CENTRIFUGAL PUMP EXPLOSIONS

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SNYOPSIS

Several explosions have occurred within the casings of centrifugal pumps within the mining industry in South Africa as well as in general industrial applications in other countries. In some cases, these have resulted in fatalities, severe burns and certainly in all cases, damage to property and loss of production. This paper attempts to understand the phenomenon and to describe the process undertaken at the AngloGold SA Metallurgical operations to eliminate or at best, minimise the potential for any reoccurrence of these events.

1. INTRODUCTION

Centrifugal pumps are used extensively in process industries throughout the world. They range in size from A frame to H frame with corresponding electric motors from 2.2 kW to 375 kW. The majority of these pumps run at between 500 to 1200 rpm being driven by overhead mounted four-pole low voltage motors. The reason for the overhead mounted motors is primarily to allow flexibility with pulley size selection to run the pump at the exact speed demanded by the process. The other benefits are to prevent the high thrust forces from the pump being passed directly on to the motor and finally, to reduce floor space.

Centrifugal pump casings are typically castings. There are essentially three basic configurations: solid casings bolted to a frame, symmetrically split casings and three piece construction. The majority of the pumps are rubber lined with rubber lined impellers.

2. DESCRIPTION OF RECORDED INCIDENTS

There are not many well documented incidents of centrifugal pump explosions available. Below is a brief description of incidents reported locally and in other countries.

2.1 Chilled water circuit brine pump at a US Nuclear Defence facility - 5 August 1998

This must have been quite an explosion as the pump that exploded normally pumps 189 l/s and is driven by a 300 kW motor. These pumps are normally left with their suction and delivery valves in the open position when not running. Due to leakage, this pump’s valves were closed. The pump was started by accident and after about two hours, it exploded. An Operator was slightly injured. There were no procedures in place.
2.2 A miner was seriously injured when a centrifugal pump that was pumping sand slurry exploded at an un-named mine in the US. Shrapnel from the pump struck the miner who was standing about 10 m away. The investigation revealed that the inlet was plugged and the discharge was restricted.

2.3 Island Creek Coal Company, Buchanan County, Virginia, USA – 28 January 2002

This fatal accident is the best documented of all the incidents reported. The tragic accident happened at about 20h30 on 28 January 2002 at a coal preparation facility. Briefly, the plant was off-line for about two days and operating personnel were in the process of re-starting the plant. The pump in question transfers fine coal (- 0,5 mm), from a tank up to a disc filter. The vertical head is about 10 m. When the pump was started, both suction and delivery lines were blocked with fine coal that had settled over

A supervisor reported seeing steam coming out of the drive shaft and sent the operator to investigate. The operator apparently stopped another pump first by mistake then stopped the transfer pump. Immediately after he stopped the transfer pump there was an explosion. The front plate of the casing blew off striking the operator fatally injuring him. A photograph of the pump with the front plate is shown in Photo 1.

The pump had been running for an estimated 40 minutes before it exploded. The investigation suggests that what happened is that when the transfer pump was stopped, the all metal pump internals were red hot. The sudden drop in casing pressure was sufficient to permit a small amount of gland service water through the gland into the casing. This instantly turned to steam causing the explosion. Note the short suction and delivery.

2.4 At an underground mine in the US, a pump exploded destroying two permanent stope workings about 50 m away. The explosion was felt 280 m from the pump. 80 kg of pump material could not be found during the clean up operation.

2.5 There is a recollection of a fatality in Australia when a Warman pump exploded.

2.6 At Bong mine in Liberia, a 14/12 rubber lined pump exploded shattering the pump house roof.
2.7 At No.8 Gold Plant at Vaal River in 2000, a 90 kW CD mill return pump exploded when the suction was blocked apparently by a piece of conveyor belting used to line the feed sump. An operator had been standing next to the pump about five minutes before the pump exploded. It also shattered the corrugated iron pump house. What was strange about this event is that the pump is the first stage of two pumps in series. There was absolutely no evidence that the delivery was blocked as the pump had been running for the whole shift. Photos 2 and 3 show the extent of the damage. There is no question that if the operator had been standing in close proximity to this pump, he would have been killed or very seriously injured.

![Photo 2. Remains of CD pump explosion](image1)

![Photo 3. Pump casing parts](image2)

2.8 In November 2001, a centrifugal pump transferring solution exploded at North 2 Gold plant at Vaal River. There had been a shutdown and the plant was in the process of being started up. The operator started the pump without checking the status of the suction and delivery valves. Both valves were in fact closed. He was apparently under the impression that he was starting the standby pump whose valves were open. All pumps performing this duty are now positive displacement with pressure relief valves.

2.9 In February 2002, a B frame calcine water pump at the West Float and Acid plant at Vaal River exploded when it was started with both suction and delivery valves closed. It was after a power failure and it was a simple case of the Operator starting the wrong pump. The pump has Saunders type valves that, from a distance, are difficult to see if they are open or closed. Photographs are shown below in Photo 4 and Photo 5.

What was significant about this incident is that the force of the explosion ripped the pump body off its base and fractured the rear half of the casing. Presumably the stiffness of the suction pipe on the front of the casing was stronger than the bolts holding down the pump base.
2.10 In January 2003, a small AB frame carbon transfer pump burst at North 1 Gold plant at Vaal River. An Engineering Apprentice was overhauling the pump adjacent to the unit that burst and was hit in the face and arms with hot carbon particles. The pump had been started with both suction and delivery lines choked with carbon. What was different in this accident is that the bolts failed before the casing and the casing opened up when the bolts stretched in tension. The subject of strength of casing bolts versus casing design is debated later on in the paper. Photographs of the failed pump are shown below in Photos 6 and 7.

2.11 At Namdeb mine in Namibia, a large sand transfer pump exploded. The damage done here was extensive as can be seen in photographs 8 and 9. Pieces of the pump were found more than one hundred meters from the building, the roof of which it shattered. Again the force ripped the pump pedestal off its base.
2.12 In October 2003, a C frame slurry pump transferring product at Modikwa Platinum exploded resulting in the Operator loosing three toes. He had been standing on the flange of the delivery line when it exploded.

Photo 8. Sands pump at Namdeb

Photo 9. Roof after explosion

3. SEARCH FOR A SOLUTION

We accepted that regardless of everyone’s best intentions, these events have happened, are happening and if nothing is done, will continue to occur. It was decided to start looking for a cost effective solution to eliminate centrifugal pump explosions and bursts. An explosion is defined as a casing failure and a burst is defined as bolt failure separating the casing without actually fracturing it.

The first thing was to try to understand the phenomenon. When a pump continues to operate with closed valves, a certain percent of the motor power is transferred to the fluid in the casing as heat energy. Eventually the fluid heats up to boiling temperature and turns to steam. The pump then effectively becomes a boiler. The pressure due to the superheated steam then reaches a point where it exceeds the strength of the casing or the casing bolts. When this point is reached, the pump either explodes or bursts under pressure.

3.1 Current detection

Our first mistake was to make conclusions about what we thought would happen under closed valve conditions. We assumed that if the pump were not pumping, the motor current drawn would drop off significantly enough to be able to fit a current sensing device. When tests were done on various sized motors, it was found that the current drawn varied from pump to pump with no relationship apparent that depended on size, duty or speed. The reason for this is that the pump still sees a head to pump against in the closed delivery valve. Short tests were conducted on eleven pumps ranging from 2.2 kW up to 110 kW. The running amps were taken then both suction and delivery valves closed. The drop in Amps drawn was noted. The percentage drop varied from 43% to −2%, in other words, the current went up in the one case.
3.2 Pressure sensing

Our next step was to look at pressure monitoring switches. Obviously as the temperature of the fluid in the casing starts to rise, so does the pressure. The problem with this is that the pressure will only rise significantly when it starts to turn to steam. The switch would have to be fitted into a small pocket in the suction or delivery pipes between the valve and the casing. For clear and process water applications, this would not represent a problem. For slurry, carbon, acid and other chemicals it would be problematic. The slurry and carbon would probably block the port and the chemicals would attack it. The cost of a pressure switch with a range of 2.8 to 28 bar is just over R3 000. Also required is a relay in the panel as well as the hard wiring. This is not an attractive option as it is costly, susceptible to damage and will only be effective when the fluid in the casing starts to turn to steam.

3.3 Temperature monitoring

The next obvious step was to look at temperature monitoring. The attractive feature of monitoring temperature is that the fluid temperature starts to rise immediately after the valves are closed or the lines are restricted. If a pump is transferring water at an ambient process temperature of 20°C, a temperature probe can be installed to trip the motor at 40°C and not risk any damage to casing or liners. The same problems as pressure sensors with the probe pockets also apply. The cost for the probe, controller, relay and hard wiring is approximately R1 800 per installation.

3.4 Strain gauges

The concept of putting strain gauges on the casings was briefly looked at. This would not be suitable as the casings are cast iron and have a non-linear stress/strain relationship. They fracture upon reaching their stress limit.

3.5 Pressure relief valves

Pressure relief valves are commonly used in process industries. Every boiler has one fitted, set at 10% above maximum working pressure. These would be the most reliable as they can handle temperature and pressure. For chemicals however, stainless steel would have to be used. These were discounted as the cost of these valves runs into thousands of Rands.

3.6 Rupture discs

We then contacted the petrochemical industries to find out how they protect their vessels and pipelines under pressure. They typically use aluminium rupture discs mounted in special holders as shown in Photos 10 and 11.
These are precision engineered discs and holders that would work in most instances. Those working in chemicals would have to be Teflon lined. There was a doubt about how the thin aluminium would stand up to carbon particles. Again these are very expensive at about R600 per disc and R1 400 per holder set. The method of installation would be a short piece of pipe welded into the suction line with a flange welded on. The holder with disc fitted would be sandwiched between this flange and another flange welded to a standpipe. This standpipe would either discharge back to the feed tank or into a drain if the fluid were harmless. They were essentially discounted due to the high cost and the delicate construction, which would be prone to damage.

3.7 Fusible plugs

The concept of fusible plugs was looked at. There are essentially two types—heat sensitive and pressure sensitive. Again a pocket would have to be welded onto the suction or delivery lines as close to the pump casing as possible. A small plug of fusible material would be fitted into the pocket to fail when a certain temperature of pressure was reached.

A brief search in the market place shows the lowest temperature rated fusible plug available was about 80° Celsius. This was considered too high and too close to boiling temperatures, especially at altitude where our operations are situated. There is also the concern that with carbon, the pocket will become blocked and not be exposed to the full temperature in the casing. The concept of pressure sensitive plugs is similar to the rupture discs and would be attractive if something robust and cost effective could be found.

3.8 Graphite bursting discs

We managed to contact a company who manufacture graphite rupture discs. They could offer a 50 mm disc that would burst at 600 kPa for R232. It is essentially a blank disc of graphite with the centre machined out to give the desired pressure rating. A photograph of a 600 kPa disc is
shown in Photo 12. The majority of our pumps not in series have suction pressures less than 300 kPa.

The idea was to fit these discs between two flanges similar to the aluminium rupture discs. A small section of 50 mm pipe is welded into the suction pipe as close to the pump flange as possible. The exhaust pipe can feed back to the tank feeding the pump or, where the fluid is harmless like process water, it can discharge into a drain. When each unit was risk assessed, cognisance was kept of the fact that when these discs burst, the fluid will have turned into steam and hence must discharge safely. Where there are sets of pumps in series, only the first pump has the disc fitted. Typical installations of discs sandwiched between two flanges are shown in Photos 13 and 14.

Each plant calculated how many discs to install. We decided not to fit them to pumps that already have some method of protection through status protection like flow meters giving a signal to a PLC. In total 415 pumps were identified to be fitted with graphite disc protection.

4. DISC INSTALLATION

Some initial problems were experienced with the discs. As they are pure graphite and hence quite soft, a few disc plates were accidentally scratched. When the pumps were started, these discs failed. Some were also damaged whilst being placed between the flanges. Rubber gasket material was then inserted between the discs and the steel flanges. All discs ordered initially were 600 kPa rating. We found that on some pumps with higher suction pressures, these discs burst when there were slight surges or water hammer type effects. Discs rated at 800 kPa were installed on these pumps.
5. TEST RIG

It was decided to construct a test rig to conduct some simple tests to try establish some more information about the phenomenon. A small AB frame pump was connected up at North 1 Gold Plant. The objective was to take temperature and pressure readings for various scenarios with different lengths of suction and delivery pipes. A photograph of the pump is shown in Photo 15.

The objective of these tests is to determine what percentage of energy is transferred to the fluid whilst trapped within the suction pipe, casing and delivery pipe. The pump with pipes fitted was filled up on each occasion with buckets of water to measure exact volumes.

Photo 15 Test AB frame pump

6. TEST RESULTS

A total of four tests were conducted. In all tests the gland service water port was plugged. During the tests the gland was pulled up on occasions when water was seen leaking out. In all cases when the delivery pipe blank flange was opened, it was observed that some water had escaped during the test. This is accepted as in real life situations, the fluid will escape through the gland when the pressure in the casing exceeds that of the gland service water, which is usually only 10 to 80 kPa above delivery pressure.

6.1 Test 1 – 500 mm suction and 500 mm delivery

This first test ran for 28 minutes with readings being taken every 2 minutes. The temperatures in the suction and delivery both rose from 7°C to 58°C in this time. There was no change in delivery pressure from 180 kPa through the test. The suction pressure remained at zero for this and all tests as expected. It can be seen from the graph below that the temperature rose as expected in a linear fashion.
The current drawn by the motor during the test was a constant 7.3 Amps. Approximately 0.5 litre out of the initial 15 was lost during the test. The formula to calculate heat energy addition is simply

\[
(Volume \ of \ fluid) \times (Specific \ heat \ of \ fluid) \times (change \ in \ temperature) \times m \ (kg) \times Cp \ (kJ/kg/°C) \times \Delta T = \text{Total heat in kJ}
\]

To convert this to power, the figure in kJ must be divided by the total time in seconds.

For each of the tests, the volumes are separated into pipe sections and the casing section as the change in temperatures in each varied. The casing temperature changes were taken as an average of the suction and delivery pipes immediately next to the casing. The motor power drawn is calculated using the formula:

\[
\text{Power (kW)} = 1.732 \times \text{Voltage (V)} \times \text{current} \times \text{Power factor}
\]

**Volume 1 – Suction pipe**

\[
\text{Power} = 2.51 \times 4.18 \times (58 - 7)/28/60 = 0.317 \text{ kW}
\]

**Volume 2 – Pump casing**

\[
\text{Power} = 101 \times 4.18 \times (58 - 7)/28/60 = 1.269 \text{ kW}
\]

**Volume 3 – Delivery pipe**

\[
\text{Total power transferred to fluid}
\]
Power = 2.51 x 4.18 kJ/kg°C x (58 – 7)/28/60
= 0.317 kW

Power = 0.317 + 1.269 + 0.317 kW
= 1.9 kW

**Electrical power**

Power = 1.732 x 525 x 7.3 x 0.8
= 5.31 kW

**Percentage of motor energy transferred to the fluid entrapped = 1.9/5.31 = 35.8%**

The remaining 64.2% is lost in friction in the bearings, pulleys, V belts, gland as well as radiant heat of the piping which we monitored with a thermographic camera. It remained at about 30°C Celsius throughout the test as it is effectively insulated with the rubber linings.

It took 28 minutes to raise the temperature by 51°C degrees. The rate of rise of temperature is therefore 1.82°C per minute. Under similar operating conditions for such a pump with a suction and delivery valve positioned 500 mm from the casing, it would take 44 minutes to reach 100°C from 20°C. It is assumed that only when the fluid reaches boiling point does the pressure start to increase dramatically. After the fluid reaches boiling point, heat continues to be added at the same rate by the motor which in this case was 1.9 kW or 1.9 kJ/s.

This pump’s casing has a rating of 850 kPa with a manufacturers factor of safety of 2.5. This gives a theoretical failure pressure of 2 125 kPa. Using a pressure/enthalpy graph for a boiler and a starting casing pressure of 180 kPa, the water started to turn into steam at 111°C. The enthalpy at this point is 490 kJ/kg. Assuming it converts into steam at constant pressure, the enthalpy at fully dry steam state is 2 702 kJ/kg. The energy added is therefore 2 212 kJ per kg. From here it superheats up to the casing failure pressure of 2 125 kPa. The temperature at which this happens is 215°C. The enthalpy at this point is 3 281 kJ/kg. The enthalpy added to failure is therefore 579 kJ/kg. At this point most of the water and steam would have escaped through the casing, flanges and gland. Using the Rankin formula:

\[ m = \frac{PV}{RT} \]

\[ = \frac{2 125 000 \text{ Pa} \times 0.015 \text{m}^3/287 \text{ x (215 + 273)}}{287 \times (215 + 273)} \]

= 0.28 kg

Therefore only 0.28 kg of steam is present at failure. The total energy added is

579 kJ/kg x 0.28 kg = 162 kJ

It is being added by the motor at a rate of 1.9 kJ per second and therefore will explode in 162/1.9 = 85 seconds.

Time taken for water to heat from 7°C to 117°C = (117 – 7)/1.82 = 60 minutes
Time taken to convert to steam = (2 702 – 490) x 7.5kg / 1.9 = 145 minutes
Time taken to superheat to failure = 162/1.9 = 1.5 minutes
Total time to failure = 206.5 minutes or 3 3/4 hours.

6.2 Test 2 – 1 000 mm suction and 500 mm delivery

An additional 500 mm length was added to the suction and the test conducted in a similar fashion to the first test. The results were:

Volume 1 – Suction pipe 1 = 0.308 kW
Volume 2 – Suction pipe 2 = 0.261 kW
Volume 3 – Delivery pipe = 0.327 kW
Volume 4 – Pump casing = 1.263 kW

Total power transferred to fluid = 2.159 kW

Electrical power = 1.732 x 525 x 7.2 x 0.8 = 5.24 kW

Percentage of motor energy transferred to the fluid entrapped = 2.159/5.24 = 41.2%

As expected, the temperature rise in the second length of suction pipe furthest from the suction had a lower temperature rise than that nearer the suction of the pump. The temperature in the delivery was marginally higher than the immediate suction. The percentage of motor power transferred to the fluid is also relatively consistent with the first result. The current was only 0.1 of an Amp below the first test.

Also as expected, the rate at which heat is added remains similar to the first test. There is just more fluid to heat up and the process is the same but would just take longer to fail. There was an interesting observation in the second suction temperature at t = 24 minutes. The temperature was observed to increase by almost eight degrees in less than the two minute interval. It subsequently returned to its linear relationship. It does indicate that there are perhaps, vortices or thermal waves moving in the pipe.

6.3 Test 3 – 1 500 mm suction and 500 mm delivery

A third length was added to the suction and the test repeated. The results were:
Volume 1 – Suction pipe 1 = 0.267 kW
Volume 2 – Suction pipe 2 = 0.197 kW
Volume 3 – Suction pipe 3 = 0.180 kW
Volume 4 – Delivery pipe = 0.314 kW
Volume 5 – Pump casing = 1.161 kW
Total power transferred to fluid = 2.119 kW

Electrical power = 1.732 x 525 x 7.1 x 0.8 = 5.16 kW

Percentage of motor energy transferred to the fluid entrapped = 2.119/5.16 = 41.03 %

This is again consistent with the second test with the heat addition being a constant 2.1 kW. The power drawn was again only 0.1 Amps down on Test 2 at 7.1 Amps.

6.4 Test 4 – 500 mm section and 1 000 mm delivery

The last test was different to the first three in that the delivery was now increased.

Volume 1 – Suction pipe = 0.292 kW
Volume 2 – Delivery pipe 1 = 0.317 kW
Volume 3 – Delivery pipe 2 = 0.255 kW
Volume 4 – Pump casing = 1.219 kW
Total power transferred to fluid = 2.084 kW
Electrical power  \[ = 1.732 \times 525 \times 7.5 \times 0.8 = 5.46 \text{ kW} \]

Percentage of motor energy transferred to the fluid entrapped \[ = \frac{2.084}{5.46} = 38.2\% \]

In this test the suction and delivery temperatures closest to the casing rose at essentially the same rate. This has been the trend throughout all the tests. The second delivery pipe temperature was less than that closest to the pump although it did seem to start increasing towards the end of the test. The power added by the motor is again just over 2 kW but the current drawn rose slightly to 7.5 Amps, up slightly from the other tests.

7. DISCUSSION OF THE RESULTS

What was consistent throughout the tests is that the heat addition remains almost constant at about 2 kW regardless of the volume of water. This is expected as it can be compared to a domestic heater in a room. The heat from the heater remains the same regardless of the size of the room it is put in. The bigger the room, the longer it takes to reach the same temperature. It is not necessary to do the enthalpy calculations to realise that the smaller the entrapped volume, the quicker the failure.

What is significant is the percentage of the motor power that transfers heat into the fluid. It would appear that 40% can be used as a rule of thumb for this application. Water did leak out during the tests which is realistic as this will also happen in operation. Gland service water will not enter the casing as soon as the pressure increases slightly as the differential is only slight.
8. PUMP DESIGN PARAMETERS

One of the major issues that determine what will happen when a pump runs with closed valves is the strength of the casing versus the strength of the casing bolts. What complicates the matter is the determination of what projected area does the pressure in the casing act on. In the majority of cases the body of the pump does not move and the force exerts itself on the projected area of the pump casing half. The pump manufacturers design a pump on volume and pressure. When the physical size of the pump is complete, a casing rating is selected depending on the intended operating pressure. A factor of safety of between two and two and a half is usually used. They do not consider which is stronger—the casing casting or the combined strength of the bolts. Consider three test cases:

8.1 Test case 1

The AB carbon transfer pump that burst at North 1 gold plant in January 2003 stretched the bolts before the casing could fail. There were 9 x M12 grade 4.6 bolts holding the casing together. The projected actual area of the casing including the suction opening is 145 500 mm². As with the test pump, the casing design rating is 850 kPa with a factor of safety of 2.5 = 2 125 kPa.

The force required to fracture the casing = Internal pressure x Projected cross sectional area

\[ = 2 125 000 \text{ Pa} \times 145 500 \text{ mm}^2 = 309 \text{ kN} \]

The combined strength of the bolts in tension = No. bolts x cross sectional area x Yield strength

\[ = 9 \times 113 \text{ mm}^2 \times 240 \text{ MPa} = 244 \text{ kN} \]

In this case the bolts should and did actually fail before the casing. Had these been grade 8.8 bolts, the combined strength changes to 650 kN which means the casing would have failed first possibly killing the Learner.

8.2 Test case 2

The second test case was an enigma. By chance when visiting the pump manufacturers, there was a large pump that had the casing fractured on the pressure test bed. It had a working casing pressure of 3 000 kPa. It is procedure to hydraulically pressure test each pump to twice the rated pressure, in this case 6 000 kPa. The person conducting the test was distracted and the test pressure rose to 7 500 kPa at which point the front part of the casing fractured and fell on the ground. It is held together with eight M20 grade 8.8 bolts. The effective area is approximately 280 000 mm².
The force required to fracture the casing = Internal pressure x Projected cross sectional area
= 7 500 000 Pa x 280 000 m² = 2 100 kN

The combined strength of the bolts in tension = No. bolts x cross section area x Yield strength
= 8 x 314 mm² x 640 MPa = 1 608 kN

In theory, the bolts should have failed before the casing at a calculated pressure of 5 750 kPa. When this was brought to the manufacturers' attention, they did an investigation. The test was repeated at a slower pace giving time for the bolts to stretch. Sure enough when the test pressure reached about 6 000 kPa and stopped, the bolts were observed to have stretched a few millimeters.

8.3 Test case 3

Returning to the one pump that sheared the bolts on its pedestal, we can calculate the force required to rip out the four bolts as well as the time to explosion. It is a B frame calcine water transfer pump. What is different to a clear water pump is that the feed temperature is already at 70°C. The relative density of the fluid being pumped is 1.1. Using quartzite as an estimate for the solids specific heat, the effective specific heat of the fluid drops from 4.18 kJ/kg C for water to 3.71 kJ/kg C for this calcine water.

With the valves closed, this pump draws 18.7 Amps from the 30 kW motor. Using the 40% we calculated in the test pump, the heat transferred into the fluid is estimated at:

\[ P = 0.4 \times 1.732 \times 18.7 \times 525 \times 0.8 = 5.44 \text{ kW} \]

For the estimated 45 l entrapped within the casing and short suction and delivery lines, it would take

\[ \text{Time} = 45 \text{ l} \times 3.71 \text{ kJ/kg C} \times (117° C - 70° C)/5.44 \text{ kW} = 20 \text{ minutes} \]

for the fluid to reach 117°C and start converting to steam.

As the volume and heat added are three times that of the test pump, the time to reach failure from here will also be 147 minutes or a total of 167 minutes, just short of three hours.

These pumps have their suction and delivery valves very close to the casing of the pump, which expedited the failure. They have now all been fitted with the graphite disc protection.

The other unique feature of this failure was that it was the only one where the force of the explosion sheared the pump base bolts. It is evident that things such as stiffness of suction and delivery pipes will determine where the force is released. With a casing force of about 390 kN at failure, the 30 kN resistance of the two front 16 mm bolts is minimal.
9. SUMMARY AND CONCLUSIONS

This paper has tried to highlight the fact that centrifugal pumps explode and burst all around the world and evidently more often than we thought. Efforts were made to obtain more information from the places where these incidents occurred with little success. The fact that only two people were killed in the twelve incidents listed above is a matter of pure chance.

The search for the best solution explored many options. There is an even simpler solution in the case of a simple installation. A simple installation is defined as a pump that is fed from a tank or open vessel adjacent to the pump. A 50 mm standpipe can be welded into the suction pipe immediately before the pump and can feed directly back to the feed tank as shown below. The fluid in the standpipe can never under normal operating conditions exceed that of the tank. Under closed valve conditions, the casing is effectively vented to atmosphere. The pressure in the casing will never exceed the static head of the feed tank. This eliminates the need for the graphite disc. This system should only be used where it is practical to feed back to tank.

The question remains for the pump designers of whether they should design the strength of the casing bolts to be weaker that the fracture strength of the casing. The obvious reason for the strong bolts is to hold the casing together. Pumps with weaker bolts will burst and pumps with stronger bolts will explode. In the one instance where the pump burst, the damage, apart from the unfortunate injury, was minimal compared to those that exploded.

10. RECOMMENDATIONS

There is no doubt that centrifugal pumps represent a danger. Most modern operations will have some pumps with flow meters or pressure switches or some other means of detecting that the pump is running but not delivering. The majority of pumps do not have this protection and the reliance is on the human element.
The author therefore recommends for pumps with no protection at present:

1. Where possible, standpipes should be fitted to all centrifugal pump suction lines between the valve and the casing.

2. Those where it is not practical to fit a standpipe should have the graphite discs fitted.

3. In installations where a pump is fed from another pressure source such as another pump some distance away; a pressure relief valve can be fitted. The alternative is to ask the disc supplier for a higher rated disc and install it.

4. Suction and delivery columns should be as long as practical. There are usually space constraints at most installations. This will not prevent the explosion; it will just buy you more time.

5. Where flow is measured electronically, the PLC should be programmed to trip the pump if there is current but no flow for a continuous period of ten minutes.

Over and above these measures, the author believes that more research needs to be done and pump manufacturers should have a closer look at their design philosophy on strength of bolts versus strength of casing.

**AngloGold Ashanti:**

Risk assessment identified critical pumps. Standpipes installed on all simple systems. Initially carbon discs installed but were found to wear in some instances. There were also problems with surges prematurely bursting discs. Pressure relief valves installed. All new installations fitted with protection.

**Anglo Platinum:**

Various plants at different stages of implementation. All new pumps fitted with protection.

**Alcoa:**

All high-risk slurry pumps are being fitted with thermal cut-out switches. All new pumps are to be double cased in construction and casting are to be capable of fitting the thermal cut out probe.
Barrick:

Bulyanhulu Mine Tanzania –

1. Additional instrumentation on valve positioning.
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3. Thermal fusible plugs fitted in discharge section rated at $65^\circ$ C.

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