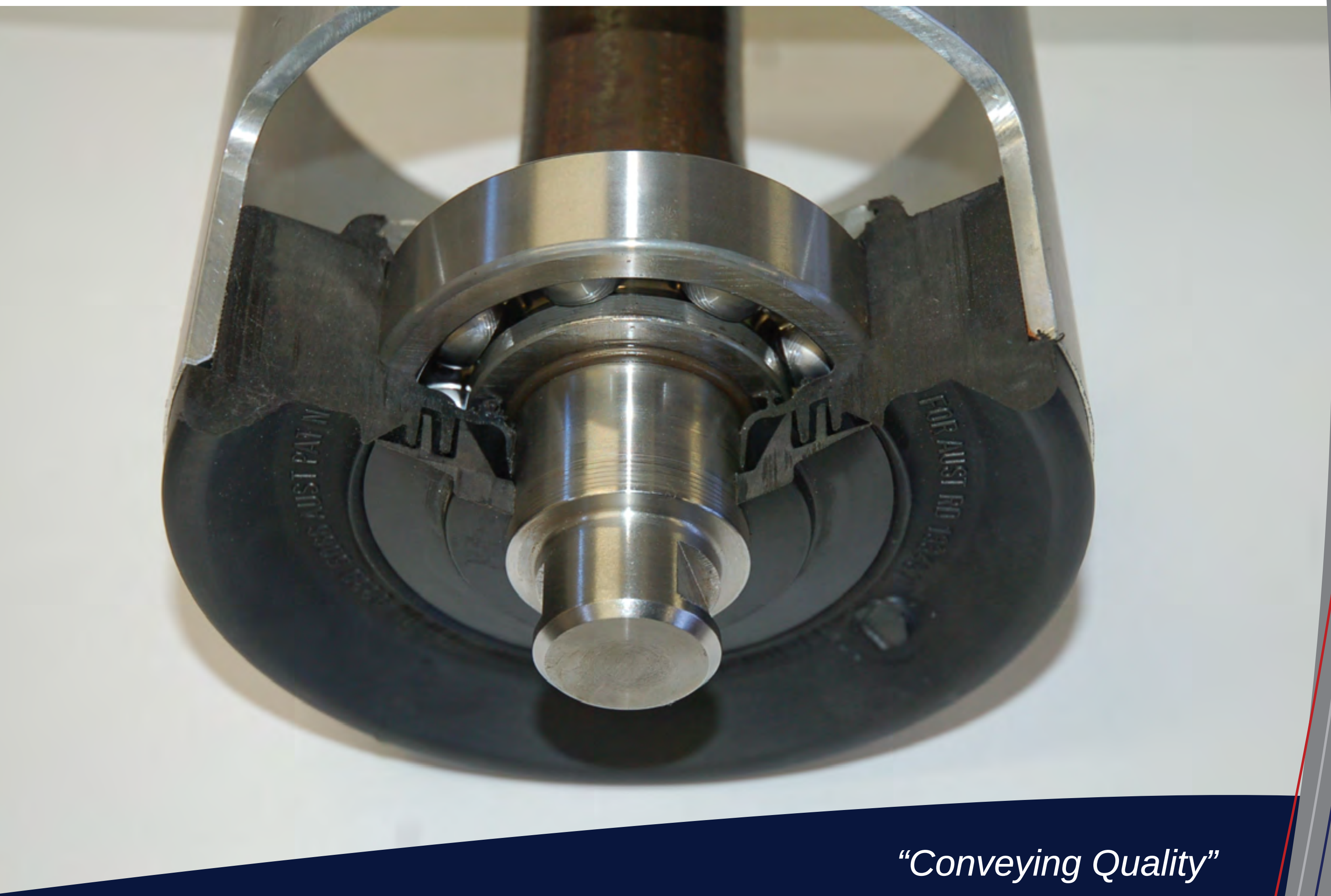
JLV Industries Pty Ltd



"Conveying Quality"

Glideseal Idler Roller Technical Analysis

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INTRODUCTION

JLV Industries Pty Ltd is proud of the developments made with the Glideseal Idler Roller. We have spent a lot of time improving the performance of the product. Our existing customers can see for themselves first hand the improvements shown by using Glideseal Idler Rollers.

This document is for potential customers and other interested parties to view some of the technical performance details of the Glideseal Idler Roller. Over the years the Glideseal Idler Roller has been subject to extensive testing by ourselves, our clients and by independent consultants. The reports contained within this document are all from independent consultants.

The Glideseal Idler Roller was specifically designed to provide a cost effective solution to inherent problems associated with conventional idler rollers. This has been proven at locations with harsh operating environments.

The main design characteristics of the Glideseal Idler Roller are;

- Efficient sealing arrangements - leading to extremely long service life.
- Low noise emissions - for applications in close proximity to personnel or populated areas.
- Ease of handling - providing a light weight safety friendly product.

CONSULTANTS DETAILS

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ALUMINIUM SHELL STRESS & FATIGUE

152mm OD Aluminium Idler Rollers Stress and Fatigue Analysis

Prepared By: ***John Shedley - Engineering Dynamics Consultants Pty Ltd***
Date: ***August 2005***
Report Number: ***EDC Report 663-01***

Executive Summary

JLV Industries Pty Ltd (JLV) is manufacturing a series of 152mm diameter idler roller designs consisting of an aluminium shell with moulded polymer end caps. As part of the design verification of these rollers, Engineering Dynamics Consultants Pty Ltd (EDC) was contracted to undertake a finite element analysis based stress and fatigue assessment of the aluminium shells for a limited number of design configurations. Analysis of two shell thicknesses' and two roller lengths was undertaken. This report presents the details of the analysis conducted and the results determined. The key results of the analysis are as follows:

• Peak static stress levels determined for each roller design investigated are below the allowable static stress level of 170 MPa for 6061 / 6005A Aluminium Alloys.

• The alternating stress ranges for the aluminium shells were also found to be less than the design limit of 70 MPa.

Based on these results, the available material properties and the applied loading the proposed idler designs were found to be acceptable from strength and fatigue considerations.

Objectives

The work scope and main objectives of this study were as follows:

• Construct finite element analysis (FEA) models of each of the four roller configurations being considered from the solid models and dimensional details supplied by JLV.
• Apply the supplied loading information to each model.
• Solve each model for stresses and deflections due to the applied load. Simulate the alternating stress experienced by the roller during rotation.
• Assess the strength and fatigue integrity of the aluminium shells.
• Additionally, compare the results of the FEA assessment to those determined through manual calculation to establish if empirical formulae can be derived to simplify evaluations of other shell designs.
• Prepare a report documenting the assessments undertaken and the results determined.

Technical Approach

Finite Element Model

Detailed finite element models of each of the four designs were developed using the I-DEAS finite element analysis software. The four model configurations had the following shell details:

1. 3.75mm wall thickness, 524 mm long.
2. 3.75mm wall thickness, 796mm long.
3. 5.00mm wall thickness, 524 mm long.
4. 5.00mm wall thickness, 796mm long.

Details of a typical polymer (Texin 255) end caps and approximate details of the bearing outer race and shaft were also included in each model to approximate actual support stiffness conditions for the shells. The models were developed from geometry supplied to EDC by JLV and consisted of a combination of linear brick and beam finite elements. Each model consisted of approximately 40,000 nodes and elements. Half models of each roller design were developed, utilising the symmetry of the rollers and their loading to reduce overall model sizes. The contact between the shell and end cap was modelled as a locked connection (i.e. no slip occurs). This is consistent with the results of a previous contact based analysis that showed no slip of these connections occurs.

Loads

JLV supplied two loading scenarios. These consisted of a uniform loading along the length of the roller of 383.33 kg/m and 690kg/m. These loads were applied as a parabolic pressure over an assumed contact arc of approximately 10mm.

Strength Assessment

The strength of the shell designs was assessed by comparison of the Von Mises stresses determined to the relevant material strengths as specified in AS/NZS 1664.2 Aluminium Structures Part 2: Allowable Stress Design. Based on clause 3.4 and the formulae in Table 3.4 (C) of AS1664 an allowable stress of 170 MPa was determined for both tensile and compressive stresses.

Fatigue Assessment

The assessment of the fatigue resistance of the aluminum shell was also made to the guidelines of AS1664.2. This requires weld and parent metal details to be classified, with appropriate fatigue life curves (S-N) curves being provided, for the assessment of cyclic stresses experienced by these details. In this instance no weld material is present in the design. Parent metal has been assessed as a Detail Category A, with an allowable design stress range of 70 MPa. This fatigue limit is independent of material grade. Stress ranges used for the assessment were established as the maximum change in either of the axial or circumferential stress. Plots of stress (in each direction) vs. rotational position were produced at the key locations at the middle and the end of the shell (for both inside and outside shell surfaces).

Results

Strength Assessment

Summary of Maximum Von Mises Stress (MPa)

Model Detail	Max Stress – Load 1 383 kg/m	Max Stress – Load 2 690 kg/m
3.75mm Wall x 500mm	19.2	34.5
3.75mm Wall x 800mm	24.3	43.8
5.00mm Wall x 500mm	11.7	21.1
5.00mm Wall x 800mm	14.3	25.7

The allowable stress for the strength assessment is 170 MPa. All designs are easily below this limit.

Fatigue Assessment

This table presents a summary of the longitudinal and circumferential stress ranges determined for each shell design for the highest load scenario. The allowable stress range for each design is 70MPa. The peak stress range determined was 67.5 MPa for the 3.75mm x 800mm long design.

Summary of Maximum Fatigue Stress Ranges (MPa) at Max. Load for each Design

Model Detail	Circumferential Stress Range	Longitudinal / Axial Stress Range
3.75mm Wall x 500mm	51.5	31.2
3.75mm Wall x 800mm	67.5	38.6
5.00mm Wall x 500mm	32.5	18.7
5.00mm Wall x 800mm	39.7	22.4

Conclusions

Considering the above results, the following conclusions are provided:

• Each shell design considered has acceptable static strength when compared to the requirements of AS1664.2.

• For each design the maximum alternating stress range is less than the allowable design stress range of 70 MPa, for all load cases.

• Resultant stresses are not readily determined from empirical formulae. Peak stresses are dominated by section 'squashing' effects rather than global bending stresses as might be expected.

END CAP INTERFERENCE ANALYSIS

152mm OD Aluminium Idler Rollers End Cap Interference Analysis

Prepared By: ***John Shedley - Engineering Dynamics Consultants Pty Ltd***
Date: ***August 2007***
Report Number: ***EDC Report 801-01***

Executive Summary

JLV Industries Pty Ltd (JLV) manufactures a series of 152mm diameter idler roller designs consisting of an aluminium shell with moulded polymer end caps. Engineering Dynamics Consultants Pty Ltd (EDC) has previously undertaken an assessment of the relative stress and fatigue performance of two alternate end cap designs, being a solid option and a hollow cored option, as detailed in EDC report number 663-01.

This report presents the results of some further comparative analysis of the two end cap designs considering the initial interference fit between the end cap and shell as well as that between the end cap and bearing outer raceway. The purpose of the analysis being to determine resulting contact conditions between the mating components and the resulting stresses given that starting tolerances differ between the two end cap designs.

Analysis was undertaken for both 6308 and 6309 bearing types installed in a 152mm OD x 708 mm long aluminium shell. A comparative review of the stresses, displacements and contact pressures determined was made for each bearing size.

The following key conclusions were made:

Shell Stresses
• Shell stresses in all designs are well below the acceptable level for strength.
• The maximum alternating stresses in the shell for all configurations were of similar magnitude and less than the design acceptable level of 70MPa.

End Cap Stress
• All stresses in the polymer end caps of each configuration are well below the tensile strength of 48MPa provided for the Texin 255 material.
• Alternating stress ranges determined for the plastic end caps are considered to be low for all designs, however due to the lack of specific fatigue properties of the materials adopted the ultimate acceptability of these levels could not be determined.

Contact Pressures
• The solid end cap design has a much more uniform and higher bearing contact pressure distribution than the hollow design. This would likely result in greater resistance to the bearing spinning and a larger margin for aging/creep of the material before spinning may initiate.

END CAP INTERFERENCE ANALYSIS

Objectives

The work scope and main objectives of this study were as follows:

- Construct finite element analysis (FEA) models of each of the four roller configurations being considered from the solid models and dimensional details supplied by JLV.
- Apply the supplied loading information to each model.
- Solve each model for stresses and deflections due to the applied load. Simulate the alternating stress experienced by the roller during rotation.
- Assess the strength and fatigue integrity of the aluminium shells.
- Additionally, compare the results of the FEA assessment to those determined through manual calculation to establish if empirical formulae can be derived to simplify evaluations of other shell designs.
- Prepare a report documenting the assessments undertaken and the results determined.

Technical Approach

Finite Element Model

Detailed finite element models of each of the four design configurations were developed using the I-DEAS finite element analysis software. The four model configurations were as follows:

1. 5.00mm wall thickness, 708 mm long, Solid End cap, 6308 bearing.
2. 5.00mm wall thickness, 708 mm long, Hollow End cap, 6308 bearing.
3. 5.00mm wall thickness, 708 mm long, Solid End cap, 6309 bearing.
4. 5.00mm wall thickness, 708 mm long, Hollow End cap, 6309 bearing.

The models were developed from geometry supplied by JLV and consisted of a combination of linear brick and thin shell finite elements. Each model consisted of approximately 85,000 nodes and 75,000 elements. Quarter models of each roller design were developed, utilising the two planes of symmetry of the rollers and the loading to reduce overall model sizes.

The contact between the shell and end cap and between the outer race of the bearing and the end cap was modelled utilising contact elements. The contact elements adopted were set up based on the actual level of interference present in the design, as supplied by JLV.

Boundary Conditions

In each model, nodal restraints were applied to the inner surface of the outer raceway. These restraints simulated the radial support of the bearing whilst allowing axial float and rotation. Appropriate symmetry boundary conditions were developed and applied on the two planes of symmetry of the idlers. These boundary conditions also ensured the model was adequately restrained from rotating and floating axially.

END CAP INTERFERENCE ANALYSIS

Contact conditions

Contact elements were used to define the interference fits between the shell – end cap and the end cap - bearing connections based on the unassembled component tolerance data supplied by JLV. A friction coefficient of 0.2 was assumed for each of the following interference joints considered:

Solid End Cap – 6308 Bearing
    Bearing to End Cap interface 0.5mm on radius
    Shell to End Cap interference 0.2mm on radius.

Hollow End Cap – 6308 Bearing
    Bearing to End Cap interface 0 mm on radius
    Shell to End Cap interference 0.55mm up to 0.7mm on radius.

Solid End Cap – 6309 Bearing
    Bearing to End Cap interface 0.5 mm on radius
    Shell to End Cap interference 0.2 mm on radius.

Hollow End Cap – 6309 Bearing
    Bearing to End Cap interface 0 mm on radius
    Shell to End Cap interference 0.55 mm up to 0.7mm on radius.

Strength Assessment

The strength of the shell designs was assessed by comparison of the Von Mises stresses determined to the relevant material strengths as specified in AS/NZS 1664.2 Aluminium Structures Part 2: Allowable Stress Design. Based on clause 3.4 and the formulae in Table 3.4 (C) of AS1664 an allowable stress of 170 MPa was determined for both tensile and compressive stresses.

Fatigue Assessment

The assessment of the fatigue resistance of the aluminum shell was also made to the guidelines of AS1664.2. This requires weld and parent metal details to be classified, with appropriate fatigue life curves (S-N) curves being provided, for the assessment of cyclic stresses experienced by these details.

Results

Shell Stresses

Stresses were determined for two conditions, being the interference effects alone and secondly the interference plus the applied load.

Summary of Shell Von Mises Stresses (MPa)

Model Detail	Stress Adjacent to End Cap - due to		Max. Stress – away from end cap
	Load + Interference	Interference fits only	Load + Interference
Solid End cap – 6308 Brg.	51.6	38.1	36.7
Hollow End cap – 6308 Brg. (1.1mm interference).	50.9	37.5 (42.4[1])	36.7
Solid End cap – 6309 Brg.	65.5	51.9	38.1
Hollow End cap – 6309 Brg. (1.1mm interference).	44.7	33.1	36.6

1. Peak stress

Summary of Shell Stress Results

• The allowable stress for the strength assessment is 170 MPa. All designs have stresses well below this limit.

• Alternating stresses have not been fully extracted and reviewed, however are considered to be less than the design guideline level of 70 MPa.

The highest cyclic stressed area of the shell is away from the end cap, consistent with the highest stressed region in the previous EDC analyses (EDC report 663-01). Further the maximum Von Mises stress determined in this area for the current analysis is 38.1 MPa (Solid - 6309 design) which is less than the highest fatigue stressed design in the previous analysis (at 43.8 MPa). Given all configurations in the previous assessment were acceptable for fatigue, the stresses from this analysis are also considered acceptable.

END CAP INTERFERENCE ANALYSIS

End Cap Stresses

Summary of Peak End Cap Von Mises Stresses (MPa)

Model Detail	Interference fits only	Load + Interference	Load Effect only (i.e. Alternating stress)
Solid End cap – 6308 Brg.	6.9	8.9	2.0
Hollow End cap – 6308 Brg. (1.1mm interference).	6.0	7.3	1.4
Solid End cap – 6309 Brg.	7.2	8.6	1.7
Hollow End cap – 6309 Brg. (1.1mm interference).	5.9	7.0	1.4

Summary of End Cap Stress Results

• The highest peak stresses of up to 8.9 MPa occur on the solid end cap - 6308 design. This is however a very localised effect (possibly a mesh related anomaly) with the nominal stresses similar or lower than those of the corresponding hollow designs at around 4 to 5 MPa.

• Alternating stresses of up to 2 MPa were predicted for the solid - 6308 design however this is also highly localised. Typical alternating stress levels are in the order of 0.5 to 1 MPa for all designs with the highest being on the Hollow - 6309 design.

Contact Pressures

The contact pressures between mating faces of the shell / end cap and end cap / bearing were compared to aid understanding of the likelihood of slip on these interfaces.

Summary of Contact Pressures (MPa)

Model Detail	Bearing Surface (nominal)	Bearing Surface (peak)	Shell Surface (nominal)	Shell Surface (peak)
Solid End cap – 6308 Brg.	4.8	9.3	1.5-3.0	5.8
Hollow End cap – 6308 Brg. (1.1mm interference).	0.7-3.3	6.6	2.0	7.2

END CAP INTERFERENCE ANALYSIS

Model Detail	Bearing Surface (nominal)	Bearing Surface (peak)	Shell Surface (nominal)	Shell Surface (peak)
Solid End cap – 6309 Brg.	5.4	9.5	1.5-4.0	7.1
Hollow End cap – 6309 Brg. (1.1mm interference).	0.2-3.6	8.8	2.5	7.2

Conclusions

Considering the above results, the following conclusions are provided:

Shell Stresses

• Shell stresses in all designs are well below the acceptable level for strength.
• Stresses in the shell of the solid end cap designs are higher than the corresponding hollow design as a result of the higher level of interference in the press fit of the bearings and the greater rigidity of the end cap design.
• The solid design with the 6309 bearing has the highest stresses as it has the least material in the end cap to accommodate the distortion introduced by the interference fits.
• The maximum alternating stresses in the shell of all configurations were of similar magnitude and less than the design acceptable level of 70MPa.

End Cap Stress

• All stresses in the polymer end caps of each configuration are well below the tensile strength of 48MPa provided for the Texin 255 material. A safety margin of at least 6 exists over the peak stresses determined. Nominal stresses in the solid design are generally similar or slightly lower than the hollow design, however the peak local stresses are higher (These are highly localised and may be partially mesh related anomalies).
• Alternating stress ranges determined for the plastic end caps are considered to be low for all designs, however due to the lack of specific fatigue properties of the materials adopted, the ultimate acceptability of these levels could not be determined.

Contact Pressures

• The solid end cap design has a much more uniform and higher bearing contact pressure distribution than the hollow design. This would result in greater resistance to the bearing spinning and a larger margin for aging/creep of the material before spinning may initiate.
• The magnitude and distribution of the pressure on the mating faces of the shell - end cap connection are much more consistent between designs. As such, similar joint performance could be expected between designs.

END CAP, SEALS, BEARING & GREASE ANALYSIS

Examination And Analysis Of Idler Components From Port Waratah Coal Services Limited

Prepared By: ***Dr. A Zurhaar - Zedcon Scientific Services***
Date: ***July 2009***
Report Number: ***090618***

Introduction

Six (6) 152.4 mm x 4mm - 910 mm galvanised steel shell idlers idler assemblies were received from JLV Industries for disassembly, examination and analysis. The idlers were labelled ”1-6” and have apparently been in service since 1999 on Conveyor #5.31 at Port Waratah Coal Services Limited’s Kooragang Coal Terminal. The conveyor belt width is 2500mm conveying coal at a maximum tonnage of 10,500 tph. The polyurethane end caps and seals, bearing assemblies and greases were forensically assessed for their condition and any evidence of deterioration or wear.

Condition Of Polyurethane End Caps And Seals

In all cases the polyurethane of the end caps and seals was in excellent condition with no evidence of deterioration or aging. The polymer has performed well and would still have a long service life of many years.

Effectiveness Of Dust Seals

The polyurethane seals which close the end caps and bearing housing to the axle have performed very well with five out of the six idlers examined showing seals that have been totally effective in preventing any ingress of fine coal dust which is clearly present in their operating environment. It is not possible to determine why the sixth idler experienced ingress but it is likely that the seal was mechanically disrupted from its seat causing an avenue for contaminant ingress. The design of the seal system is such that a properly seated seal will provide a totally effective barrier against any external ingress.

Grease Analysis

Samples of used grease were taken from one end cap of each of the six idler units and analysed for condition, wear metals and contaminants. A sample of new grease was also received for reference purposes. All of the used greases, including that of Idler No.6, showed no signs of deterioration or break-down with both chemical and physical properties being identical to that of the new grease. Analysis by Inductively Coupled Plasma - Mass Spectrometry techniques produced the following results.

END CAP, SEALS, BEARING & GREASE ANALYSIS

Analyte	New Grease	Idler No.1	Idler No.2	Idler No.3	Idler No.4	Idler No.5	Idler No.6
Iron (Fe)	<1	14	8	11	16	27	>1000
Lead (Pd)	<1	<1	<1	<1	<1	<1	<1
Copper (Cu)	<1	<1	<1	<1	<1	<1	<1
Molybdenum (Mo)	<1	<1	<1	<1	<1	<1	<1
Chromium (Cr)	<1	2	3	2	4	3	35
Aluminium (Al)	<1	<1	<1	<1	<1	<1	<1
Nickel (Ni)	<1	<1	<1	<1	<1	<1	<1
Silicon (Si)	8	10	13	11	9	8	25
Sodium (Na)	<1	<1	<1	<1	<1	<1	<1
Zinc (Zn)	<1	<1	<1	<1	<1	<1	<1
Calcium (Ca)	5	6	5	5	6	6	5
Magnesium (Mg)	<1	<1	<1	<1	<1	<1	<1

All results are expressed as mg/kg. Values of less than 100 mg/L may be considered insignificant and inconsequential in these greases. The iron and silicon results reflect the level of contaminant ingress that is visible in the bearing of Idler No.6. The levels of other metals indicate any wear condition. In all cases, with the exception of Idler No.6, there are no significant wear metals present. The elevated chromium level in the grease from Idler No.6 is consistent with the slightly elevated wear expected from such a high level of contamination with mineral particles.

END CAP, SEALS, BEARING & GREASE ANALYSIS

Condition Of Bearings

The bearing races and ball cages were sectioned from several idler axles in order to examine their wear state. All components of the bearing assemblies were examined under magnification to detect evidence of wear from service or accelerated wear and scoring due to dust ingress. For the five idlers that exhibited no contaminant ingress, the bearing races and balls were in excellent condition with no significant wear or scoring noted. For the bearing in Idler No.6, which was subjected to a heavy loading of mineral dust in the grease, the bearing assembly was still in fair condition with only minor wear. The low level of wear is testament to the effectiveness of the grease in suspending and lubricating an otherwise abrasive environment. On the basis of the bearing conditions, all of the idler units (except No.6) would have had a substantial service life of many years still left. We would not expect that it is possible to obtain better performance from a bearing system in this type of application. The grease being used and the end cap design is clearly providing an effective system to maximise service life.

Digital Images Of Components

Digital images were taken from the sectioned idlers and bearing assemblies. The images were taken of the assemblies at various stages of disassembly and cleaning.

Summary

With the exception of Idler No.6, all of the idler end cap and bearing assemblies that were submitted are in excellent condition and all have a substantial service life left. There was no evidence of dust ingress or any significant contamination in any of the bearings. All bearings were in excellent condition. The idlers submitted are understood to be approximately 10 years old. For idlers exhibiting no signs of grease contamination, we would expect that they have at least another 5 years of service life after which we recommend a thorough inspection and sampling of a further set of idlers to assess their wear condition.

Comparison of Conveyor Idler Noise Levels
DBCT Yard Conveyors - JLV Glideseal Vs Standard Steel Idlers

Prepared By: *Michael Caley - Ron Rumble Pty Ltd*
Date: *September 1999*
Report Number: *MC/99/3018.Rpt2*

Introduction

On 25th August 1999 our office undertook a series of noise measurements along DBCT yard conveyors, to compare noise levels produced by different types of conveyor idlers.

Testing was carried out on the following conveyors:

R4 Yard Conveyor fitted with standard steel idlers - loaded & empty
R5 Yard Conveyor fitted with JLV glide-seal idlers - loaded & empty

Other than the type of idler fitted, the R4 and R5 conveyors are comparable, having the following features in common:

1.4km length
1.6m belt width
Same design belt speed
Comparable structures with 45° troughs

Test Conditions

Measurements were conducted between 1145 and 1400 hours. Noise levels were measured when the conveyor was running stably and loaded, and when running empty.

Test Instrumentation

The test instrumentation consisted of

Precision sound level meter	Rion NA27 (S/N 0038 0650)
Portable calibrator	Rion NC73 (S/N 108 13 189)

CONVEYOR IDLER NOISE LEVELS

Test Procedure

JLV Glideseal idlers are fitted along the full length of the R5 yard conveyor. This conveyor has been in service for a period of approximately 1 year.

Standard steel idlers are fitted along the full length of the R4 yard conveyor. This conveyor has been in service for approximately 5 years.

There are two yard conveyors located on each bund, separated by a service road-way. Noise measurements for a particular conveyor were measured on the outer-side of the conveyor, and remote from operating stacker-reclaimers, so that measurements were uncontaminated by noise from other conveyors.

The measurement position was 1m horizontally from the top of the belt lip.

When the belt was empty, 30 second L_{Aeq} noise samples were taken at 5 locations along the belt separated by 30m intervals. Measurements when the belt was full were undertaken at 10 locations.

Test Results

The test results for the JLV Glideseal idlers are around 10dBA quieter than the results for standard steel idlers.

The correlation between dB noise level increases and the change in subjective loudness, as heard by the human ear, are as follows:

3dB increase	just noticeable
10dB increase	twice as loud

CONVEYOR IDLER NOISE LEVELS

Comparison of Noise Levels

Yard Conveyor	Averaged Noise Level (L_{eq} dBA)	
	Belt Empty	Belt Loaded
R4 (Standard Steel Idlers)	82.8	83.0
R5 (JLV Glideseal Idlers)	72.9	72.8

Averaged Octave-band test data

Yard Conveyor	Load Status	Overall Level (dBA)	A-weighted Octave Band Levels								
			31.5	63	125	250	500	1k	2k	4k	8k
R4	empty	82.8	37.0	56.5	68.4	73.4	78.9	76.1	75.2	66.7	57.3
R4	loaded	83.0	35.2	57.8	67.1	71.9	78.4	77.3	76.3	69.2	60.1
R5	empty	72.9	36.2	52.7	58.0	61.7	66.6	67.7	67.1	60.5	52.3
R5	loaded	72.8	34.6	51.6	58.0	61.1	66.0	67.8	67.0	61.0	54.5

ACCELERATED SALT CORROSION TESTING

Accelerated Salt Corrosion Testing of Idler Shaft And Stubs

Prepared By: ***Dr. A Zurhaar - Zedcon Scientific Services***
Date: ***October 2009***
Report Number: ***090911***

Introduction

Eight (8) samples were received from Mr Mark Maplesden of JLV Industries for accelerated salt spray corrosion testing. The sample batch consisted of the following;

Sample Identification	Item Description
1	Nylon shaft bush
2	Acetal shaft bush
3	304 stainless steel stub
4a,b	316 stainless steel stub (2off)
5	Powder coated shaft
6	316 stainless steel pushed on sleve
7	K1045 steel stub

The above items were to be subjected to accelerated corrosion testing in a salt spray chamber with the objective of assessing the tolerance ofthe materials to corrosion induced by salt spray attack.

Method

The samples were placed in a Heraeus Salt Fog Chamber and exposed to 500 hours of continuous neutral salt fog at 35°C in accordance with Australian Standard 2331.3.1. All items were digitally photographed after exposure to the salt spray.

Results

The salt spray test is a particularly aggressive method of inducing an accelerated corrosion environment and the fine mist is very penetrating to parts. After exposure, the samples were visually inspected and the following observations were noted;

Summary

On the basis of the tests conducted, the stainless steel stubs have shown virtually no signs of corrosion or salt attack. This was expected for the 316 grade. The 304 grade performed better than expected given this grade is generally considered less suitable for salt environments.

The nylon and acetal shaft bushes have both resisted corrosion but clearly the mild steel component has severely corroded as expected.

The powder coated shaft sample showed some resistance to salt attack but polyester powdercoats are generally only of limited protection in salt environments as saline vapour permeation of the coating still occurs.

The 316 stainless steel pushed on sleeve resisted corrosion whilst the underlying steel shaft severely corroded as expected.

The K 1045 steel stub was completely non-resistant to salt attack and accordingly was severely corroded.

EXAMINATION AND ANALYSIS OF IDLER COMPONENTS

Examination And Analysis Of Idler Components

Prepared By: ***Dr. A Zurhaar - Zedcon Scientific Services***
Date: ***April 2004***
Report Number: ***040414***

Introduction

Two (2) idler assemblies were received from Mr D. Sealey of JLV Industries for disassembly, examination and analysis. The idlers were labelled "A" and "B" and have been in service since March 2001 on Conveyor P106 at bhp billiton Nelson Point, Port Hedland. The polyurethane end caps, bearing assemblies and greases were throughly assessed for their condition and any evidence of deterioration or wear.

Condition Of Polyurethane End Caps

In all cases the polyurethane of the end caps was in excellent condition with no evidence of deterioration or aging. The polymer has performed well and would still have a long service life of many years.

Effectiveness Of Dust Seals

The polyurethane seals which close the end caps and bearing housing to the axle have performed very with all seals having been totally effective in preventing any ingress of the fine iron oxide dust which is clearly present in their operating environment. All of the greases showed no visible evidence of dust ingress whatsoever.

Wear Condition Of Aluminium Idler Casing

The wall thickness of the aluminium idler casings was measured along its length with a calibrated vernier gauge to assess any reduction in thickness due to wear. The manufactured wall thickness is 5.00mm ± 0.38mm. The following measurements were recorded for each idler casing;

Idler "A"	4.92	4.93	4.94	4.94	4.93	4.92	4.92	4.94	4.92	4.92
Idler "B"	5.03	5.06	5.06	5.06	5.07	5.07	5.07	5.07	5.07	5.06

The above results show that both idler casings exhibit no significant reduction in wall thickness after three (3) years of service.

Grease Analysis

Samples of used grease were taken from one end cap of each of the two idler units and analysed for condition, wear metals and contaminants. A sample of new grease was also received for reference purposes. The used greases showed no signs of deterioration or break-down with both chemical and physical properties being identical to that of the new grease. Analysis by Inductively Coupled Plasma – Mass Spectrometry techniques produced the following results.

Analyte	New Grease	Idler “A”	Idler “B”
Iron (Fe)	<1	7	5
Lead (Pb)	<1	<1	<1
Copper (Cu)	<1	1	1
Molybdenum (Mo)	<1	<1	<1
Chromiun (Cr)	<1	2	2
Aluminium (Al)	3	4	3
Nickel (Ni)	<1	<1	<1
Silicon (Si)	13	15	12
Sodium (Na)	<1	<1	<1
Zinc (Zn)	<1	<1	<1
Calcium (Ca)	5	6	5
Magnesium (Mg)	<1	<1	<1

All results are expressed as mg/kg. Values of less than 100 mg/L may be considered insignificant and inconsequential in these greases. The iron and silicon results reflect the level of dust ingess whilst the levels of other metals indicate any wear condition. In all cases there are no significant wear metals present. There is no evidence of significant bearing wear in any of the greases.

Condition Of Bearings

The bearing races and ball cages were sectioned from the idler axles in order to examine their wear state. Two sets of sectioned bearings have been forwarded with this report for your reference. One set represents Idler “A” and the other set represents Idler “B”.

All components of the bearing assemblies were examined under magnification to detect evidence of wear from service or accelerated wear and scoring due to dust ingress. In all cases, the bearing races and balls were in excellent condition with no significant wear or scoring noted. On the basis of the bearing conditions, all of the idler units would have had a substantial service life of many years still left. We would not expect that it is possible to obtain better performance from a bearing system in this type of application. The grease being used and the end cap design is clearly providing an effective system to maximise service life.

Summary

All of the idler end cap and bearing assemblies that were provided are in excellent condition and all have a substantial service life left. There was no evidence of dust ingess and contamination in any of the bearings. All bearings were in excellent condition. The grease is designed to be very effective in suspending any ingressed dust and protecting the bearings from wear, however, in the case of these idlers the polyurethane seal is so effective that no contamination was able to occur..

This assessment has shown that after approximately 3 years of service, all materials of the idler units are in excellent condition and all have a substantial service life left as wear and deterioration has been insignificant in all respects.

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