Australian Government
Australian Transport Safety Bureau

ATSB TRANSPORT SAFETY INVESTIGATION REPORT
Marine Occurrence Investigation No. 186 and 191
Final

Independent investigation into the equipment failure on board the Australian registered bulk carrier Goliath

in Bass Strait
22 September 2002
and
off Jervis Bay, New South Wales
12 February 2003

Australian Government

Australian Transport Safety Bureau

ATSB TRANSPORT SAFETY INVESTIGATION REPORT
Marine Occurrence Investigation
No. 186 and 191
FINAL

Independent investigation into the equipment failure on board the Australian registered bulk cargo carrier *Goliath*

in Bass Strait on 22 September 2002 and
off Jervis Bay, New South Wales, 12 February 2003

Released in accordance with regulation 16 of the *Navigation (Marine Casualty) Regulations 1990*

Published by:	Australian Transport Safety Bureau
Postal address:	PO Box 967, Civic Square ACT 2608
Office location:	15 Mort Street, Canberra City, Australian Capital Territory
Telephone:	1800 621 372; from overseas + 61 2 6274 6590
	Accident and serious incident notification: 1800 011 034 (24 hours)
Facsimile:	02 6274 6474; from overseas + 61 2 6274 6474
E-mail:	atsbinfo@atsb.gov.au
Internet:	www.atsb.gov.au

ISBN and formal report title: see 'Document retrieval information' on page v.

CONTENTS

DOCUMENT RETRIEVAL INFORMATION

Report No.	Publication Date	No. of pages	ISBN	ISSN
186 & 191	June 2006	88	1 921092 46 7	1447-087X

Publication Title

Independent investigation into the equipment failure on board the Australian bulk carrier *Goliath* in Bass Strait on 22 September 2002 and off Jervis Bay, New South Wales, on 12 February 2003.

Prepared by

Australian Transport Safety Bureau
PO Box 967, Civic Square ACT 2608 Australia
www.atsb.gov.au

Abstract

On 22 September 2002 the catastrophic failure of the main engine turbocharger disabled the cement carrier *Goliath* in Bass Strait. The replacement turbocharger failed in a similar manner on 12 February 2003, only four and a half months later, when *Goliath* was off Jervis Bay, again disabling the ship.

At 0107 on 22 September 2002, the Australian flag bulk cement carrier *Goliath* experienced a main engine turbocharger failure while the vessel was en route from Newcastle to Devonport when the turbocharger ‘exploded’ and disabled the ship’s main engine.

At 1543 on 12 February 2003, while *Goliath* was en route from Devonport to Sydney, the turbocharger failed again. This time, the failure was witnessed by the second engineer who heard the rapid acceleration of the turbocharger before it exploded. Once again, the turbocharger had been damaged beyond repair and the ship was disabled.

The investigation report concludes that both turbocharger failures were similar and had occurred when the compressor discs burst due to overspeed. While it is not possible to state with certainty, the most likely mechanism leading to both overspeeds was a scavenge fire in the engine.

The ATSB delayed the release of these final investigation reports because of litigation involving the parties and because the investigations preceded and were not protected under the *Transport Safety Investigation Act 2003*.

THE AUSTRALIAN TRANSPORT SAFETY BUREAU

The Australian Transport Safety Bureau (ATSB) is an operationally independent multi-modal Bureau within the Australian Government Department of Transport and Regional Services. ATSB investigations are independent of regulatory, operator or other external bodies.

The ATSB is responsible for investigating accidents and other transport safety matters involving civil aviation, marine and rail operations in Australia that fall within Commonwealth jurisdiction, as well as participating in overseas investigations involving Australian registered aircraft and ships. A primary concern is the safety of commercial transport, with particular regard to fare-paying passenger operations. Accordingly, the ATSB also conducts investigations and studies of the transport system to identify underlying factors and trends that have the potential to adversely affect safety.

The ATSB performs its functions in accordance with the provisions of the *Transport Safety Investigation Act 2003* and, where applicable, relevant international agreements. The object of a safety investigation is to determine the circumstances to prevent other similar events. The results of these determinations form the basis for safety action, including recommendations where necessary. As with equivalent overseas organisations, the ATSB has no power to implement its recommendations.

It is not the object of an investigation to determine blame or liability. However, it should be recognised that an investigation report must include factual material of sufficient weight to support the analysis and findings. That material will at times contain information reflecting on the performance of individuals and organisations, and how their actions may have contributed to the outcomes of the matter under investigation. At all times the ATSB endeavours to balance the use of material that could imply adverse comment with the need to properly explain what happened, and why, in a fair and unbiased manner.

Central to the ATSB's investigation of transport safety matters is the early identification of safety issues in the transport environment. While the Bureau issues recommendations to regulatory authorities, industry, or other agencies in order to address safety issues, its preference is for organisations to make safety enhancements during the course of an investigation. The Bureau is pleased to report positive safety action in its final reports rather than make formal recommendations. Recommendations may be issued in conjunction with ATSB reports or independently. A safety issue may lead to a number of similar recommendations, each issued to a different agency.

The ATSB does not have the resources to carry out a full cost-benefit analysis of each safety recommendation. The cost of a recommendation must be balanced against its benefits to safety, and transport safety involves the whole community. Such analysis is a matter for the body to which the recommendation is addressed (for example, the relevant regulatory authority in aviation, marine or rail in consultation with the industry).

1 EXECUTIVE SUMMARY

At 0107 on 22 September 2002, the Australian registered bulk cement carrier, *Goliath,* experienced a main engine turbocharger failure while the vessel was en route from Newcastle to Devonport. The turbocharger had 'exploded' and in the process had disabled the ship's main engine. At the time of the incident, the vessel was some 26 miles south of Point Hicks on the south-east coast of Victoria.

With no possibility of running the main engine, the vessel's managers (in consultation with the hull and machinery underwriters) organised a tow through United Salvage. The weather at the time, and for the ensuing three days, was good. A tug was dispatched from Melbourne and eventually arrived at *Goliath*'s position some 39 hours later. *Goliath* finally arrived at Station Pier in Port Melbourne at 1700 on 25 September after an uneventful tow. While the ship was berthed at Station Pier, the main engine was fitted with a new turbocharger assembly.

Some four and a half months later, *Goliath* was en route from Devonport to Sydney with a cargo of cement. By the afternoon of 12 February, the ship was off the south coast of New South Wales in good weather with the voyage proceeding normally.

At 1540 the second engineer was working in the engine room adjacent to the main engine when he heard the engine speed increase slightly followed by the rapid acceleration of the turbocharger. The turbocharger's speed sounded as though it was increasing exponentially. He realised that the turbocharger was overspeeding dangerously so he started to run for the control room a short distance away. As he was approaching the control room he encountered the third engineer coming in the opposite direction and pushed him forcefully back into the control room. As the two engineers hit the floor of the control room, the main engine turbocharger exploded. The turbocharger had been damaged beyond repair again and the ship was effectively disabled with no way to run the main engine.

The master contacted the ship's management company to inform them of the situation and by 1755 the ship's managers had reached a towage agreement with Adsteam Marine. A Sydney based tug was mobilised and by 0956 on 13 February the tug had arrived at the ship's position and *Goliath* had been taken in tow. *Goliath* arrived at its berth at Glebe Island in Sydney at 1200 on 14 February after an uneventful tow.

The report's conclusions include:

- Both turbocharger failures were attributable to the radial fracture of the centrifugal compressor disc/impeller.
- The fracture characteristics of both compressor disc failures were similar and typical of material failure under centrifugal overload conditions.
- Several crew members of *Goliath* witnessed the turbocharger undergo an uncontrolled transient overspeed event immediately before the second failure.

- The similarity of the compressor disc failures supports the likelihood that the first turbocharger underwent an uncontrolled transient overspeed event similar to the second failure.
- The condition of some of the 'scavenge' spaces[1] at the time of each failure was poor.
- While it is not possible to state with certainty, the two possible mechanisms which led to the turbocharger failure were a slight slip of the compressor disc or a scavenge fire but was more likely to have been a scavenge fire based on the evidence.
- The vessel's maintenance regime prior to each turbocharger failure was probably inadequate with respect to scavenge cleaning.

The report recommends that shipowners, managers and ship's engineers should ensure that the maintenance regime applied to slow speed diesel engine scavenge spaces is thorough and takes into account the engine's history, condition and conditions of service.

The ATSB delayed the release of these final investigation reports because of litigation involving the parties and because the investigations preceded and were not protected under the *Transport Safety Investigation Act 2003*.

1 Under piston spaces that provide fresh air for each cylinder to clear them of exhaust gas.

2 SOURCES OF INFORMATION

The master and crew of *Goliath*

CSR Shipping, Sydney

Australian Maritime Safety Authority (AMSA)

ABB Turbo Systems

HSD Engine Company

Wartsila-NSD

References

Sulzer RTA-U Engine Selection and Project Manual

3 NARRATIVE

3.1 *Goliath*

Goliath is an Australian flag, self-discharging bulk cement carrier of 15 539 deadweight tonnes at its summer draught of 8.335 m (Figure 1). The vessel was owned by Goliath Portland Cement Company, Tasmania, and was managed by CSR Shipping, Sydney. It is classed ✠100A1, Cement Carrier strengthened for heavy cargoes, with ✠ LMC[2], UMS[3] and CCS[4] notations, with Lloyd's Register.

Goliath was built in 1993 by Hanjin Heavy Industries Company in Ulsan, South Korea. The vessel has an overall length of 143.00 m, a moulded breadth of 23.50 m and a moulded depth of 11.90 m. Bulk cement is carried in four holds, forward of the accommodation superstructure. The cargo is discharged using ship's equipment.

The vessel trades mainly between Devonport in northern Tasmania and the mainland ports of Melbourne, Newcastle and Sydney.

Figure 1: *Goliath* alongside Station Pier, Melbourne

Goliath has a crew of 18 comprising a master and three mates, chief and three engineers, chief and six integrated ratings, two catering staff and a trainee. At the time of the incidents all of the crew were Australian nationals.

2 Notation assigned when machinery is constructed and installed under Lloyd's Special Survey in accordance with Lloyd's rules.

3 Notation denotes ship may be operated with the machinery spaces unattended.

4 Central control station.

The mates on *Goliath* maintain the traditional four hours on, eight hours off watchkeeping system. The engineers operate a 24 hour duty roster system with the engine room unmanned outside of normal daytime working hours which are between 0800 and 1700 each day.

Goliath's master at the time of both incidents held a master class one certificate of competency issued by the Australian Maritime Safety Authority and had been at sea since 1960. He had been master since 1971 and had been master on the vessel since 1993. The chief engineer on board the vessel at the time of the first incident had been at sea since 1975 and held a class one (motorship) certificate of competency issued by the Australian Maritime Safety Authority and had been chief engineer on *Goliath* for the previous six years. The chief engineer on board the vessel at the time of the second incident had been at sea since 1987 and held a class one (motorship) certificate of competency issued by the Australian Maritime Safety Authority. He had been chief engineer on *Goliath* for the previous two years but had also sailed on the vessel as first engineer some seven years previously for a period of three years.

3.2 The main engine

Goliath's main engine is a Sulzer 5RTA52 of 6 080 kW built under licence by Korean Heavy Industries Company (Figures 2 and 3). The engine is a five cylinder, two-stroke, single acting, 'crosshead' type with a bore of 520 mm and a stroke of 1 800 mm. The engine is directly reversible, has a normal operating speed of 120 rpm, and drives a single fixed-pitch propeller to give the ship a service speed of 14.5 knots.

As a crosshead engine, each of *Goliath*'s main engine pistons are connected to individual sliding crosshead assemblies via fixed piston rods. The motion of each piston is then transmitted from its crosshead assembly to the crankshaft via a connecting rod. Each piston rod slides through a stuffing box which seals the scavenge space around the bottom of the cylinder liner from the crankcase. Pressurised scavenge air is provided by an exhaust gas driven turbocharger via a charge air cooler.

Scavenging (clearing) of cylinder exhaust gas is achieved as each piston nears the bottom of its power stroke and the piston uncovers the scavenge ports located towards the bottom of the cylinder liner. When the scavenge ports are uncovered, pressurised and cooled scavenge air flows from the scavenge space into each cylinder and travels upward to exit the combustion chamber via a single camshaft driven exhaust valve fitted to the cylinder cover. The exhaust valve opens just before the piston uncovers the scavenge ports and closes just after the piston moves back upward to re-cover the scavenge ports. As the piston continues to travel upward towards top-dead-centre (TDC) the air in the cylinder is compressed. Fuel injection takes place just before the piston reaches TDC, with ignition taking place as, or just after, the piston passes TDC. The power stroke then takes place until the exhaust valve opens just before the piston uncovers the scavenge ports again.

Figure 2: Main engine top

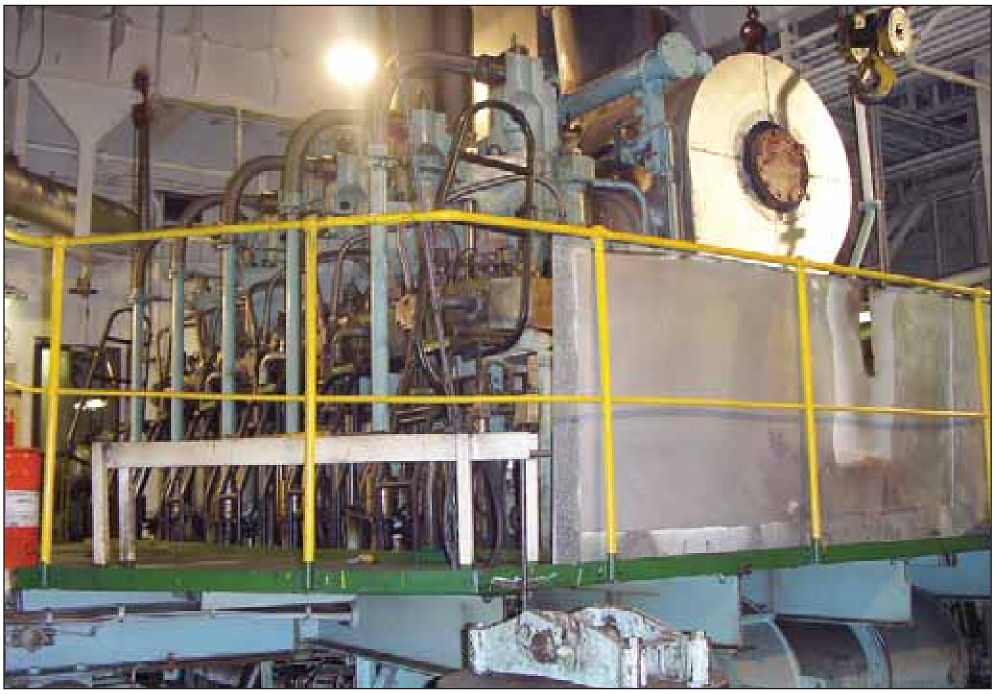
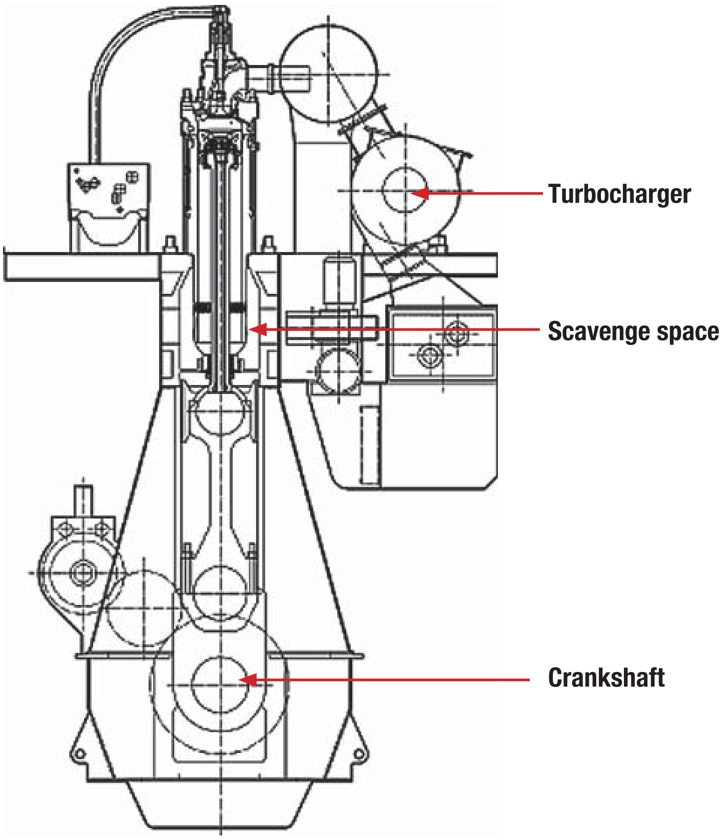


Figure 3: Cross-sectional drawing of RTA Sulzer



Goliath's main engine is fitted with a single ABB designed, VTR 454A-32 exhaust gas driven turbocharger built under license by Ishikawajima Harima Heavy Industries (IHI) Company in South Korea.

The main components of the turbocharger are (refer to fig.1, annex 1);

- the rotating element or rotor (a steel shaft with a turbine disc on one end and an air inducer/compressor impeller assembly on the other),
- an air filter/silencer,
- compressor casing,
- gas inlet casing and,
- gas outlet casing.

Exhaust gas from the main engine flows through the gas inlet casing to a nozzle ring. The exhaust gas is expanded by the nozzle ring and directed onto the blades of the turbine disc. The exhaust gas then flows to atmosphere via the gas outlet casing, exhaust uptake and an economiser. The rotor is spring mounted on rolling contact bearings and is free to rotate. The exhaust gas energy imparted to the turbine disc turns the rotor which in turn drives the air inducer/compressor impeller assembly. Air is drawn into the rotating inducer/compressor impeller, via the air filter/silencer, where its velocity is greatly increased. The high velocity air then flows via the diffuser into the volute of the compressor casing (or scroll) where the air's velocity is converted to pressure. The compressed air then flows to the charge air cooler, where it is cooled, and from there to the scavenge manifold via a water separator. The cooled and compressed air then flows from the scavenge manifold into each of the scavenge spaces surrounding the bottom of each cylinder liner. Each scavenge space is fitted with three sets of reed valves (non-return valves) which prevent the flow of air back into the scavenge manifold.

Turbocharger rotor revolutions are directly related to main engine load. When the engine load increases so do the turbocharger revolutions and the scavenge (charge air) pressure. On *Goliath* this means an increase in main engine revolutions (and thus main engine load) increases the turbocharger revolutions, 120 engine revolutions per minute corresponds to approximately 15 000 turbocharger revolutions per minute and a charge air pressure of approximately 2.3 bar. *Goliath*'s main engine turbocharger has a maximum rated speed of 17 400 revolutions per minute.

3.3 The incident

3.3.1 Incident 22 September 2002

Goliath left Newcastle, New South Wales, on 21 September 2002 bound for Devonport, Tasmania, to load a cargo of powdered cement.

At approximately 0107 on 22 September, when the ship was 26 miles south of Point Hicks, Victoria, the main engine turbocharger exploded. Many of the crew were awoken by the explosion including the chief engineer, second engineer (who was duty engineer for the day) and third engineer. As the second engineer got out of bed the alarm panel in his cabin started to annunciate several alarms followed by the fire alarm. He quickly dressed and went to the engine room meeting the third engineer on the way.

By this time the chief engineer had arrived in the engine room and found the space filled with exhaust gas. Water, lagging and other debris were strewn around the main engine top plates. The main engine was still running but had slowed to approximately 50 rpm. On closer inspection the chief engineer saw a large hole in the main engine turbocharger gas outlet casing and realised that the turbocharger had failed catastrophically. He went to the control room where he telephoned the bridge to request that they stop the main engine.

When the other engineers arrived in the engine room they set about checking and stabilising other engine room systems and cleaning up the mess made by the turbocharger explosion.

After the main engine had stopped the chief engineer made a closer inspection of the damaged turbocharger (Figure 4). He found that the gas outlet casing had been shattered and the rotor was lying in the bottom of the casing without its compressor disc. He then rang the master to inform him that the turbocharger had failed completely and that there was no way the turbocharger could be repaired and so the main engine could not be run.

At 0150 the master made contact with CSR Shipping and informed them of the situation and the need to organise a tow. He then contacted various tug companies and the Rescue Coordination Centre (RCC) in Canberra to inform them of the situation.

By 0920 the master and vessel's managers (in consultation with the hull and machinery underwriters) had arranged a lump sum contract through United Salvage to tow *Goliath* to Melbourne. The tug *Keera* was dispatched from Melbourne later that day. The weather at the time of the turbocharger failure and for the ensuing three days was good.

While waiting for *Keera* to arrive, *Goliath* drifted towards the coast and the master had become concerned that the ship may ground. In response to these concerns, *Goliath*'s managers had organised a rig support vessel in the area, *Lady Elizabeth*, to stand-by the ship.

On the morning of 23 September, when it became clear to the master that there was some danger of the ship grounding, *Lady Elizabeth* took *Goliath* in tow. At 1620 on 23 September the two vessels rendezvoused with *Keera* and the tow was passed to the Melbourne tug which continued towing the ship into Melbourne.

Goliath finally arrived at Station Pier in Port Melbourne at 1700 on 25 September without further incident.

While the ship was in Melbourne, a replacement turbocharger was fitted to the main engine. All components of the turbocharger assembly were completely renewed except the air filter/silencer unit which was overhauled and then refitted to the new turbocharger.

After the repairs were completed, *Goliath* departed Melbourne and sailed to Devonport on 30 September 2002. At about 0230 on 1 October, after the vessel had arrived in Devonport, high temperature alarms in the scavenge space for number three cylinder indicated that a scavenge fire had occurred.

Figure 4: Failed turbocharger 22 September 2002

3.3.2 Incident 12 February 2003

At 0940 on 11 February 2003 *Goliath* left Devonport en route to Sydney with a cargo of cement. By the afternoon of 12 February the ship was off the south coast of New South Wales in good weather with the voyage proceeding normally.

At 1540 the second engineer was working in the engine room adjacent to the main engine when he sensed the engine speed increase by perhaps one RPM followed by the rapid acceleration of the turbocharger. The turbocharger's speed sounded as though it was increasing exponentially. He realised that the turbocharger was overspeeding dangerously so he started to run for the control room a short distance away. As he was approaching the control room he encountered the third engineer coming in the opposite direction and pushed him forcefully back into the control room. As the two engineers hit the floor of the control room, the main engine turbocharger exploded.

After the second and third engineers had risen from the control room floor, they found the engine room full of smoke with cooling water and debris lying all around the main engine top plates. The turbocharger outlet casing had ruptured and the air casing was lying approximately two metres from the turbocharger (Figure 5). The fire alarm was sounding. The second engineer went to the engine side and brought the main engine fuel rack control to zero to stop the main engine.

By 1545 the engineers had called the bridge to indicate that there was no fire but that the main engine turbocharger had exploded. The master was on the bridge by this time and he contacted the ship's management company to inform them of the situation and to indicate that a tow might be required. The chief engineer arrived in the engine room and, after inspecting the damaged turbocharger, informed the master that the main engine could not be run.

The master made contact with the RCC in Canberra at 1600 to report the incident.

By 1755 the ship's managers had reached a towage agreement with Adsteam Marine. The Sydney based tug *Wonga* was mobilised to rendezvous with *Goliath* and then tow the disabled ship to Sydney.

At about 1830, the Chief Engineer inspected the scavenge spaces through the fuel pump side scavenge doors. He noticed that number one and number two scavenge spaces were relatively dry. A subsequent inspection was done after the ship had berthed in Sydney which is discussed in Section 4.6.1. of this report.

At 0956 on 13 February *Wonga's* tow line was secured on *Goliath* and the tow began. *Goliath* finally arrived at its berth at Glebe Island in Sydney at 1200 on 14 February after an uneventful tow.

Figure 5: Failed turbocharger 12 February 2003

4 COMMENT AND ANALYSIS

4.1 Evidence

Following the failure in September 2002 the ATSB initiated an investigation into the circumstances and causes of the turbocharger failure. An investigator from the Australian Transport Safety Bureau (ATSB) attended *Goliath* at Station Pier in Port Melbourne on 25 and 26 September 2002. A variety of documentary evidence was obtained from the ship including copies of the ship's navigation charts, log books, movement book, passage plan, engine room logs, maintenance records and procedures. The master, chief engineer, duty engineer and marine manager from CSR Shipping were interviewed and provided accounts of the incident.

The investigation was underway when the ATSB was notified of the second turbocharger failure in February 2003. It was decided that the two incidents should be investigated in combination and two investigators from the ATSB attended *Goliath* while the ship was berthed at Glebe Island on 14 February. Once again a variety of documentary evidence was obtained from the ship and the relevant crew were interviewed. Investigators examined the failed turbocharger and took possession of samples of various parts including the compressor disc and turbine blades. The samples were subsequently examined in the ATSB metallurgical laboratory to determine the possible turbocharger failure mode. Samples from the first turbocharger failure were also obtained and subject to the same analysis.

Advice was sought from the turbocharger manufacturer including; the incidence of similar failures in the past, possible failure mechanisms and past failures of the VTR 454-32 turbocharger. ABB Turbo Systems provided advice including:

> Such incidents of overspeed are rather rare, but they can happen.
>
> Most common mechanisms are excessive energy to the turbine (e.g. malfiring of engine, burning of slobbery oil in the manifold, etc.) or reduced energy absorption on compressor side (e.g. due to excessive back pressure from after cooler).
>
> We have no special negative records from this type of engine series and the VTR454-32 turbocharger.

The main engine manufacturer was also consulted and similar advice was sought regarding the failure and possible mechanisms. They provided the following:

> Up to now, lots of RTA.2/2U engines are being produced and operated in the world but we have not experienced in turbocharger explosion. On rare occasions, turbocharger trouble was reported to us but not turbocharger explosion. Typically, bearing damage, turbocharger blade broken etc.
>
> …In principle, we can say that a rapid increase in the turbocharger speed to a very high speed, which finally destroyed the turbocharger can only be reached by an additional energy caused for example by exhaust pipe fire or piston underside fire. If such a fire occurs, something must be wrong in the engine operation…
>
> We have produced a number of RTA.2/2 engines but we have not yet experienced in fuel and exhaust cam damage that leads to turbocharger explosion.

4.2 Failure mechanisms and technical analysis

The complete technical analysis report by the ATSB Technical Analysis Unit is contained in Annex 1. In summary the ATSB's technical analysis concluded:

- Both turbocharger failures on board *Goliath* were very comparable in terms of the extent of damage sustained.
- Both turbocharger failures were attributable to the radial fracture of the centrifugal compressor disc/impeller.
- The fracture characteristics of both compressor disc failures were similar and typical of material failure under centrifugal overload conditions.
- No evidence of prior cracking or other defects was found within the examinable fractures of both compressor discs.
- The metallurgical and physical properties of the first failed disc were considered satisfactory and typical of the material used to produce the disc.
- Several crew members of *Goliath* witnessed the turbocharger undergo an uncontrolled transient overspeed event immediately before the second failure.
- The similarity of the compressor disc failures supports the likelihood that the first turbocharger underwent an uncontrolled transient overspeed event similar to the second failure.

4.3 The turbocharger failures

Goliath experienced two major main engine turbocharger failures within six months of each other.

The ship's maintenance records indicate at the time of the first failure, the turbocharger had run 5 600 hours since its previous major overhaul (which included the complete re-blading of the turbine wheel) and that routine maintenance on the unit was up to date. After the first failure, the turbocharger assembly was completely renewed with the exception of the air filter/silencer unit which was overhauled and then refitted to the new turbocharger. At the time of the second failure, the new turbocharger had run for only 1 540 hours.

When the turbocharger failed in September 2002, there were no witnesses to the event as it occurred at 0107 with the engine room operating unmanned. There were no engine alarms immediately before the failure and the main engine monitoring equipment did not record any abnormality in the time leading up to the failure or the actual turbocharger speed at the time of the failure. There was no evidence to indicate the mechanism which led to the turbocharger failure aside from what could be ascertained from the analysis of its failed components.

The turbocharger failure in February 2003, however, occurred while the engine room was manned and was witnessed, in particular, by the second engineer. The second engineer had sailed on the ship for a total of almost two years and was 'in-tune' with the normal running condition of the main engine and turbocharger. His observation of the slight main engine speed increase just before the turbocharger's rapid speed increase and subsequent failure is considered to be reliable. He estimated that the whole event took approximately seven seconds which at 120 rpm corresponds to approximately 14 revolutions (and 14 firing strokes for each cylinder). Once again the main engine monitoring plant did not record any abnormality with the turbocharger or main engine in the time immediately before the failure or the turbocharger speed at the time of the explosion.

4.4 Turbocharger overspeed mechanisms

There are several possible mechanisms which may result in a transient turbocharger overspeed. These mechanisms can be largely separated into two categories:

- an increase in the effective energy driving the turbine or,
- the effective unloading of the compressor.

4.4.1 Turbine driving mechanisms

An increase in the effective energy driving the turbine may result from increased energy in the engine's exhaust gas, most commonly as a result of partially combusted fuel being exhausted from the engine's cylinders. An example of this phenomenon is a scavenge fire which has the potential to increase the energy in the engine exhaust by starving the combustion process in the cylinders. Turbocharger 'surging' is a common symptom of a significant scavenge fire and is caused by a higher level of exhaust gas energy output from the variously affected cylinders. The increase in exhaust gas energy increases the turbine rotor speed with a resulting increase in the air mass flow from the compressor assembly. The scavenge pressure usually increases until the air flow from the compressor 'stalls' and the breakdown of air delivery is followed immediately by a backward wave of air (or 'bark') through the air volute, compressor and air filter/silencer. In most scavenge fires, the increase in exhaust gas energy is not sufficient to result in a catastrophic turbocharger overspeed.

Another mechanism for increasing the exhaust gas energy at the inlet to the turbocharger is a fire or after burning in the exhaust manifold before the turbocharger. Exhaust manifold fires are not common on large slow speed diesel engines but possible if sufficient uncombusted fuel or lube oil (from cylinder lubrication for example) is ignited within the exhaust manifold plenum. The phenomenon has been documented in smaller two stroke diesel engines powering rail locomotives where it has led to turbocharger overspeed failures. If a fire, or sufficient after burning occurs within the exhaust manifold, the manifold becomes a high volume combustion chamber with the fire rapidly heating and expanding the hot gases exhausted from the cylinders before they enter the turbocharger.

An increase in the effective energy driving the turbine may also result from a change in the conditions in the exhaust uptake. A reduction in pressure in the exhaust trunk after the turbine will increase the effective energy driving the turbine and consequently the turbocharger speed. An example of this phenomenon is an economiser fire which may create conditions of increased up-draught by heating the exhaust uptake which will decrease the effective backpressure at the turbine.

4.4.2 Compressor unloading mechanisms

The turbocharger's compressor assembly may be effectively unloaded in a number of ways which may result in a transient overspeed of the turbocharger rotor. These include:

- a sudden significant increase in the pressure drop across the air inlet casing i.e. a restriction,
- a sudden significant rise in the air pressure on the outlet side of the compressor, i.e. a blockage at the charge air cooler or after the cooler at the inlet to the scavenge manifold,
- a sudden significant decrease in the air pressure at the compressor's outlet i.e. a large leak from the scavenge manifold, charge air cooler or air trunking,
- the loss or partial loss of one or more compressor impeller or inducer blades,
- sustained relative slip between the compressor disc and the rotor shaft, or,
- a slight slip of either the compressor or inducer disc resulting in misalignment between the two rotating elements.

All of these mechanisms will effectively unload the turbocharger rotor and will result in a higher rotor speed for a given exhaust gas energy input. Any decrease in air mass flow from the turbocharger to the engine for a given load setting will also result in rapidly rising exhaust gas temperatures which may further increase the turbocharger speed.

4.5 Main engine performance data

Main engine performance data recorded in *Goliath*'s engine room logs provides useful information on the general running condition of the main engine and turbocharger before each failure. The main engine performance parameters drawn from past periodic data logs, (at times when the main engine was running with a normal service load), in the voyages prior to each turbocharger failure are reproduced in Table 1. Main engine trials data is also reproduced for reference in Table 2.

The main engine data log on 21 September 2002 was taken automatically at 1530, more than nine hours before the first turbocharger failure. At the time, the ship was on a relatively long voyage from Devonport and from Newcastle and the engine load and other parameters would have been very similar to those immediately before the turbocharger failure. Of note are the exhaust temperatures from number three cylinder at the time and also over the previous twelve day period covered by the logs in Table 1. Figure 6 is a graph of the cylinder exhaust temperature deviations for voyages in the twenty days prior to the first failure when the engine was running at a normal service load. The graph clearly shows number three cylinder exhaust temperatures, while well within normal limits, were up to

15 degrees higher than the other four cylinders, and had progressively increased over the period.

The data log on 12 February 2003 was taken at 1530, only 10 minutes before the second turbocharger failure. This data is very relevant as it indicates that all of the critical engine and turbocharger parameters at this time were within normal limits. While still within normal limits it is of note that the turbocharger exhaust inlet and outlet temperatures (allowing that the thermocouple was reading approx 50 degrees high according to the chief engineer's subsequent advice) and the cylinder exhaust temperatures, in particular from number two cylinder, were somewhat higher than usual. Like the exhaust temperatures from number three cylinder before the first turbocharger failure, the exhaust temperatures from number two cylinder were up to 30 degrees higher than the other cylinders and had progressively increased in the period prior to the second turbocharger failure (Figure 7). Figure 7 is a graph of the cylinder exhaust temperature deviations for voyages in the sixty days prior to the second failure when the engine was running at a normal service load.

After the first failure, the turbocharger was completely renewed, with the exception of the air filter/silencer. The data from the logs prior to the second failure indicates that engine performance with the new turbocharger was somewhat worse at similar load settings than it was with the original turbocharger. The cylinder exhaust gas temperatures were somewhat higher and scavenge pressures lower for similar engine load settings.

Table 1: Excerpts from engine room logs showing main engine seagoing performance parameters

Performance data for main engine from engine room logs														
Date	*Cylinder exhaust temperatures (°C)*					*T/C Inlet temp (°C)*	*T/C Out temp (°C)*	*ME load indic*	*Scav Press (bar)*	*Scav temp (°C)*	*Turbo RPM*	*Eng RPM*	*ER temp (°C)*
	1	*2*	*3*	*4*	*5*								
9/9/02	312	323	330	319	316	432	356	6.28	2.2	46.9	15 104	118	34
12/9/02	310	326	334	328	326	439	367	6.47	2.3	48.1	15 015	120	35
14/9/02	301	309	319	308	302	422	336	6.41	2.2	43.9	14 559	117	30
18/9/02	311	320	338	322	322	441	367	6.20	2.3	46.8	15 101	119	34
21/9/02	302	321	335	326	323	441	371	6.25	2.3	47.7	15 159	120	37
First turbocharger failure at 0107, 22 September 2002													
31/1/03	314	344	320	323	332	452	419*	-	2.1	42.8	14 229	-	36
1/2/03	302	329	310	311	325	454	419*	6.34	2.1	41.7	14 472	-	36
3/2/03	307	339	319	317	331	463	424*	6.36	2.1	42.0	14 528	-	38
6/2/03	332	367	326	336	337	460	432*	5.93	1.9	41.7	13 984	-	42
8/2/03	327	361	320	330	331	454	419*	6.13	2.1	41.9	14 427	-	35
11/2/03	324	359	319	327	325	452	420*	5.99	2.1	41.3	14 267	-	36
10 mins prior to failure	339	380	333	343	346	479	436*	6.21	2.1	42.6	14 625	-	42
Second turbocharger failure at 1540, 12 February 2003													

* Turbocharger outlet temperature thermocouple was apparently reading about 50° high.

Table 2: Excerpts from main engine trials at engine works

	Main engine trial data* (19/6/92)													
Load	*Cylinder exhaust temperatures (°C)*					*T/C Inlet temp (°C)*	*T/C Out temp (°C)*	*ME load indic*	*Scav Press (bar)*	*Scav temp (°C)*	*Turbo RPM*	*Eng RPM*	*ER temp (°C)*
	1	*2*	*3*	*4*	*5*								
50	280	300	280	300	290	330	287	5.3	0.85	42	11 100	95.1	21
78.5	310	325	310	320	310	370	278	6.6	1.65	38	14 300	110.7	22
90	320	340	330	330	325	390	280	7.1	2.00	36	15 500	115.9	22.5
100	340	370	360	365	350	425	287	7.5	2.31	37	16 500	120.0	23.5

* Trials were conducted on a test bed using diesel fuel (10 122 Kcal/kg, 3.512 cst @ 40°C) with the engine loaded by a mechanical brake.

Figure 6: Cylinder exhaust temperature deviations from 1 to 21 September 2002 before first failure

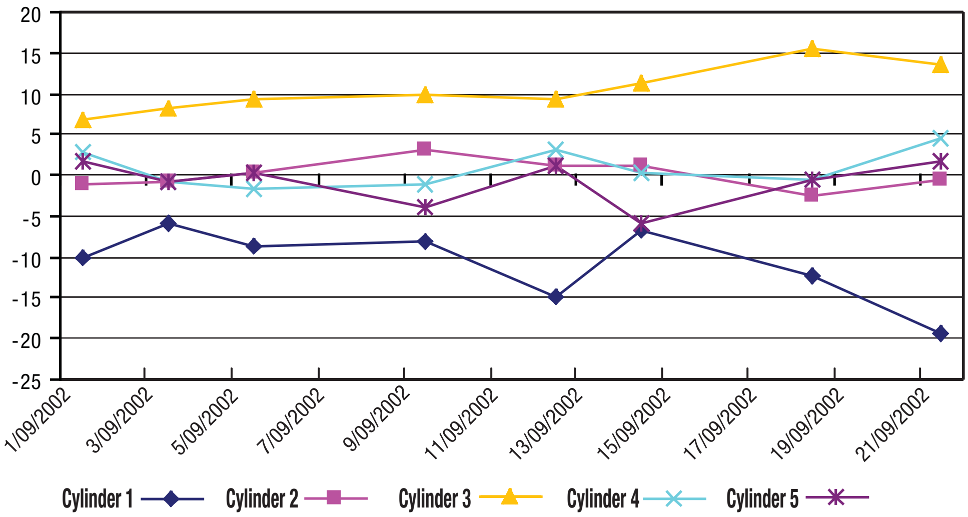
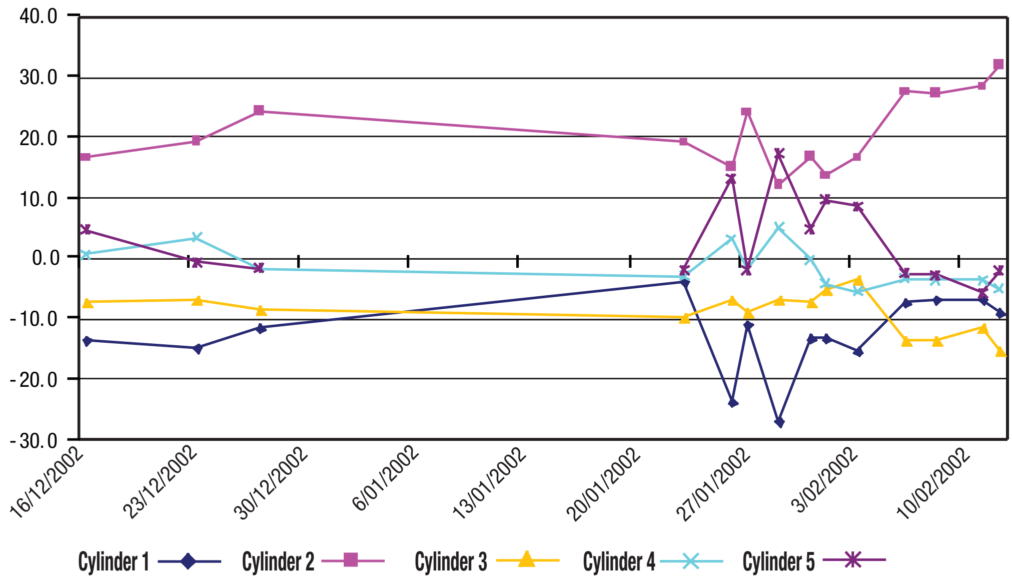


Figure 7: Cylinder exhaust temperature deviations from 16 December 2002 to 10 February 2003 before second failure



While the engine performance data prior to each failure does not reveal a definitive cause for the turbocharger failures they do reveal some information of relevance. It is significant that both turbocharger explosions occurred when the vessel was on longer voyages (i.e. not between Melbourne and Devonport) and the main engine was running in the higher range of normal load settings. In each case one cylinder had an exhaust temperature which was significantly higher than the average, although still within normal limits, and had an increasing trend in the days leading up to each turbocharger failure. The absence of any particularly abnormal data or any alarm prior to each failure is also significant as it indicates that the mechanism which caused the turbocharger overspeeds occurred over a very short time period.

4.6 Main engine inspections following turbocharger failures

After each of the turbocharger explosions, the ship's engineering staff performed a number of inspections and tests on the main engine. These inspections were particularly detailed after the second failure when it was evident from the observations of the second engineer that the turbocharger had over-sped. Inspections were conducted of the economiser, the main engine's exhaust system, scavenge spaces, charge air cooler, piston rings and exhaust valves. All of the main engine's fuel injectors were tested and the timing of the fuel pumps was checked.

The piston rings were all found to be in a satisfactory condition as were the fuel injectors and the timing of the fuel pump. The exhaust valves were also found to be in a satisfactory condition although there were some heavier carbon deposits (up to five mm thick) in the area above the valve seat on the exhaust manifold side of number two valve. The exhaust manifold was relatively clean (aside from turbocharger debris) and dry. The economiser and exhaust uptake revealed no signs of a fire.

The charge air cooler inspection revealed a number of damaged cooling tubes. The damage was consequential, (the result of the disintegration of the turbocharger), and thus not indicative of the cause of the overspeed

4.6.1 Scavenge spaces

Three hours after the second failure, the main engine scavenge spaces were opened and inspected by the chief engineer via the doors on the fuel pump side of the engine. He noted that the number one and two scavenge spaces in particular appeared to be relatively dry.

The chief engineer inspected the scavenge spaces again several days later, this time from the scavenge manifold side. In this inspection he found a degree of fouling on all of the cylinder liners, with the exception of the recently overhauled number three, and that the scavenge manifold side ports of the number two cylinder were almost totally blocked. In addition, two of the three reed valve banks on number two unit were totally blocked and the third 70 per cent blocked with scavenge residue. The two blocked reed valve banks also exhibited charring on the tips of the valve plates. The chief engineer took photographs of the scavenge spaces during his inspection (Figures 8 to 12).

In submission the chief engineer at the time of the second failure stated:

> Prior to the second turbocharger failure the No 2 exhaust temperature was seen to be running 25 to 35 degrees higher than the other units. During the repair period after the second failure all scavenge valves were cleaned, cylinder inlet ports cleaned and all reed valve banks changed. After repairs were complete I noted that the exhaust temperature on No 2 unit was back in line with the other units and it remained that way until I paid off and went on leave on the 4/03/2003. I rejoined the vessel in Sydney on the 25/04/2003 to find that the No 2 units exhaust temp again running 25-35 degrees C above the others. Upon reaching the next port (Devonport 27/04/2003) I carried out a scavenge inspection of that unit and found that the reed valves were almost completely choked up and the inlet ports on the scavenge side of that unit were in pretty much the same condition as they were after the second failure. There wasn't much residue on the floor of the scavenge though and drainage was good. No reason for this clogging up of valves and ports was immediately evident so the scavenge and ports were cleaned and reed valves changed. But prior to closing up the scavenge the 1st engineer…noticed that cylinder oil was still dripping from the liner which should have stopped by that time. It was then realized that the piston was leaking oil. The leakage was only very slight and I judged that the rate was about one drip every ten seconds. No 2 piston was removed reconditioned piston fitted on 04/05/2003 upon disassembly of the piston it was found the crown O'ring was brittle and had failed.

Given the discovery of the leaking piston crown O-ring some three months after the second turbocharger failure, it is probable that the excessive fouling in the scavenge space of this cylinder at the time of the second incident was caused by leaking piston cooling oil.

Figure 8: Number one cylinder liner scavenge ports (scavenge manifold side)

Figure 9: Number two cylinder liner scavenge ports (scavenge manifold side)

Figure 10: Number three cylinder liner scavenge ports (scavenge maniflod side)

Figure 11: Number four cylinder liner scavenge ports (scavenge manifold side)

Figure 12: Number five cylinder liner scavenge ports (scavenge manifold side)

4.7 Analysis of turbocharger overspeed mechanisms

There is no conclusive evidence indicating what caused *Goliath*'s main engine turbocharger to overspeed on two occasions. However many of the possible overspeed mechanisms can be ruled out by the subsequent examinations of the turbocharger and main engine.

When considering the compressor unloading mechanisms, the examinations of the air filter/silencer, inlet casing, air outlet trunking from the turbocharger and charger air cooler revealed no evidence to support the scenarios of a sudden change in the air inlet or outlet conditions to, or from, the compressor assembly. In both incidents, the rotor shafts where the compressor discs were seated did not exhibit any significant scoring or enough deposited disc material to indicate that sustained relative slippage of the compressor discs had occurred. In both failures, the inducer disc remained firmly secured to the rotor shaft to indicate that it had not slipped on the shaft. This, however, does not rule out the possibility that there may have been a partial slip of the compressor or inducer disc on the shaft resulting in misalignment between the two rotating elements.

The loss, or partial loss, of blade(s) from either the compressor impeller or inducer, while unlikely, cannot be ruled out. However, given the close tolerances between these rotating elements and the adjacent air casing means that the loss of a blade or blades would probably result initially in partial rotor lock, further blade loss and disruption of the air casing before an overspeed could occur. The second engineer's statement about the sequence of events before the second failure, i.e. no abnormal turbocharger sounds preceding the rapidly increasing speed, does not appear to support this proposition.

When considering the turbine driving mechanisms, the inspections of the exhaust uptake and economiser did not reveal any evidence of a fire or other abnormality which could have significantly altered the turbocharger exhaust conditions. Similarly the internal inspection of the main engine exhaust manifold revealed no evidence of unburnt fuel, oil or abnormal carbon deposits so it is unlikely that an exhaust manifold fire led to the turbocharger overspeed.

4.7.1 Compressor/inducer misalignment

Neither of the rotor shafts from the turbocharger failures displayed evidence in the form of significant scoring or sufficient deposited material on the compressor disc seats to indicate that a sustained slip of the compressor disc occurred. This, however, does not preclude the possibility that the compressor disc rotated a small distance on its seat as a minor slip of perhaps several millimetres would not have led to significant scoring or the deposition of a large amount of compressor material on the seat. A small slip would have resulted in the misalignment of the compressor and inducer discs and consequently the disruption of the flow of air from the inducer into the compressor. In this way the compressor may have been unloaded which in turn led to the turbocharger over speeding.

The submission from the chief engineer at the time of the second turbocharger failure contained a copy of an ABB after sales service bulletin applicable to VTR

454A-32 series turbochargers. The bulletin, number 5/98, related to turbochargers with two-piece compressor wheels (i.e. separate compressor and inducer wheels), grey cast iron casings and delivered from ABB's Baden/Switzerland works prior to 1997. The service bulletin's introduction stated:

> Field experience has shown that critical operating conditions may occur if the defined values/limits stated in the operation manuals are not strictly observed.
>
> In rare cases, this could lead to irregular situations, for example a compressor wheel burst.

Such critical situations could arise due to:

> …Air intake temperature to turbocharger higher than 35°C (95 F) if not otherwise specified.

The service bulletin then goes on to indicate that where any doubt exists as to whether the prevailing operating conditions could lead to a 'critical situation' owners and operators should consider fitting the turbocharger with 'an external protection cover'. The purpose of the protection cover is to contain any flying debris in the case of a catastrophic turbocharger failure in order to protect engine room personnel and equipment from possible injury/damage.

The publication and content of this service bulletin suggests that there have been a number of turbocharger failures in the past which have resulted in a compressor wheel burst, similar to those on board *Goliath*. One of the apparent preconditions of the failures in the past has been an air intake temperature higher than 35°C.

ABB Turbo Systems were asked to provide information and advice regarding the service bulletin, the past turbocharger failures (if any) which led to its publication, and their research regarding the effect of air intake temperatures higher than 35°C. A submission was received from ABB in response to this request that states:

> Our conclusion is, that the findings in reports No. 186 and 191…are very accurately describing a compressor wheel burst due to overspeed resulting from a scavenge fire.
>
> When issuing the After Sales Service Information we used 35°C as the point, where the customer should think about precautionary measures. By no means this represents the real limits of the turbocharger. There are turbochargers in commercial applications with aluminium wheels which exceed 50°C regularly, without such problems as faced on the Goliath
>
> Running a compressor wheel permanently at a very high intake temperature and high turbocharger speed at the same time, would lead to accelerated aging of the material due to creep in the highly loaded zones (centre hub of the wheel, not clamping hub) and thus reduce exchange intervals.

It should be noted that on vessels operating in temperate or tropical climates the engine room temperature (from which the air is drawn by the turbocharger) would regularly exceed 35°C for extended periods. This makes the implicit ABB recommendation to operate the turbochargers specified in the bulletin within this limit impractical for most vessels.

The assembly of the aluminium alloy compressor disc on the turbocharger rotor shaft is an interference fit which requires the disc to be heated to a predetermined temperature during assembly. When it is heated, the disc expands which allows it to be slid onto its seat on the rotor shaft. After assembly the compressor disc cools on the shaft and the required interference is obtained. The seat for the compressor disc on the rotor shaft is 'microblasted', (a process where the sections of the steel shaft are blasted with grit to give a specified surface 'roughness'), which further increases the friction between the shaft and the disc and allows more torque to be transmitted from the shaft to the compressor disc.

In service the compressor disc's temperature rises. This is due to some degree by heat transferred from the rotor shaft (the turbine end of the shaft is hot as it is heated by the hot exhaust gases) but is mainly a result of heat transfer from the air being accelerated, compressed and thus heated, by the rotation of the compressor disc. As the disc's temperature rises, it expands at a greater rate than the steel rotor shaft and consequently the amount of clamping force due to the interference fit on the shaft becomes progressively less. In service, the compressor disc temperature will reach a point of equilibrium which is dependant on the engine load (which governs the exhaust temperature inlet to the turbocharger, the turbocharger speed, scavenge pressure etc.) and the ambient engine room air temperature (temperature of the air entering the turbocharger).

If the compressor disc becomes too hot in service there is a risk that it will expand enough to allow it rotate on its seat on the rotor shaft resulting in either sustained slip or a slight slip resulting in misalignment with the inducer disc. This effect is likely to be manifested when the turbocharger is operating close to its maximum revolutions (i.e. when the engine load is high), with the maximum torque on the compressor disc, when the ambient engine room air temperature is relatively high. This was addressed by ABB in their submission in that:

> It is a fact, that the clamping seat is the coolest area during the operation. The temperature of the shaft is higher than the temperature of the clamping hub, which is permanently kept cool by the fresh air.

The engine room air temperature is logged by *Goliath*'s engineers each day when they do their morning engine room inspection at about 0800. The engine room air temperature was logged on 21 September 2002, around seventeen hours before the first failure, as 37°C. It is unknown what the engine room temperature was at 0107 when the turbocharger failed but the outside air temperatures were similar at the time of the failure and when the engine room air temperature was previously logged.

The engine room air temperature recorded in the morning before the second failure was 42°C and probably would have increased during the course of the day as the outside air temperature rose. (The deck officers on watch recorded the outside air temperature rising from 22°C during the morning to 26°C just before the failure). It is probable that the engine room air temperatures at the time of each failure were higher than the 35°C stipulated in the ABB service bulletin.

4.7.2 Scavenge fire

A short period of time after the turbocharger was rebuilt in September 2002 (approximately 22 running hours), *Goliath* experienced a fire in the scavenge space of number three unit. The fire led to a scavenge space inspection, cleaning and the replacement of all of the reed valve assemblies between the scavenge manifold and the cylinder scavenge chambers with overhauled spares. The chief engineer at the time noted in his subsequent handover notes (to the incoming chief engineer) that the scavenge spaces were dirty and that he thought that they had not been cleaned at the time of the previous reed valve service. At the time, the scavenge fire was thought to be unrelated to the turbocharger failure on 22 September. However, it is possible when considering this event in the context of the evidence found after the second failure, that a scavenge fire (possibly in the scavenge space of number three unit) preceded the first turbocharger failure. There are no photos or records available about the state of the scavenge spaces at the time of the first failure, only a mention of their condition after the scavenge fire on 1 October 2002.

In submission the chief engineer at the time of the second failure stated:

> I did see some of the reed valve[5] banks removed from the engine following the one documented scavenge fire in 2002. I do not know from which units the reed valves had been removed, but I was amazed to see that quite a few of the individual reeds were physically warped due I assume to heat damage. None of the reed valves removed from the engine following the second turbocharger failure in 2003 displayed any physical warping due to heat. A number had breakages due to fatigue but that is normal.

The build up of residue in the scavenge spaces and on almost all of the main engine cylinder liners found by the chief engineer after the second turbocharger failure was excessive (particularly number two). The residue was also unusually dry in the area of number one and two liners.

The poor condition of the piston crown O-ring, found some three months after the second failure, was the apparent cause of the poor state of number two scavenge space. A leak of piston cooling oil from the defective O-ring would have allowed oil to leak into the scavenge space under the piston. When heated this oil would have formed a gummy residue which would have progressively accumulated on the reed valves and in the liner ports. Over time the accumulation of residue would have impeded the flow of scavenge air to the cylinder and resulted in declining scavenge efficiency and consequently poor combustion in the cylinder. The elevated exhaust temperatures from the cylinder prior to the second turbocharger failure and the build up of carbon residue on the exhaust valve found after the failure are indicative of incomplete combustion.

While there were no scavenge temperature alarms before either turbocharger failure, the scavenge temperature sensors for units one, two, three and four had triggered alarms soon after the second failure. The chief engineer also noticed that the temperature of one, two and three scavenge spaces remained relatively high for some time after the engine had shutdown at 90°C, 95°C and 85°C respectively. While these high scavenge temperatures may be partially attributed to the loss of

5 Reed valve – a set of spring steel plates that acts as a non return valve.

scavenge air when the turbocharger failed (with the engine running for a short period) and the rapid main engine stoppage, the fact that the higher temperatures were centred around number two unit is significant. This evidence, in addition to the occurrence of the suspected scavenge fire following the first turbocharger failure, indicates that it is possible that there was a scavenge fire, probably centred around number two cylinder, which may have led to the second turbocharger overspeed.

While most scavenge fires will not result in a catastrophic turbocharger overspeed it may be possible if the fire leads to a large enough increase in the exhaust gas energy into the turbocharger. In the case of the second turbocharger failure, a fire in the number two scavenge space may have further starved the cylinder's already relatively poor combustion and led to a rapid increase in the amount of energy in the form of unburnt fuel being exhausted from the cylinder. In these circumstances a scavenge fire would also have been accompanied by a significant loss of power from the cylinder.

A loss of power from one cylinder would have caused the *Goliath*'s main engine to slow momentarily which would have been compensated by the main engine governor increasing the fuel pump setting to maintain the engine speed. It is possible that the governor may have settled at a slightly higher speed setting and the variation in engine speed in this fashion may account for the change in engine sound noted by the second engineer immediately prior to the turbocharger's rapid increase in speed. The combination of the unburned fuel leaving number two cylinder and the increased exhaust energy from the other four cylinders may have been sufficient to cause the turbocharger to overspeed. Whether or not there would have been sufficient energy in 14 or so firing strokes (seven seconds) to cause the 'exponentially increasing' turbocharger speed is a matter for some conjecture.

Main engine maintenance

The scavenge space inspection after the fire in number three unit, shortly after the first turbocharger failure, apparently revealed a high level of scavenge fouling. Similarly the condition of the scavenge spaces after the second turbocharger failure was poor around number two cylinder, albeit as a result of piston cooling oil leaking from the defective O-ring. Whether or not the condition of the scavenges led to a fire, which in turn caused the turbocharger failures, cannot be concluded with any certainty however their condition does indicate that the vessel's main engine maintenance regime in this respect may have been deficient.

The ship's main engine maintenance records indicated that the scavenges had been cleaned following the scavenge fire on 1 October 2002, after the first turbocharger failure, 1 518 running hours prior to the second failure. Number three scavenge space and reed valves had been cleaned with the cylinder overhaul 372 running hours prior to the second failure. The usual service period for scavenge cleaning and overhauling the scavenge reed valves is 3 000 hours. While this period is based on the manufacturer's recommendations, the condition of scavenge spaces should be carefully monitored as other factors such as fuel quality, prolonged or frequent periods of low-load running, piston ring and cylinder liner wear can significantly increase the rate of scavenge fouling. In *Goliath*'s case the vessel's trade, the relatively high number of hours since overhaul for some of the pistons, and the findings of past scavenge inspections probably should have led to more frequent and thorough cleaning of the scavenge spaces. The maintenance regime on board every ship must be modified as needed to take into account specific service conditions and any faults which become evident in the condition based monitoring.

The first engineer at the time of the second turbocharger failure submitted that:

> After the second failure, but not because of it, the ship's scavenge maintenance schedule was changed from a 3000 hour service for the entire engine to 3000 hour services done for each cylinder based on each cylinder's running hours. This, coupled with regular 2000 hour piston ring inspections, has improved the maintenance quality of the scavenge spaces, scavenge ports and reed valves.

Goliath's schedule means that there are many short voyages particularly the ones between Melbourne and Devonport. As such, the main engine is started and stopped, warmed up and cooled down, relatively frequently with protracted periods of relatively low-load running in between. This type of operation adversely impacts on the rate at which the scavenge spaces become fouled.

As piston rings wear and loose their tension over time, the amount of combustion product and unburnt cylinder lubrication oil deposited in the scavenge spaces increases. Prior to the first turbocharger failure, number three piston had the highest number of running hours and underwent a 10 000 hour overhaul on 16 November 2002. At the time of the overhaul, the chief engineer noted that the liner's scavenge ports were about 30 per cent blocked after only 1 146 running hours since the previous scavenge clean. The piston crown was also burnt down about 20 mm on one side. The piston cooling temperature for this cylinder dropped by about two degrees when the fuel injectors were exchanged before this cylinder overhaul. This cylinder was overhauled again on 18 January 2003.

Number two cylinder liner's scavenge ports were found to be in a poor state after the second failure after only 1518 running hours. At the time, this piston had the second highest number of hours since its previous overhaul at 7 219 hours. While some of the scavenge fouling may be attributed to the leaking piston crown O-ring found some three months later, any wear in the bore of the cylinder liner and on the piston rings due to their time in service would have exacerbated the problem.

The amount of fouling in the scavenge spaces and in the liner ports has a significant impact on the efficiency of cylinder scavenging. The relatively high exhaust temperatures of number three and number two cylinders in comparison with the other cylinders for similar loads in the weeks prior to the two turbocharger failures was indicative of this.

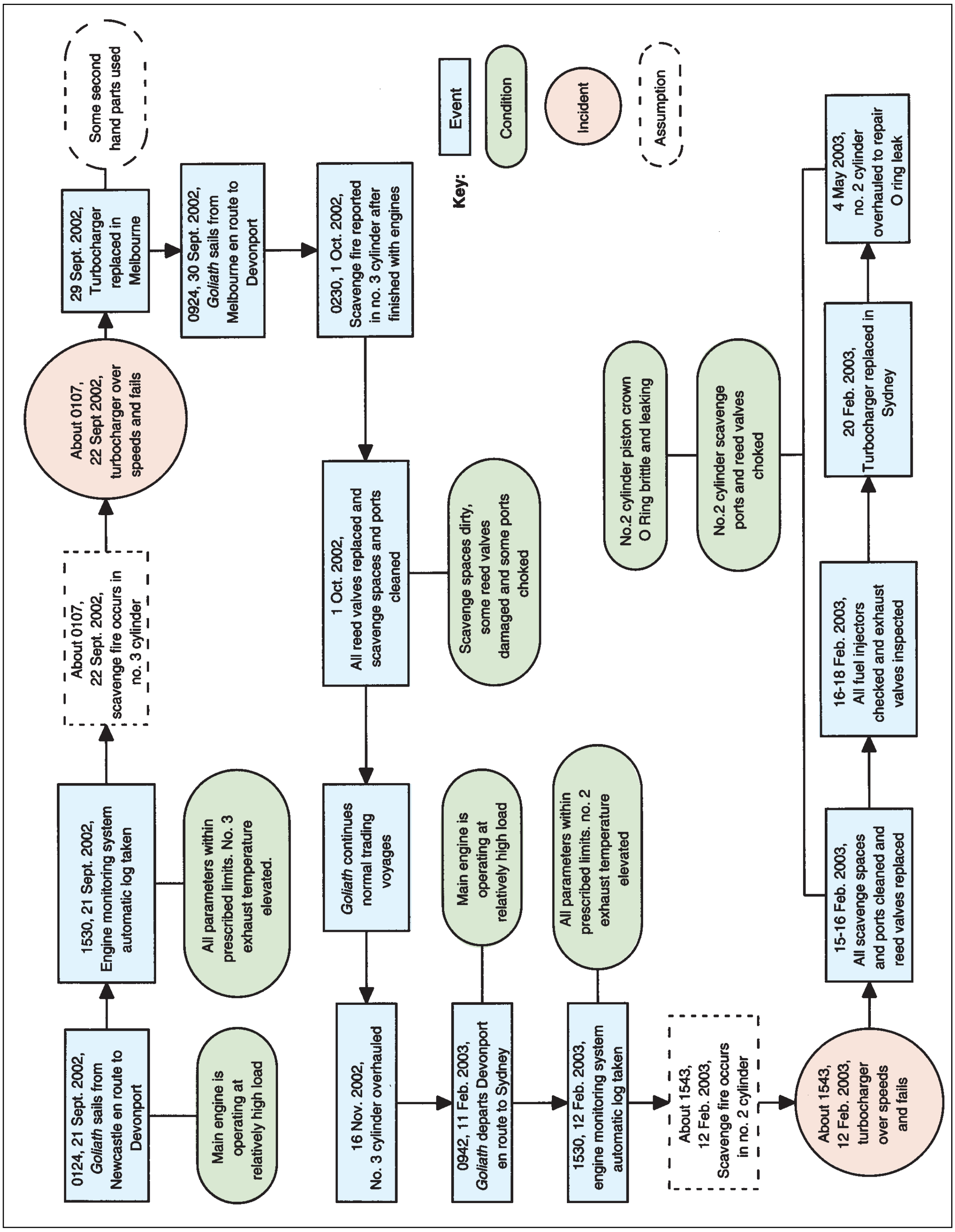
4.7.3 Other incidents

The turbocharger overspeed failures which occurred on *Goliath* are unusual. A search for other similar incidents revealed two recent incidents, one in August 2000 involving the Turkish bulk carrier *Marine Express,* the second in January 2001 involving the Maltese chemical tanker *Aral*. Both incidents occurred in Canadian waters.

The *Marine Express* incident involved the failure of both of the vessel's main engine turbochargers in two separate incidents. After the first turbocharger failure the vessel was proceeding at low speed to port. The failure of the second turbocharger was attributed to overspeed after the main engine experienced a scavenge fire.

Aral's main engine turbocharger failed in an uncontained manner similar to *Goliath*'s. The compressor wheel was found to have disintegrated which resulted in the cast iron turbocharger casings exploding. Pieces of the casings penetrated and ruptured piping and the control room doors and windows in a 180 degree arc around the forward end of the turbocharger. Investigation of the *Aral* incident did not reveal a definitive cause although the turbocharger was not fitted with a tachometer and it was suspected that it had been operated for a long period of time in an overspeed condition. There was no indication that there had been a scavenge or uptake fire which had resulted in a transient overspeed.

Figure 13: Events and causal factors chart



5 CONCLUSIONS

These conclusions identify the different factors contributing to the incident and should not be read as apportioning blame or liability to any particular organisation or individual.

The following conclusions are made with respect to the turbocharger failures aboard *Goliath* on 22 September 2002 and 12 February 2003:

1. Both turbocharger failures on board the *Goliath* were very comparable in terms of the extent of damage sustained.
2. Both turbocharger failures were attributable to the radial fracture of the centrifugal compressor disc/impeller.
3. The fracture characteristics of both compressor disc failures were similar and typical of material failure under centrifugal overload conditions.
4. No evidence of prior cracking or other defects was found within the examinable fractures of both compressor discs.
5. The metallurgical and physical properties of the first failed disc were considered satisfactory and typical of the material used to produce the disc.
6. Several crew members of the *Goliath* witnessed the turbocharger undergo an uncontrolled transient overspeed event immediately before the second failure.
7. The similarity of the compressor disc failures supports the conclusion that both turbochargers underwent an uncontrolled transient overspeed event similar to the second failure.
8. While it is not possible to state with certainty, the two possible mechanisms which led to the turbocharger failure were a slight slip of the compressor disc or a scavenge fire but was more likely to have been a scavenge fire based on the evidence.
9. The engine room temperature at the time of each failure was probably higher than the maximum stipulated in ABB Turbo Systems after sales service bulletin 5/98 applicable to VTR 454A-32 series turbochargers.
10. The condition of some of the scavenge spaces at the time of each failure was poor.
11. The vessel's maintenance regime prior to each turbocharger failure was probably inadequate with respect to scavenge space inspection and cleaning.

6 RECOMMENDATIONS

MR20060027

Shipowners, managers and ship's engineers should ensure that the maintenance regime applied to slow speed diesel engine scavenge spaces is thorough and takes into account the engine's history, condition and conditions of service.

MR20060028

It is recommended that ABB Turbo Systems release a further service bulletin pursuant to service bulletin 5/98 detailing instances of turbocharger overspeeds with appropriate warnings and guidance for the safe operation of the affected turbochargers.

7 SUBMISSIONS

Under sub-regulation 16(3) of the Navigation (Marine Casualty) Regulations, if a report, or part of a report, relates to a person's affairs to a material extent, the Inspector must, if it is reasonable to do so, give that person a copy of the report or the relevant part of the report. Sub-regulation 16(4) provides that such a person may provide written comments or information relating to the report.

The final draft of the report was sent to the master, both chief engineers, the second engineer at the time of the second incident, CSR shipping, ABB Turbo Systems, HSD (Doosan) Engine Company and MTQ Engine Systems (Australia).

Submissions were received from CSR Shipping, the chief engineer at the time of the second failure, MTQ Engine Systems, Doosan Engine Company and ABB Turbo Systems limited.

At the time of publication, the turbocharger failures on board *Goliath* were the subject of on-going civil litigation in the Supreme Court of Queensland between the supplier of the turbocharger assembly fitted after the first failure, MTQ Engine Systems, and CSR Shipping. Both companies commissioned extensive technical reports in 2004 and 2005 on the turbocharger failures and submitted these reports as a part of their respective submissions.

7.1 Doosan Engine Company

The Doosan Engine Company submitted the following:

> With regard to the investigation report No.s 186 and 191 for the turbocharger failure on MV Goliath, we would like to inform you that it is so difficult for us to assume the possible reason with some pictures and measured data.
>
> But basically, we don't have any objection on your investigation result.
>
> Based on the status of damages in the pictures and measured data in your report, we can assume the turbocharger failure mechanisms below.
>
> – It is assumed that the initial cause of the failure is extra amount of carbon residues in scavenge air receiver which blocked scavenge air ports.
>
> – The air amount was decreased and gas temperature was increased due to blocking of scavenge air ports.
>
> – Higher gas temperature increased the turbine power.
>
> – The compressor wheel was overloaded because of high turbine power and high air outlet temperature.
>
> – The compressor wheel was operated under the overload condition during long period.
>
> – A sudden engine load change led to the unloaded condition of compressor wheel and it resulted in a transient over speed condition.

Considering the low turbocharger RPM and high temperature energy in your report, the overloaded condition of the compressor wheel can be easily concluded.

Therefore, the compressor wheel was weakened owing to the overloaded operation during long period and the turbocharger over speed can be concluded as the main cause of the turbocharger failure.

7.2 ABB turbo systems

In submission ABB Turbo Systems stated:

Thank you for sending us a copy of your investigation report covering your findings after the 2 failures on bulk cargo carrier MV Goliath.

We share your conclusion that the most likely cause of the major breakdown was caused by a severe over speed of the turbocharger rotor. The observation by the crew just prior to 2nd breakdown and the later findings on the engine ultimately confirm your conclusion. In this sense we acknowledge your report and congratulate you for finding the initial cause.

7.3 CSR shipping

In 2003, the solicitors acting for CSR shipping commissioned BMT Murray Fenton (marine consultants) to investigate the turbocharger failures on board *Goliath* and prepare a report as part of their submission to the Supreme Court of Queensland. Three reports were ultimately prepared as a result of this investigation, "*Goliath*" 8719, "*Goliath*" Supplementary Report 8719A and "MV *Goliath* – failures of main engine turbochargers", and were submitted as evidence to the court on 10 December 2004. CSR Shipping tendered these reports as their submission to the ATSB.

The supplementary report (8719A) deals with the specifications of the turbocharger supplied by MTQ Engine Systems following the second failure on 12 February 2003 and consequently has no relevance to the two turbocharger failures investigated by the ATSB. The report "*Goliath*" 8719 contains in Appendix 1 a report by ABB Turbo Systems on the turbocharger which failed on 12 February 2003 (this report was not provided to the ATSB during the investigation). Relevant conclusions from all of the reports are reproduced below.

7.3.1 ABB report

ABB Turbo Systems (Switzerland) undertook an investigation into the turbocharger which failed on 12 February 2003 including macroscopy, fractography and metallography on the compressor wheel fragments and some detailed inspections of the inducer wheel and turbine shaft. In their report, finalised in July 2003, they concluded that:

Shaft and Inducer wheel

Roughness and diameter of the microblasted area of the compressor/inducer wheel seat are acceptable for an older shaft. The stamped specification does not represent the design feature of the failed investigated shaft (not an existing ABB specification), also the actual specification stamped on the inducer wheel does not correspond to the design features determined by dimension measurements.

The inner diameter of clamping hub is above our max. limit for new inducer wheels. The shrinkage of shaft and inducer wheel is below our minimum limit, which result in a reduced torque transmission from shaft to inducer wheel.

Due to the above findings we conclude that the first turbocharger breakdown occurred in Sept. 2002 was repaired with second hand parts.

Material investigation on compressor wheel segments

The failure investigation revealed crack starting inside the compressor wheel bore at the clamping hub seat. No indications of LCF or creep failure mode were detected on investigated parts. The fracture mode was identified as ductile forced rupture. The crack initiation mode at the clamping hub is unknown. The crack initiation must result form additional induced high stress accumulation in this area or due to insufficient material strength (which, however, could not be proven by the samples taken and investigated) of the compressor wheel. The reason for high stressors could be:

- Surface flaws (machining, handling or assembly)
- Dimensional error (shrink fit)
- Internal additional loading (excessive shaft motion, bearing failure)
- External additional loading (engine vibration, excitation of compressor blades and wheel)
- Abnormal operation (excessive speed, excessive surging, excessive torque or moment of momentum)

The low amount of Cu, Mg and Si found in the investigated material is responsible for a lower amount of hardening phase, which reduces the strength of the compressor wheel. The low electrical conductivity indicate an insufficient heat treatment during the manufacturing process. Nevertheless, the tensile tests still showed satisfactory results. The material properties of the investigated compressor wheel point to a non ABB manufactured spare part. We have no experience with the material properties of spare parts not supplied by ABB.

Considering the extent of damage on the shaft (missing turbine blades) the turbocharger clearly failed due to overspeed. The reason for this overspeed can not be determined with this investigation.

7.3.2 MV *Goliath* – failures of main engine turbochargers report

BMT Murray Fenton's investigation included the commission of detailed failure analysis and metallurgical work which was performed by materials engineer and failure analyst Dr D.R.H Jones of Cambridge University. The results of this analysis were detailed in the report titled '*MV Goliath – failures of main engine turbochargers*'. Dr Jones was instructed to investigate both turbocharger failures which occurred on *Goliath* (22 September 2002 and 12 February 2003) and 'express an opinion as to the most likely sequence of events leading up to the failures'. In addition to being provided with a number of documents, references and access to some of the parts from the failed turbochargers, Dr Jones viewed a video recording made by ABB of an overspeed test to destruction on a similar turbocharger. In the opinion of Dr Jones, 'on the balance of probabilities, the most likely failure sequence is as follows.'

1) Regarding the second failure, slip occurred between the compressor wheel clamping hub and the mating shaft, caused by inadequate assembly interference (and possibly also inadequate creep properties). As a result, the drag on the rotor shaft was decreased, allowing the turbocharger to reach a speed at which the compressor wheel failed by multiple ductile tensile fracture. Once the compressor wheel had burst, the inducer wheel exerted little drag on the rotor shaft, allowing the turbine to speed up to a point at which it disintegrated as well.

2) Regarding the first failure, because the relevant pieces of clamping hub are not available for examination, I have no direct evidence that slip occurred between the compressor wheel clamping hub and the mating shaft. Consequently, I am not able to construct a failure sequence with any degree of certainty.

7.3.3 '*Goliath*' 8719 report

BMT Murray Fenton was instructed to ascertain the probable cause of the second turbocharger failure on *Goliath* on 12 February 2003 by examining reports, crew statements, log extracts and photographs. A representative from BMT Murray Fenton also attended the ABB Turbo Systems inspection of the failed components. Whilst the comments are specifically relating to the 12 February 2003 failure, additional contributions are made with reference to the two previous turbocharger failures which occurred on 22 September 2002 and 2 July 2001 (a turbine blade failure). The conclusions from the report '*Goliath*' 8719 were:

> The following conclusions are made concerning the turbocharger failure on 12th February 2003.
>
> 11.1 The impeller wheel was the turbocharger component to have initially failed in service resulting in momentary loss of compressor load, turbocharger overspeed, impeller bursting and catastrophic damage sustained by the turbocharger. There is also evidence to suggest that slippage of the interference fit between the inducer wheel and rotor shaft occurred at an undetermined time.
>
> 11.2 The investigation carried out by ABB revealed that a crack had started at the impeller bore at the clamping hub. Thus, feasibility of fatigue cracking having occurred in the area sufficiently to result in a loss of drive to the impeller wheel cannot be discounted. Similarly, inadequate shrinkage allowance of the impeller wheel during the fitting to the rotor shaft would have allowed rotation on the shaft, during service, placing both impeller and inducer vanes out of alignment. The inducer wheel could no longer effectively induce air into the impeller wheel efficiently resulting in a change of compressor characteristics and a loss of compressor loading, and speeding of the rotor assembly to the point of bursting the impeller.

11.3 The turbocharger type installed on "GOLIATH" is designed with the engine matching and loading to achieve, for a new impeller, a service life of 100,000 hours before renewal. The impeller installed on September 2002 by MTQ failed on 12th February 2003 after 1540 hours service. The original fitted impeller was in service for 9 years without any reported operational problem before failing after running for 36482 hours.

11.4 The documents provided for examination do not evidence that the main engine overspeeded prior to the turbocharger overspeed therefore it is unlikely that causation of failure was borne from either defective engine components or engine system failures. The alarm history display data prior to the failure time does not support the theory that an engine problem occurred to cause the engine overspeed. The main engine was operating at approximately 87% of full load at 113 RPM when the turbocharger failed. Any significant change in engine speed would have been controlled by the engine governor.

11.5 Identification markings overstamped on the inducer wheel suggest that the inducer, and probably the impeller, were not new when fitted to the turbocharger in September 2002 or alternatively the parts were not originally supplied for a turbocharger with a specification the same as installed on "GOLIATH". It is unlikely that genuine new impeller and inducer wheels, if supplied to Kemklen Technical Services, allegedly by an ABB Licensee, would warrant further non destructive testing to be carried out by them on 24th September 2002 to establish hardness values and dimensional checks before assembling onto the rotor shaft. If a complete rotor assembly was supplied to Kemklen as the certification would suggest, it would not be a normal procedure or necessary to remove the inducer and impeller, carry out dimensional and hardness checks, rebuild the rotor and check balance prior to supply to MTQ. It is probable that only a bladed rotor shaft without compressor wheels was supplied by the ABB Licensee to Kemklen. The complete rotor assembly was then built up using a hybrid source of turbine blades to satisfy the "GOLIATH" blade specification, compressor wheels and partition wall.

11.6 The severity of the failure, produced component debris which were not contained within the structures of the gas outlet casing, presented a clear safety risk to the engine room crew. The gas outlet casing should be of material specification GGG40, which was introduced in 1997 to supersede the previous material GG20, and is required to withstand containment testing of up to 125% maximum loading. It would be prudent to establish the material specification of the gas outlet casing currently fitted and in service since February 2003. The certification of supply received from Clayton UTZ does not specify the casing material.

11.7 The failure investigation carried out by ABB Switzerland, revealed a crack starting inside the impeller bore at the clamping hub seat. ABB state that the crack must have resulted from "additional induced high stress accumulation in this area". Five possibilities for additional stress were considered, three were engine borne and two of which were related to the supply, handling and assembly of the impeller by MTQ. Of the three reasons for increased levels being primarily engine borne, there are no documents which I have examined that suggest that such occurrences such as turbocharger surging, excessive engine speed, torque or engine vibration has been experienced during the five months of engine operation between the turbocharger repair in September 2002 and the failure in February 2003, to cause increased stress on the impeller. In my opinion, based upon the evidence provided in the engine log documentation, it is unlikely that causation of the failure was borne from either engine components or engine system failures.

11.8 The two reasons for additional stress on the impeller which are related to the turbocharger parts and service supply includes the handling and assembly during repair which could have caused an increase in stress onto the clamping hub of the impeller. Incorrect assembly of the impeller, with uneven gaps between the inducer and impeller, or alternatively the impeller not being fitted correctly against the shaft shoulder, due to excessive bore clearance could have resulted in an increase in stress on the clamping hub, which is one of the most critical points on the impeller wheel. An incorrect shrink fit allowance between the impeller bore and rotor shaft would have increased hoop stress at the impeller bore if it was too high, or alternatively, if the shrinkage allowance was low, as was the case with the inducer wheel, a reduced torque capability could have resulted in radial slippage of the impeller on the shaft. The actual shrinkage allowance was unable to be determined due to the fractured impeller bore and thus any claim that the shrinkage fit of the impeller was incorrect cannot be evidenced. However, slip should not occur between the impeller and shaft with genuine ABB components which have been assembled according to ABB's limits and clearances.

11.9 The MTQ damage report dated 8th April 2003, stated that during the rotor build prior to fitting in September 2002 the recorded maximum temperatures obtained whilst fitting the impeller and inducer wheels in the factory were 118°C and 112°C respectively. Examination of Dr Jones' report, paragraph 4.2, calculates that at such temperatures obtained during fitting the ABB stated assembly interference fit of 0.30mm would not have been achieved. It is possible that the assembly temperatures were recorded incorrectly or that they were not measured at all during assembly.

11.10 Approximately one third of the impeller wheel was missing from the damaged parts recovered from the turbocharger failure on 12th February 2003 and therefore was not available for metallurgical examination. It is difficult, therefore to evidence or determine if the used impeller wheel fitted by MTQ during the turbocharger repair in September 2002 had a latent (or patent) critical defect at the clamping hub area. However, failure after 1,540 hours service compared to an expected service life range of 50,000 to 100,000 hours depending on the engine's load profile suggests that if the main engine has not been operated in a manner that could have contributed to the additional stress levels encountered by the impeller wheel to manifest in failure, then in the absence of any other indications, on balance of probabilities, either a defect was inherent at the impeller clamping hub area or that improper fitting/alignment of the compressor wheel caused increased stress or slippage at one of the most critical areas, resulting in failure at the impeller bore at the clamping hub seat.

7.4 MTQ Engine Systems

In 2005 the solicitor representing MTQ engine systems, the supplier of the turbochargers fitted after both failures, commissioned WBM Pty Ltd to examine various aspects of the two turbocharger failures on *Goliath*. They were instructed to prepare a report for submission to the Supreme Court of Queensland. WBM were provided with various documents relating to the failures as well as some of the components from the failures which were subjected to another detailed metallurgical examination by UQ Materials Performance of the University of Queensland. On 17 November 2005, the WBM report titled *Investigation of Aspects of Failures of Turbochargers on the MV Goliath (Matter of MTQ Engine Systems and CSR Ltd)* and the associated UQ Materials Performance report titled *Metallurgical Aspects of Turbocharger Failures* were provided to the ATSB as the MTQ Engine Systems submission.

7.4.1 Metallurgical Aspects of Turbocharger Failures report

UQ Materials Performance undertook a metallurgical examination of the failed components of the turbochargers from both incidents (22 September 2002 is referred to as TC1 and 12 February 2003 is referred to as TC2). Their report *Metallurgical Aspects of Turbocharger Failures* concluded that:

> **Circumferential Sliding**
>
> I do not believe there is any strong evidence for the compressor wheel from the second failure (February 2003) having experienced circumferential sliding on the shaft. The three pieces of evidence cited by Dr Jones are better explained in terms of other phenomena, as follows:
>
> - Softening of the clamping hub is better explained by overheating during installation;
> - Signs of melting and re-solidification are better explained by post failure deposition of molten material generated by rubbing of the vanes on the housing;
> - Circumferential striations on the clamping hub bore appear to have been present prior to installation.
>
> **Differences in Condition of Components from the Two Failures**
>
> I do not believe there is any evidence of differences in the condition of the compressor wheels between TC1 and TC2 that would be considered significant with regard to the cause of failure of TC2-C.
>
> - The bulk hardness values of the two compressor wheels are not significantly different.
> - The clamping hub of the compressor wheel from TC2 is softened by comparison with the compressor wheel body, but we do not know whether

similar softening occurred in TC1 because no fragments of the compressor clamping hub are available from that failure. The majority of the clamping hub from TC2 remains within the reported ABB acceptable hardness range, although the end-face region falls below this acceptable range.

- There is some moderate softening around the guide bore region at the TE end of the compressor wheel in TC1, but no comparable softening in TC2. I am unable to discern any significance or importance in this minor difference.
- The compressor clamping hub bore surface from TC2 shows a crust of material consisting of re-solidified aluminium, aluminium oxides, iron oxides and iron-aluminium intermetallic compounds. However, a crust matching this description is also present on fracture surfaces, both in TC2 and TC1. Therefore, there is nothing unique to TC2 in this crust, and it is evidently consequential to the failure.

The only significant difference that I have noted between the inducer wheels of TC1 and TC2 is that the TE face of the inducer wheel of TC1 shows deep circumferential scoring whereas there is no such damage on the TE face of the inducer of TC2.

All four of the compressor and inducer seating areas on the shafts from the two failures show evidence of transfer of aluminium from the mating component. The compressor seat from TC2 does not show more aluminium than the other three seating areas examined.

The only significant difference that I have noted between the compressor seats from TC1 and TC2 is that the compressor seat from TC1 shows some evidence of circumferential sliding on the compressor shaft whereas that from TC2 does not show such evidence.

Differences in Material Composition

It is difficult to see much significance in the minor differences that exist in material chemical composition between the compressor wheels from the two failures (TC1-C and TC2-C).

Contrary to the statement in the report of Brem and Marty that TC2-C is out of specification in four of the seven specified elements, the current work indicates that it is well within the standard specification for grade 2618 in six out of seven elements. It is however out of specification in respect of Zn as reported by Brem and Marty.

I do not know whether the deviation from specification in respect of Zn for TC2-C should be regarded as evidence that the component is not of genuine ABB or ABB-licensee manufacture.

7.4.2 Aspects of Failures of Turbochargers on the MV Goliath (Matter of MTQ Engine Systems and CSR Ltd) conclusions

The conclusions of the WBM report were as follows:

1. The catastrophic failures of two main engine turbochargers on the M.V Goliath which occurred on 22 September 2002 and 12 February 2003 were immediately caused by overspeeding the turbocharger.

2. In each case the major component which was the first to fail was the compressor wheel and in each case it failed through bursting, or forced fracture under the influence of centrifugal loads associated with the high rotating speed. In neither case is there any indication that fatigue was a causative factor for the failure of the compressor wheel.

3. Macroscopic visual examination of the hub of the compressor wheel from the second failure and of the mating shaft seat from both failures, shows no sign of sustained, high speed circumferential slip of the wheel hub on the shaft.

4. In the case of the clamping hub of the compressor wheel on the shaft from the second failure, microscopic examination by Dr Gates also shows no sign of sustained, high speed circumferential slip of the wheel hub on the shaft. Dr Jones comes to a different view, but based only on observations under the Scanning Electron Microscope of indications which are not duplicated or visible under lower magnifications.

5. In my opinion, slippage of the compressor wheel on the shaft at relative speeds of several thousand revolutions per minute, and over at least two seconds, would leave significant macroscopic signs on the aluminium wheel hub and possibly on the shaft, such as circumferential scores or marks, galling or deposits of molten metal.

6. In both cases the failure modes of the turbochargers, and their compressor wheels, are extremely similar. The damage caused and even the pattern of fracture in the compressor wheels shows no significant difference between the two failures. The report of Dr Gates confirms that there is no significant difference between the materials of the compressor wheels in the two cases, or the materials of the inducer wheels in the two cases.

7. For these reasons I believe that the two failures have a common cause. I believe it is highly unlikely that the second failure has a completely different cause from the first.

8. Overspeed of the turbocharger rotor must result from a significant energy imbalance over the machine. This could occur either through major unloading of the compressor, or from significant addition of energy to the exhaust gas which drives the turbine. Unloading of the compressor could occur either through blockage of the inlet or outlet air, or through significant slip of the compressor wheel on the shaft. Addition of energy to the exhaust gas could occur through the combustion of leaked fuel or lubricating oil added to the exhaust gas passages, the combustion of carbon deposits in the exhaust passages, or the passage through the engine of unburnt fuel which subsequently was combusted in the exhaust gas passages.

9. Blockage of the compressor inlet or outlet would cause significant disruption to the operation of the main engine, because of lack of inlet air. It is extremely unlikely that this occurred in either case. Slippage of the compressor wheel on the shaft could cause unloading of the turbocharger and hence turbocharger overspeed, but it is extremely unlikely that such significant slippage at the connection between wheel hub and shaft could simultaneously support the transmission of sufficient torque to accelerate the compressor wheel to

its bursting speed. As stated at 5 above, I do not believe that the physical condition of the compressor wheel hub from the second failure, or the shaft from either failure, is consistent with significant slippage. Therefore it is extremely unlikely that unloading of the compressor was the cause of overspeed in either failure.

10. There is evidence in the case of the second failure that the scavenge ports and reed valves to the No 2 cylinder were significantly blocked. This could be expected to cause elevation of the exhaust gas temperature from that cylinder, because of decreased air mass flow to that cylinder. This is reflected in the elevation of that cylinder's exhaust gas temperature above the others, when the engine was under load, in the period leading up to the second failure.

11. The logs of engine parameters and alarms indicate that immediately prior to the first failure elevated temperatures and scavenge fire alarms indicated the possibility of a scavenge fire. In the absence of evidence indicating damage to or malfunction of these instruments, it is reasonable to accept these indications at face value. Shortly after the first failure a scavenge fire was suspected to have occurred, and maintenance of the scavenge ports and reed valves was undertaken. Following the second failure, similar maintenance was undertaken, particularly on the No 2 cylinder.

12. A scavenge fire could cause depletion of oxygen in the inlet air to the affected cylinder, and in turn cause incomplete combustion of the fuel injected into the affected cylinder. This unburnt fuel would then pass into the exhaust passages and mix with excess oxygen in the exhaust gases from the other cylinders. Combustion of this fuel would then be possible. The energy from even a small amount of unburnt fuel when added to the exhaust gas flow entering the turbocharger would be sufficient to cause a significant energy imbalance, leading to uncontrolled acceleration of the turbocharger rotor.

13. In the case of the turbocharger runaway caused by addition of energy to the exhaust gas stream feeding the turbocharger, the compressor would feed more air than usual to the engine inlet. This would allow the engine to continue running, and would increase the effectiveness of the engine and its combustion processes, causing acceleration of the engine. However, the governor would act to reduce the fuel input and stabilise main engine speed. This is consistent with observations at the time of the second incident.

14. In my opinion the most likely cause of both turbocharger overspeeds was addition of energy to the exhaust gas stream feeding the turbocharger. The most likely source of this energy would be a scavenge fire.

7.5 Comments on submissions

All of the submissions received by the ATSB investigation were carefully considered before this investigation report was finalised. The various interested parties agreed that the turbocharger compressor discs failed due to overspeed and that there are no significant differences between the two failures. There is some difference of opinion in the submissions about the events that led to the second turbocharger overspeed.

ABB Turbo Systems, and WBM agree with the ATSB's investigation conclusion that a scavenge fire was the most probable event which resulted in a significant energy imbalance in the turbocharger that led to the overspeed failures. The Doosan engine company also has no objection to this finding.

The Cambridge University report suggests that the compressor wheel in the second failure slipped on the shaft causing the turbocharger to unload resulting in an overspeed. This conclusion is not borne out by any macroscopic examination of the compressor parts and it does not explain the first failure. BMT Murray Fenton also does not discount slippage of the impeller on the shaft in the second failure but they also do not explain the cause of the first failure. ABB Turbo Systems (Switzerland) also contends that non-genuine and possibly second-hand parts were used in the repair after the first failure although this does not explain the first failure either.

All parties agree that the turbochargers failed due to an overspeed failure of the compressor disc. The evidence from both failures suggests that the two failures had a similar event sequence. While it is not possible to state with any certainty, there are two possible mechanisms which could have led to the turbocharger failure, a slight slip of the compressor disc on the shaft due to temperature or a scavenge fire. Based on the evidence, the most likely scenario leading to the both failures was a scavenge fire.

8 *GOLIATH*

IMO Number	9036430
Flag	Australian
Port of Registry	Devonport
Classification society	Lloyds Register of Shipping (LR)
Ship Type	Bulk cement carrier (self discharging)
Builder	Hanjin Heavy Industries Company, Ulsan, Korea
Year built	1993
Owners	Goliath Portland Cement
Ship managers	CSR Shipping, Sydney (at the time of the incidents)
Gross tonnage	11 754
Deadweight (summer)	15 539 tonnes
Summer draught	8.335 m
Length overall	143.00 m
Moulded breadth	23.50 m
Moulded depth	11.90 m
Engine	KHIC Sulzer 5RTA52
Total power	6 400 kW
Service speed	14.5 knots
Crew	18 Australians

Report No. 21/03
Task No. BE/200200016

Examination of Main Engine Turbocharger Failures

MV Goliath

22 September 2002 and 12 February 2003

CONTENTS

EXECUTIVE SUMMARY

Following the in-service destructive failure of the main engine turbocharger of the MV *Goliath*, the Australian Transport Safety Bureau Technical Analysis Unit was tasked to investigate the failure with a view to determining the probable cause of the event. During the course of the investigation, the *Goliath* sustained a second turbocharger failure and in view of the similarity of the two failures, the investigation was extended to encompass both events.

Both turbocharger failures resulted in the destruction of the entire mechanical assembly and the explosive liberation of debris within the confines of the ship's engine room. As a result of the turbocharger failures, the main engine was not capable of continued operation and the ship was disabled.

An analysis of the evidence gathered from the physical examination of both failures concluded that both had resulted from the centrifugal overload and rupture of the main compressor impeller disks. The liberated, high energy disk segments impacted and disrupted the compressor housing, destroying the bearings and displacing the main shaft which led to the fracture and release of the turbine blades from interference with the turbine housing. An apparent absence of contributory physical or metallurgical deficiencies within the compressor disks led to the further conclusion that the most probable cause of the disk ruptures was a transient, high-level over-speed event, stressing the disk beyond the physical strength of it's parent alloy. That conclusion was supported by witnesses to the second failure, who reported hearing the turbocharger accelerate rapidly immediately before the event.

NR Blyth

Senior Transport Safety Investigator
Technical Analysis

1 FACTUAL INFORMATION

1.1 Introduction

On 22 September 2002 and again on 12 February 2003, the Australian merchant vessel *Goliath* sustained the catastrophic, explosive failure of the main engine turbocharger unit. The turbocharger failure prevented the continued operation of the main engine, thus disabling the vessel and necessitating the taking of a tow to return the vessel to port.

Following the first failure, the Australian Transport Safety Bureau commenced an investigation into the circumstances surrounding the incident, in view of the potential threat of injury to crew and the loss of control of the vessel. That investigation was subsequently extended to include the second failure, given the apparent similarities between the two events.

1.2 Examination brief

The ATSB Technical Analysis unit was tasked to undertake an investigation of the engineering and metallurgical aspects of both turbocharger failures. To that end, an inspection was made of the wreckage of the first unit at the premises of Dynamic Turbocharger Services (Australia) Pty Ltd, Melbourne. The wreckage of the second turbocharger was inspected in-situ aboard the *Goliath*. Information relating to the operation and behaviour of both turbocharger's and the ship's main engine was obtained during both inspections.

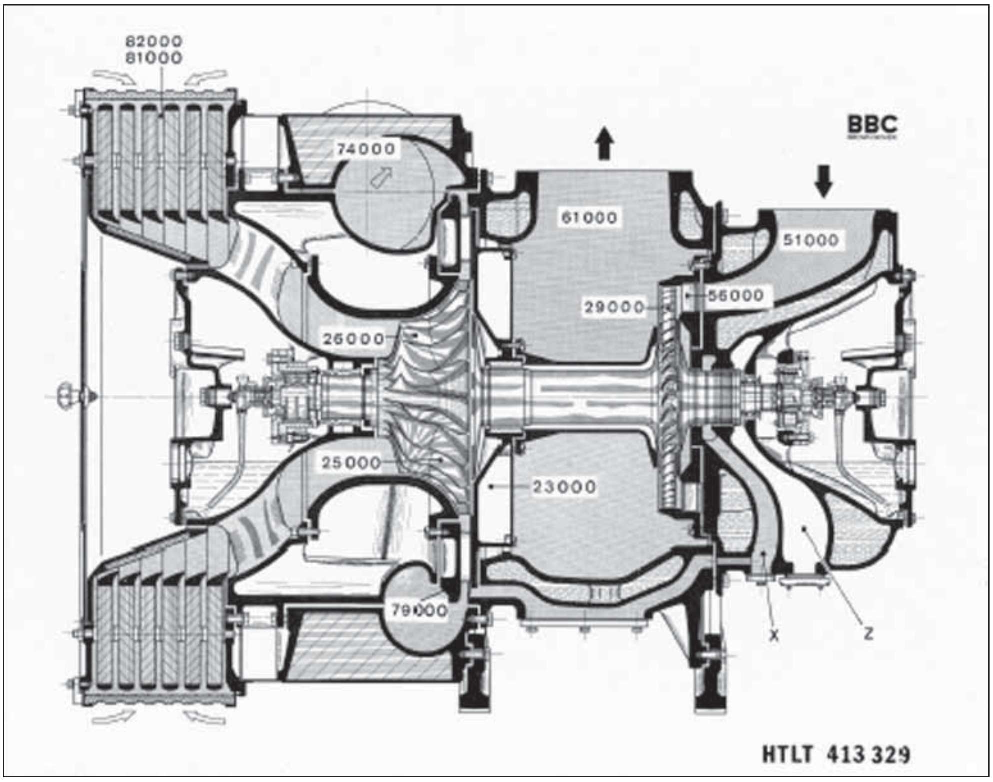
1.3 Failure one, 22 September 2002

The failed turbocharger unit was identified as an ABB Turbo Systems Ltd model VTR 454A-32, manufactured under licence by Ishikawajima-Harima Heavy Industries Company Ltd. The unit was a conventional design with an axial, single-stage exhaust driven turbine coupled to a single-stage centrifugal compressor (fig. 1). Information received indicated the turbine stage had operated for approximately 5,600 hours since overhaul and installation of new blades. The compressor rotor was reported to have accumulated approximately 36,500 operational hours from new, with the nominal replacement age for that component being 50,000 hours[1]. The duty of the turbocharger was described as being 'high cycle', with a vessel history of short voyages, corresponding with short main engine running intervals. At the time of the failure, the engine room was unoccupied.

1.3.1 Visual examination

An inspection of the turbocharger wreckage recovered from the *Goliath* was made in a progressive manner, examining each primary section for evidence of the failure mechanism/s.

Figure 1: Diagrammatical view of the VTR 454 turbocharger type



1.3.1.1 Turbine housing & nozzle ring

The turbine section of the turbocharger comprised two adjoining cast-iron housings. The engine exhaust gases were ducted into a housing located at the end of the assembly (fig. 2) and through a full circumference nozzle assembly into the rotor case. The gas inlet casing and nozzle ring, while having sustained significant impact damage (fig. 3), had not liberated any components or cracked/fractured in a manner that may have been a precursor to the failure. The remnants of the turbine rotor shroud assembly that surrounded the rotor within the gas outlet casing (fig. 4) had sustained multiple and extensive impacts from liberated turbine blade fragments.

Figure 2: Exhaust inlet casing and turbine nozzle ring

Figure 3: Heavy impact damage to the otherwise intact turbine nozzle ring

Figure 4: **Remanents of the turbine shroud assembly**

1.3.1.2 Turbine rotor

The turbine rotor comprised an integral shaft-disk (fig. 5), with a set of 45 blade elements installed within conventional fir-tree slots. All bar one of the turbine blades had fractured transversely from the base of the airfoil section (fig. 6), with that single blade fracturing from the top of the root 'fir-tree' (fig. 7). An inspection of the fracture morphology showed a similar semi-crystalline appearance across all blades, including the blade that fractured from the fir-tree root. Most of the blade set showed transverse rubbing contact against the nozzle-facing platform edges (fig. 7) around the full rotor circumference. A selection of five turbine blades was subsequently removed from the rotor disk for laboratory examination. Those included the fir-tree root failed blade and two additional blades from either side.

Figure 5: **Turbocharger shaft with (left to right) inducer, missing impeller disc, partition and turbine disc**

Figure 6: 'Cobbed' turbine rotor with all blades fractured

Figure 7: Left shingle turbine blade fractured at the top of the fir-tree root

1.3.1.3 Turbine rotor Shaft bearings

The turbine-end shaft bearing was a straight roller type and had sustained the destruction of the inner race with clear brinelling (indentation) of the race surface by the rolling elements. The compressor-end bearing assembly comprised a dual ball bearing set, with the outer bearing carrying the shaft axial thrust loads. The compressor end bearings were in comparatively good condition, with both units free to rotate smoothly by hand. Neither of the shaft bearing units showed any evidence of operational distress or premature breakdown, nor was there any evidence of lubricant deprivation or overheating.

1.3.1.4 Compressor rotor

The compressor rotor was made up of a forward inducer coupled to the main centrifugal impeller. While the inducer section had remained on the turbocharger shaft and was essentially intact (fig. 8), the impeller had been liberated and was recovered in multiple fragments. The impeller had fractured radially into at least four comparably sized segments (fig. 9), with several smaller pieces also recovered. A section comprising approximately one-quarter of the disk was not recovered. The surfaces of the shaft from where the impeller disk was located were studied and showed no evidence marks or scoring that may have suggested the insecurity or rotation of the impeller on the shaft. The matching impeller face did not display scoring or damage of the type shown by the mating inducer face (fig. 10). All of the rotor fragments were subsequently retrieved and examined in the ATSB's Canberra laboratories.

Figure 8: Compressor end of the turbocharger shaft. Missing the impeller disc

Figure 9: Recovered segements and fragmnets of the impeller disc

Figure 10: Rear face of the inducer element showing circumferential scoring consistent with continued rotation after the failure of the impeller disc

1.3.1.5 Compressor housings

The VTR 454 compressor stage was contained by two housings that provided for the induction, compression and outlet of the engine charge air. The inner housing that surrounded the compressor impeller had sustained extensive mechanical damage from the impact of the impeller fragments (fig. 11). The internal ducting that fed the compressor had been completely broken away, exposing the diffuser and outlet plenum and several of the smaller fragments of the failed impeller were recovered from that area. Corrosion and other evidence of water ingress (fig. 12) was noted within the discharge casing, indicating the cooling water galleries around the turbine casing had been compromised. The entire housing body had fractured circumferentially around the flange adjoining the centre housing.

Figure 11: Looking into the remnants of the compressor housing

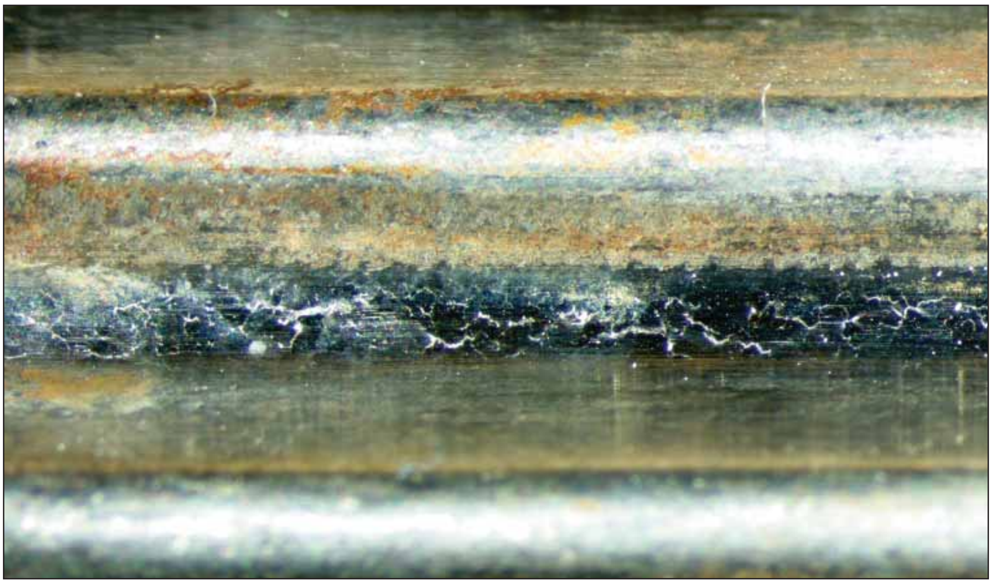
Figure 12: Debris and corrosion within the lower compressor housing body, indicating break-up of the cooling channels and the ingress of water

Figure 13: **Selection of turbine blades taken from around the blade that failed at the fir-tree root**

Figure 14: **Carbonaceous, sooty deposit over a turbine blade fracture surface**

Figure 15: **Same blade as fig. 14, cleaned and showing a uniform fracture typical of gross bending overloads**

Figure 16: An example of cracks and fissures within the fir-tree root of the turbine blades – typical of bending overload forces



1.3.2 Laboratory examination

The five turbine rotor blades and the fragments of impeller disk were inspected in detail at the ATSB Canberra laboratories.

1.3.2.1 Turbine blades

The five retrieved blades (fig. 13), which included the blade fractured at the fir-tree root were examined unaided and using the low-power stereomicroscope. While some of the fracture surfaces were partly discoloured as-received (fig. 14), this was suspected to be carbonaceous materials from the exhaust. Indeed, light cleaning removed the discolouration and presented bright, uniform fractures in all cases (fig. 15). The general fracture surfaces presented a coarse, interdendritic morphology, which was consistent with overload failure of cast high-temperature nickel alloys. Examination of the fir-tree root beneath the primary fracture showed multiple cracks and tears at the base of the root-form on one side (fig. 16). These features were again consistent with the effects of a rapidly applied bending overload force.

The fractures examined showed no evidence or indications of prior cracking, pre-existing defects or other anomalous features.

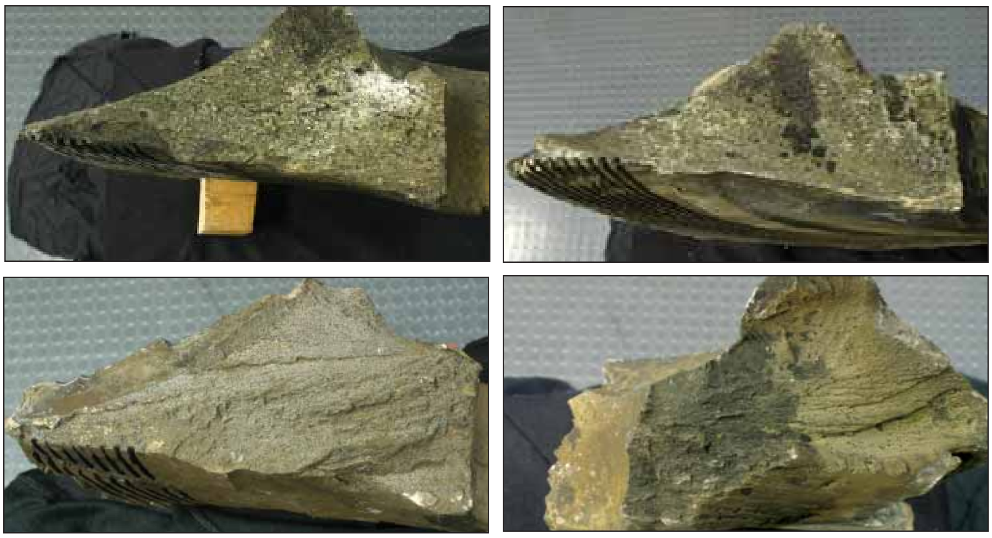
1.3.2.2 Compressor impeller

A total of five fragments of the ruptured impeller disk were examined (fig. 17). A large segment of the disk, representing approximately one-quarter of the impeller circumference was not recovered. All of the fragments had sustained extensive post-failure mechanical impact and abrasion damage over both forward and aft faces. The fracture and other surfaces had also been extensively stained with dark carbonaceous combustion products and a yellow-green product.

Figure 17: Segments and smaller fragments of the compressor impeller disc. Note the predo minant radial orientation of the primary fractures

The primary disk fractures were predominantly radial in orientation. Although partially obscured by the staining, it was apparent that the dominant morphology was one of rapid brittle overload, with the riverline / chevron markings suggesting the commencement of fracture at the disk bore. Close visual and stereomicroscopic scrutiny of the fractures did not reveal any unusual fracture morphologies, nor did it find any evidence of the presence of pre-existing cracking or other defects along the planes of failure. Figures 18 to 21 present the primary fracture surfaces of the disk and show the direction of fracture where this was determinable.

Figures 18 & 19 (top), 20 & 21 (bottom): Primary fracture surfaces of the impeller disk fragments. All show features typical of gross overload



1.3.3 Chemical analysis

A sample of the compressor disk was spectrographically analysed[2] to determine the chemical composition of the disk alloy (weight %).

Al	Si	Cu	Fe	Mg	Zn	Cr	Ni	Mn	Ti	Sr	Zr
~bal	0.21	2.42	1.05	1.53	0.03	<.005	1.05	0.01	0.07	<.001	<.005

"~ bal" denotes balance of composition

The analysis results conformed to the general specification for a UNS A92618 or Aluminium Association 2618 age-hardenable aluminium forging alloy.

1.3.4 Metallographic examination

A sample for general assessment of the representative disk microstructure was taken from an area on the disk approximately two-thirds of the radius towards the outer rim. The sample was oriented so as to enable examination of the full through-thickness microstructure in that position.

When examined in the unetched (as-polished) condition, the structure was typically comprised of an irregular distribution of rounded blue-grey particles within an amorphous matrix (fig. 22). A reference[3] for the AA2618 alloy indicated these particles to be an insoluble FeNiAl9 phase. Etching the specimen in a 0.5% HF solution revealed a fine distribution of CuMgAl2 precipitate throughout the matrix (fig. 23). Microstructural anomalies or other potentially detrimental features were not observed within any of the areas examined.

Figure 22: Unetched and etched microstructural views of the impeller disk alloy. The coarse grey particles in figure 22 are the FeNiAl9 phase

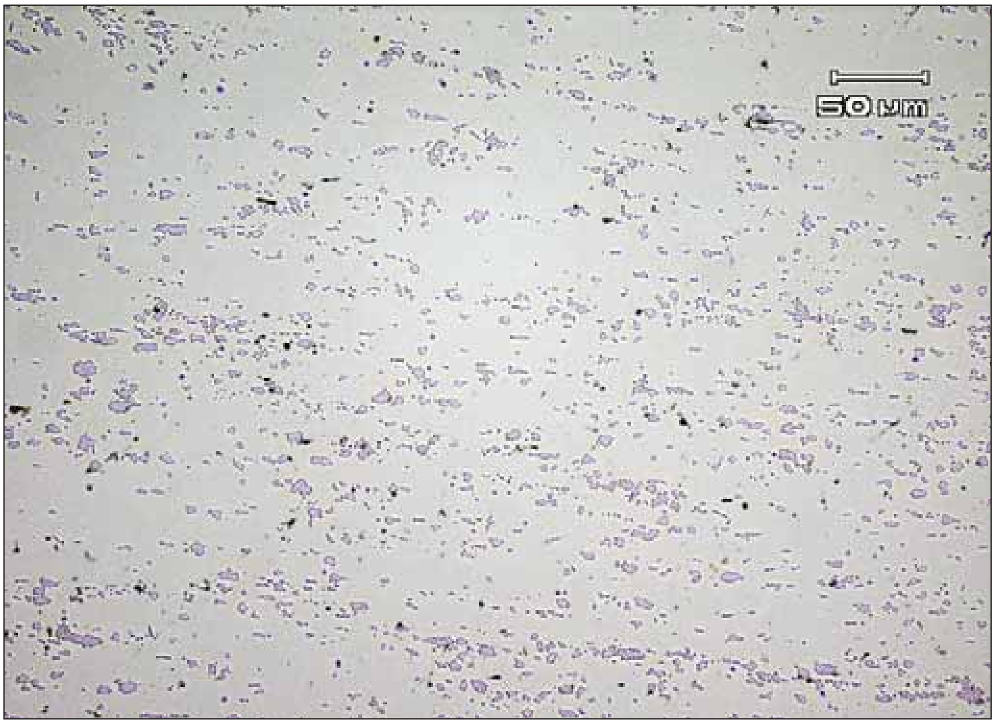
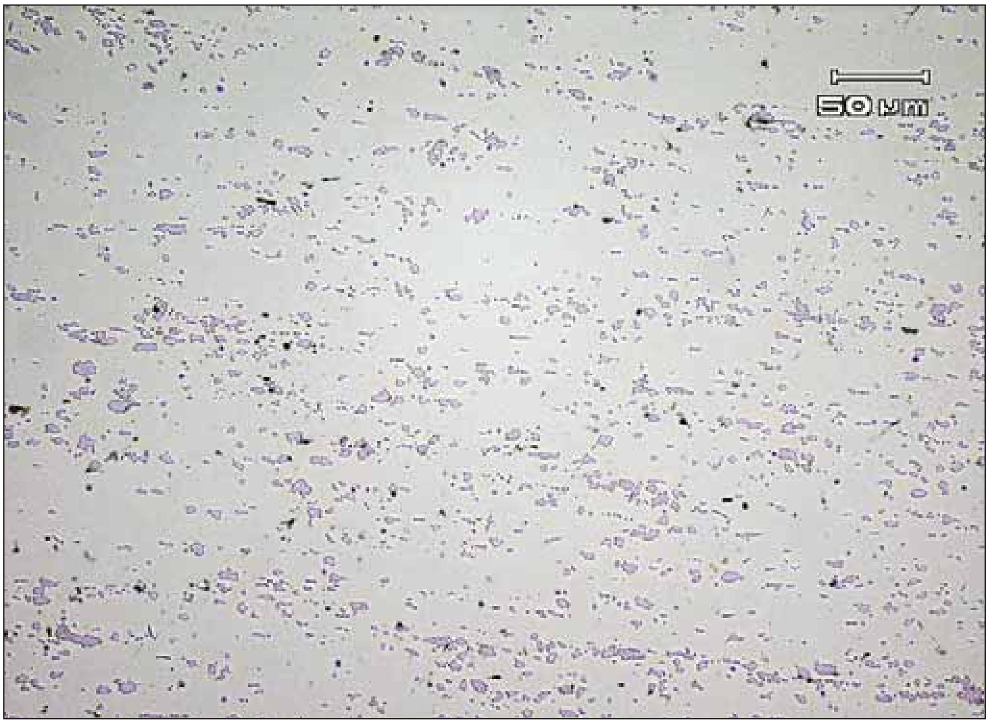


Figure 23: General mottled fine phase of the CuMgAl2 age-hardening precipitate



The observed metallographic structures were consistent with the alloy being in the solution-treated and aged condition, which is generally the nominal state for engineering components produced from the 2618 alloy.

1.3.5 Tensile tests

Three circular section tensile test specimens were machined from one segment of the failed disk and tested in accordance with AS1391 – 1991[4]. The specimens were oriented tangentially to the disk axis and were spaced so as to represent the material at the disk bore, mid-section and outer section. All specimens were taken from the mid-thickness position at each location. A reference for the typical properties of the 2618 alloy in the T61 heat treatment condition was included[5] for comparison against the test results.

Sample ID	*0.2% Proof Stress (MPa)*	*Ultimate Tensile Stress (MPa)*	*% Elongation over 50 mm*	*% Reduction in area*
Inner	421	445	4.5	14
Mid	424	452	6	18
Outer	414	452	6	10
2618-T61[5]	372	440	10	-

1.4 Failure two, 12 February 2003

The second turbocharger failure occurred in a catastrophic and explosive manner, very similar to the first failure. At the time of the failure, the *Goliath* was underway and the main engine was reported as operating normally. The ship's second and third engineers were in the engine room at the time of the failure and provided similar recounts of the event. Both crewmembers reported hearing the turbocharger accelerate immediately before the failure, with the normal 'whine' of the turbine and compressor increasing in pitch over approximately seven seconds. The increase in pitch was reported as exponential in nature and continued unabated until the moment of failure. The second engineer also reported hearing what he thought was a slight increase main engine speed during the turbocharger acceleration. Alerted by the abnormal turbocharger behaviour, both crewmembers were able to escape the engine room before the failure occurred.

The turbocharger assembly had been installed new following the first failure three months earlier and had accrued 1,541 hours of operation since that time. During its short service life, the turbocharger had reportedly performed normally, with no noted instances of abnormal operation or performance.

1.4.1 Visual Examination

The wreckage of the second turbocharger unit was examined within the *Goliath*'s engine room, essentially where it came to rest following the failure. As had occurred previously, the casings of the turbocharger had sustained extensive damage and fracture and in this instance, the primary shaft had fractured at the turbine end bearings, allowing the entire shaft and rotors to displace axially out of position (figs 24 & 25).

Figure 24: View of the second failed turbocharger showing the ejected shaft and exposed exhaust outlet casing

Figure 25: Opposite side of the second failed turbocharger, showing the exhaust inlet casing

1.4.1.1 Turbine housing and nozzle ring

The turbine housing, shroud and nozzle guide vanes all showed extensive mechanical deformation consistent with multiple energetic impacts. The exhaust inlet casing had broken away from its mounts, as had the adjoining exhaust outlet casing (fig. 26), fracturing the housing cooling jackets and supply pipe-work. The outlet casing had dropped approximately 150 millimetres and broken away from the outlet manifold connection. The inlet manifold flexible coupling (bellows) had sustained considerable distortion but remained connected to the inlet casing. The guide vanes were all present and in place within the nozzle ring (fig. 27) and none of the associated components showed any evidence of having cracked, fractured or separated before the main turbocharger failure.

Figure 26: Failed housing mounts

Figure 27: Intact nozzle ring and guide vanes

1.4.1.2 Turbine rotor and shaft

The turbocharger shaft had fractured approximately 150 millimetres from the turbine end and the entire assembly had moved axially toward the compressor. Fracture and complete separation of the compressor housing had allowed the shaft to fall almost completely out of the casing and onto the adjacent deck (fig. 28). Examination of the shaft fracture surfaces showed gross plastic deformation and ductile shear failure – features consistent with rapidly applied bending overloads. The inspection found no evidence of prior or progressive cracking or other defects.

The turbine rotor had shed all of the airfoil blades, with most having liberated completely from the fir-tree slots; the remainder fracturing within the root fir-tree (fig. 29). Inspection of the slots showed burring and deformation of the ribs, consistent with the liberation of the blades under a combination of in-plane bending and centrifugal loads. None of the fractured blade stubs that were examinable showed any evidence of fatigue cracking or pre-existing defects.

Figure 28: Shaft and turbine wheel ejected from the casing due to the failure of the shaft at the bearing housing (arrowed)

Figure 29: Completely de-bladed turbine disk

1.4.1.3 Shaft bearings

Both compressor and turbine end bearing assemblies had been completely destroyed during the failure. The fragments of each unit showed no visible evidence of thermal distress or overheating, however the level of mechanical disruption of the assemblies prevented any further examination of value.

1.4.1.4 Compressor rotor

As had occurred in the first failure, the compressor rotor disk had fractured into multiple radial segments, some of which were recovered from within the turbocharger wreckage. Again, a section comprising approximately one-quarter of the disk was not recovered. The disk fractures (fig. 30 & 31) showed a uniform brittle overload morphology (fig. 32) that was very similar to the fracture features exhibited by the first failed disk. None of the fractures showed any visible evidence of pre-existing cracking or material defects, nor did the disk bore or mating shaft surfaces show any sign of relative rotation of the disk on the shaft while the assembly was still operational (fig. 33). The inducer rotor remained intact and in place on the shaft (fig. 34).

1.4.1.5 Compressor housing

The entire compressor housing, comprising the intake, rotor shroud, diffuser and outlet had broken away from the turbocharger body during the failure (fig. 35) and had sustained very extensive mechanical damage. The inner shroud walls surrounding the rotor had fractured into multiple small fragments, consistent with the energetic impact of the rotor segments (fig. 36). A survey of the housing fracture surfaces found no indication of pre-existing cracking or other features that may have structurally weakened the assembly and contributed to the break-up.

Figures 30 & 31: Segmented failure of the compressor disk, very similar to the first failure (fig.17)

Figure 32: Typical brittle overload fracture exhibited by the impeller disk

Figure 33: Shaft surface showing no evidence of impeller insecurity or rotation

Figure 34: Inducer in place adjacent to the normal disk location

Figure 35 & 36: **Compressor housing completely separated from the turbocharger body and exhibiting extensive internal damage**

1.5 References

[1] Discussions on 11 October 2002 with Mr Rod Iliff, State Manager, Dynamic Turbocharger Services (Australia) Pty Ltd.

[2] Analysis conducted by Spectrometer Services Pty Ltd, Coburg Victoria. Report number 11741.

[3] ASM Handbook, Vol. 9 'Metallography & Microstructures', page 366, figures 55 – 58.

[4] Machining and testing of tensile test specimens conducted by Amec Technical Services, Welshpool WA. Report number 3A70/M2.

[5] ASM Handbook, Vol. 2 'Non-ferrous Alloys and Special Purpose Materials' page 84, table 39.

2 ANALYSIS

2.1 Failure mechanisms

The investigation found that both turbocharger failures sustained on-board *Goliath* were comparable in terms of:

- **Lack of debris containment**
  During both failure events, the turbocharger body and casings were compromised and allowed the liberation of energetic debris into the engine room, presenting a grave hazard to personnel working in the area.
- **Breakage and loss of turbine blades**
  Both events were characterised by the partial or complete loss of blades from the turbine rotor. Analysis of the blade failures revealed a commonality in terms of the failure mechanism – most showed evidence of fracture or liberation from the rotor under backward bending forces. That in itself was evidence of impingement or entrance of materials or debris into the plane of blade rotation, or a loss of blade rotational clearance caused by casing disruption.
- **Casing disruption**
  In both instances, the rotor alignment has been compromised by the mechanical break-up and separation of the major turbocharger casing sections.
- **Bearing failure**
  Where they had failed, the turbocharger shaft bearings showed no evidence of overheating, premature breakdown, inadequate lubrication or other operational deficiency. All failures were characterised by gross mechanical disruption of the bearing assembly, attributable to the casing disruption and movement.
- **Compressor disk failure**
  The compressor disks from both turbocharger's had failed by fracturing in multiple radial locations through the body section. The fracture surfaces were characterised by features typical of brittle overload failure, with no visible evidence of anomalous structures, prior cracking or material defects. In both cases, the compressor disks showed no indication of insecurity or rotation on the shaft before the failures.

2.2 Compressor disk failures

A general overview of the evidence gathered during both investigations suggested that the fracture and liberation of the compressor disks was most probably the initial and primary cause of the catastrophic and destructive turbocharger failures. When in operation and spinning on its rotational axis, the dominant loads on the disk are generated by mass centrifugal forces and present as circumferentially oriented tensile stresses. The mode of disk failure (multiple radial hub fractures) and the absence of contributory physical defects or prior cracking suggested that the failures were stress / overload related and attributable to one or a number of the following factors:

- Gross overspeeding of the turbocharger, producing greatly elevated levels of operational stress and the resultant overload failure of an otherwise sound disk.
- Over-temperature operation of the disk, leading to a progressive over-ageing of the disk alloy and loss in strength to a point where the normal operational stresses produce failure of the disk material.
- Sustained mild overspeeding, producing accelerated microstructural creep and eventual stress-rupture of the disk.

The results of the mechanical testing and metallurgical evaluation of the material from the first disk failure indicated that the disk had not sustained any appreciable thermal or creep damage during its operational life. While the specification for the disk manufacture and properties was not available for confirmation, the general microstructural and mechanical properties exhibited by the disk material were well within the expected and 'typical' range for the determined disk alloy (AA2618) and likely heat-treatment condition (T61).

On the basis of the findings it was concluded that the failure of the first turbocharger compressor disk was a product of a gross overspeed transient, sufficient to directly produce the dynamic overload and failure of the disk. That conclusion was directly supported by the evidence available from the second turbocharger failure, where crewmembers heard the unit accelerate significantly beyond its normal operational speed range immediately before the destructive failure. Further support to the conclusion came from the very comparable nature of the compressor disk fractures and the fact that the second failure occurred in a unit with only three months operational history and was thus unlikely to have accumulated any significant material degradation in that short time.

3 CONCLUSIONS

3.1 Findings

- Both turbocharger failures aboard the *Goliath* were very comparable in terms of the extent of damage sustained.
- Both turbocharger failures were attributable to the radial fracture of the centrifugal compressor disk / impeller.
- The fracture characteristics of both compressor disk failures were similar and typical of material failure under centrifugal overload conditions.
- No evidence of prior cracking or other defects was found within the examinable fractures of both compressor disks.
- The metallurgical and physical properties of the first failed disk were considered satisfactory and typical of the material used to produce the disk.
- Several crewmembers of the *Goliath* witnessed the turbocharger undergo an uncontrolled transient overspeed event immediately before the second failure.

Independent investigation into the equipment failure aboard the
Australian registered bulk cargo carrier *Goliath*
in Bass Strait on 22 September 2002
and off Jervis Bay, New South Wales 12 February 2003