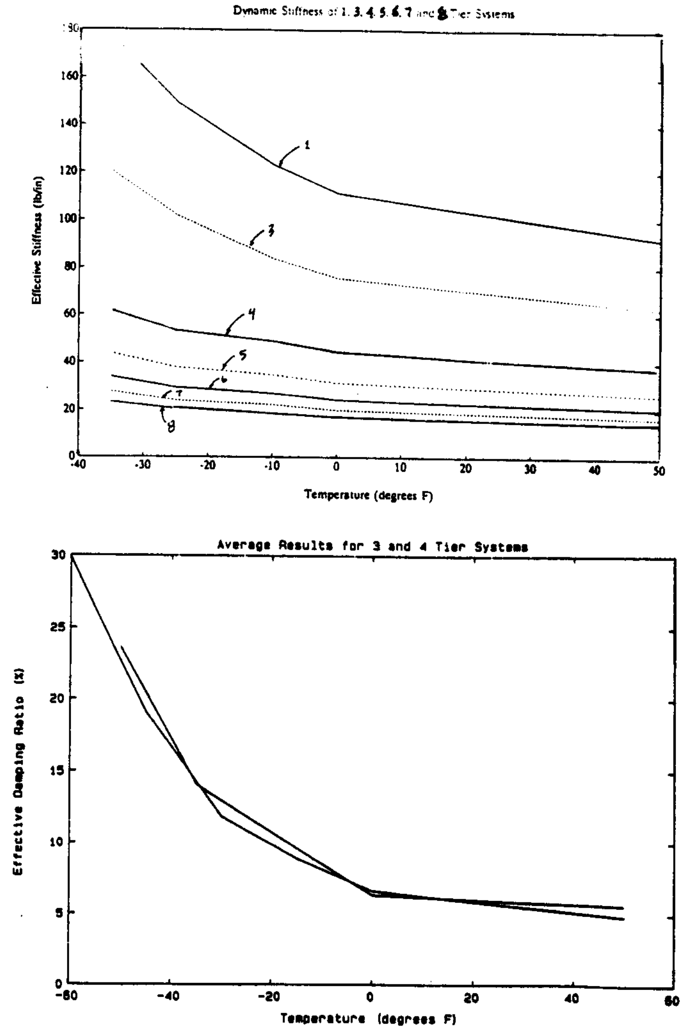

Figure 11 Temperature Dependance of PVD Dynamic Properties

was installed in another 20-span section, and the third 20-span section was unmodified. The most effective prototype PVD configuration consisted of one damper in each span using a pattern of high frequency (3-tiers and 50 lbs), medium frequency (4-tiers and 50 lbs), and low frequency (5-tiers and 75 lbs) dampers in alternate spans (i.e., triple tuning). The field evaluation allowed for a "side-by-side" comparison of the effectiveness of the mitigation systems and provided a basis for evaluating the reduction provided by each system under similar wind conditions. Free-vibration tests were conducted on both the unmitigated spans and on the spans with the PVD system. WIV data was also gathered automatically for several months.

The data acquisition system and instruments were similar to that used in the initial field studies. Within each 20-span section, six spans were instrumented with a vertically oriented accelerometer at midspan. In the damped section, two additional vertically oriented accelerometers were attached to the damper mass in order to measure the motion of the damper mass. In the straked and bare sections, an additional horizontally oriented midspan accelerometer was installed to measure of the potential effects of in-line drag forces. A meteorological station was placed adjacent to the pipeline in each 20-span section.

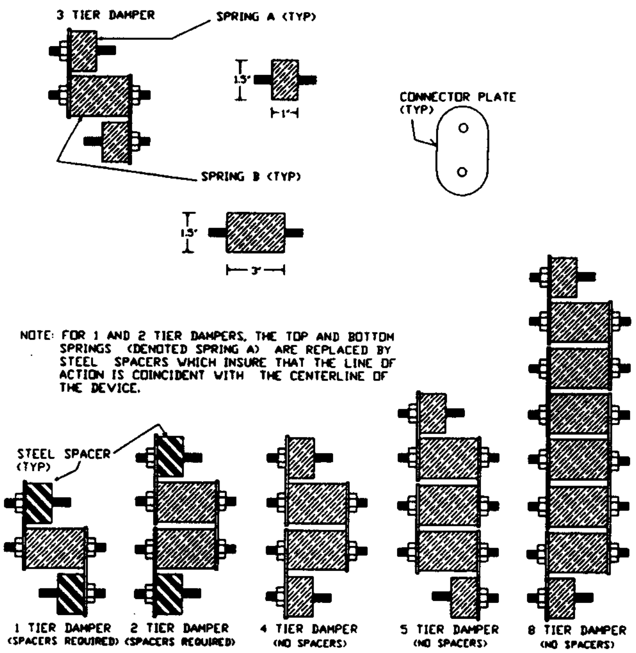

Figure 10 Illustration of PVD Configuration

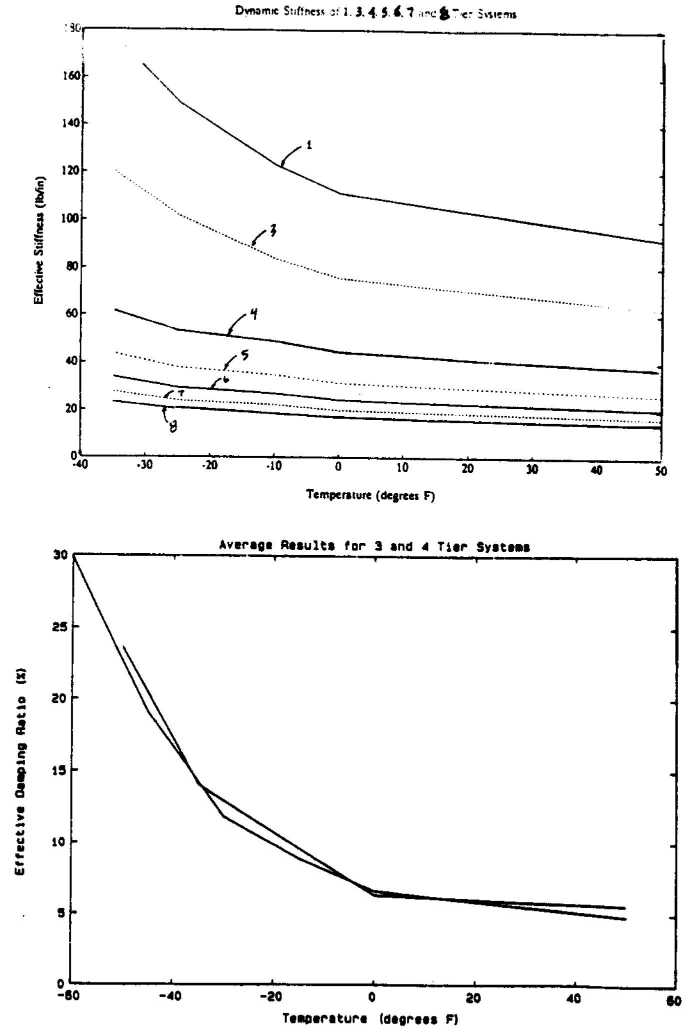

Figure 11 Temperature Dependance of PVD Dynamic Properties

Several damper configurations were laboratory tested at ARCO's research facility in Plano, Texas. The objective was to establish the vibration properties of different configurations for a range of temperatures, displacement amplitudes, and frequencies. The testing involved controlled displacement cycling in a dynamic testing machine equipped with a cold box. The tests considered a temperature range of +50° F to -50° F, a displacement range of +/- 0.1 to +/- 0.5 inch, and a frequency range of 1.5 to 4 Hz. The force and displacement histories provided the effective stiffness and effective damping ratio of the tested configuration. The tests were conducted on 3 and 4 tier systems, with the results for systems with more or less tiers obtained by modeling the stiffness of the system as a series of springs. Additional tests were conducted on a 3 tier configuration after it had been exposed to ultraviolet (UV) bombardment to simulate the effects of aging and weathering and to establish long term performance and replacement criteria.

A summary of the test results is presented in Figure 11 which illustrates the dramatic increase in effective stiffness and damping ratio with decreasing temperatures. The low temperature results indicate that the increase in stiffness is associated with a dramatic increase in damping ratio. Thus the potential loss of tuning due to stiffening may be compensated by extremely high damping.

7. Field Evaluation of Promising Mitigation Systems

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The data acquisition system and instruments were similar to that used in the initial field studies. Within each 20-span section, six spans were instrumented with a vertically oriented accelerometer at midspan. In the damped section, two additional vertically oriented accelerometers were attached to the damper mass in order to measure the motion of the damper mass. In the straked and bare sections, an additional horizontally oriented midspan accelerometer was installed to measure of the potential effects of in-line drag forces. A meteorological station was placed adjacent to the pipeline in each 20-span section.

7.1 Free-Vibration Testing

Free-vibration tests were conducted in the damped and unmitigated sections. The test procedure was identical to that used in the initial field investigation. The data gathered during the free-vibration test was processed to determine the vibration frequencies of the system and to estimate the effective damping ratio based on the free-vibration decay. Samples of typical free-vibration response of the undamped and damped sections are illustrated in Figure 12. This figure clearly indicates the dramatic

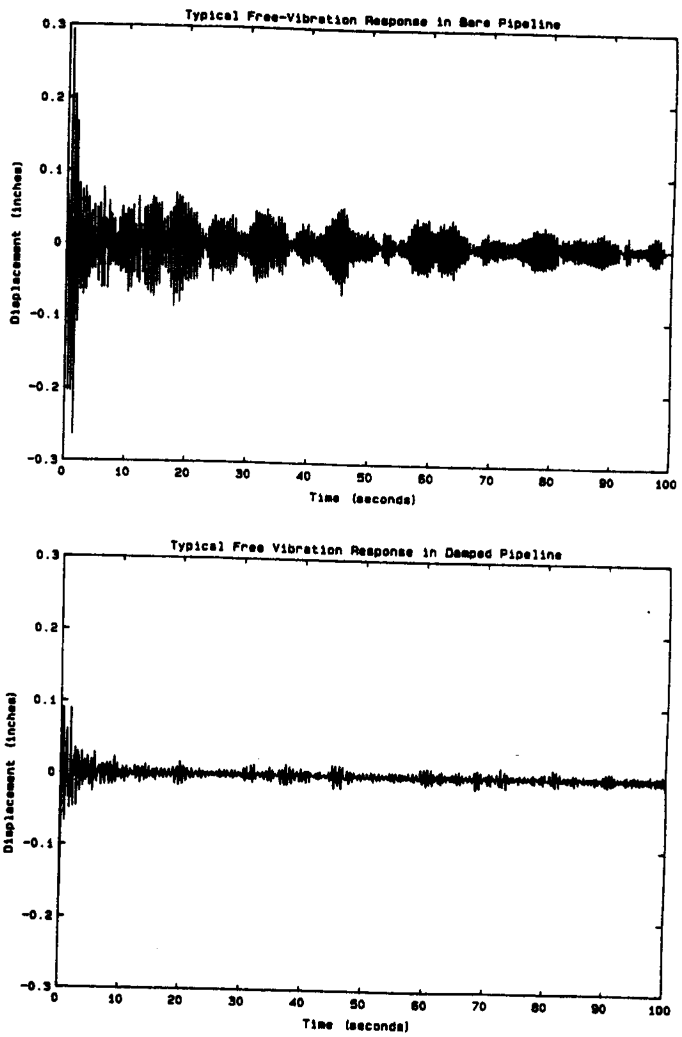

Figure 12 Free Vibration Response in Pipeline With and Without PVD's

increase in damping provided by the dampers. For the undamped section, damping ratios ranging from 0.1% to 0.5% of critical were obtained. These results are consistent with the results obtained in the initial investigation where a damping ratio of 0.4% was obtained. For the damped section, effective damping ratios ranging from 1.8% to 2.5% of critical were obtained. As shown in Figure 12, the pipeline free-vibration recorded in the damped pipeline spans decays quite rapidly, and hence provides a relatively poor definition of the vibration frequencies participating in the response. However, the vibration frequencies range from approximately 1.5 to 4 Hz, which is consistent with the predicted frequency range.

7.2 WIV Data

The data acquisition system was programmed to record data after initiation by either a displacement trigger or a wind trigger. A displacement trigger occurs when the root-mean-square (RMS) of the integrated displacement of the pipeline at one midspan accelerometer station falls within a specified range. A wind trigger occurs when the average wind speed, resolved perpendicular to the pipeline alignment, at one meteorological station falls within a specified range.

Nearly 1000 events were recorded over several months. Processing of the data was focused on a few events that represented more extreme vibration conditions. Each event was characterized based on wind properties and vibration in the unmitigated, straked and damped pipeline sections. Approximately 35% of the data was previewed and classified using the following procedure:

(1) Obtain the average, standard deviation, and the minimum and maximum values of the wind speed perpendicular to the pipeline at meteorological stations for the entire event and determine a two minute segment of perpendicular wind speed data that most nearly corresponds to steady-wind conditions.

(2) Filter and integrate the acceleration data from midspan stations to obtain displacement time histories for the "steady wind" portion of the event. Compute the minimum, maximum and standard deviation (RMS) of integrated displacement histories.

(3) Develop RMS reduction measures with the following procedure:

RMS_bare=max of[RMS(bare displacement histories)]
RMS_straked=max of[RMS(straked displacement histories)]
RMS_damped=max of[RMS(damped displacement histories)]

<u>RMS reductions:</u>
BAREoverSTRAKED = RMS_bare/RMS_straked
BAREoverDAMPED = RMS_bare/RMS_damped
STRAKEDoverDAMPED = RMS_straked/RMS_damped

(4) Plot the perpendicular wind speed at the meteorological stations, the integrated displacement history, and the corresponding FAS from one bare span, one straked span, and one damped span.

7.3 Comparison of Systems

An assessment of the helical strake and PVD systems, which outlines the costs, advantages, and disadvantages is presented below. It should be noted that, although the field evaluation provided a side-by-side comparison of the mitigated sections with respect to an unmitigated section, it is a limited basis for conclusions since the evaluation was conducted on a single pipeline system over a short duration with respect to the life of an oil field.

Helical Strake System:

(1) the cost of construction and installation was approximately 500$ per span installed,

(2) the complicated strake pattern and geometry makes field installation difficult, especially in cold weather,

(3) in many of the installed strakes, variation in the strake pitch along the span length was observed,

(4) several of the installed strakes were not perpendicular to the surface of the pipe jacket, and in some cases were laying nearly flat on the jacket surface.

(5) the strakes tended to induce a more broad-banded (multi-modal) response than that observed in the bare pipeline,

(6) the strakes provided reduction measures that ranged from 0.6 to about 4 (a reduction measure less than 1.0 indicates an <u>increase</u> in vibration amplitude),

(7) when the straked pipeline was upwind of other pipes on the rack, reduction measures greater than 1.0 were observed,

(8) when the straked pipeline was downwind of other pipes on the rack, reduction measures less than 1.0 were consistently observed (indicating an <u>increase</u> in vibration amplitude).

Pipeline Vibration Dampers:

(1) the cost of construction and installation was approximately 130$ per span installed,

(2) the dampers can be pre-assembled and easily attached to the pipeline and suspended weight,

(3) the deformed lengths of the dampers range from about 6 inches for the 3 tier system to about 14 inches for the 5 tier system; these lengths are acceptable.

(4) the dampers tended to induce a more broad-banded (multi-modal) response than that observed in the bare pipeline,

(5) the PVD system provided reduction measures that ranged from 1.5 to about 7, and an increase in vibration amplitude was <u>never</u> observed (an example of damper performance is given in Figure 13),

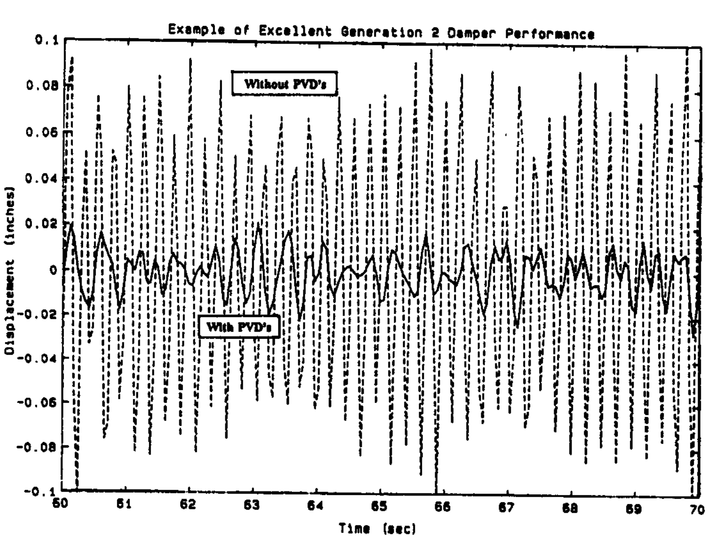

Figure 13 Example of PVD Performance

(6) the performance of the PVD system was not influenced by wind direction,

(7) as illustrated in Figure 14, the performance of the PVD system improved with the amplitude of the bare pipeline (i.e., the larger the amplitude in the bare pipeline, the better the damper performance),

Comparison of Strake and Damper Systems:

(1) STRAKEDoverDAMPED reduction measures ranging from 1.2 to 4.8 were computed, thus the strake system <u>never</u> outperformed the PVD system,

(2) the strakes consistently <u>increased</u> the vibration amplitudes when they were downwind of adjacent pipelines on the rack while the PVD's <u>never</u> increased vibration amplitudes under any wind conditions,

(3) the PVD system is significantly less expensive and easier to install than the strakes.

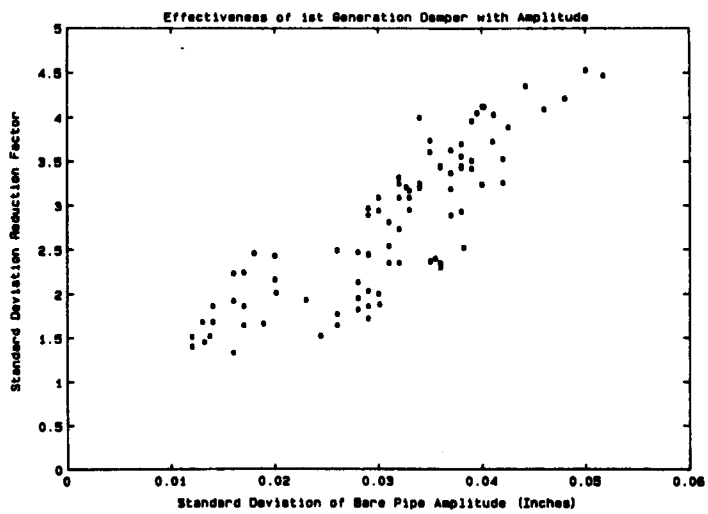

Figure 14 Effectiveness of PVD's with Amplitude

7.4 Conclusions from the Field Evaluation

Based on the observations reported in the previous sections, it was concluded that: (1) the costs, the construction and installation difficulties, the increased amplitudes when attached to a pipeline downwind of other pipes on a rack, and the uncertain influence on downwind pipelines, make the helical strake system unsuitable for field implementation; and (2) the performance, the compactness in both the deformed and undeformed configurations, the modularity (with the tiers and weights) that can provide tuning over a broad range of frequencies, and the cost, make the PVD suitable for field-wide implementation. Furthermore, the side-by-side field evaluation indicated that the PVD system significantly outperformed the strake system at one-quarter of the cost.

8. Design and Analysis of Field-Wide PVD System

Based on the field evaluations, it was determined that the PVD system was suitable for field wide implementation to reduce WIV amplitudes. The mitigation effort was directed at cross country and drill site pipelines with 6 to 12-inch nominal diameters and spans ranging from 30 to 90-feet aligned perpendicular to the prevailing wind directions (see Figure 1). This classification represents over 14,000 spans.

8.1 Design Methodology

Design Objectives. Although the PVD system is effective at reducing vibration amplitudes, it cannot completely eliminate vibration. The field wide design effort had to consider a range of vibration frequencies for each pipeline, and the broad temperature range to which the systems are exposed. The following design objectives were established, based on evaluations of regional weather data, the results of earlier field evaluations, and the weld test data:

(1) Damper placement and frequency tuning aimed at reducing vibration in the mode shapes with corresponding frequencies bounded by frequencies for a single span with pinned- and fixed-end boundary conditions. These vibration shapes are illustrated in Figure 4. Higher frequency vibration shapes were not considered.

(2) Use of dampers of three different frequencies (i.e., triple tuning) on each pipeline system to strive for wind-induced stress levels below a ceiling value of 3.4 ksi for all of the modes in the specified frequency range.

(3) Performance of the damper at a full range of relevant temperatures, especially in the low temperature regime where the stiffness and damping ratio both increase dramatically. The low temperature stiffness increase can limit the effectiveness of the damper due to loss of frequency tuning, but this effect may be compensated for by the corresponding increase in damping. Analysis of wind data associated with low temperatures over a 20 year period indicates that the loss of damper effectiveness at -45° would mean that the pipelines were threatened by WIV 0.9% of the time. Based on this observation, a low end design temperature of -45° F was selected. The high end design temperature was 50° F.

Design Procedure. Application of the dampers to a pipeline system involves tuning the natural frequency of the damper to the natural frequency of the pipeline system. There are two main issues to be considered in applying the damper: (1) the stiffness of the elastomeric springs depends on the ambient temperature, the age of the elastomer, the magnitude of the strain induced by movement of the mass, and the frequency of vibration; (2) the pipeline system will typically have numerous natural vibration frequencies and associated modes of vibration that need to be controlled. These issues are addressed as follows:

(1) A characteristic damper frequency is predicted based on the average ambient temperature (about 10° F), the average elastomeric stiffness considering aging (stiffening) over the damper service life, the maximum expected strain in the elastomer, and the required damper frequency. The characteristic damper frequency is tuned to the required frequency by varying the number of elastomeric springs and the weight of the suspended mass.

(2) For typical above-ground pipelines, which exhibit bands of closely-spaced natural frequencies and associated modes (i.e., the band of modes with frequencies between the single span pinned- and fixed-end modes) which require vibration control, three dampers are employed. A low frequency or L damper is tuned to lowest natural frequency in the band, a high frequency or H damper is tuned to highest natural frequency in the band, and a medium frequency or M damper is tuned to a frequency approximately half way between the lowest and highest frequencies in the band.

Design Evaluations The performance of each design was evaluated with respect to the design objectives using the SSD aerodynamic model. For a given pipeline system (defined by the diameter, wall thickness, contents and span length) the procedure was: (1) build a finite element model of the undamped multi-span pipeline system; (2) estimate the uni-modal response of the undamped system using lift coefficients corresponding to laminar and turbulent wind conditions, considering ambient temperatures of -45° F, +10° F, and +50° F to account for the effect of temperature on Reynold's number and air density; (3) select a system of high (H), medium (M), and low (L) frequency dampers (defined by the number of tiers and suspended weight) based on the frequencies obtained from the undamped model; (4) develop three additional finite element models to represent the pipeline including the added dampers using the damper stiffness values at -45° F, +10° F, and +50° F; (5) use a specially developed multi-degree-of-freedom procedure to compute the effective modal damping ratios of the multi-span pipeline system with dampers attached at midspan locations at temperatures of -45° F, +10° F, and +50° ; and (6) estimate the uni-modal (stress) response of the damped pipeline system using lift coefficients corresponding to laminar and turbulent wind conditions, and considering temperatures of -45° F, +10° F, and +50° F to account for the effect of air temperature on the Reynold's number, air density, and damper properties. Wind-induced stress spectra were then compared with the ceiling value of 3.4 ksi.

The results from these evaluations indicate that, under turbulent wind conditions all of the designs satisfied the stress ceiling value of 3.4 ksi. However, under laminar wind conditions many of the designs exceed the stress ceiling value of 3.4 ksi; although the criteria is usually exceeded for only one or two modes of vibration at the bounding temperatures (i.e., at -45° F or +50° F rather than at +10° F). A few of the designs are highly stressed at high modal frequencies, but these results are at frequencies well above the frequencies corresponding to the random shedding threshold and thus can be discounted.

Because the stresses under laminar wind conditions exceed the 3.4 ksi stress ceiling for certain designs, there was a need to further document the effectiveness of the damper system. For several designs, spectra of the RMS stress response of the damped and undamped pipelines were compared. The reduction in WIV stress levels demonstrated the effectiveness of the PVD system designs.

8.2 Assessment of Field Wide Design

There were many difficulties involved in the development of a set of PVD systems that would protect the extensive inventory of 6 to 12-inch nominal diameter pipelines. The difficulties arose because the pipeline inventories include a large variety of pipeline diameters, wall thicknesses, span lengths, and contents as well as variations in the number of spans within straight line runs. Several simplifications were made in evaluating the damper systems. Many of these simplifications were investigated by sensitivity studies, others were not because of time limitations.

Based on these considerations and on the design evaluation and sensitivity studies, the following performance can be expected from pipeline systems included in the field wide design:

(1) Stresses due to WIV under turbulent wind conditions should remain below the ceiling of 3.4 ksi. Since 3.4 ksi is the infinite fatigue life stress range for welds containing critically positioned typical weld defects, the damper systems should provide pipelines (with or without typical weld defects) adequate protection from fatigue damage during vibration under turbulent wind conditions.

(2) Under laminar wind conditions, stresses due to pipeline vibration will exceed the 3.4 ksi ceiling in certain modes of vibration. However, these stresses should remain below 6.8 ksi. Since 6.8 ksi is the infinite fatigue life stress range for welds without defects, such welds should be adequately protected by the damper system from fatigue damage during vibration under laminar wind conditions. Vibration under laminar wind conditions may cause fatigue damage and failure in welds with critically positioned defects.

In order for a pipeline with a PVD system to accumulate fatigue damage in a weld with defects during vibration in laminar WIV, the weld must be near a location of maximum stress along the pipeline. The location of maximum stress along the pipeline depends on the mode of vibration, and the stress exceeds the 3.4 ksi ceiling for a few modes, only. Vibration under laminar wind conditions does not appear to be common in the Kuparuk oil field. Vibration under laminar conditions was not observed during data collection in the field, however qualitative observations suggest that these vibration events may occur. Thus the potential for fatigue damage cannot be discounted.

In conclusion it should be noted that the damper systems will significantly reduce pipeline vibrations stress in both turbulent and laminar wind conditions. The objective of reducing the stress to a level below the 3.4 ksi ceiling was not achieved under all environmental conditions for all pipelines included in the field-wide design. Thus a conclusive assessment of the possibility of future fatigue damage and fatigue failures cannot be made without further study. However, the PVD is clearly effective in reducing WIV amplitudes and the associated fatigue threat.

9. Conclusions

The pipeline wind-induced oscillation studies undertaken at the Kuparuk oil field are summarized in this paper. The paper has presented the dynamic properties of above ground pipelines and characteristics of wind induced pipeline vibration determined from field data, the results from analytical models and correlation with field data, and the results of the pipeline testing and the field weld survey. This information outlines the nature of the wind-induced pipeline vibration problem and the threat it poses. The paper reviewed WIV mitigation systems, and presented the design and results from field evaluation of two systems.

It was concluded that the helical strake system was not suitable for field-wide application. The PVD system was found to be suitable for field-wide application. The field-wide design proved to be challenging because large variations in numerous design parameters had to be considered. Evaluation of the PVD system designs indicated that the dampers will significantly reduce pipeline vibration stress. The objective of reducing the stress to a level below the 3.4 ksi ceiling was not achieved under all environmental conditions for all pipelines include in the field-wide design. However, the PVD system is remarkably effective in reducing WIV amplitudes and the associated fatigue threat.

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