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Feasibility Study of a Bi-directional Centrifugal Pump for DBT class 45 CST Gearbox Used in Underground Coal mining Operation

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Abstract
This paper presents a feasibility study of using a bi-directional centrifugal pump into DBT's Series 45 CST gearbox. The suitability of other pumps for cooling and the design of a new symmetrical centrifugal pump that would be suited to the series 45 CST gearbox have been reviewed with financial versus functionality and usability. The analysis and results of this study indicate that by introducing the newly designed bi-directional pump, DBT may save over \$370k in production costs over 10 years. This equates to a savings of \$1850 per gearbox which is about a 26% saving on the current set-up, and thus bi-directional pump is reasonably feasible.

Introduction
Underground coal mining is one of the major operations of Australia's coal industry. Significant benefits can be gained by improving the design of any equipment used to extract coal from underground mines. The underground mining industry is a very specialized industry where conventional surface mining methods are ineffective and unique equipment is needed. In Queensland, longwall mining system is commonly used for hauling coal and rock from the longwall face. Figure 1 shows the longwall system as it would sit on the coal face. The shearer, located under the shields, is the cutting instrument which is always the main features of a lonwall system. In such system, coal is removed by shearer and is conveyed by a chain conveyor. The heavy duty chain conveyor is sprocket driven (Figure 2) by the Controlled Start Transmission (CST) gearboxes and they are required to transmit an enormous amount of power for bulk materials handling.

There are over 250 of the 45 CST Gearboxes in operation around the world today [1]. Up to 3 of the 45 CST Gearbox's can be used simultaneously to harness up to 2500 kW of power for hauling coal and rock from the longwall face. Each face can support up to three 45 CST gearboxes, two at the maingate (1 P45 and 1 KP45) and 1 KP 45 on the tailgate. The typical maingate, takes advantage of the P45 and KP 45 gearboxes. Even though it is protected by guards the gearbox is still exposed to debris on all sides. Such a heavy duty conveyor and capacity of mining today requires special features when controlling the enormous amounts of power being transmitted into bulk materials handling. The CST Gearboxes fill this role.

Because of limited room underneath the shearer shields, the shearer is forced to ride on the chain conveyor. This means that if there is a cave in on the coal face, some large rocks for example, may not fit under the shearer when it makes its return journey back down the face. It is in these special circumstances that the conveyor and gearboxes will be required to run the opposite way, so that the large rocks can be removed from the conveyor from the tailgate end. The current design of using centrifugal pumps with freewheeling assemblies for cooling gearboxes has shown problems and failures. Therefore, redesigning to a reversible (a bi-directional pump) has been proposed. The feasibility of this bi-directional centrifugal pump into DBT's Series 45 CST gearbox is investigated in this paper. This bi-directional pump should improve the gearbox operating and cooling system by simplifying the drive train to the gearbox cooling oil pump.

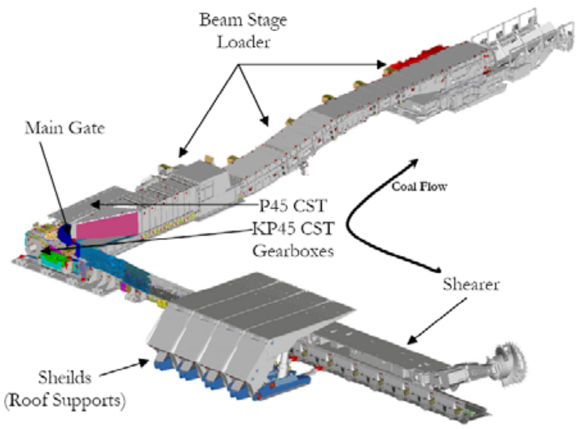


Figure 1: The Longwall System

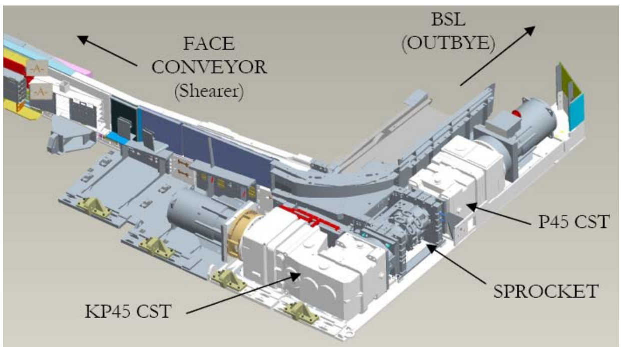


Figure 2: Typical maingate using two(2) CST Gearboxes

Background and Significance
Since the 45 CST gearbox's first design the most suitable pump available has been a right hand centrifugal pump with

two free wheeling assemblies as part of the gear reduction drive train, because no pump that could suit the application was available.

At Drummen Mine in U.S.A. the first sign of a problem surfaced in September 2002. A gearbox failed and after further investigation it was found that the pump freewheeling assemblies were failed. The freewheeling assemblies kept failing and replaced at every overhaul. DBT America thought the failure was because the conveyor had to keep running backwards. This was due to the fact that the mine was a unique multi-layer seam where there was a couple of feet of coal, a foot or two of clay rock in the middle and another couple of feet of coal on the bottom. This occasionally caused large rocks to fall on the conveyor and block the shearer's path. This meant that they would reverse the conveyor and remove the large rocks from the tail gate end. It was this reversed use of the 45 CST gearbox that prompted America to experiment with using the MP PUMPS 200 series reversible pump from the 30 CST and test its performance in the 45 CST. DBT Germany tested this pump at higher speed but as the temperature rose the pump could not hold the 2.5 bar of pressure necessary to maintain the flowrate quota [2, 3]. So they then proceeded to try 2 pumps as shown in Figure 3.

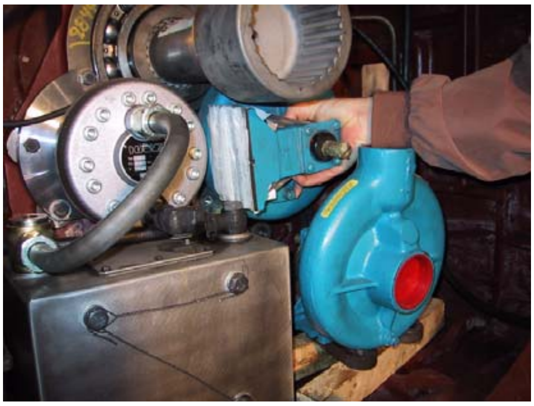


Figure 3: Drummen Attempt to fit two MP 200 Series Reversible Pumps.

The problem surfaced at Speed Mining U.S.A. was as a failed gear. DBT America replaced the broken gear with a wider face design and then the narrow gears adjacent to it broke. They then proceeded to replace all the gears in the train with wider designs. The gearbox now, was able to survive a panel but the gearing would be severely worn. This then led onto breaking pump shafts. Speed Mining never reversed their conveyor so this was not their problem.

The problem next showed up at Monteray Mining. They survived panels with widened gears. Then the pump shaft failed so they made that thicker too. The gearbox then made it through the block just barely. People within the company and customers claimed it was a mechanical problem. After months of investigation the problem was found to be caused from transient re-switching torques [4] in the large AC induction motors, where the electric motor was cutting out for a second or two at near synchronous speed which means it would have had the highest torque available to try and get back to speed. It was this torque acting against the inertia of the fluid in the cooling system that created enormous forces in the mechanical components of the pumping system, causing the weaker gears and shafts to fail. This could explain why the bigger gears and thicker shafts were not coping. This problem was solved by installing capacitors on the motor side of the switch to eliminate voltage drop and install timers to let the motor slow right down before a restart.

Redesigning to a reversible pump will probably not prevent these types of failures though some of the gear train could be eliminated and therefore some inertia will be reduced. Redesign would provide a cleaner, less expensive gear train and save money in the long run. Two free wheeling assemblies and a couple of gears could be eliminated.

Description of the Series 45 CST Gearbox

The planetary gear CST has been developed for driving underground chain conveyors. It is designed for installation on heavy duty drive frames which can withstand equal or greater torque, with the shaft and sprocket supported by separate bearings. The transmission gear is: P45 – a two stage planetary design; KP45 – a two stage planetary coupled with crown and pinion and straight helical gear train design. Transmission ratios are 28, 33, 39, 45:1, and 50:1. These transmission ratios are achieved by changing the input planetary stage. The maximum transmissible torque is 450,000 Nm. The entire gearbox (gears, bearings and pumps) is immersed in oil. The gearbox is designed to operate at certain angles of inclination and special oil levels are designated for each angle of inclination. General assembly of a KP45 CST is shown in Figure 4. The supply unit is the location of the cooling oil pump.

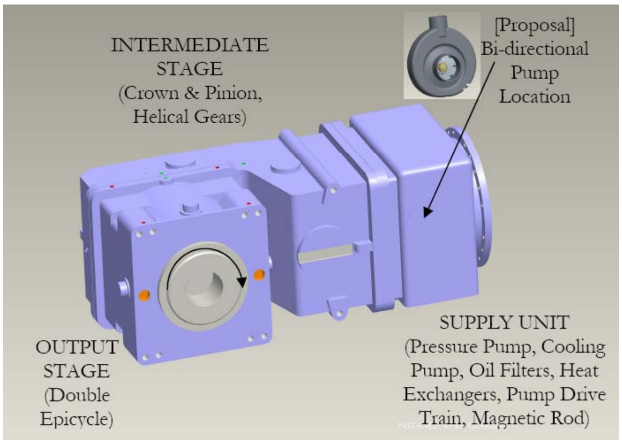


Figure 4: KP 45 CST Gearbox, General Assembly and Section Contents

The operating conditions in this gearbox are:

- Cooling Water Flow rate: 25 L/min (sufficient even if the gearbox is entirely buried)
- Cooling Oil Flow rate : 1500 L/min (sufficient to cool the clutch at maximum torque on start up)
- Oil Capacity is around 800 Litres
- Cooling Oil Circuit operates at around 2.5 bar of pressure

The cooling oil pump (Figure 5) sits near the bottom of the gearbox so to make sure that all of the oil in the tank is circulated to transfer heat from the clutch to the casing and heat exchangers. There is a general rule of thumb for this; the suction should sit around 1/3 of the Suction Diameter away from the bottom of the tank.

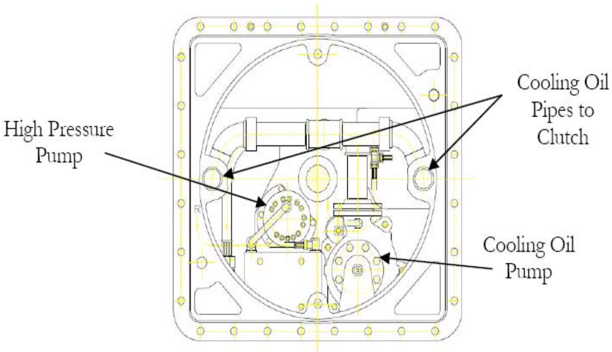


Figure 5: Supply Unit 45 CST Physical Geometry of Pump

The cooling pump drive train consists of three shafts (Figure 6). Two gears of the main drive shaft transmit power to two free-wheeling assemblies (opposite direction of sprags) on the idler shaft. This is so because the direction of the idler shaft is always going one way. A helical gear on the idler shaft transmits power to the pump shaft. When the Main drive shaft is operating at 1450 rpm the pump shaft is turning at 2900 rpm.

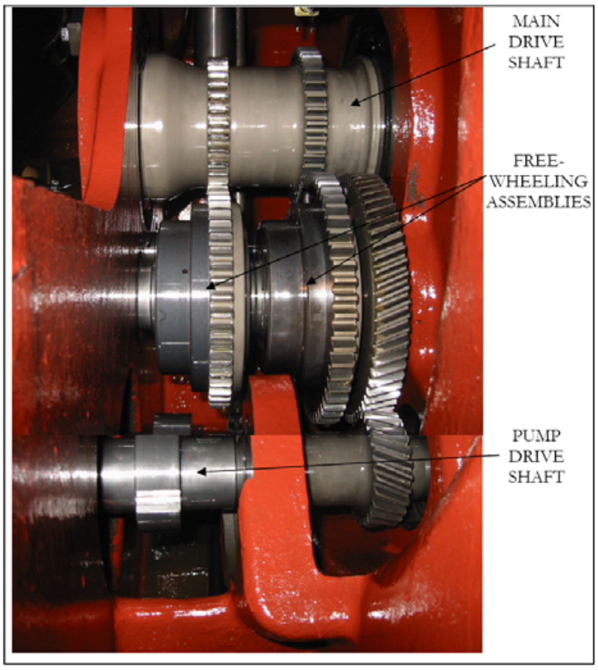
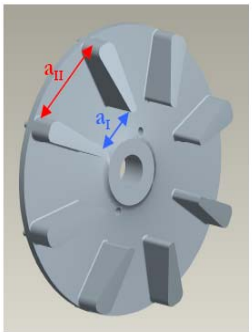


Figure 6: Cooling pump drive-train 45 CST

Proposed Bi-directional Pump

Calculations

Preliminary calculations for the model of a basic pump are used to obtain basic dimensions of the pump impeller and housing. The detailed calculation can be found elsewhere [6]. The next step is to refine the vane layout for a symmetrical impeller. Normally the cross-sectional area of the channel increases gradually from inlet to outlet of the vanes. If the area ratio is too high then flow separation may be increased. By working on increasing the depth of the inlet side of the vanes and decreasing the depth of the outlet of the vanes, the increase of cross-sectional area can be tailored while working with this difficult star type impeller. The actual area of flow ratio is calculated to be 0.98 as shown in Figure 7. In the equation in Figure 7, a_I and a_{II} are the flow areas between vanes which are marked on the diagram of Figure 7.



$$\frac{a_{II}}{a_I} = \frac{48.8 \times 15}{30 \times 25} = \frac{732 mm^2}{750 mm^2} = 0.98$$

When a safe value lies between 1 and 1.3

Figure 7: Impeller - Cross-Sectional Area Ratio

The diffuser system is where a large part of the dynamic head is created. This diffuser system was interesting because it had to be symmetrical to conform to the goal of equal efficiencies in either direction. This is why an 180˚ volute was chosen instead of the conventional 360˚. The fluid is directed to the outlet at the top of the volute via a small radial triangular guide. The fluid transmission is designed as smooth as possible. The area of flow ratio between the outlet of the vane channel (a_{II}) and the outlet throat (a_{thr}) was calculated to be 0.82 as shown in Figure 8. Four vanes were chosen because there are only four vanes dispensing fluid into the volute at any one time. The other 4 vanes on the opposite side would put energy into the fluid from which some of it may escape out the top from the vacuum, some will squeeze through the bottom of the volute and some would be recycled through the pump. Further testing could verify what happens to this excess charged fluid in this symmetrical system.

$$\frac{a_{thr}}{a_{II}} = \frac{\pi 27.5^2}{48.8 \times 15 \times 4_{vanes}} = \frac{2403 mm^2}{2928 mm^2} = 0.82$$

Figure 8: Pump throat area / impeller outlet area Ratio

This value gives the appearance that the outlet is way too small although this is necessary to speed the velocity of the fluid up so that we can achieve our goal flow rate which was calculated in the preliminary calculations. It is easy to see how this pump can be less efficient than a conventional centrifugal pump but this is a minuet sacrifice compared to the suitability for an application that requires a pump to be driven both ways. Some features were added to generally make the pump look aesthetic and smooth flowing.

Physical and Mechanical Features of Design

During the literature review on centrifugal pumps some helpful features were found that could be included in the design. First of all a radial, semi-open impeller (Figure 9) was chosen to be implemented because of its ease of manufacturing and helpful feature. It allows the manufacturer to cast and machine the product for a smaller cost than that of pure machining.

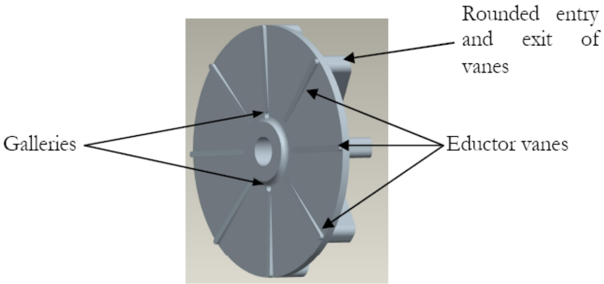


Figure 9: Features of Impeller

In Figure 9 you can notice the eductor vanes on the back of the impeller. If it is decided that packing with wearing rings should be used as the seal mechanism, eductor vanes can reduce pressure on the packing box, giving longer packing life and less maintenance [5]. Other design features include; a single DIN style nut for fastening; Nominal inlet and outlet pipe diameters for BSP pipe threads; and appropriate metric tolerances according to the Pump Handbook [5]. Items left out are; the design of seals, bearings, bearing housing, footing flange and suction funnel.

Reviewing the 90˚ bend in flow (Figure 10), more time could have been spent on counter-sinking the fastening nut so that is acts as less of a restriction. Another consideration was the wall thickness of the housing. The author could not find any document about it, but would assume that it would have to be twice as thick as it is shown here so that it may withstand any pressure spikes in the system and not develop any fatigue cracking throughout its life.

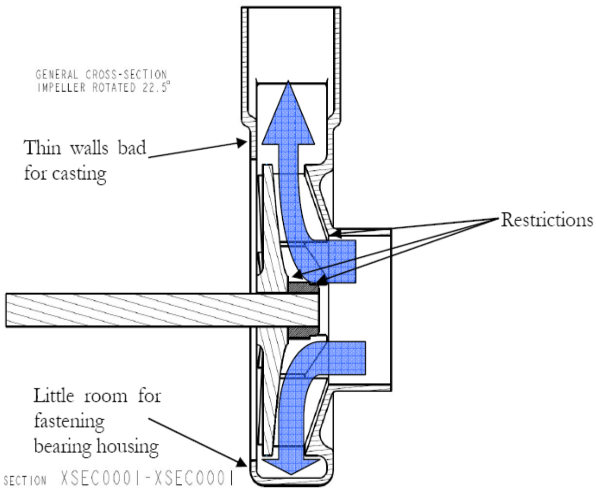


Figure10: Pump Review – Cross-Section

Method of Manufacture

The method of manufacture should be similar to most brands of centrifugal pumps. A large quantity of about 100 units would need to be produced to make the exercise worthwhile. Real quotes from casting foundries and machine shops around Australia may have been obtained with the completion of detailed drawings.

Symmetrical Impeller: The impeller was the simpler of the two parts to manufacture and was chosen to be made of **cast steel** (preferably a more ductile material) and then machined to tolerance.

Symmetrical Housing: The housing was a much more intricate design and if required to be cast in two halves, machined back to a reference and bolted together. Although this design does not allow for the bolting of the two halves, it may be able to be cast as one piece. The chosen casting material would not have to have high impact strength nor be corrosion resistant. A cast grey iron should give the machinability and strength required as ductility is not a big concern with the housing because it only has to withstand the hydraulic pressure. The other advantage of cast grey iron is the strength in thin walled structures.

Predicted Performance

We can predict the performance of a pump using formulas and charts, derived from industry test samples, located in the Pump Handbook.

Hydraulic Efficiency: The hydraulic efficiency depends on the design and the execution of the flow passages. However we can predict hydraulic efficiency by using the chart given in the Pump Handbook which shows hydraulic efficiency versus capacity [5].

$$Q = 0.025 m^3/s$$

$$\eta_H \approx 1 - \frac{0.071}{Q^{0.25}} = 1 - \frac{0.071}{(0.025)^{0.25}} = 0.82$$

Where Q represents flow rate (in m^3/s) and η_H represents hydraulic efficiency (in %).

Volumetric Efficiency (η_V) is defined by,

$$\eta_V = \frac{Q}{Q + Q_{Losses}}$$

Using the chart given in the Pump Handbook which shows volumetric efficiency as a function of specific speed of the pump (N_{sm}) and capacity (in gallon per minute, gpm) we can predict the volumetric efficiency to be:

$$N_{Sm} = 40,\ Q = 400 gpm$$

$$\text{from chart reads to be } 1 - \eta_V = 0.025$$

$$\therefore \eta_V = 0.975$$

Mechanical Losses: The mechanical losses can only be calculated when the bearings and seals are available. We can use the “Ratio of mechanical power (P_M) loss to water power (P_W) as a function of specific speed and capacity” as an estimate. This is located in the Pump handbook. This gives:

$$\frac{P_M}{P_W} = 0.015$$

Impeller Disk Friction: It is explained in the Pump Handbook that for pumps above specific speed of N_{sm} = 39 the disk friction power (P_{DF}) is relatively small and can be approximated by,

$$\frac{P_{DF}}{P_W} \approx 0.02$$

Maximum Pump Efficiency: We can calculate the maximum pump efficiency from the estimated losses as given below.

$$\eta = \frac{1}{\frac{1}{\eta_H \eta_V} + \frac{P_{DF}}{P_W} + \frac{P_M}{P_W}} \approx \frac{1}{\frac{1}{0.82 \times 0.975} + 0.02 + 0.015} = 0.78$$

Power Needed: With 78% maximum pump efficiency, the power necessary (P_S) to operate the pump can be calculated by,

$$P_S = \frac{1}{\eta} 9.80 QH\, sp.gr. = \frac{9.80 \times 0.025 \times 31 \times 1}{0.78} = 9.73 kW$$

Where H is the pump head (in meter), Q is the flow rate (in m^3/s) and sp.gr is the specific gravity.

Basis of Application: The proposed design is somewhat similar to the design of the reversible MP Pump used in the class 30 CST gearbox. When we compare pump performance characteristics of the MP Pumps 200 Series Reversible pump to the 200 Series Left or Right hand pumps we can notice a definite drop in head production in the reversible series of about 10%. We can also notice an approximate 30% increase of power consumption which also indicates the kind of efficiency decrease we would get by using the bi-direction/ reversible configuration. Considering these estimated performance figures and comparing them to the MP Pumps 200 Series, it is fair to say that the calculated performance estimates are valid for this type of pump and are not greatly different to a high performing pump of that size.

Financial Viability

The financial viability was studied by gathering cost estimates from different suppliers of pumps. The initial development investment, for first year only, costs estimated was approximately $30,000 and $5,000 per manufacture new pump – assuming cheaper manufacturing with a Chinese casting foundry and machine shop [7]. The savings of roughly $5,500 for pump and $1,500 for free wheeling assemblies, totaled about $7,000 per gearbox are possible. Assuming DBT will sale at least 20 Gearboxes per year, DBT will save approximately $370,000 over the 10 year period of the underground coal mining boom as shown in Figure 11. This equates to a saving of $1,850 per gearbox which is about a 26% saving on the current set-up.

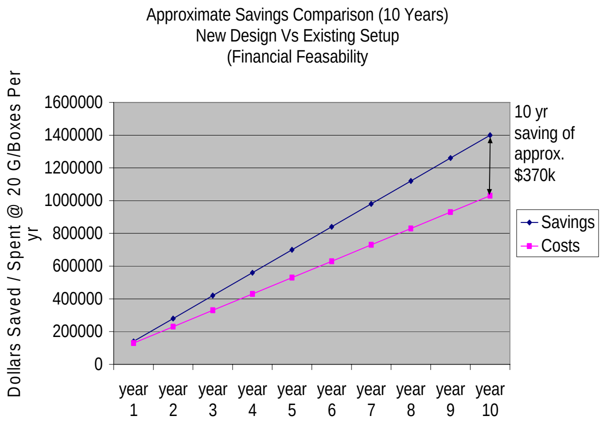


Figure 11: Financial Feasibility

Conclusions and Recommendations

This study assessed the feasibility of installing a bi-directional centrifugal pump into the 45 CST. A reversible centrifugal pump would best suit the clutch cooling system in the 45 CST. When comparing the financial capitol cost of manufacturing the pump and the long term savings of a reduced drive-train in the supply unit of the gearbox, manufacturing a new pump is definitely the more expensive option in the first couple of years. However, DBT will save approximately $370,000 over the 10 years which equates to a saving of $1,850 per gearbox which is about a 26% saving on the current arrangement. So it is concluded that making the change to a bi-directional centrifugal pump is a financially feasible option.

If the underground coal mining industry survives for at least 10 more years, the CST 45 gearbox will be most popular because of feasibility of a bi-directional pump. This means any improvements made to it would be large scale. Because the feasibility of the bi-directional pump installed on a 45 CST is concluded as 'reasonable', the author would recommend DBT puts further research into this first stage design.

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