THE EXPERIMENTAL STUDY AND SIMULATION OF TUBE RUPTURE IN SHELL-AND-TUBE HEAT EXCHANGERS

A.M. Thyer, A.J. Wilday and G. Bankes *
Health and Safety Laboratory, Harpur Hill, Buxton. SK17 9JN
*** Health and Safety Executive, Offshore Safety Division, Merton House, Stanley Rd., Bootle. L20 3DL**

This paper summarises the findings of a Joint Industry Project to determine the effectiveness of computer simulation models used to predict the consequences of tube rupture in high pressure gas coolers. This was achieved by undertaking an experimental test programme in which tubes carrying a high pressure gas were ruptured inside a full-scale shell-and-tube heat exchanger. The facility was extensively instrumented with strain guages and pressure transducers to allow records of short and long-term pressures/strains in the heat exchanger and relief pipework to be measured. This information was then compared against a number of blind simulations performed by other participating companies to determine the level of accuracy. The experimental data obtained support the use of one-dimensional fluid dynamic models to predict the flowrates, shell pressures and forces on the heat exchanger and connected lines. The relief system opening time and the shell-side friction are important input data to such models. The measurement of opening times of typical bursting discs and safety valves is recommended. The experimental data also support the possibility of using finite element modelling to take account of the effect on the shell of the high pressure pulse being of short duration rather than a static load.

Keywords: heat exchanger, shell-and-tube, tube rupture

INTRODUCTION

There are many shell and tube heat exchangers operating on oil production platforms in the North Sea where high pressure gas is either cooled or heated by the use of a low pressure utility fluid, such as sea water. Such heat exchangers typically have gas on the tube side at 100-250 bar gauge pressure and sea water on the shell side at 5-20 bar. The shell side of exchangers is usually designed to withstand a pressure just above the operating pressure of the liquid. It therefore needs to be protected against tube failure by fitting either bursting discs or pressure relief valves. However, a critical problem for the engineer has been to select a suitable design pressure for the utility (low pressure) side of the exchanger, which will be able to withstand the pressures generated before the chosen pressure relief system is fully effective.

Incidents which have occurred over the past decade clearly illustrate the highly detrimental engineering and financial consequences that can result from a tube rupture. These have ranged from catastrophic failure of the heat exchanger shell with associated mechanical damage to nearby process plant (Brent Delta, Jan 1989) to the ingress of flammable gasses into accommodation modules and process areas (Fulmar Alpha August 1991), fires, personnel injury and mechanical damage as a result of pressurisation of the cooling water system. In many instances the consequential financial losses have been considerable.

The industry has sought to tackle the problem by carrying out quantitative risk assessments as well as employing specialist consultants to undertake computerised dynamic simulations of the pressures generated within the units in the event of a tube failure. These simulations make a number of simplifying assumptions, but the accuracy of their modelling predictions is unknown, because the models have never been validated against experimental data obtained from realistic experiments.

The Institute of Petroleum Heat Exchanger Task Group was therefore set up to progress work to develop confidence in the design methods used by the offshore industry and thereby ensuring that high-pressure heat exchangers are able to withstand possible tube rupture. The terms of reference of the Task Group called for experimental work to provide data sets for validation of the computer codes, the specification of their accuracy, and to establish the range of applicability. A simple "rule of thumb" design criteria for heat exchangers[1,2] was also investigated. Guidelines for use by industry have been produced by the Institute of Petroleum[3].

Work proceeded in phases. The Institute of Petroleum (IP) funded a literature review and survey by Trident Consultants and Foster Wheeler Energy[4], and HSE carried out its own literature review[5]. In the second phase the IP funded work by AEA Technology aimed at producing a simple "rule of thumb" assessment method[1,2], and HSE funded a series of shock tube experiments to allow partial validation of codes in a one-dimensional geometry. Very good agreement was obtained between code predictions and shock tube test[6]. This work showed the one-dimensional computer model was accurate in predicting pressure peaks and profiles in a one-dimensional (shock-tube) geometry. However, experimental results from a representative heat exchanger, which is strongly three-dimensional, were thought necessary to fully validate the codes, establish greater confidence in their predictive capability, and further aid the development of design guidelines. This is the subject of this present paper.

In August 2000 the preceeding five-six years work on heat exchangers came to fruition when The Institute of Petroleum published its guidance document: “guidelines for the design and safe operation of shell-and-tube heat exchangers to withstand tube failure”. This document is available through the IP.

DESCRIPTION OF THE TEST FACILITY

The experimental programme was carried out on a typical offshore shell-and-tube heat exchanger, a photograph of which appears as Figure 1. The shell for the heat exchanger was designed and fabricated by Motherwell Bridge Thermal Ltd. following a typical offshore design, and was made from 10 mm carbon steel with an internal diameter of 740 mm and an overall length of 3750 mm. Two main nozzles were fitted along with a number of smaller nozzles to allow internal inspection. The main nozzles were: one of 150 mm diameter for a graphite bursting disc with a specified failure pressure of 10 bar (g), and a 203 mm diameter nozzle to represent part of the cooling water circuit. Other dimensions and nozzle positions are given in Figure 2.

The tube bundle for the heat exchanger was a surplus four-pass bundle containing 566, 19 mm Monel tubes, which was purchased for a nominal fee from Texaco North Sea. Prior to the programme the header was sawn off leaving a flat tubesheet. At the same time the U's were removed leaving straight open ended tubes. This served a dual purpose:

a) it allowed the easy removal of straight tubes from the bundle; and;
b) it allowed the tubes to flood with water.

The latter point was necessary to avoid complications for the modellers as a partially air-filled system would give ‘spongy’ pressure pulses.

Twelve tubes were removed from the tube bundle; 6 were replaced with specially weakened carbon steel tubes to be failed in the test programme, and six with tubes carrying miniature pressure transducers. These were positioned in close proximity to the failure point. Tubes not in use for instruments or compressed air injection were plugged with tapered steel plugs.

The appearance of outlet pipework for the 'cooling water' and relief pipework is illustrated in Figure 3. In order to withstand forces exerted by liquid slugs during tests, this pipework was secured to 10 one tonne concrete blocks and further weighed down with 12 one tonne bags of gravel.

The compressed air system comprised a 0.91 m^3 / minute compressor, feeding a bank of five 0.258 m^3 cylinders. The total reservoir volume at 1 bar was 1.29 m^3 giving 345 m^3 at the maximum working pressure of 275 bar. Air was delivered to the heat exchanger through a 50 mm nominal bore (NB) pipe.

During the tests the heat exchanger was isolated from the air receivers by an air-operated quarter-turn ball valve. The cylinders were then taken to between 20 and 50 % of the nominal failure pressure for the tube under test. At this point air was slowly introduced into the unpressurised tube via a by-pass line fitted with a flow restriction. This avoided a sudden shock-loading of the weakened tube which could have caused it to fail prematurely. Once the pressure was balanced on both sides of the main supply valve, it was opened and the compressor run until the weakened tube in the heat exchanger eventually failed. For the highest pressure test at 150 bar the compressor was run for 3.5 hours.

The mechanical strains in the shell were measured using fourteen two-axis 'tee' type gauges, having a frequency response of around 80 kHz (giving ±1% strain reading accuracy). Pressures were monitored using up to 12 externally mounted transducers on the shell (Kister 701 H, and Druck, PDCR 930), 4 internal transducers (Kistler 601H), with a further 6 transducers being fitted to the discharge pipework.

Two separately-controlled and triggered Nicolet digital logging systems were used. These were configured to log at rates of 100 or 12.5 kHz depending on the frequency response of the instrument. This gave a total logging time ranging from 3.5 to 7 seconds, which was ample to observe short-term transients, a steady state flow period, and pressure decay in the system once the majority of water in the shell had been ejected.

In addition to instrumentation on the shell and discharge pipework, one further pressure transducer was fitted to the compressed air feed pipe. The output from this instrument was logged and used to determine the exact failure pressure of the tube, and to follow the discharge characteristics of the air receivers.

The final feature of the test facility was the use of a large water tank at the end of the 8" water discharge pipe to determine the relative proportion of water venting through this pipe and the bursting disc.

THE TEST PROGRAMME

Four tests were undertaken at failure pressures of 71, 102, 130 and 146 bar.

EXPERIMENTAL RESULTS

The results for Test Three, with a failure pressure of 145.9 bar are described as they are typical of those obtained.

The results obtained were tabulated to include, peak instantaneous pressures/strains, peak long-term values over a 10 ms period, values at 0.5 s, and values at 2.5 s. These are still subject to a nondisclosure period and can be obtained through the Institute of Petroleum. Peak instantaneous pressures are given in Table 1.

Table 1
Test 3 instantaneous peak pressures: burst pressure 145.9 bar

Tranducer	**1**	**2**	**3**	**4**	**5**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**int. top**	**int. bottom**
Peak pressure / bar	20.02	21.09	23.9*	25.7*	17.2*	20.18	34.35 11.8*	18.9*	14.53	145**	16.25	18.98	93.5	37.55
Time to peak / ms	0.88	1.22	-	-	-	2.98	15.53	-	4.88	3.42	2.55		0.33	0.76
Distance from failure point / m	0.84		0.60	0.81	0.65	2.17	2.08	2.85	2.78	2.09	0.29	0.03	0.55	0.27

* Pressure transducer faulty, reading derived from strain gauge results
** Pressure transducer showed two small peaks of around 30-40 bar followed by one at 145 bar, after which it failed.

Examination of the video record for this test indicated a flow velocity in the bursting disc line of 19 $m.s^{-1}$, with a period of slug-flow lasting approximately 3.9 s. This corresponds reasonably well with the output obtained from the pressure transducer at the mid point of the horizontal run of 150 mm NB pipe, where the pressure peak was around 3.25 s long.

REVIEW OF RESULTS FROM EXPERIMENTAL PROGRAMME

The results obtained show that the initial effects of tube failure are to create a high pressure gas bubble in the vicinity of the failure point, in which the pressure may approach a significant proportion of the source pressure, as proposed by Goyder[2]. During the first few milliseconds after failure a ringing pressure profile is obtained with peaks of around 50 μs duration. These decay rapidly leaving a long-term pressure 0.5 s after failure of between 3 - 4 % of the original source pressure. This may persist for approximately 2 s, after which the pressure falls to approach zero 3.5 s after failure. During this period the compressed air reservoir pressure falls to 80 - 83 % of the source pressure.

Strain profiles show similar trends with an initial peak followed by long-term decay. Here it is likely, however, that the decay in strain readings is both a function of falling pressure and venting of the water in the shell. It was noticed during preparation for the tests that the strain gauges required rebalancing after the shell was filled as it sagged slightly under the load. This was more pronounced in the commissioning test than in the actual experimental tests, as the shell was supported under the tubesheet and the saddle near the domed end,

(Figure 1), rather than on both saddles and the tubesheet. This induced an artificially high bending moment along the exchanger longitudinal axis. However, it is likely that loss of water would result in a slight decrease in the baseline strain reading for the gauges.

Good correlation has been obtained between data sets obtained for the four tests. It was found that pressure pulses in excess of 10 ms duration rose as the tubeside pressure increased, with that at 71 bar being about 6.1 % of the tubeside pressure, and that for 146 bar being 7.7 %. Similar trends were seen in the long-term at 0.5 s after failure (Table 2).

In contrast to the results in Table 2, correlation between peak shell-side and tubeside pressures is poor and no real trend was evident, as can be seen from the data in Table 3 below. This lack of a clear trend will be due, at least in part, to the response and accuracy of the pressure transducers. The Druck strain gauge type transducers tended to overestimate the magnitude of the original pressure pulse due to their inability to withstand the severe mechanical acceleration to which they were exposed during the tests.

Table 2
Mean shell-side pressures as a function of tube failure pressure

Test No.	Failure pressure / bar	Highest pressure over a 10 ms period / bar	% tubeside pressure	Pressure at 0.5 s / bar	% tubeside pressure	Pressure at 1.0 s / bar	% tubeside pressure
1	70.9	4.3 ± 3.8	6.1	2.5 ± 2.0	3.4	2.4 ± 2.0	3.4
2	102.3	8.2 ± 2.9	8.0	4.6 ± 1.4	4.5	4.3 ± 1.2	4.2
4	130 ± 1.4	10.0 ± 4.9	7.7	5.6 ± 3.6	4.3	4.7 ± 3.7	3.6
3	145.9	10.4 ± 3.8	7.1	6.0 ± 1.4	4.1	4.8 ± 1.6	3.2

Table 3
Correlation between tubeside pressure and peak shell-side pressures

Test number	Tubeside pressure / bar	Mean peak shell-side pressure /bar	Mean peak shell pressure as % tubeside pressure
1	70.9	6.8	10
2	102.3	27.1 ± 19.3	26
4	130 ± 1.4	10.7 ± 4.6	8
3	145.9	20.8 ± 6.0	14

Similar trends to those above were seen for the strain gauge results.

Measurement of the volume of water retained in the shell after the test and venting through the relief line also provided useful data in the form of the volume of water venting per unit area of pipe. For the 203 mm NB pipe which was permanently open (apart from a polythene disc to retain a static head of water), the amount vented was 1.4-1.6 $dm^3.cm^{-2}$. For the 150 mm NB pipe attached to the bursting disc a larger volume of 2.3-2.6 $dm^3.cm^{-2}$ was vented. This difference was due to the inertia of the water in the completely filled larger pipe, thus decreasing the rate of discharge. Whereas the 150 mm NB pipe was dry downstream of the bursting disc, thus the only impedance to flow was due to the convoluted path and internal surface roughness of the pipework. Despite this apparently higher contribution for the 150 mm pipework, the quantity venting from each pipe was approximately the same, at about 33 % of the water in the shell.

With the exception of peak pressure, which varied between tests, a reliable trend was observed between the percentage tubeside pressure and the long term pressures in the shell. For instance, pressure peaks exceeding 10 ms duration, range from 6-8 % of the original pressure, and long term pressures 0.5 and 1.0 s after tube rupture vary from 3.5 - 4.5 % and from 3.2 - 4.2 % of the original gas pressure respectively. These pressures are significantly below those originally anticipated , and are well below the design pressure for the shell. Should only one exit have been available, however, considerably greater pressures would have developed with an increased chance of damage occurring to the shell or tube bundle.

Results from the strain gauges displayed similar trends to the pressures and can in many cases be used to back-calculate a reliable causative pressure, especially for pressure peaks over 10 ms duration, or in the longer term at 0.5 s or so. For instantaneous pressures, however, the highly transient nature of the pressure peaks means that the shell has insufficient time to react and generate the full potential strain reading. For different shell materials and thicknesses, however, this will be different and this effect may be a characteristic of the chosen test shell. However, as this was constructed to be representative of units in service, it appears likely that many shells will only develop a fraction of the strain they could potentially develop.

Post-test examination of the shell and tube bundle have shown that although the support baffles for the tube bundle flex, no permanent damage occurred. Thus under the test conditions utilised it is extremely unlikely that baffles would be displaced to slide along the tubes and completely block the shell. A similar conclusion can be drawn for the shell. However, a note of caution should still be aired as the shell was specifically designed and tested to ensure that, whilst it was representative of those in industry, it would not fail prematurely. The highest strain recorded arose in the fourth test, where the failure point was directly facing the shell and approximately 2.5 cm from the inner wall. This strain of 1336 με, corresponds to a causative pressure of 74.7 bar (57 % of the tubeside pressure). Whilst this was within the shell hydrotest pressure of 84.7 bar, it was approaching it and the point where plastic deformation would occur. However, plastic deformation is not failure, and whilst some enlargement may have resulted, the shell did not fail. Should the shell have been an old one with evidence of internal corrosion (rather than a new unit manufactured specially for the test programme) or should the shell already have been under pressures in excess of 10 bar from cooling water pumps, damage to the shell could have occurred.

Forces in the water discharge pipework on the shell were in line with that predicted prior to the programme. Velocities in the 6" pipe on the bursting disc ranged from 19 -22 ms^{-1}, corresponding to a load of 10-12 tonnes.

Whilst the heat exchanger did not fail catastrophically during this work, the possibility of failure still exists, should the pressures developed be greater, or the shell more responsive towards high speed strain. It is therefore recommended that an understanding of which materials would be more susceptible to damage from strain transients is developed. The studies carried out by Sheffield University in parallel to this project may be used as a basis for this[7].

Apart from the possibility of damage to the heat exchanger, damage to associated pipework could occur. It is therefore recommended that the maximum shell pressure should be used in the calculation of pipe loads, i.e. forces developed with the minimum number of exits from the shell. This would avoid the situation where comparatively small loads were predicted with multiple event paths, all of which were not present when a tube did fail. If this is a potential problem for systems known to have weak pipe supports it may be necessary to upgrade these, or to specify in a permit to work system that the heat exchanger is completely shutdown if valves are closed on pipework leading from it.

These tests have investigated the effectiveness of a bursting disc on the relief system only. There may be many units protected by combinations of bursting discs and pressure relief valves (PRVs), or solely by PRVs. It is widely known that PRVs have a much slower response than bursting discs. This slow response time may allow the development of higher shell pressures, especially if the PRV is the only exit. A continuation study substituting the bursting disc with a PRV is therefore recommended to determine if units fitted with PRVs have a greater chance of catastrophic failure of the shell or downstream pipework.

USE OF EXPERIMENTAL RESULTS

COMPARISON WITH FLUID DYNAMIC MODEL PREDICTIONS

The IP Heat Exchanger Task Group invited those consultants who had the capacity to model the fluid dynamics of tube failure in heat exchangers to join the Task Group and to model the tube rupture experiments. Modelling was to be done with the code they would use commercially and using the same assumptions as appropriate. These consultants all used models which predict the pressures in the heat exchanger shell as a function of time following tube rupture. The modelling was conducted in two phases:

a) "Blind", i.e. without access to the experimental results, modelling of the first three experiments. HSL provided the modellers with details of the experimental rig and operating procedure together with the actual burst pressure of the tube. The purpose of this phase was to test the level of agreement of the codes when used (as they are used commercially) to predict the pressures in the shell following a tube rupture.
b) Further modelling with access to the experimental results. The purpose of this phase was to draw out any conclusions concerning how the models could be improved.

One-dimensional models tend to be used commercially as their accuracy is believed fit for purpose. The alternative of CFD programmes require significantly greater computing time and have higher associated cost. The one-dimensional codes all differ in their detailed implementation, but have certain assumptions in common. All model the compressibility of the water and usually take account of shock waves and their interactions. Some explicitly model the growth of the gas bubble. All make one-dimensional approximations to the flow within the LP side of the heat exchanger. The overall friction is modelled in terms of a loss coefficient which can be obtained from the pressure drop during normal operation.

Three computer models (those of Fluor Daniel, Hydraulic Analysis and W S Atkins) took part in the validation exercise. In phase (a), all were conservative when compared with the experimental results for peak pressure in the shell when run "blind" using commercial modelling assumptions (see Figure 4). These results therefore support the use of a one-dimensional modelling approach.

In phase (b), it was possible to investigate some of the factors giving rise to the spread of predictions shown in Figure 4. It was found that good input data for the models on disc burst time and shell-side friction are important. The results were less sensitive to tube burst time and this work identified that a tube burst time in the range instantaneous to 0.7 ms is appropriate (for carbon steel tubes). Shell-side friction information will usually be available from the heat exchanger designer. Disc burst time, or safety valve opening time, is important and it was recommended that further work be done to measure such parameters. When input assumptions were changed to the agreed best estimate values, the predictions of the models converged and gave good agreement with the experimental data. Some of the differences between model predictions and experimental results could be explained by the fact that the models assume that all the water is driven out of the shell by the gas, but in the experiments significant water remained within the shell.

Guidance has been developed by the Institute of Petroleum[3], giving recommendations for the assumptions to be made when modelling the fluid dynamic aspects of tube rupture in heat exchangers. This should help to ensure that sufficiently accuarate predictions can be obtained using any of the available one-dimensional models.

COMPARISON WITH FINITE ELEMENT MODEL PREDICTIONS

Associated work was carried out at Sheffield University to study the failure criteria for heat exchanger shells subjected to short duration pressure pulses[7,8]. A finite element model of a heat exchanger shell was developed and was validated using both small-scale shock tube data and the large-scale experimental results reported here. Good agreement was obtained.

A finite element model allows failure criteria to be developed, whereby the failure pressure is higher than for static loading if the duration of the pressure pulse is very short. However, if this were to be used for design, such a finite element model would need to be developed for the specific application (shell dimensions, material etc.). The model could then be used to predict the maximum stress developed in response to the short duration high pressure pulse predicted by the fluid dynamic model. This stress could then be compared with the allowable stress within the pressure vessel design code. This possible approach is further discussed in the Institute of Petroleum guidance[3].

CONCLUSIONS

1. Large-scale tube rupture experiments have been carried out on a shell-and-tube heat exchanger, typical of those in use in the North Sea. Experimental data suitable for the validation of both fluid dynamic modelling of the flows/pressures and finite element modelling of the stress in the shell have been generated.
2. The experimental data support the use of one-dimensional fluid dynamic models to predict the flowrates, shell pressures and forces on the heat exchanger and connected lines. The relief system opening time and the shell-side friction are important input data to such models. The measurement of opening times of typical bursting discs and safety valves is recommended.
3. The experimental data also support the possibility of using finite element modelling to take account of the effect on the shell of the high pressure pulse being of short duration rather than a static load.
4. The results of this work have been used in the development of guidance on safe design to take account of tube rupture in shell and tube heat exchangers (Institute of Petroleum, 2000).

ACKNOWLEDGEMENTS

This work was carried out as a Joint Industry Project sponsored by the Institute of Petroleum; the Health and Safety Executive; Amerada Hess; BHP Petroleum; CMPT; Esso Engineering; Fluor Daniel Applied Computer Solutions; Marathon Oil; Mobil North Sea; Phillips Petroleum; Shell International Oil Products; Texaco North Sea; and Total Oil Marine. In-kind support was given by: Fluor Daniel Applied Computer Solutions; Foster Wheeler Energy Ltd.; Hydraulic Analysis Ltd.; IMI Marsden Ltd.; Marathon Oil; Motherwell Bridge Thermal Ltd.; Shell International Oil Products; Texaco North Sea; Trident Consultants Ltd.; and W S Atkins. Related work carried out by the University of Sheffield was funded by the Engineering and Physical Sciences Research Council. We should like to express particular thanks to Mr C Weil (consultant); Dr B Ewan and Mr M Moatamedi (Sheffield University); IMI Marsden Ltd.; Foster Wheeler Energy Ltd.; and Messrs P Kerry, D Bagshaw and E Belfield (HSL).

REFERENCES

1. Goyder HGD (1996). Structural integrity assessment of heat exchangers following a tube rupture. AEA Technology Report AEAT-0347, April 1996

2. Goyder, H.G.D., (1997). Tube rupture in high pressure heat exchangers", ASME Conference, Dallas, 1997.

3. Institute of Petroleum (2000) Guidelines for the design and safe operation of shell and tube heat exchangers to withstand the impact of tube failure, ISBN 0 85293 286 3

4. Trident (1993) " Report to the Institute of Petroleum on the Development of Design Guidelines for Protection Against Over-Pressures in High Pressure Heat Exchangers: Phase One", Trident Consultants Ltd and Foster Wheeler Energy, Report J2572, known as "The Trident Report".

5. Wilday A.J. and Thyer A. M. (1995). The protection of heat exchangers against tube rupture - literature review. HSE Internal Report, IR/L/RAM/ES/95/11.

6. Wilday A.J., Thyer A. M. and Ewan, B.C.R., (1996). Shock tube experiments to partially model tube rupture in high pressure shell-and-tube heat exchangers. HSE Internal Report, RAS/96/12, FS/96/12

7. Ewan, B C R, (1999), "A study of the failure criteria for vessels under transient pressure loadings", Report to EPSRC

8. Ewan, B C R & Moatamedi, M, (2000), "Design aspects of chemical plant exposed to transient pressure loads", submitted to Chem. Eng. Research & Design, Trans IChemE Part A

Figure 1
The instrumented heat exchanger before installation in bunker

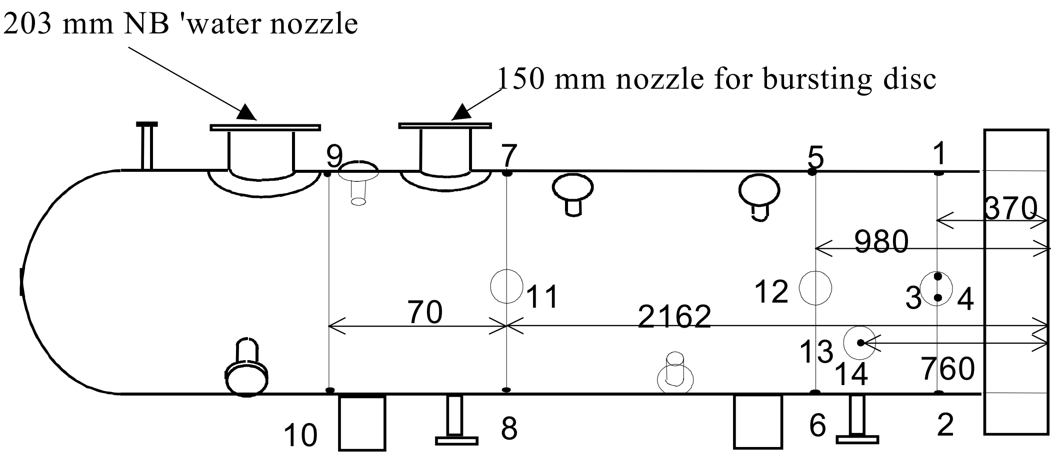

Figure 2
Instrument positions on shell and their separation

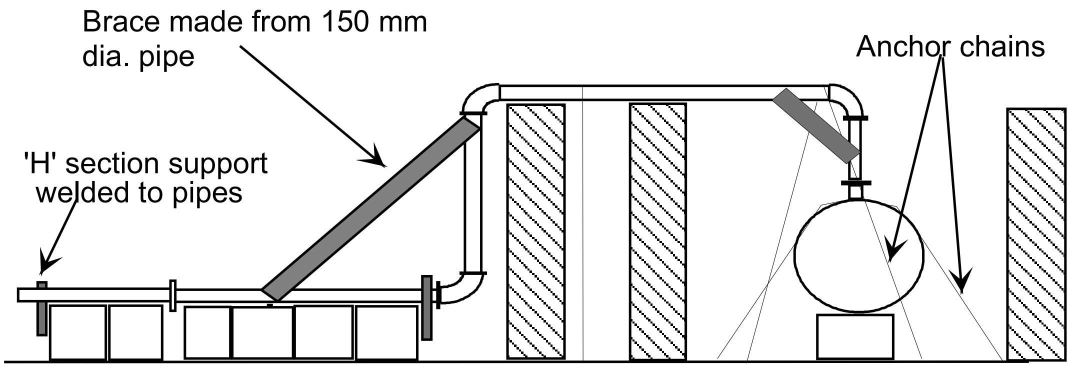

Figure 3
Sketch showing 150 and 203 mm water discharge pipework and method of securing pipes/heat exchanger in bunker

One tonne bags of gravel on horizontal pipe run outside bunker not shown

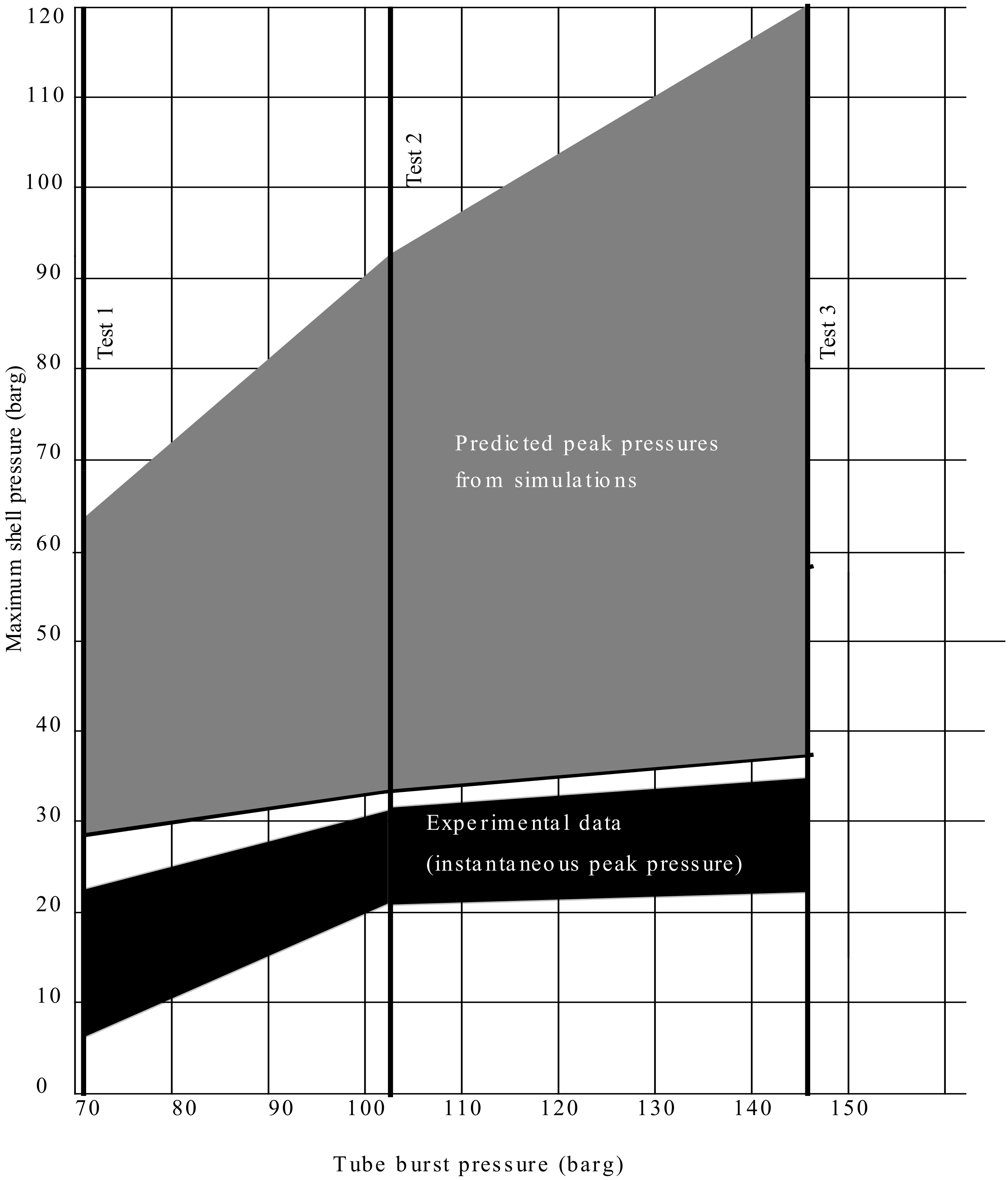

Figure 4
Comparison of experimental data with one dimensional fluid dynamic model predictions