

# **Evaluation Report: E 1937 X 12**

**Published by: Evaluation International, December 2012**

Index classification: 2.1

## **THERMOWELL VALIDATION TESTS VortexWell**

**Manufacturer: Okazaki, Japan**

**INTERNATIONAL INSTRUMENT USERS' ASSOCIATIONS  
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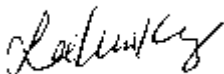

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## THERMOWELL VALIDATION TESTS

A Report for

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Date: 8 May 2012

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## **Thermowell Validation Tests - VortexWell**

Manufactured by Okazaki, Japan

Evaluated by TUV SUD NEL, Glasgow, UK

Author: W K Lee

TUV SUD NEL reference: Project No: EVI003 - Report No: 2012/194 (Revised)

Evaluation International Project No: 139

Report No: E 1937 X 12

Index classification 2.1

Publication date: December 2012

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### **SUMMARY**

Up to now, if a thermowell failed the ASME Performance Test Code (PTC 19.3, 2004), the manufacturer has been left with several options: either to shorten the thermowell immersion, or to increase the diameter of the thermowell, neither of which is often very practical or cost effective for the user. The only other option used by the majority of thermowell suppliers is to incorporate a velocity collar on the thermowell in order to move the point of vibration or resonance.

Okazaki has developed a unique design of thermowell, the VortexWell®, which does not require a velocity collar and is cost effective for the end user in terms of purchase, installation and maintenance costs (whole lifecycle costs). VortexWell® incorporates an innovative helical strake design, very similar to the helical strakes seen on car aerials and cooling towers. By using the latest CFD software to visualise the flow behaviour, Okazaki was able accurately to compare a standard tapered thermowell and its new VortexWell®. In the analyses, the standard tapered thermowell showed classic shedding behaviour as expected, whereas the VortexWell® demonstrated no signs of regular flow behaviour. The VortexWell® helical strake design disturbed the flow sufficiently to interrupt the regular formation of vortices. Whilst a small vortex was observed in the wake of the VortexWell® this was a localised stagnation point and did not shed.

In 2011 Evaluation International contacted NEL Ltd to discuss a series of laboratory tests and analyses to evaluate the resilience of two types of thermowell, namely standard thermowell and Okazaki VortexWell, to be operated in the oil and gas flow facilities. The initial proposal (NEL-7794) was established in 2011, but the work scope was subsequently revised and agreed on 1<sup>st</sup> March 2012 prior to the start of this project. Consequently, the laboratory tests were commissioned, refined and conducted in NEL's oil flow facility in East Kilbride, Glasgow.

In parallel with the laboratory tests, both finite element (FE) and closed form techniques were employed to verify test results. Having obtained the test and analytical data, a comparative study was also instituted to evaluate the dynamic performance and mechanical strength of the designated thermowells. Apparently, the test results indicate that the VortexWell has significantly outperformed the standard thermowell. However, care must be exercised when interpreting the test data summarised in the report, as the standard thermowell was installed 10 diameters downstream of the VortexWell. As such, the performance of the standard thermowell might be distorted by the wake of the VortexWell because both thermowells exhibited more or less the same excitation frequencies, although the excitation magnitudes of the standard thermowell were observed to be much higher.

During the data analysis phase of this study, it was observed from the strain data that the standard thermowell experienced some phase changes and, in some cases, it reached a 180° phase change. This indicated that the standard thermowell was excited and operated within the range of first natural frequency. This phenomenon was also predicted by both FE and closed form modal analyses. Indeed, a much higher excitation magnitude of the standard thermowell was noted throughout the laboratory tests.

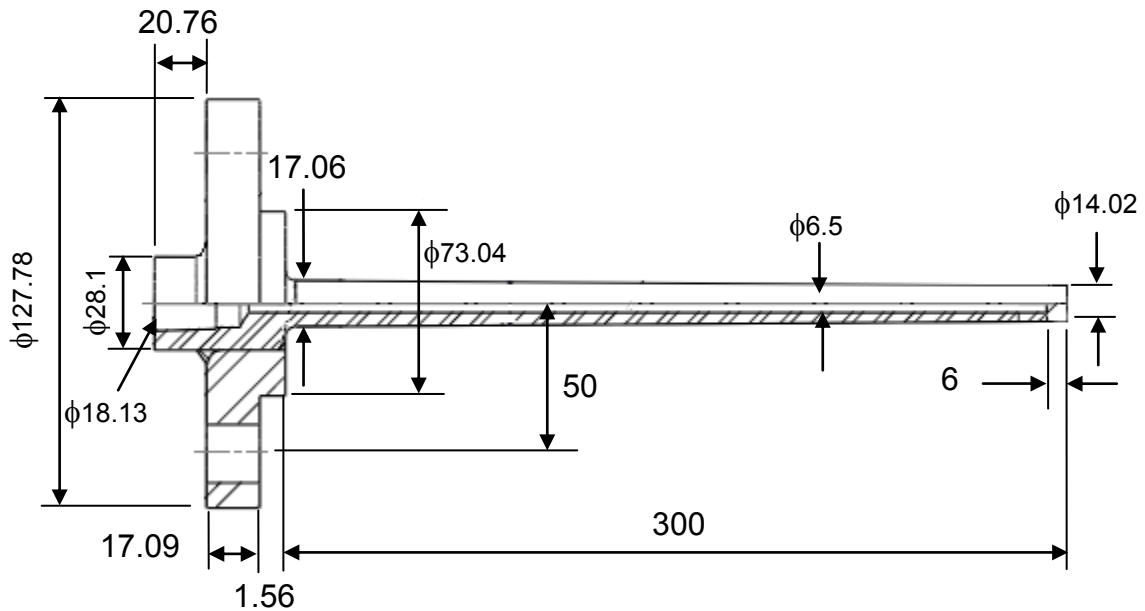
To objectively verify and hence confirm the findings, it is recommended that:

1. a computational fluid dynamic simulation (CFD) be undertaken to investigate if unsteady flows and installation effects could affect the dynamic performance and hence induce higher loading magnitude over the two thermowells;
2. further laboratory tests be carried out to individually and independently test the VortexWell and standard thermowell on NEL's flow process line;
3. an experimental modal analysis be carried out to verify if natural frequencies of these two thermowells could be influenced by installation effects and unsteady flows.

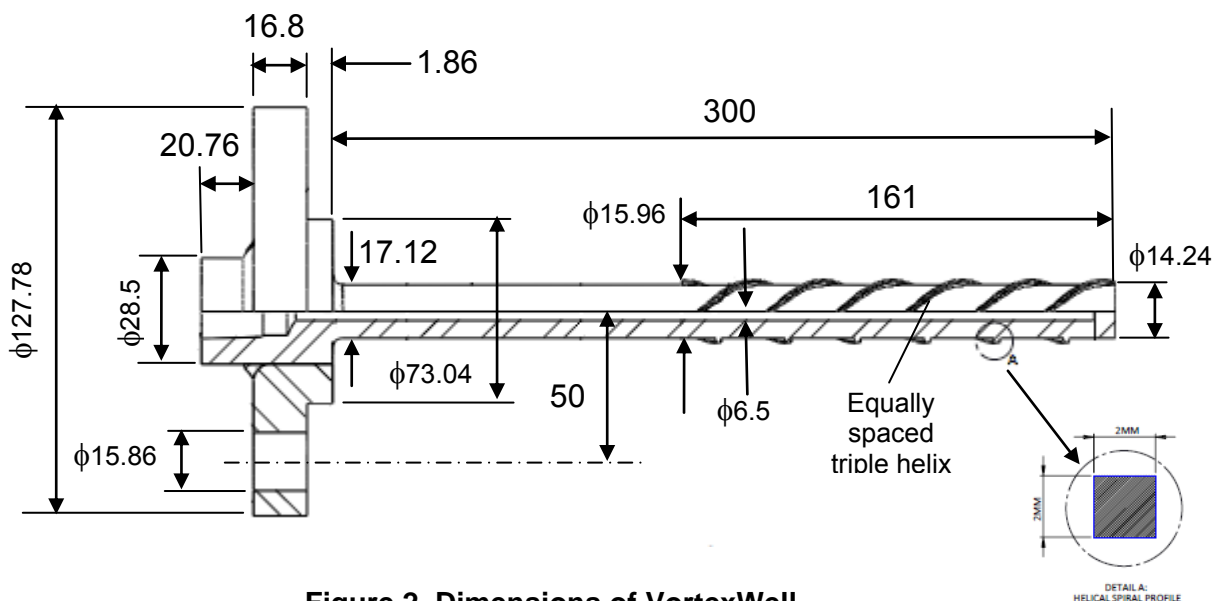
# 1 INTRODUCTION

A thermowell is an intrusive metallic housing inserted into a pipeline or a vessel so as to enable operation of a thermometer or thermocouple for the purpose of measuring the temperature of the flow or content. A thermowell is designed to provide a pressure / temperature / structural boundary without introducing unacceptable measurement uncertainties and time lags [1].

In 2011, NEL was contracted by this client to commission an experimental and analytical investigation to assess and compare the resilience (i.e. dynamic performance and mechanical strength of the structural boundary) of two types of thermowell, namely the standard and Okazaki VortexWell. Figures 1 & 2 present the dimensions, tapering and spiraling features on the specimens supplied by the client.



**Figure 1 Dimensions of Standard Thermowell**



**Figure 2 Dimensions of VortexWell**

In parallel with laboratory tests at NEL's flow facilities, analyses were also conducted in April 2012. This report which forms the deliverable of the project, outlines the methodology, laboratory tests and desk top analyses adopted to derive results, discussions, conclusions and recommendations.



## 1.1 Scope

By agreement with Evaluation International, the following activities were contracted and undertaken:

- To each designated thermowell body single element strain gauges were bonded at three locations along the outer body, Figure 6;
- The strain-gauged thermowells were inserted in a pipe section supplied by the client and installed in a 6 inch diameter flow process line at NEL;
- The two strain-gauged thermowells were subjected to a series of flow tests as summarised in the methodology section and Appendix A of this report;
- Modal and stress analyses w based upon the strain data obtained from the laboratory tests were undertaken using a computational software suite entitled SolidWorks / CosmoWorks Professional;
- A report summarising the work, its findings and any recommendations would be provided and would constitute the deliverable for this project.

## 1.2 Objective

The object of this work was to obtain information on the possible effects of vortex shedding upon the dynamic performance and mechanical strength of a standard tapered thermowell and an Okazaki VortexWell. Fundamentally, this study compares both excitation frequencies and stresses imposed upon the structural boundaries of the designated thermowells under a set of predetermined laboratory test conditions.

## 1.3 Methodology

To undertake this programme, two thermowells supplied to NEL by the client were instrumented using single element strain gauges positioned in the same locations, Figure 6. Each strain gauged thermowell was individually inserted into a pipe section which was then installed in a 6 inch diameter oil flow process line of NEL's flow facility. To minimise flow interactions between the two thermowells, they were separated by a distance of around 10 pipe diameters in the process line.

For the laboratory tests, 30 test points were generated from each individual specimen. At each test point, the flow rate was maintained for a minimum of 5 minutes with the frequencies and strains recorded from each thermowell. The test parameters of flow rate, fluid density, fluid viscosity, Reynolds numbers, frequencies and strain measurements were tabulated for each of the tests. The test parameters and conditions are summarised in Appendix A, Tables A1 to A3 of this report.

In conjunction with the test work, desk top modal and stress analyses which based upon the strain data obtained from the laboratory tests were conducted using SolidWorks and CosmoWorks Professional. The measured and predicted stress levels were then compared to determine if there was an improvement of dynamic performance and level of stress of the VortexWell over the standard thermowell.

Lastly, technical issues concerning the techniques employed and findings were discussed and recommendations were made on completion of this programme.

## 2 TEST SETUP

A schematic of NEL's oil calibration flow line is shown in Figure 3. As can be seen, the working fluid was re-circulated around the test facility using two variable speed pumps and maintained within a large supply tank. In addition, two reference meters were fitted downstream of the pumps providing live data for monitoring the flow conditions of the facility.

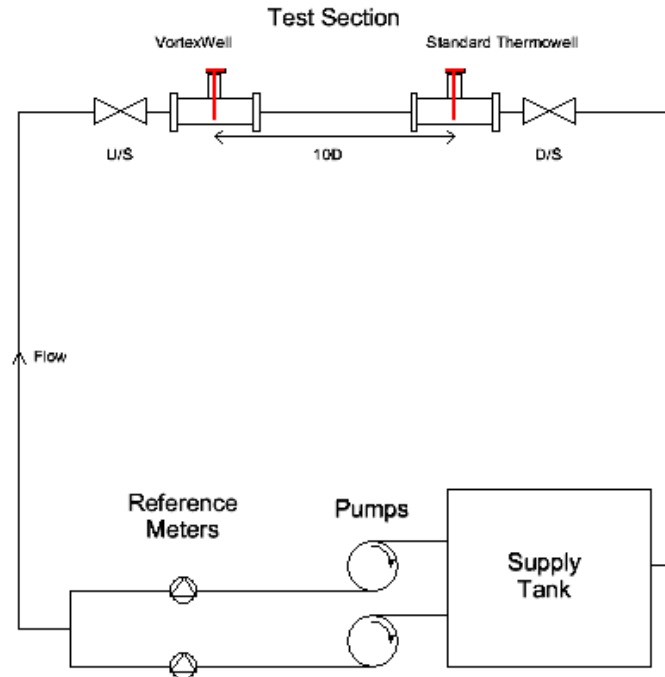


Figure 3 Recirculation Test Setup

### 2.1 Set Up for Testing Thermowells

Also shown in Figure 4 is a photograph of the test assemblies installed in the oil calibration flow line with a separation of 10 pipe diameters between the strain gauged thermowells. The flow rate was determined from pre-calibrated reference flowmeters, which could be operated singly, or in parallel, to cover the required range of flow rate.

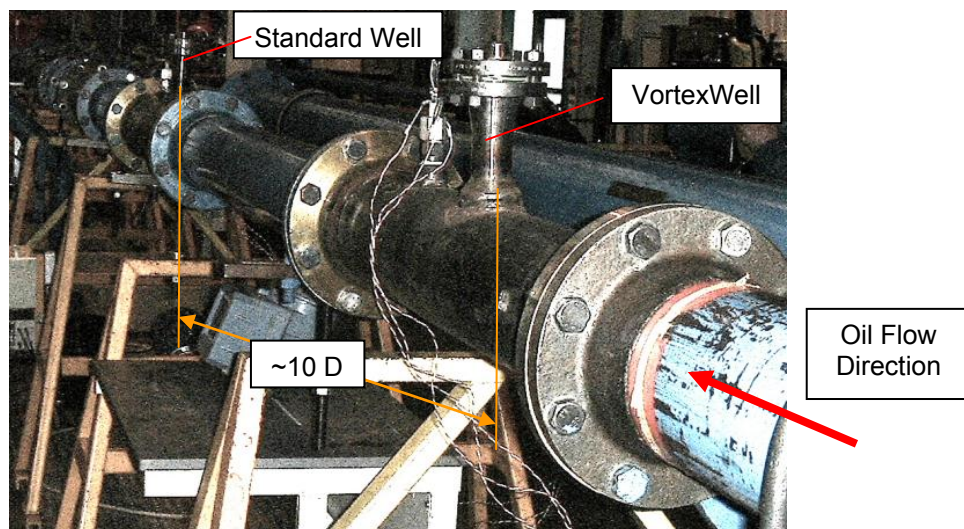
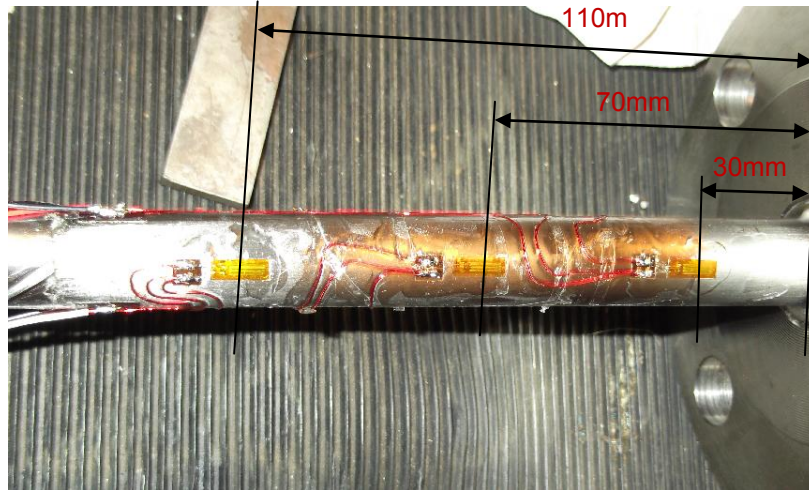


Figure 4 Test Layout

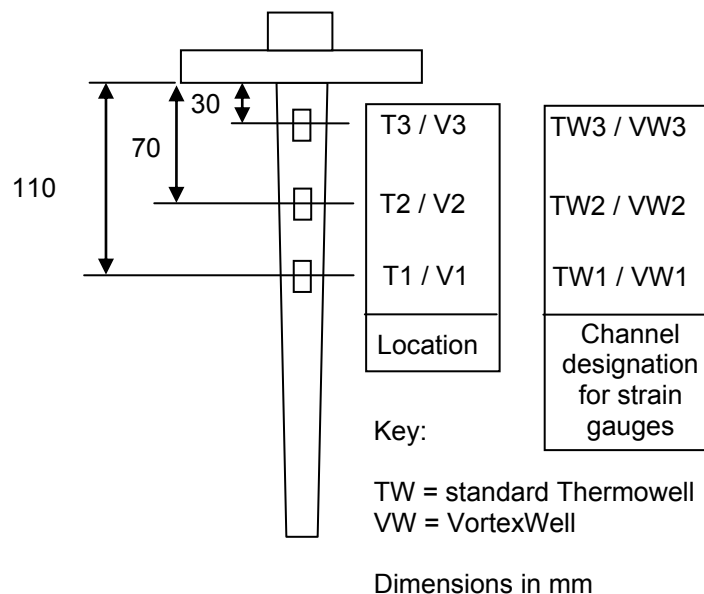
## 2.2 Strain Gauging

Three single element strain gauges were bonded on each thermowell outer shell. Figure 5 shows the three strain gauges bonded on the VortexWell. For the purpose of comparing dynamic performance and mechanical strength, the three strain gauge locations were chosen to be bonded on three different locations which were identical for each thermowell.



**Figure 5 Single Element Strain Gauges bonded on the VortexWell**

A schematic of the thermowell strain gauge locations and the designations for identifying strain gauge channels is shown in Figure 6.



**Figure 6 Schematic Diagram of Strain Gauges positioned on Each Thermowell**

Please note that T1, T2, T3, ...V3 are numbers for distinguishing strain gauge locations and TW1, TW2, TW3,...VW3 are designations of strain gauge channels.

### 2.3 Initial Verification Tests

To confirm that the bonded strain gauges could be accurately operated throughout the laboratory tests, all gauges have been tested using a calibrated strain gauge tester. Prior to the laboratory tests in the 6" flow process line, the strain gauges installed on the thermowells were independently validated by using a 1kg static load test as shown in Figure 7. The strain measurements were then validated using CosmoWorks simulations and MathCad calculations based upon closed form cantilever formula. The initial verification results were compared and are presented in Table 1 below.



Figure 7 Preliminary Strain Measurement for Validating FE Meshes

TABLE 1

STRESSES AND STRAINS OBTAINED FROM 1KG STATIC LOAD TEST  
(Standard Thermowell & VortexWell)

	Location ID	CosmoWorks Simulation		MathCad Verification	
		Stress N/mm <sup>2</sup>	Micro Strain	Stress N/mm <sup>2</sup>	Micro Strain
Standard Thermowell	T1	5.40	25.7	5.02	23.9
	T2	6.03	28.7	6.07	28.9
	T3	6.70	31.9	7.19	34.3
VortexWell	V1	5.43	25.8	4.92	23.5
	V2	5.94	28.3	5.96	28.4
	V3	6.32	30.1	7.09	33.8

## 2.4 Manufacturer's Comments and Test House's responses

**Note** – the TUV-NEL responses are given in red

### 2.4.1 Velocity of Oil

The offer letter says "Max Velocity : 11M/sec". However, the actual flow was only up to 5.36M/sec. Because of this, neither VortexWell nor standard thermowell reached its resonance zone.

**Test House Response:** Unfortunately, due to strain gauge failure the test was stopped before maximum velocity could be achieved.

### 2.4.2 Direction of strain gauge

Although the locations of strain gauges are clearly mentioned in the report, it does not say if they are parallel or perpendicular to the flow direction. Guessing from the test result, they were facing to the flow, which means they were only checking in-line oscillation. In hindsight, the strain gauges should have been bonded to the stem X and Y directions in order to observe both in-line and transverse forces.

**Test House Response:** The strain gauges were perpendicular to the flow direction in accordance with discussions between NEL and Okazaki. We accept that possibly the strain gauges should have been bonded to the stem X and Y directions in order to observe both in-line and transverse forces.

### 2.4.3 Locations of two thermowells

It was wrong to place these two thermowells in line. We notice that TUV kept a sufficient distance (approx 10D) between the two thermowells which appears to be a standard practice in terms of fluid dynamics. Having said that, each thermowell should have been tested one at a time.

There is one important thing to confirm. ASME PTC19.3-2010 recommends  $f_s/f_n$  should be less than 0.4 under a certain condition. This is because the in-line forces get highest around 0.5, according to ASME PTC19.3-2010. According to the test report on pages 12 and 13, the stress and strain level suddenly gets high at 3.22m/sec (According to Kazaoka san's calc,  $f_n/f_s$  is 0.338) and gets higher at 4.28m/sec ( $f_s/f_n$  0.443). Based on this result, ASME PTC19.3-2010 could be wrong since the in-line forces got highest around  $f_n/f_s$  0.4, not  $f_n/f_s$  0.5.

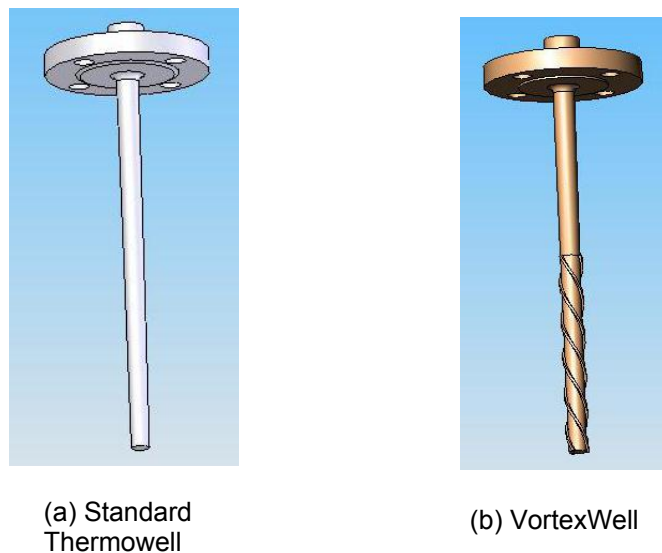
**Test House Response:** TUV-NEL has now included additional CFD work (Appendix 4) undertaken to verify no vortex influence from the upstream Thermowell was experienced.

### 3 FINITE ELEMENT (FE) MODELS

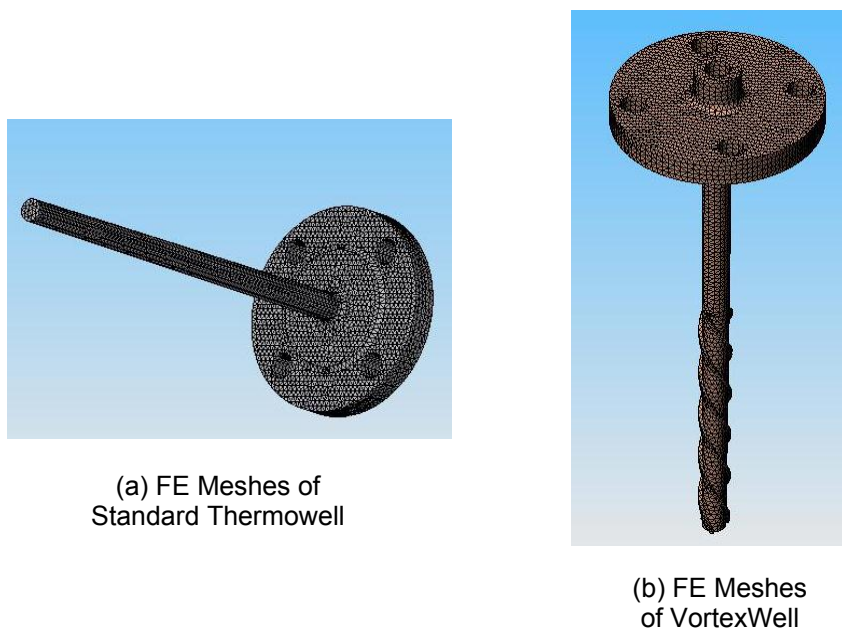
The FE models of the two thermowells were developed using the dimensions taken from the specimens supplied to NEL by the client. These models were developed for predicting natural frequencies, mode shapes, principle stresses, Von Mises stresses and factor of safety of the designated thermowell designs. Alternatively, one can use classical elastic-plastic theories for determining the stresses over the structural boundaries of the thermowells and the techniques were reported in Reference [1].

#### 3.1 Generation of FE Models and Meshes

Figures 8 and 9 respectively illustrate the CAD models and FE meshes of the designated thermowells generated using SolidWorks and CosmoWorks professional.



**Figure 8 SolidWorks Models of Thermowells**



**Figure 9 FE Meshes of Two Designated Thermowells**

### 3.2 Mechanical Properties of Thermowells

In order to simulate the mechanical strengths and dynamic performance, the mechanical properties of materials for the two thermowells, as agreed, were assumed to be homogenous plain carbon steel, which was selected from the CosmosWorks' material library, Table 2.

TABLE 2

MECHANICAL PROPERTIES OF MATERIAL

	Symbol & Unit	Quantity
Young's Modulus	E (N/mm <sup>2</sup> )	210e3
Poisson's Ratio	$\nu$	0.28
Ultimate Strength	$\sigma_{UTS}$ (N/mm <sup>2</sup> )	399.8
Density	$\rho$ (kg/m <sup>3</sup> )	7800
Yield Strength	$\sigma_{yld}$ (N/mm <sup>2</sup> )	220.6

### 3.3 Boundary Condition and Load Simulation

As can be observed, the boundary condition chosen for the FE models simulated, as close as possible, the test configuration and dynamic behaviour of the two thermowells. Figure 10 shows the boundary condition and load applied to the standard thermowell model. These conditions have been applied to both modal and stress analyses throughout this study.

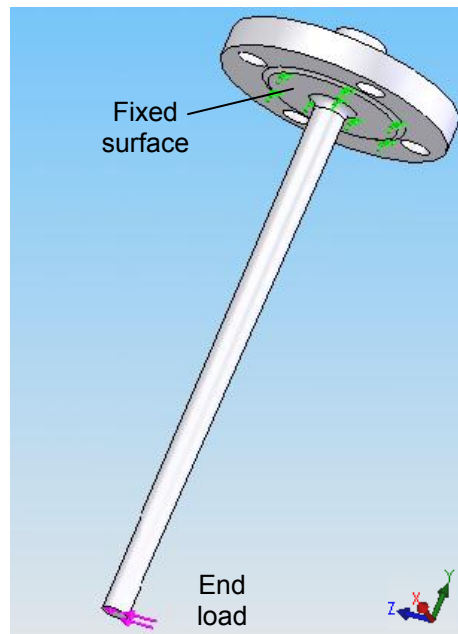
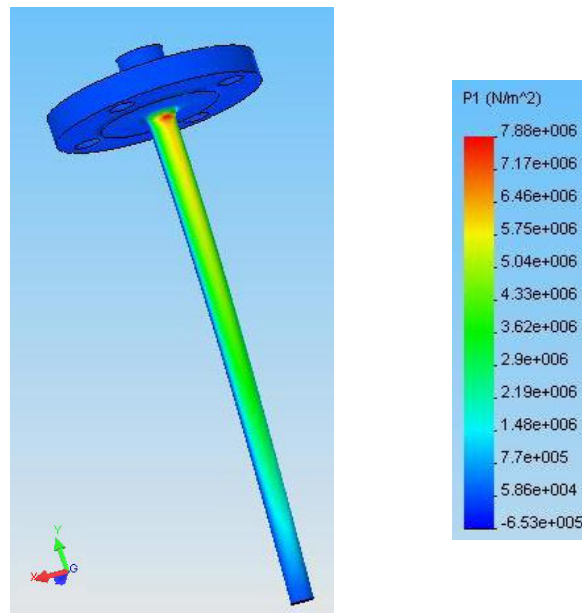


Figure 10 Load and Boundary Condition

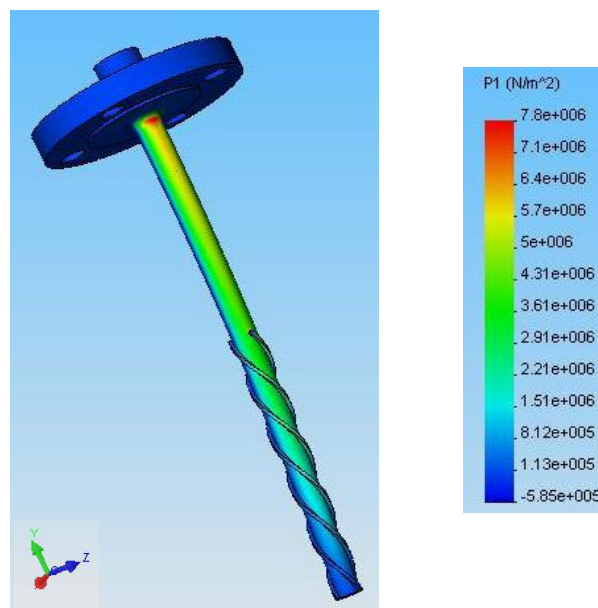
As illustrated in Figure 10, end loads applied to the two FE models were chosen to simulate the fundamental deformation of a cantilever beam under the influences of the dynamic excitation as observed during laboratory tests.

### 3.4 Validation of FE Models

Instead of undertaking a series of convergent exercises to establish representative computational mesh sizes for FEA iterations during solution runs, the mesh sizes selected for the FE models validated against strain readings measured via the 1kg static load tests as presented in Figure 7. Good agreements between the strain data, MathCad calculations and FE predictions were achieved prior to the commission of any solution runs for simulating stress contours induced over the two thermowells operating under the test conditions. Figures 10 & 11 show the resultant stress contours mapped over the two thermowells.



**Figure 11 Stress Contour over Standard Thermowell**



**Figure 12 Stress Contour over VortexWell**

Note that  $1\text{e}+006 \text{ N/m}^2 = 1 \text{ N/mm}^2$ .



## 4 RESULTS

This section presents results obtained from laboratory tests and FE simulations. It should be noted that the agreed test conditions are presented in Appendix A of this report. The tests were carried out at a nominal fluid temperature varying between 19.96°C and 23.302 °C. As such the fluid density and viscosity varied slightly throughout the tests (Appendix A Table A1).

### 4.1 Test Results

The strains and corresponding stresses obtained from the laboratory tests are summarised in Table 3 through Table 7. For the purpose of reporting, the stresses and strains are summarised and presented in terms of their maximum and minimum values.

**TABLE 3**  
**MAXIMUM AND MINIMUM STRESS AND STRAIN LEVELS**  
**(Fluid Velocity = 1.07 m/s)**

Strain Gauge Channel	Maximum Level		Minimum Level	
	Micro-strain	Stress (N/mm <sup>2</sup> )	Micro-strain	Stress (N/mm <sup>2</sup> )
TW1	10.9	2.3	6.6	1.4
TW2	14.6	3.1	9.0	1.9
TW3	11.2	2.3	5.2	1.1
VW1	11.0	2.3	4.7	1.0
VW2	14.4	3.0	5.6	1.2
VW3	3.9	0.8	-4.3	-0.9

**TABLE 4**  
**MAXIMUM AND MINIMUM STRESS AND STRAIN LEVELS**  
**(Fluid Velocity = 2.14 m/s)**

Strain Gauge Channel	Maximum Level		Minimum Level	
	Micro-strain	Stress (N/mm <sup>2</sup> )	Micro-strain	Stress (N/mm <sup>2</sup> )
TW1	18.7	3.9	11.9	2.5
TW2	22.9	4.8	15.4	3.2
TW3	22.4	4.7	12.6	2.6
VW1	20.0	4.2	15.3	3.2
VW2	22.9	4.8	15.8	3.3
VW3	12.4	2.6	5.2	1.1

**TABLE 5**  
**MAXIMUM AND MINIMUM STRESS AND STRAIN LEVELS**  
**(Fluid Velocity = 3.22 m/s)**

Strain Gauge Channel	Maximum Level		Minimum Level	
	Micro-strain	Stress (N/mm <sup>2</sup> )	Micro-strain	Stress (N/mm <sup>2</sup> )
TW1	223.1	46.86	-219.2	-46.04
TW2	333.5	70.03	-230.5	-48.40
TW3	378.4	79.47	-294.9	-61.93
VW1	13.7	2.87	-0.5	-0.10
VW2	28.3	5.95	9.8	2.05
VW3	22.0	4.61	2.4	0.51

**TABLE 6**

**MAXIMUM AND MINIMUM STRESS AND STRAIN LEVELS  
(Fluid Velocity = 4.28 m/s)**

Strain Gauge Channel	Maximum Level		Minimum Level	
	Micro-strain	Stress (N/mm <sup>2</sup> )	Micro-strain	Stress (N/mm <sup>2</sup> )
TW1	371.6	79.03	-337.4	-70.85
TW2	521.5	109.51	-380.9	-79.98
TW3	601.1	126.23	-413.6	-99.16
VW1	34.7	4.00	11.7	2.46
VW2	54.2	7.28	25.4	5.33
VW3	54.2	11.38	22.5	4.72

**TABLE 7**

**MAXIMUM AND MINIMUM STRESS AND STRAIN LEVELS  
(Fluid Velocity = 5.36 m/s)**

Strain Gauge Channel	Maximum Level		Minimum Level	
	Micro-strain	Stress (N/mm <sup>2</sup> )	Micro-strain	Stress (N/mm <sup>2</sup> )
TW1	292.0	61.3	-206.1	-43.3
TW2	Failed	Failed	Failed	Failed
TW3	473.6	99.5	-261.7	-55.0
VW1	58.1	12.2	32.7	6.9
VW2	83.5	17.5	53.2	11.2
VW3	92.3	19.4	57.1	12.0

Table 8 presents the excitation frequencies derived from the strain data. Each level was an average of 10 consecutive and complete strain cycles induced over the thermowell bodies. Please note that the strain data included only frequency contents falling within the range of fundamental natural frequency of the designated thermowells. To validate their dynamic performance in terms of natural frequencies and mode shapes, FE modal analyses were employed. Please also note that the natural frequencies presented in Reference [2] were lower than those predicted in the study. This was mainly because the masses and dimensions of the thermowells presented in Reference [2] were greater than those specimens supplied to NEL for this study.

**TABLE 8**

**EXCITATION FREQUENCIES DERIVED FROM CYCLIC STRAINS**

Fluid Velocity (m/s)	1.07	2.14	3.22	4.28	5.36
Std Thermowell Frequency (Hz)	Indeterminate	142.85	~142.86	~144.93	~147.06
VortexWell Frequency (Hz)	~136.99	142.85	~142.85	~144.93	~147.06

Figures 13 & 14 show the cyclic variation of strains acquired from locations T1, T2 and T3 as well as V1, V2 and V3 over a fixed period of time in seconds. This data was obtained from strain gauges operating at the fluid velocity of 3.22 m/s and 4.28 m/s. As analysed and observed, the strain data recorded within this flow range were considered to be the most

reliable because strains running outside this range were largely influenced by either transient noise or heavy structural vibrations.

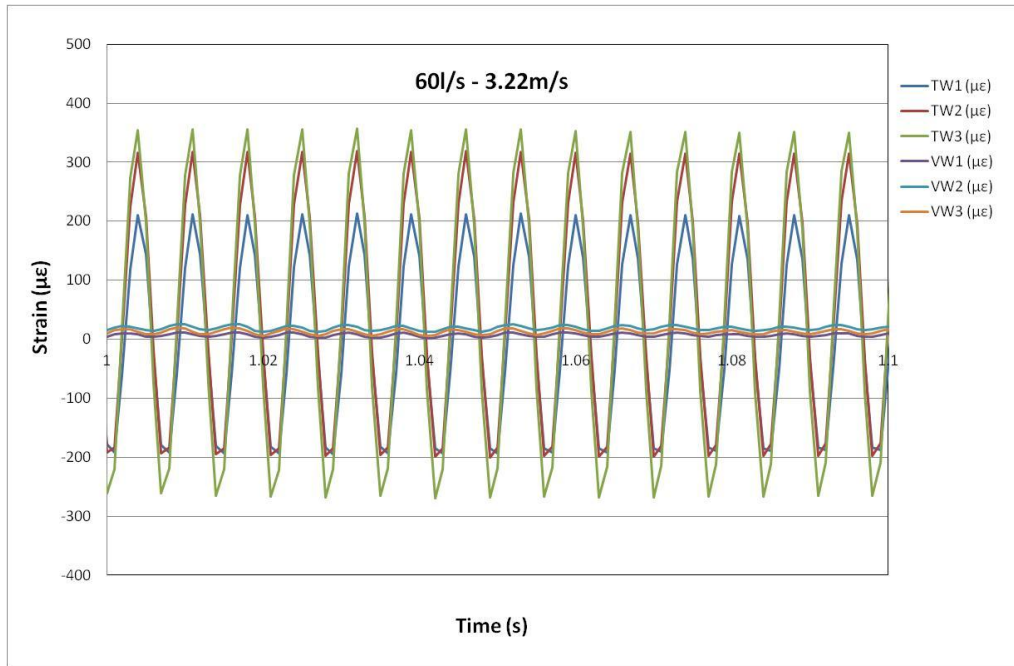


Figure 13 Cyclic Strain v Time (fluid velocity = 3.22 m/s)

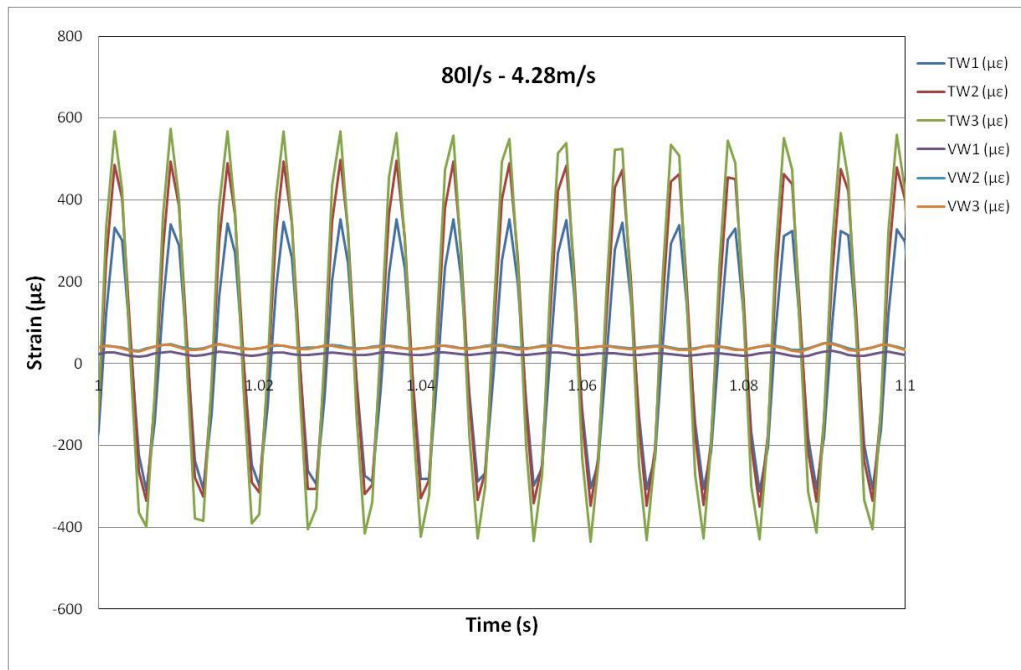


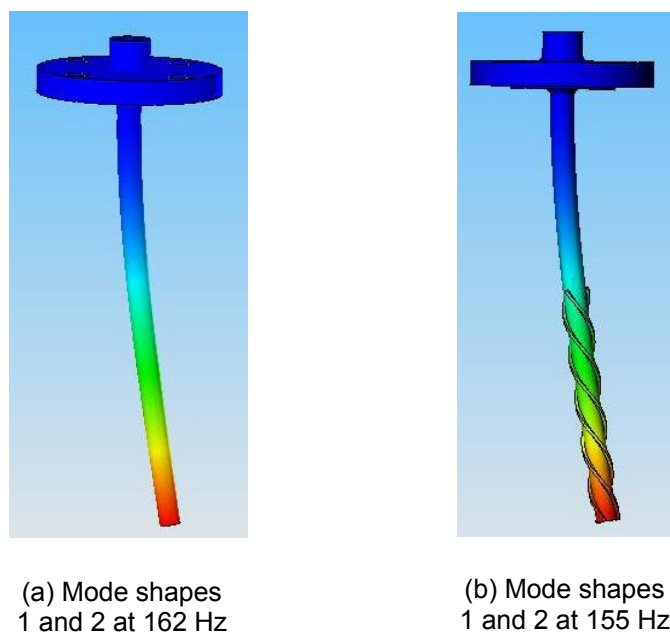
Figure 14 Cyclic Strain v Time (fluid velocity = 4.28 m/s)

## 4.2 FE Results

A series of FE simulations covering modal and static stress analyses was also conducted under this project. This sub-section summarises the FE results obtained by running CosmoWorks Professional for the study.

### (a) Modal Analysis

In order to predict the dynamic loads and hence stresses imposed on the two thermowell bodies due to excitations observed throughout the laboratory tests, a modal analysis was undertaken to predict the mode shapes and natural frequencies of the two thermowells. Figure 15 shows the first fundamental mode shapes and the corresponding natural frequencies relating to these two thermowells.



**Figure 15 First and Second Mode Shapes of Two Thermowells**

Table 9 lists the first five natural frequencies and corresponding modes obtained from CosmoWorks Professional against those derived from MathCad calculations based upon closed form cantilever equation (Appendix C, equation 4).

**TABLE 9  
NATURAL FREQUENCIES OF THERMOWELLS**

	Standard Well (FE Prediction)	VortexWell (FE Prediction)	Cylindrical Well (MathCad Solution)
<b>Mode Shape</b>	Frequency Hz		
<b>1 (x direction)</b>	161.9	154.9	136.3
<b>2 (z direction)</b>	161.9	155.1	136.3
<b>3 (x direction)</b>	876.2	880.1	852.0
<b>4 (z direction)</b>	876.9	881.5	852.0
<b>5</b>	2315.7	2342.9	2389.0

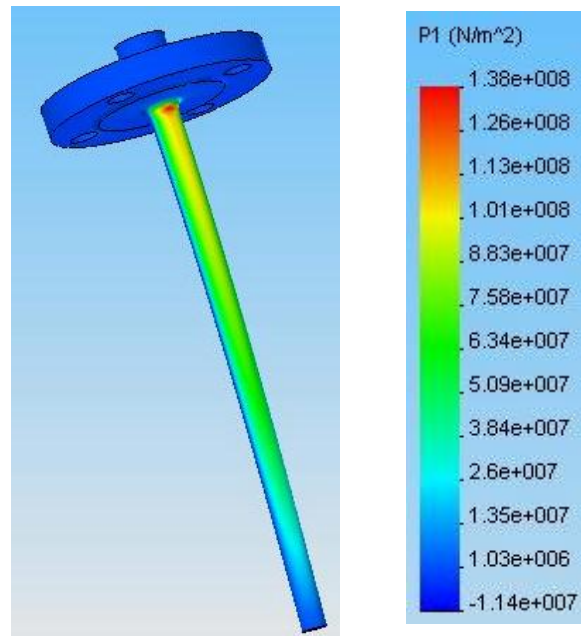
**(b) Stress Analysis**

To predict the stresses induced over the designated thermowell structures, the pressure loads on the thermowell structures were also included in the calculation, although pressure loads were measured to be relatively low in comparison with the excitation forces induced on the thermowell structures. Indeed the total mechanical load imposed on the thermowell bodies was derived from the maximum strains obtained from the laboratory tests. These loads were subsequently applied to the FE models for the purpose of predicting the mechanical stresses induced on the thermowells. Table 10 lists the mechanical (dynamic) loads derived from maximum strains measured from the two thermowells during the laboratory tests.

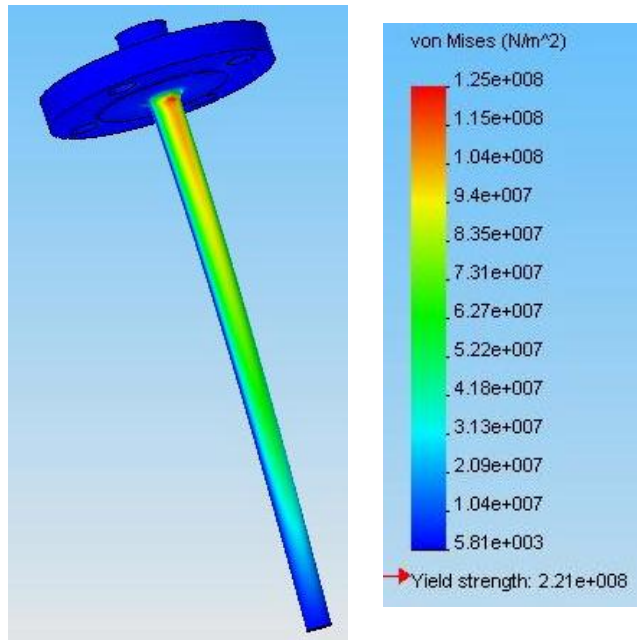
**TABLE 10**  
**DYNAMIC LOADS DERIVED FROM MAXIMUM STRAINS**

	<b>Standard Thermowell</b>	<b>VortexWell</b>
<b>Max. Strain (micro-strain)</b>	601.1	57.1
<b>Max. Dynamic Load (N)</b>	171.92	16.57

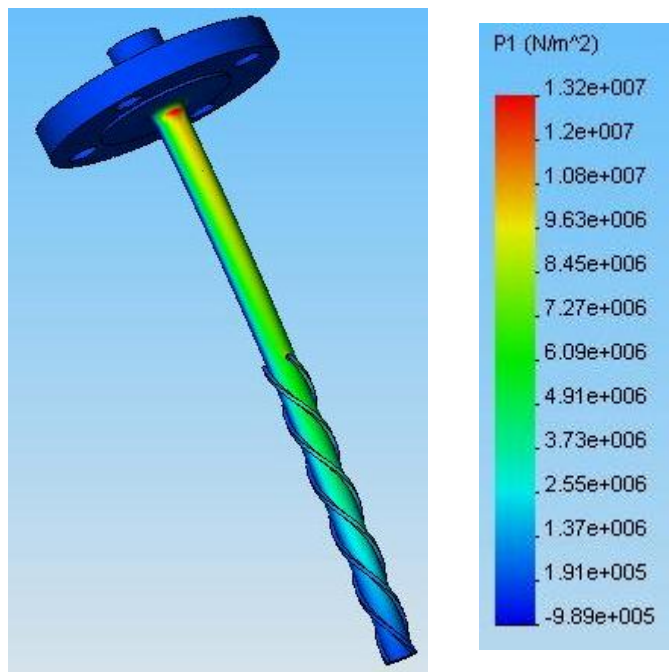
Figures 16, 17, 18 & 19 show the maximum stress contours mapped over both thermowell bodies under the influence of maximum dynamic loads.



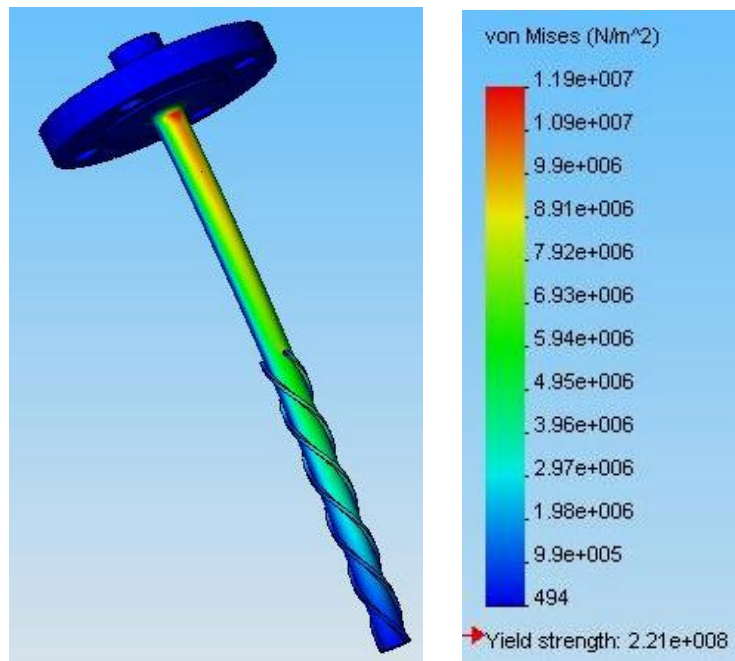
**Figure 16 Contour of Principal Stress 1 over Standard Thermowell Body**



**Figure 17 Contour Von of Mises Stress over Standard Thermowell Body**

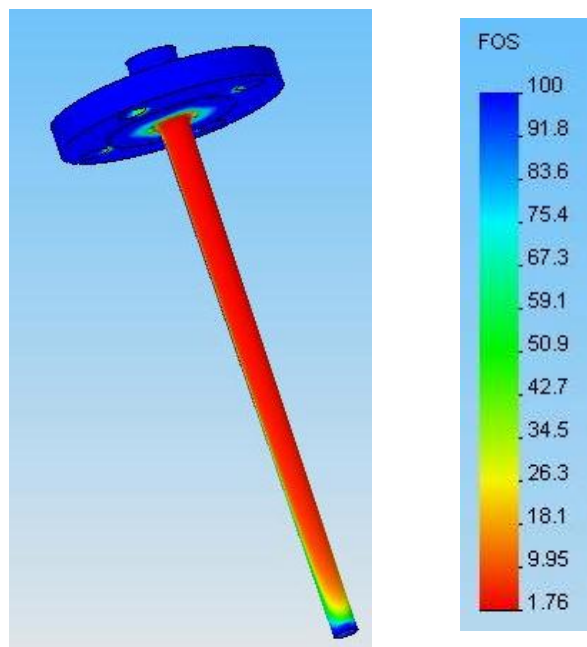


**Figure 18 Contour of Principal Stress 1 over VortexWell Body**

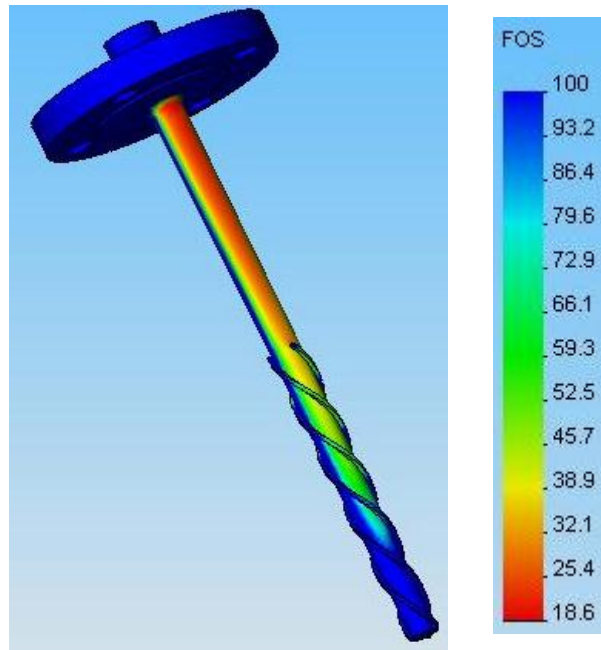


**Figure 19 Contour of Von Mises Stress over VortexWell Body**

Figures 20 & 21 compare the minimum factors of safety (FOS) between the two thermowells.



**Figure 20 Contour of Factor of Safety over Standard Thermowell Body  
(Minimum FOS = 1.8)**



**Figure 21 Contour of Factor of Safety over VortexWell Body  
(Minimum FOS = 19)**

It should be noted from the coloured scale bars that  $1 \times 10^6 \text{ N/m}^2$  is equivalent to  $1 \text{ N/mm}^2$ ,  $1 \times 10^7 \text{ N/m}^2$  is equivalent to  $10 \text{ N/mm}^2$  and  $1 \times 10^8 \text{ N/m}^2$  is equivalent to  $100 \text{ N/mm}^2$  respectively. It should also be noted from the FOS plots (Figure 18 vs Figure 21) that the VortexWell appears to significantly outperform the standard thermowell. Nonetheless, care must be taken in interpreting these results as the standard thermowell was positioned approximately 10 diameters downstream of the VortexWell. The installation effects would likely impact on the dynamic and structural performance of the standard thermowell.

**(c) Prediction of Shedding Frequency**

The shedding frequencies [3] induced by a smooth cylinder over a variety of flow regimes and Reynolds numbers can be predicted using equations 2 and 3 (Appendix B). The results which are relevant to this laboratory study are presented in Table 11 below.

**TABLE 11  
SHEDDING FREQUENCIES OF CYLINDRICAL BODY**

Fluid velocity (m/s)	Shedding Frequency (Hz)
1.07	~ 13 – 15
2.14	~ 27 – 29
3.22	~ 40 – 44
4.28	~ 54 – 58
5.36	~ 68 – 71

As can be seen from the above table, the predicted shedding frequencies of a cylindrical thermowell are much lower than those structural excitation frequencies of the two designated thermowells as observed throughout the tests (see Tables 8 and 9).



### **4.3 Discussions**

This sub-section discusses key issues concerning the tests, simulations, observations, results and findings.

#### **(a) Strain Gauge Integrity**

In order to ensure that strain measurements were sufficiently accurate, the FE simulations and closed form formula were employed to validate the 1kg static load test prior to the commission of laboratory tests. Since strain gauges are fine and delicate components, excessive structural excitations and deformations can damage both the structural and electrical integrity as can be seen in the tests with oil-gas fluid operating at 5.36 m/s. Consequently, some strain readings (data presented in Table 7) would not be representative, albeit the stress at V3 on the VortexWell was derived from the maximum strain presented in Table 7.

Since the thermowells were excited by transient loads resulting in low strain levels (data presented in Table 3 and 4), care should be taken in interpreting these results as some strains presented in these two tables could be neither accurate nor representative. It should be noted from the initial data acquisition phase that the strain readings were low and unstable as these gauges might be working within their noise regions.

#### **(b) FE Simulations**

All FE models presented in this section and boundary condition illustrated in section 3 of this report have been validated using the 1kg static load test (Figure 7) and MathCad calculation based upon closed form cantilever formula. The validated FE models and corresponding meshes were subsequently utilised to calculate the stress levels, as this approach minimised excessive convergent exercise in FEA project. The models have produced predictions which were representative of test conditions. It should also be noted that convergent exercise simply ensure numerical accuracy, it does not necessarily ensure that the simulations were representative of test scenarios. For simulating higher mode shapes, the meshes and boundary conditions would have to be modified to reflect the changes of structural deformations and loading regimes.

#### **(c) Modal Analysis and Structural Excitation**

The modal analysis was an important step for determining the dominating mode shape of the thermowells under excitation conditions and hence the nature of boundary conditions to be applied to the models. It should be noted that the natural frequency predicted by MathCad calculation based upon the closed form equation agreed well with the average natural frequency derived from the strain cycles acquired during the tests, albeit the excitation frequencies predicted by FE models appeared to be slightly higher than those calculated from the strain data. However, the excitations do include frequency contents which tie up well with the first natural frequency and corresponding mode shape of the standard thermowell predicted by the FE model.

#### **(d) Shedding Frequencies**

The shedding frequencies of the cylindrical thermowell were predicted using the closed form equations presented in Appendix B of this report. The predicted shedding frequencies appeared to be much lower than the natural frequencies of the designated thermowells. This suggests that the shedding frequencies of the two thermowells may not play a significant role in the tests. However, the shedding frequencies become critical when the natural frequencies of the thermowells are falling within the range of the shedding frequencies and a 180° phase change of the strain data is observed during the test.

### **(e) Preliminary Findings**

In comparing the FOS contours (Figures 20 and 21), it should be noted that the VortexWell has significantly outperformed the standard thermowell in terms of dynamic performance and mechanical strength. Notwithstanding, due diligence should be exercised in interpreting these analytical results as the standard thermowell was operating behind the wake of the VortexWell. Indeed, the standard thermowell was undergoing some phase changes and in some cases it exhibited a sign of significant dynamic amplification. This indicated that the standard thermowell would be operating within the range of its first natural frequency.

It should also be noted that high dynamic amplifications resulting from excitation frequencies would also shorten the fatigue and service life of thermowells.

## **5 CONCLUSIONS AND RECOMMENDATIONS**

The results of the work programme have been analysed and presented in this report showing dynamic performance and mechanical strengths of the two designated thermowells.

A comparison of dynamic performance and mechanical stresses indicates that the VortexWell has significantly outperformed the standard thermowell in this project.

To verify and hence confirm the findings, it is recommended that:

1. a CFD simulation study be undertaken to verify if unsteady flows and installation effects could affect the dynamic performance and hence impose higher loading magnitude over the two thermowells;
2. further laboratory tests be carried out to individually and independently test the VortexWell and standard thermowell at NEL's flow process line;
3. an experimental modal analysis be carried out to verify the natural frequencies of these two thermowells which might be influenced by installation effects and unsteady flows.

## REFERENCES

- 1 Brock J.E., 'Stress Analysis of Thermowells' National Technical Information Service, U.S. Department of Commerce, 11 November 1974.
- 2 Oakes T., 'Computational Fluid Dynamics Modelling', CDS Cygnet Development Services Ltd, Okazaki Report\_V3\_TO\_ApRE\_1\_300608. 30/06/2008.
- 3 Murdock J.W., 'Power Test Code: Thermometer Wells', Journal Engineering for Power, 1959.
- 4 Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full – Part 1: General principles and requirements. BS EN ISO 5167 – 1:2003, pp 3.
- 5 Massey, B.S. *Mechanics of fluids*, 3<sup>rd</sup> Edition, pp 300.
- 6 Roark, R.J., and Young, W.C. *Formulas for Stress and Strain*, Fifth Edition, pp 576.

## APPENDIX II

### Test Conditions and Data Acquisition Log

**TABLE A1**

#### SUMMARY OF TEST PARAMETERS

Collection Date & Time	Ave. Temperature	Ave. Absolute Pressure	Fluid Density	Fluid Kin. Viscosity	Fluid Dyn. Viscosity	Ref. Volume Flow Rate
	°C	Pa	kg/m <sup>3</sup>	m <sup>2</sup> /s	Pa.s	m <sup>3</sup> /s
04/04/12 11:44	19.960	247690	820.741	6.2675E-06	5.1440E-03	0.01981
04/04/12 11:52	19.994	251606	820.720	6.2617E-06	5.1391E-03	0.04000
04/04/12 12:01	20.120	259679	820.638	6.2405E-06	5.1212E-03	0.05961
04/04/12 12:11	20.143	259454	820.623	6.2368E-06	5.1181E-03	0.05964
04/04/12 12:18	20.271	271368	820.543	6.2154E-06	5.1000E-03	0.07970
04/04/12 13:54	20.401	275048	820.456	6.1937E-06	5.0817E-03	0.09959
04/04/12 14:46	23.302	319249	818.498	5.7303E-06	4.6902E-03	0.19154

**TABLE A2**

#### FLUID VELOCITIES, FLOW RATES AND REYNOLDS NUMBERS

<b>Fluid velocity – m/s</b>	1.07	2.14	3.22	4.28	5.82
<b>Flow rate – l/s</b>	20	40	60	80	100
<b>Reynolds No. of thermowells</b>	2.65x10 <sup>3</sup>	5.31x10 <sup>3</sup>	8.02x10 <sup>3</sup>	1.07x10 <sup>4</sup>	1.35x10 <sup>4</sup>
<b>Reynolds No. of 6" sch. 40, process line</b>	2.63x10 <sup>4</sup>	5.27x10 <sup>4</sup>	7.95x10 <sup>4</sup>	1.06x10 <sup>5</sup>	1.33x10 <sup>5</sup>

The Reynolds number [4] for the 6" process line is given by:

$$Re_D = \frac{VD}{\nu} \quad (1)$$

where:

$Re_D$  is the Reynolds number of the process line;

$V$  is the fluid velocity;

$D$  is the inner diameter of the pipe section;

$\nu$  is the kinematic viscosity.

**TABLE A3**

**STRAIN GAUGE DATA ACQUISITION LOG AND TEST CONDITIONS**

<b>Time Stamp</b>	<b>Velocity (m/s) / Flow rate (l/s)</b>	<b>TW1 channel</b>	<b>TW2 channel</b>	<b>TW3 channel</b>	<b>VW1 channel</b>	<b>VW2 channel</b>	<b>VW3 channel</b>
11:18:10	0/0	Strain gauge zero check at 20°C and 100 micro-strain range.					
11:36:26	0/0	Repeat zero check.					
11:43:21	1.07/20	Test began.					
11:53:38	2.14/40	Test began.					
11:02:05	3.22/60	Test began & strain gauge channels overloaded due to vibration.					
12:03:58	0/0	Test repeated.					
12:08:26	0/0	Zero check with 100 micro-strain amplifier range.					
12:09:25	0/0	Zero check with 100 micro-strain amplifier range.					
12:11:52	3.22/60	Test began with 1000 micro-strain amplifier range.					
12:19:00	4.28/80	Test began with 1000 micro-strain amplifier range.					
12:24:22	0/0	Zero check before lunch break.					
13:47:09	0/0	Zero check before restart.					
13:55:07	5.36/100	OK	Failed	OK	OK	OK	OK
13:59:03	5.36/100	Failed	Failed	Failed	OK	OK	OK
14:18:06	5.36/100	Failed	Failed	Failed	Failed	OK	OK
	10.31/192.3	All strain gauges channels failed. Strong vibration at standard thermowell and strains were not detectable from the VortexWell.					

## APPENDIX III

### Prediction of Shedding Frequency

The shedding frequency generated by a smooth cylinder is given by:

$$f = \frac{St V}{d} \quad (2)$$

Alternatively, the empirical formula [5] for calculating the shedding frequency of a cylinder is given by:

$$f = 0.198 \frac{V}{d} \left(1 - \frac{19.7}{Re}\right) \quad (3)$$

where:

$f$  is the shedding frequency in Hz;  
 $St$  is Strouhal number;  
 $V$  is flow velocity;  
 $d$  is the diameter of the cylinder;  
 $\nu$  is kinematic viscosity of the fluid;  
 $Re$  is Reynolds number of the cylinder.

Equation 2 is generally valid for the range  $250 < Re < 2 \times 10^5$ .

Figure A1 below represents the experimental relationship between Strouhal number and the Reynolds number of the cylinder utilised for the test.

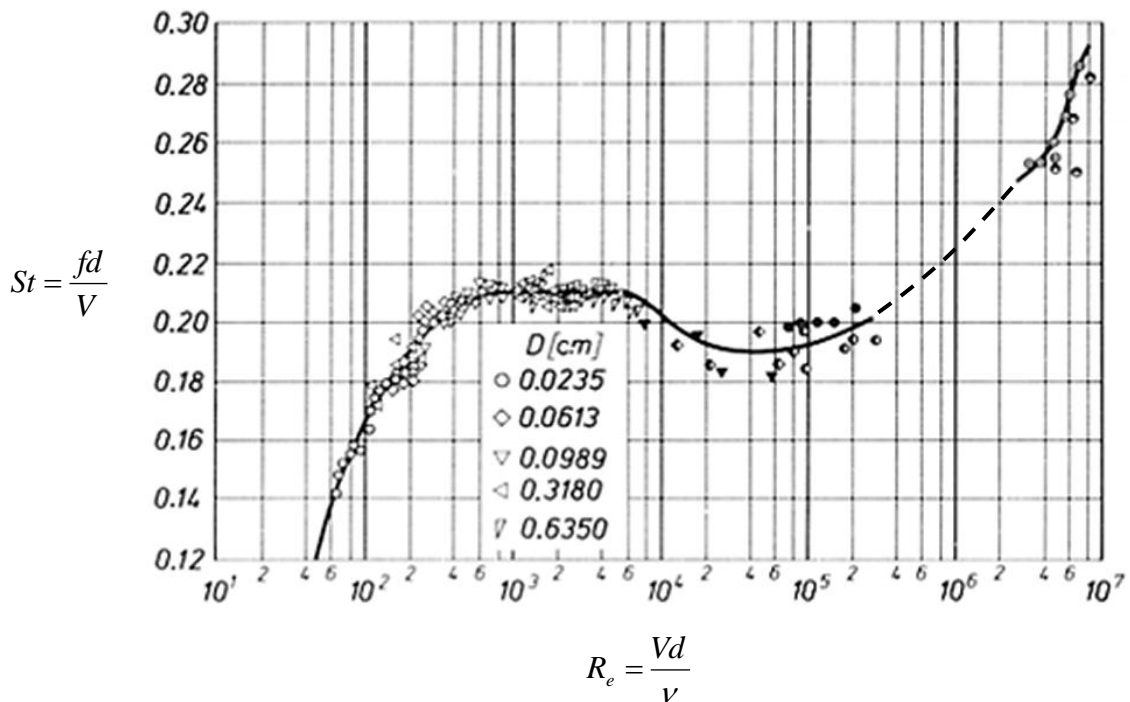


Figure B1 Strouhal Number vs. Reynolds Number  
(Experimental Data for Cylinder)

## APPENDIX IV

### Prediction of Natural Frequency using Closed Form Equations

The natural Frequencies of a cylindrical cantilever [6] can be approximated by using the closed form equation illustrated below.

$$\omega_n = \frac{K_n}{2\pi} \sqrt{EI \frac{g}{Wl^4}} \quad (4)$$

where:

$\omega_n$  is the natural frequency of the cantilever beam;  
 $E$  is the young modulus of the beam;  
 $W$  is the uniformly distributed load along the beam;  
 $l$  is the length of the beam;  
 $K_n$  is a beam constant.

TABLE C1

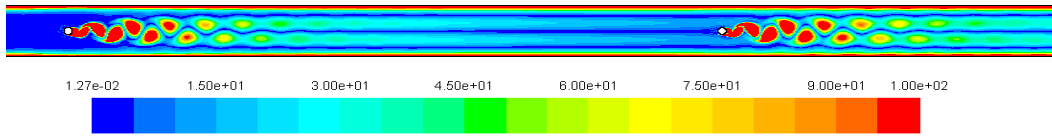
FUNDAMENTAL MODES AND  $K_n$

Mode number	$K_n$	Nodal position/ $l$
1	3.52	0
2	22.00	0, 0.783
3	61.70	0, 0.504, 0.868
4	121.00	0, 0.358, 0.644, 0.905
5	200.00	0, 0.279, 0.500, 0.723, 0.926

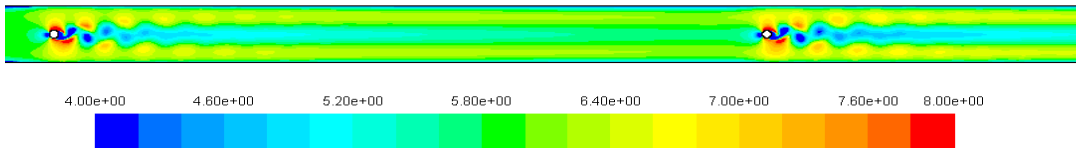
# APENDIX V

## Thermowell CFD

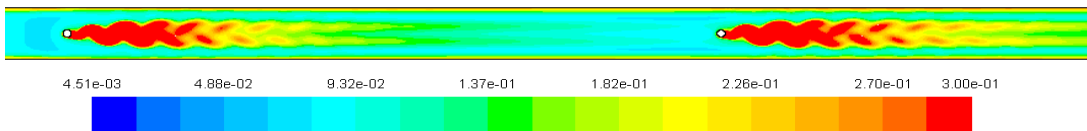
Vorticity Magnitude (1/s)



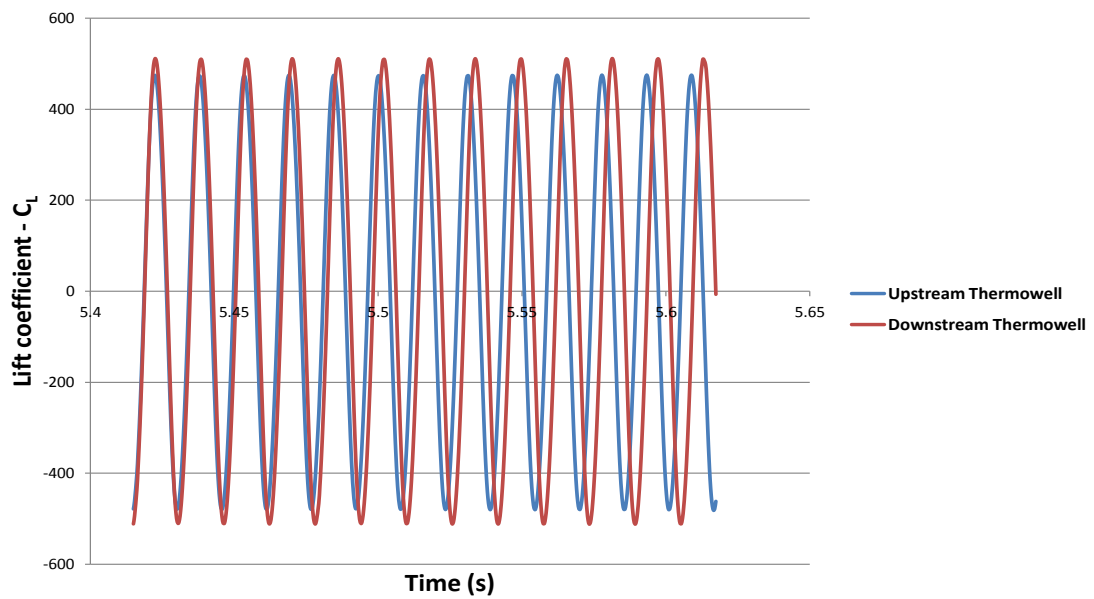
Turbulence Kinetic Energy (m<sup>2</sup>/s<sup>2</sup>)



Velocity Magnitude (m/s)



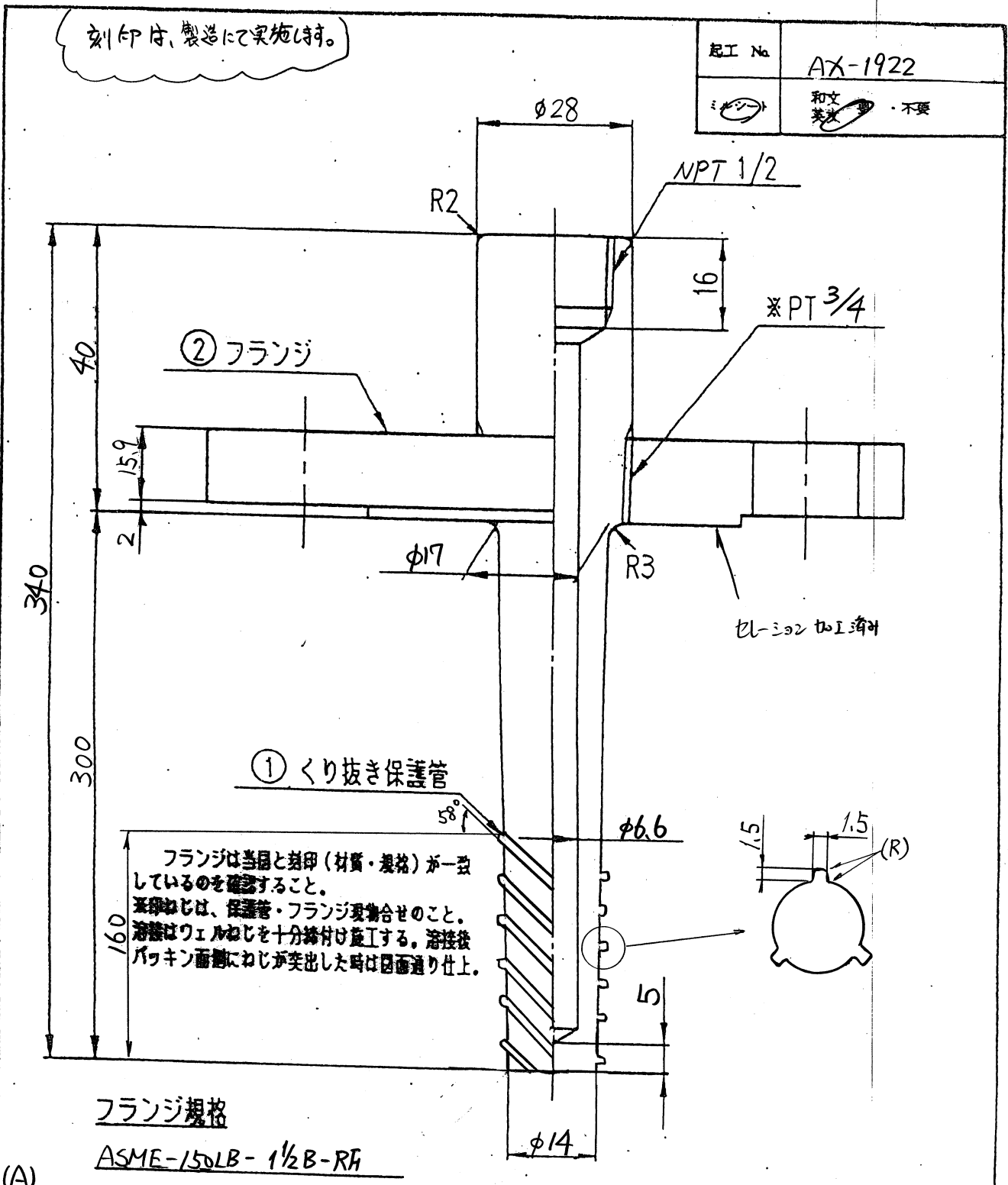
Force experienced by each thermowell varying with time





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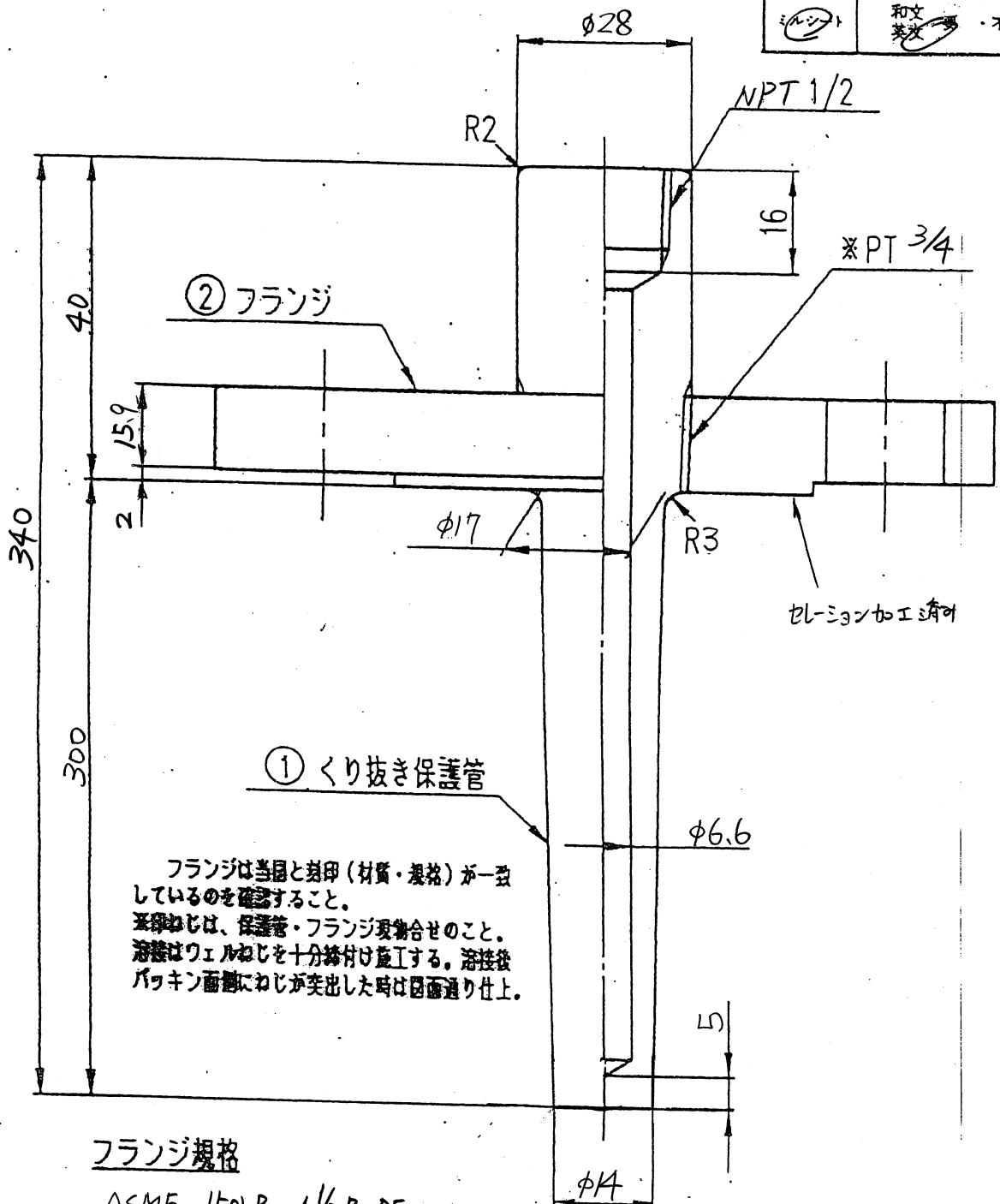


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①	くり抜き保護管	316SS	1		
符号 No	名称	TITLE	材質 M'L	個数 Q'TY	備考 REFERENCES

発注先	指定なき寸法差(削り加工)	指定なき凸部 C 0.1 ~ 0.2	未 歴 REVISION		
	JIS B 0405-'91 中級	指定なき凹部 R 0.2 ~ 0.3			
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 溶接はワエルねじを十分締め付け直工する。溶接後パッキン面裏にねじが突出した時は図面通り仕上げ。

フランジ規格  
 ASME-150LB-1 1/2 B-RF

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	①	くり抜き保護管	316SS	1		
符号 No	名	称	TITLE	材質 MTL	個数 Q'TY	備考 REFERENCES

発注先	指定なき寸法差(削り加工)	指定なき凸部 C 0.1 ~ 0.2	未 歴 REVISION				
	JIS B 0405-'91 中級	指定なき凹部 R 0.2 ~ 0.3					
ノ 級	指定なき表面粗さ(削り加工)	指定なきねじの等級 2 級又は B 級					
	JIS B 0601-'94 (63~32)a	ドリル穴には適用しない					
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