**Dynamic Simulation of 9 Dock RAH 45t Crane Rope Failure**

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## 1.0 Introduction

Strachan & Henshaw are a well-established engineering company specialising in high integrity mechanical handling equipment for the Defence & Aerospace sectors. Recently within the nuclear sector, Strachan & Henshaw have been involved in the development of Devonport Royal Dockyard Plymouth in order to de-fuel/re-fuel the Vanguard class of nuclear submarines

Part of this upgrade is the provision of a Reactor Access House (RAH), a moveable refuelling structure, which spans the width of the submarine dock and can be moved in position over the submarine reactor compartment. Within the RAH the main refuelling operations are carried out by a 45 tonne Electric Overhead Travelling (EOT) crane which provides a seismically qualified high integrity craneage facility for lifting operations during the Vanguard refit process.

As a high integrity crane a dual load path design is adopted, i.e. if there were to be a failure of the main crane load bearing mechanism a secondary load path would prevent it from dropping its load.

## 2.0 Requirement for Analysis

In order to satisfy the regulatory authorities, a general requirement was placed on Strachan & Henshaw to substantiate by calculation the safety critical hardware. The primary redundancy in this crane is a dual rope design to carry loads. During normal operation each crane rope shares the load carried, but both are able to individually handle the maximum loads and fault caseloads should the other rope fail. This can be proven easily for steady state conditions, but it is more difficult to substantiate for the transient event as the rope breaks. There is a requirement to measure the transient rope loads against a failure load of 547kN

To this end, an analysis was carried out using ADAMS. The analysis allowed for the calculation of the magnitude of the likely bounding transient loads sustained by one rope in the event of failure of the other. ADAMS was selected because it can accurately simulate the effects of all the masses in the system, the action of the compensator beam and dampers and the energy absorbing hexcel pads (contained in the head pulleys) and hoist block.

## 3.0 System Description

As previously stated, for maximum safety the crane has two rope systems in order that a load can be safely supported in the event of a single rope failure. The rope systems are isolated from each other, having separate windings on the rope barrel, isolated force paths on the hoist block and separate head pulleys on the crab.

The assembly of the 45t crane rope system is shown in Figure 1.

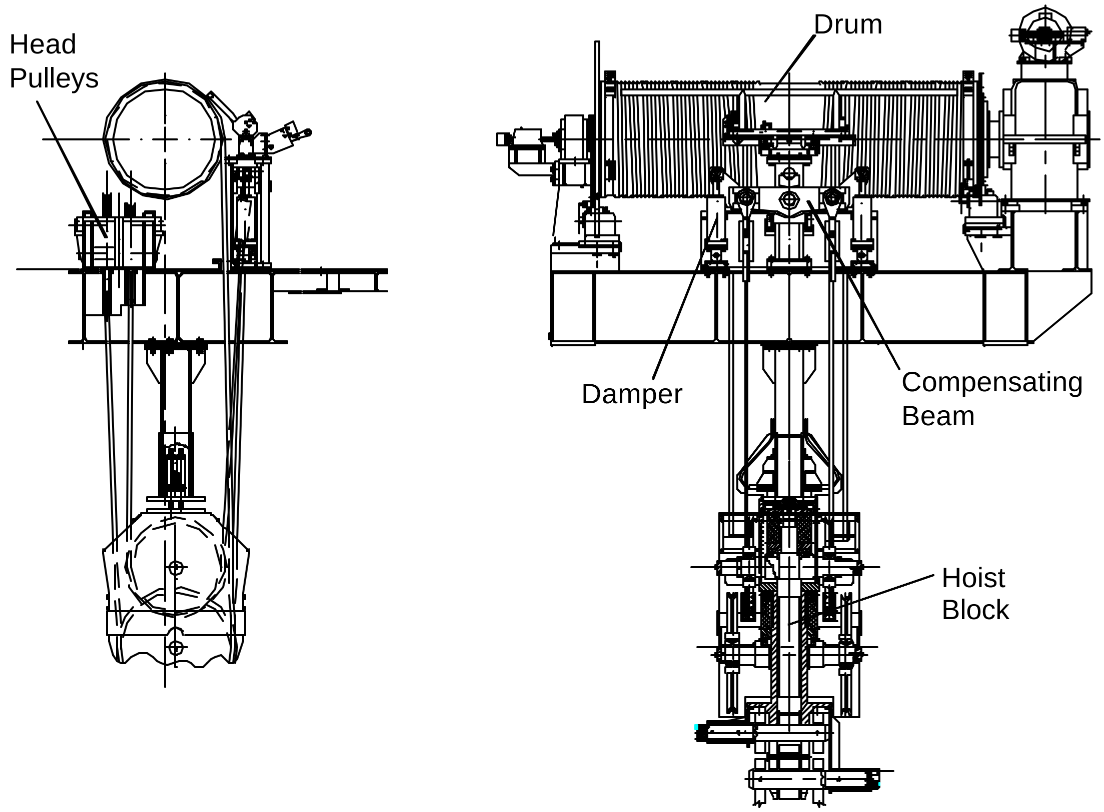


Figure 1 – General Arrangement of 9 Dock RAH Crane

Each rope is taken from the drum, around one of the hoist block pulleys, over a head pulley, round a second hoist block pulley and attached to a compensating beam (see Figure 1). This limits the load in the ropes to a an eighth of the hoist block load when operating normally, and a quarter at steady state when a single rope failure has occurred.

The compensator beam is centrally pivoted to equalise the rope loads and compensate for changes in rope length. Compression only dampers are positioned at two ends of the compensator beam such that if one of the ropes fails the load is taken up gradually in the other rope.

There are two further elements of load limitation on each rope system, provided by the predictable deformation of the frangible composite energy absorber packs. For each rope system there is one of these on the hoist block between the pulleys and the load support and one on the head pulley assembly.

## 4.0 ADAMS Model

For modelling purposes the crane was simplified into the following functional parts:

- Compensating Beam – comprising of all its mobile elements excluding the damper units, which were modelled as forces.
- Drum – a part fixed to ground providing a reference point for the model

- Hoist Block – comprising of all parts of the hoist block excluding the pulley shafts and all supported components separated from the main body by the energy absorber packs.
- Large Head Pulley – a lumped mass representing the sheave, its pulley housing and all parts that move with these items.
- Load – a part set up to represent the load hanging from the hoist block
- Lower Pulley System – a lumped mass representing the pulley shaft, the two lower sheaves and their associated bearings and fixtures isolated from the main assembly by the energy absorber pack.
- Small Head Pulley – a lumped mass representing the sheave, it's pulley housing and all parts that move with these items.
- Upper Pulley System - a lumped mass representing the pulley shaft, the two upper sheaves and their associated bearings and fixtures isolated from the main assembly by the energy absorber pack.

### 4.1 Model Assumptions

The modelling carried out is based on the following assumptions:

- Friction on the pulley bearings and pulley inertia is negligible compared to rope loadings, such that rope loads are equalised throughout all rope falls
- The crane body is itself rigid and rigidly attached to ground
- The load is rigidly attached to the hoist block
- The ropes have 2% damping
- The load on a rope is removed at the same time from all rope falls
- The compensator beam and dampers act in such a way as to keep the ropes in tension during the failure event
- Elements such as crushable pads have been represented by force elements utilising a mathematical function representative of their characteristics.
- A force function was also used to simulate the effects of the compression only dampers in the system based upon manufacturer-supplied data.

### 4.2 Rope Simulation

The action of the ropes was simulated by using a set of tangent parts that are constrained to move around pulley geometry. The rope forces are represented by an inline force acting between respective tangent parts on each rope fall. As a result of this simulation method there are a number of simplifications:

- The rope acts as a linear tension only spring acting between the tangent parts
- The rope is always in perfect contact with its pulley
- The rope force is equal between all rope falls on each rope

These were considered to be acceptable. In all cases pulley geometry is represented by a 2D circle in ADAMS aligned with the design position of the centre of the pulley groove, attached to the appropriate parts.

The tangent parts are constrained to the pulley groove by a point to curve relationship. The tangency of the tangent parts is created with two relationships, an inline relationship to the tangent marker at the opposite end of the rope fall, and a normal relationship to the centre of the pulley the tangent part is constrained to. These are forced to be perpendicular to each other, and therefore hold the tangent part at the point where the rope would leave the pulley, as shown below in Figure 2:

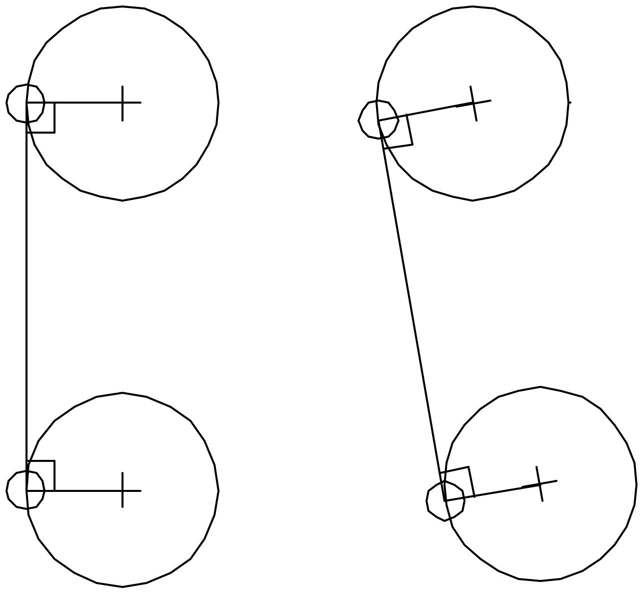

Figure 2 – Tangent Construction for Rope Forces

This construction holds to keep the ropes aligned correcly with the pulleys over the full range of motion (as shown above). In addition because the circle geometry is 2D this effectively models the action of the groove and does not allow the rope to join the pulley out of plane.

The force of the rope for each rope fall is equal and is calculated on the rope fall between the pulley on the hoist block and the compensator beam based on a spring force function.

Because of the function used to calculate rope forces the change in length of the rope is important in determining the tension in the rope at any given time. The rope fall between the compensator beam and the hoist block pulley is the most dynamic in terms of length changes during simulation, however the effects of length changes in other rope falls during simulation, e.g. head pulley energy absorber collapse is ignored by this simplification.

The resulting model is shown below in Figure 3.

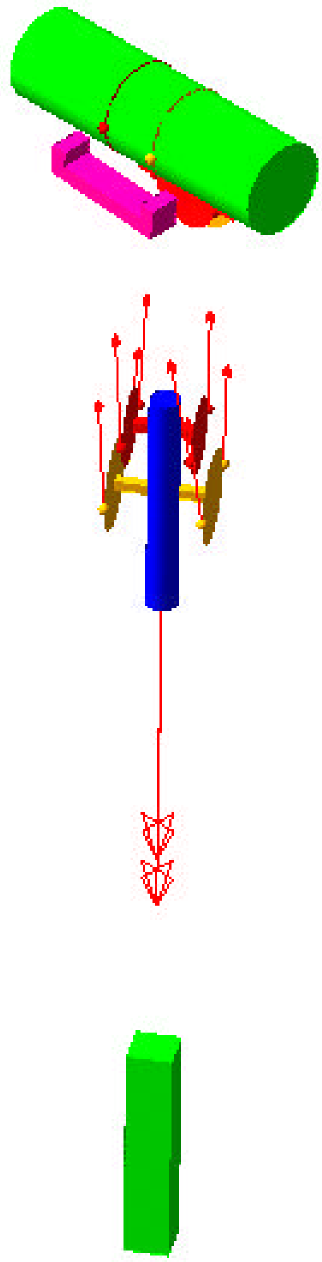

Figure 3 – ADAMS Model of 9 Dock RAH 45t Crane

**5.0 Loadcases**

The transient effect of a failed rope was considered for two crane loads The Reactor Pressure Vessel Head (RPV Head) and the Module Removal Container (MRC).

The MRC unit was modelled in two positions, lowest operating height (rope 1 length 10900mm), and greatest operating height (rope 1 length 2800mm). At each height the ropes were failed in turn. Similarly the RPV Head unit was modelled in two positions, lowest operating height (rope 1 length 13400mm), and greatest operating height (rope 1 length 2800mm). These loads and rope lengths were considered to be the envelope of worst case crane operating conditions.

Rope failure was simulated using a step function to reduce the force in one of the ropes to zero over 0.001 seconds after a short delay during which the model was allowed to stabilise. Sensitivity runs were also carried out to investigate the effects of friction, damping and rope failure time.

## 6.0 Results

A summary of the main analysis runs can be found in Table 1 below with the dynamic load factors on the remaining rope after failure in the final column

| Load Case | Rope Failed | Normal Unfailed Rope Load (N) | Peak Load in Unfailed Rope (N) | **Load Factor (Peak / Normal)** |
|---|---|---|---|---|
| RPV Short Rope | 1 Fail | 38800 | 104430 | **2.692** |
| | 2 Fail | 39600 | 98384 | **2.484** |
| RPV Long Rope | 1 Fail | 38700 | 103580 | **2.677** |
| | 2 Fail | 38700 | 103760 | **2.681** |
| MRC Short Rope | 1 Fail | 42750 | 114380 | **2.676** |
| | 2 Fail | 43700 | 107310 | **2.456** |
| MRC Long Rope | 1 Fail | 42800 | 111920 | **2.615** |
| | 2 Fail | 42800 | 111760 | **2.611** |

Table 1 Dynamic load factors for the remaining, unfailed, rope for all analyses.

Figure 4 presents a typical results plot for a RPV Head load and Figure 5 presents a typical results plot for an MRC load

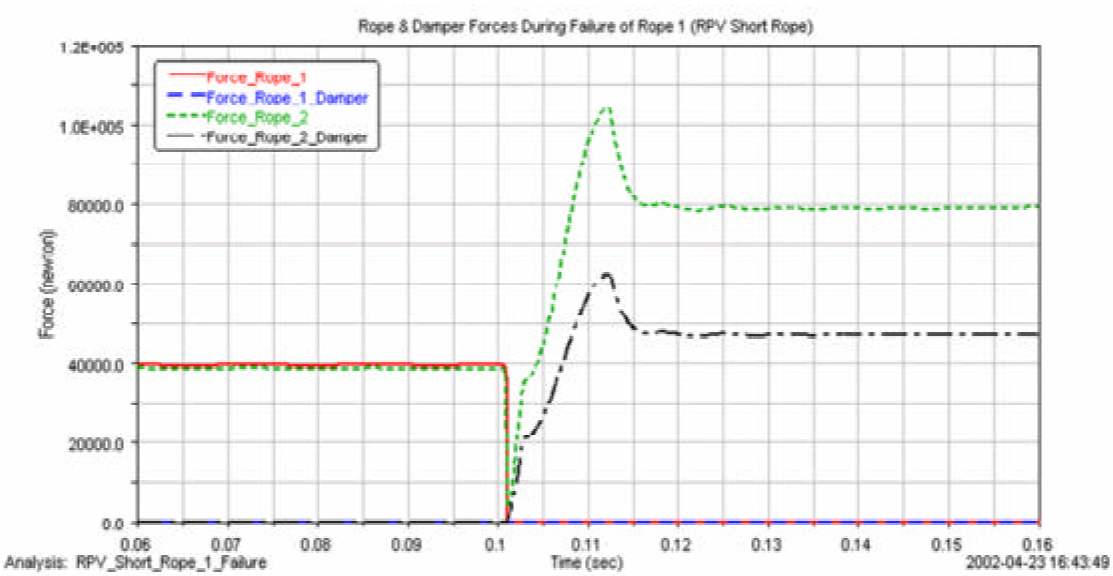


Figure 4 Rope and Damper forces during failure of Rope 1 for RPV Head s hort rope.

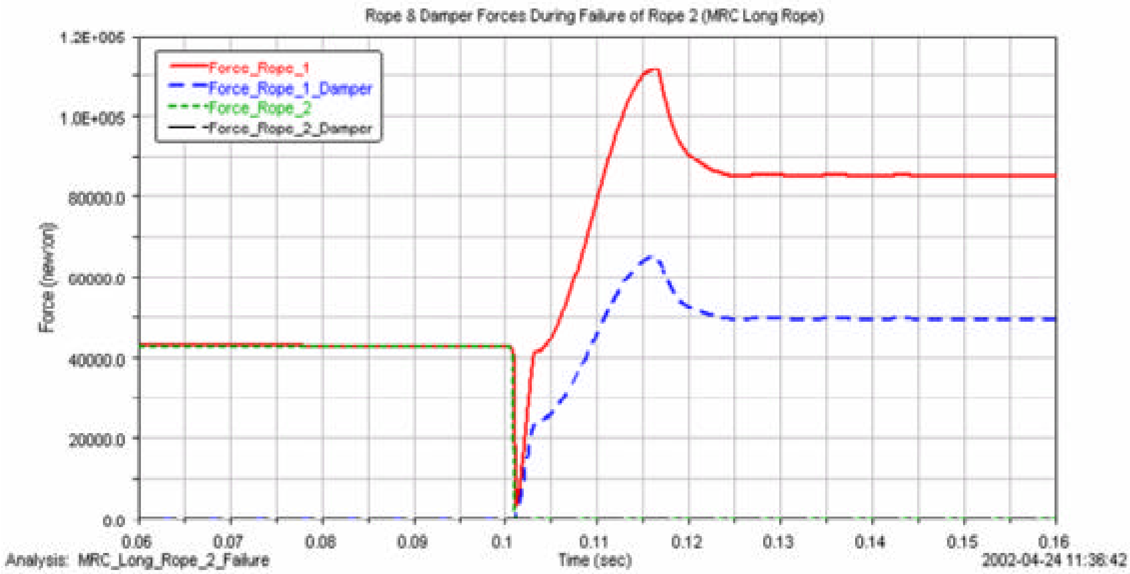


Figure 5 Rope and Damper forces during failure of Rope 2 for MRC long rope.

### 6.1 Validation

By using an energy balance, equating the potential energy due to rope failure with the strain energy required to arrest the load an approximate load factor can be estimated. This calculation is subject to the assumption that the rope mounting points are rigid.

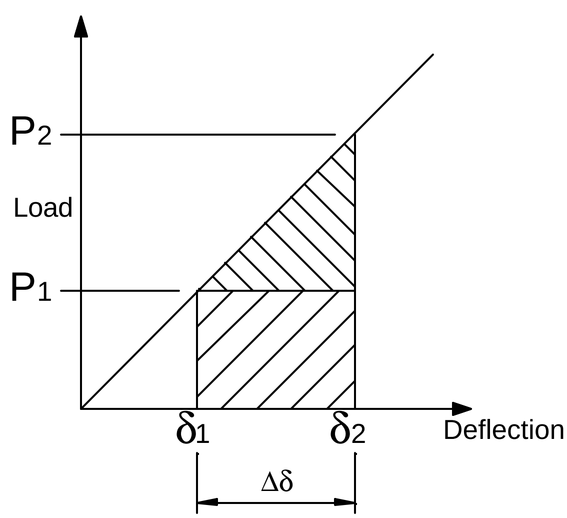


Figure 6 – Strain Energy Diagram

$P_1$ = Initial Rope Load in System (per rope system)
$P_2$ = Max Rope After Rope Breakage
$d_1$ = Extension Due to Rope Load $P_1$
$d_2$ = Extension Due to Rope Load $P_2$

*Decrease in Potential = Increase in Strain Energy*
*Decrease in Potential =* $2P_1Dd$
*Increase in Strain =* $P_1Dd + 0.5(P_1-P_2)\, Dd$
*(= Area Under Graph)*
Therefore:
$2P_1Dd = P_1Dd + 0.5(P_1-P_2)\, Dd$

$\boldsymbol{P_2 = 3P_1}$

This suggests a maximum peak load of 3 times initial rope load assuming that there is no movement of the shock absorbers.

## 7.0 Discussion

Studying Figures 4 and 5 it can be observed that, as rope failure occurs, the compensating beam on the crane rotates causing the compression only damper attached to the other rope to act, cushioning the subsequent dynamic load spike. It can also be noted that the final steady state loading the remaining rope is double that prior to the failure event.

The maximum load factor during the various rope failure events analysed is approximately 2.7, and the peak rope load experienced 114 kN, this represents a reserve factor of 4.8 against a rope breaking load of 547kN. The model results fit well with the energy balance validation presented.

## 8.0 Conclusions

- An ADAMS model has been created to substantiate the redundant load path design of a High Integrity Crane
- The model was set up to measure the dynamic load factor on a single rope during the transient event of rope failure on the other rope.
- Dynamic rope load factors of 2.7 were simulated.
- These values successfully substantiated the design.

## 9.0 References

1. Strachan & Henshaw Report 4D195/D678; “Dynamic Simulation of 9 Dock RAH 45t Crane Rope Failure” Issue 01, May 2002

## 10.0 Acknowledgements

Many thanks to Neil McGowan of MVP Ltd. for help with the initial set up of this model.