40 TONNE LOCOMOTIVE DRIVELINE UPGRADE

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Abstract

THE AUSTRALIAN SUGAR industry has more than 50 DH and 73 Class ex- Government nominal 40 tonne locomotives in operation. The first of these locomotives began service after conversion to cane field specification in 1991. At the time, the conversion of these shunting locomotives was an effective strategy to realise additional capacity in a short time frame. It was understood that a compromise with this strategy was that there was no transmission 'lock-up' function with the Voith transmission supplied with these locomotives, whereas cane haul duty requires 'lock-up' for fuel efficiency. This compromise was acceptable while the Voith transmission was reliable but the transmissions are now no longer reliable. This issue has provided an opportunity to re-design the drivelines for these locomotives to be better matched to cane field duty and thereby realise significant savings in fuel burn and improved locomotive reliability. The Voith transmission is being replaced with an Allison 4000 Series retarder model transmission coupled to a Renk-Maag vertical offset reversing transmission. The re-designed transmission can now achieve an efficiency of up to 96%, which compares with the Voith transmission highest efficiency of around 60%. The improvement in transmission efficiency has allowed the diesel engine to be downsized from a twelve cylinder 23.9 L engine to an eight cylinder 15.9 L engine (33% reduction in engine displacement). Any improvement in the 40 tonne locomotive efficiency is significant for the Australian sugar industry, as these locomotives account for more than 50% of the industry cane haul capacity. The potential for up to 15% locomotive fuel savings for the Queensland sugar industry when this project is fully implemented, combined with reduced whole life costing, is of great interest to cane transport system operators, as is the reduction in exhaust emissions combined with an expected lower radiated noise from the power module. The prototype repower is currently being installed in the locomotive 'Piralko' to begin operation in the 2017 season. A replacement program has been scheduled for the remaining 23 of Wilmar's 40 tonne locomotives, following the successful installation and testing of the prototype.

Introduction

The first two 40 tonne locomotives were introduced into the Australian sugar industry in 1991 with Victoria Mill receiving 'Clem McComeske' and Isis Mill 'No 1'. The base 40 tonne locomotives from which these were built were sourced from Queensland, NSW and Western Australian Rail. At the time, the Queensland sugar industry had a number of factors creating a requirement for improving cane haulage capacity at various mills.

These factors included:

- Increased crushing rates due to increases in assigned cane growing areas.
- Changes from Queensland Rail haulage to mill rail haulage.
- Ageing locomotive fleets.

- A need to improve the efficiency of the cane transport system, which promoted use of larger locomotives with longer rakes of wagons and in some cases using slave locomotives.

A number of modifications were required to make these locomotives suitable for cane haul operations including:

- Re-gauging to narrow gauge.
- Weight reduction.
- Engine replacement.
- Transmission gear ratio changes.
- Braking system changes.
- Cooling system changes.
- Driver cab improvements.
- Upgrading of control system.

The general arrangement of a 73 Class locomotive converted for cane field duty is shown in Figure 1.

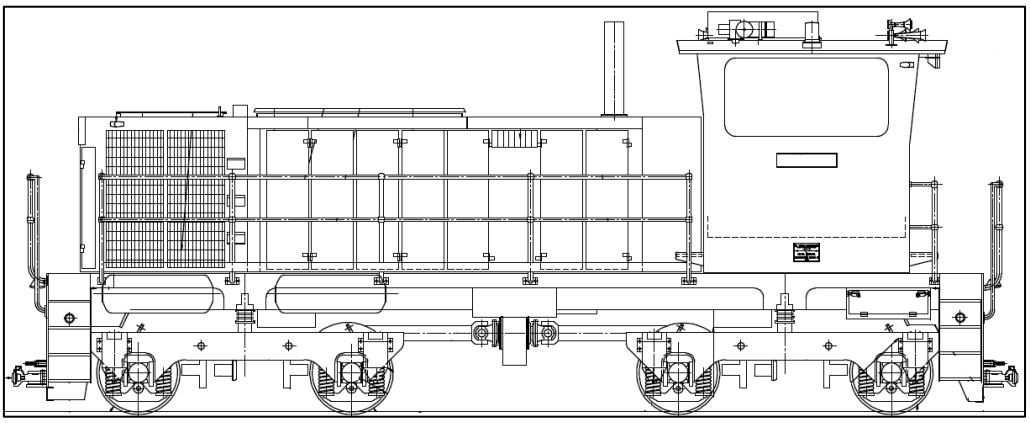


Fig. 1—General arrangement 73 Class cane field locomotive.

The need to repower

Wilmar Sugar has 13 × DH Class locomotives with the Voith L420rU2 transmission and 11 × 73 Class locomotives with the Voith L4r4U2 transmission. The Voith L420rU2 is a hydrodynamic turbo transmission with a mechanical reverse whereas the Voith L4r4U2 is a hydrodynamic turbo reversible transmission. This means that the Voith L4r4U2 transmission is more advanced in that the operator can select turbo reversing braking drive while moving.

The Voith L420rU2 and L4r4U2 transmissions (which ceased production in the 1970s) are no longer supported with a full complement of parts and the turn-around time for a transmission overhaul can be up to two years.

The efficiency of the Voith transmissions were at best about 60%–this being a constraint due to ‘torque convertor only’ operation. Whilst convertor operation is well-suited to a shunting duty, it is not efficient on main line haul duty and creates higher fuel consumption than would otherwise be the case with the more efficient ‘lock-up’ drive transmission.

Functional requirements for the driveline

The DH and 73 Class locomotives require the same or better tractive effort as available with the previously fitted engine coupled to the Voith transmission. Additionally, the driveline total life-cycle cost must be better than the current installation, with reliability being a significant consideration. To address reliability; the minimum number of high quality parts, easily maintainable at a remote sugar mill location, was a primary objective.

The original functional requirements were written with the assumption that the existing MTU 12V2000 engine at 567 kW would be required to maintain the required tractive effort for 40 tonne locomotives. After a review that considered the efficiency gains that could be achieved by using an Allison transmission (given its 'lock-up' drive, through the torque convertor), it was calculated that an engine of 448 kW would be sufficient to provide equivalent tractive effort when compared with the existing configuration.

Initial concept

The initial concept was to retain the industry-proven MTU 12V2000 engine with the capability of realising in excess of 30 000 hours service life. The transmission to suit this engine power output is the Allison 6000 Series, six speed transmission, with a convertor lockup clutch and an input retarder. This transmission package is well proven in earthmoving applications and would be suited to the locomotive duty. To enable the same speed in both directions of travel, a custom built reversing box was required. The concept is shown in Figure 2.

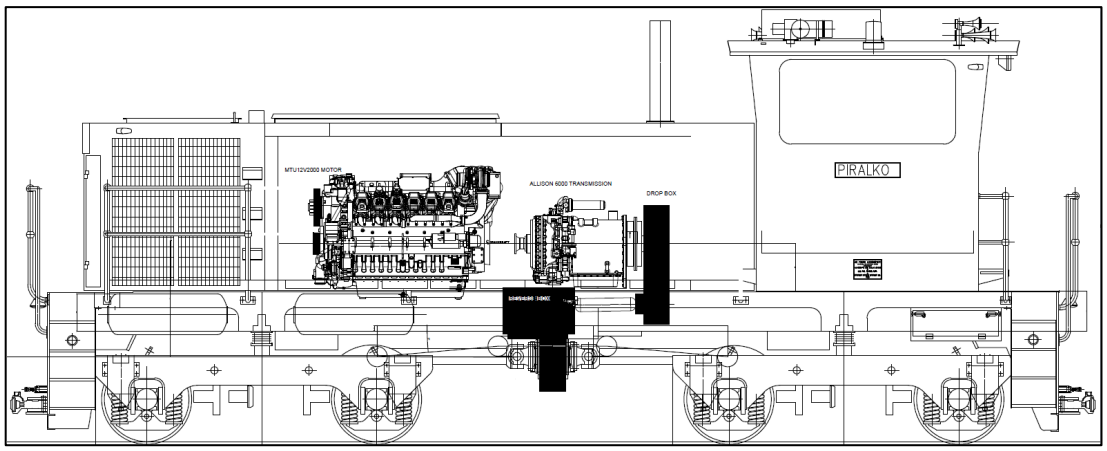


Fig. 2—12V2000 engine Allison 6000 series transmission and reversing box.

Considerations with this concept were:

- The engine and transmission are both limited production volume components which makes them relatively expensive.
- The available chassis length of the 40 tonne locomotive was marginal, when considering the overall length of the coupled-up package, with allowances for drive shafts.
- The Allison off-highway earthmoving transmissions do not have the 'smooth shifting' characteristic of the on-highway transmissions and this could lead to the steel wheel breaking traction on the steel rail when downshifting at high power settings.
- The retarder rotor is at the front of the transmission. This means the retarder rotates in relation to engine speed rather than in strict proportion to wheel speed because of the effect of gear selection. This variability could become an issue in operation and an example would be when transiting through downshifts while in retarder mode.

In summary, although this driveline was a robust and proven proposition, it was also expensive, not very fuel efficient and would transmit impact loading from the shifting process through the drive train.

Revised concept

The revised concept was to take advantage of the higher efficiency of the transmission to reduce the engine rating from 567 kW (a 23.9 litre V12 MTU unit) to a 448 kW (15.9 litre V8 Mercedes-Benz unit).

The transmission to suit this engine power output is the Allison 4000 Series, six speed transmission, with convertor lock-up clutch and output retarder. Again, to enable the same speed in both directions of travel, a custom built reversing box was required. This concept is shown in Figure 3.

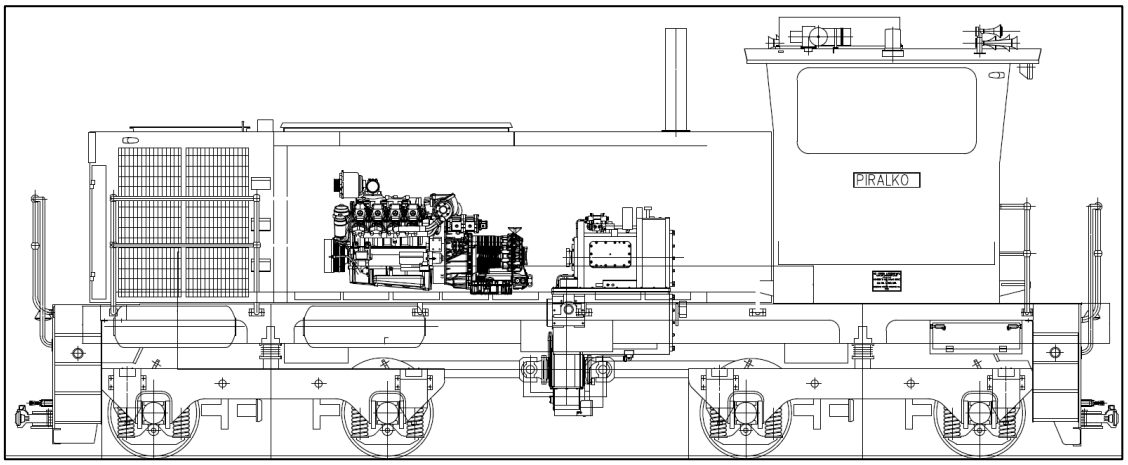


Fig. 3—OM 502 Engine, Allison 4000 series and reversing box.

Considerations with this concept were:

- The engine and transmission are both relatively large build volume components which reduces their unit cost.
- The available chassis length of the 40 tonne locomotive was more accommodating, given the reduced length of the package.
- The Allison on-highway earthmoving transmissions have adaptive shifting that provide smooth shifting characteristics and this is important to maintain adhesion during shifts at high power settings for steel wheels on steel rails.
- The 4000 Series retarder rotor is located on the transmission output shaft which implies that it rotates in direct proportion to wheel speed; therefore retardation is smooth and controllable through the entire speed range.

In summary, this driveline is readily available, more cost effective, easier to adapt to the available envelope, has advanced technology to provide less driveline impact through the shifting and retard range and is more fuel efficient.

Based on the two options for the driveline to achieve the tractive effort and performance requirement from the DH and 73 Class locomotives at the least business cost, the conclusion was clearly in favour of the 448 kW Mercedes-Benz/Allison 4000 series power package.

Engine details

The engine selected to upgrade the drivelines of the sugar industry 40 tonne locomotives was the Mercedes-Benz OM502 four stroke, direct injection diesel engine (Figure 4). This engine is a 90 degree V8 448 kW, 2 700Nm (15.9 litre) diesel engine with electronic engine management and twin turbochargers with charge air cooling.

The engine construction is a grey cast iron crankcase with wet type cylinder liners and single grey cast iron cylinder heads with four valves per cylinder (Mercedes-Benz, 2014).

This engine is widely used throughout the industrial world, with tens of thousands being manufactured each year.

Australia currently has no off-highway emission regulations. The Mercedes-Benz OM502 engine is EU Stage II emission compliant and is the better engine for greenhouse gas emissions because of the low fuel consumption compared with the current MTU 12V2000 EU Stage I engine.

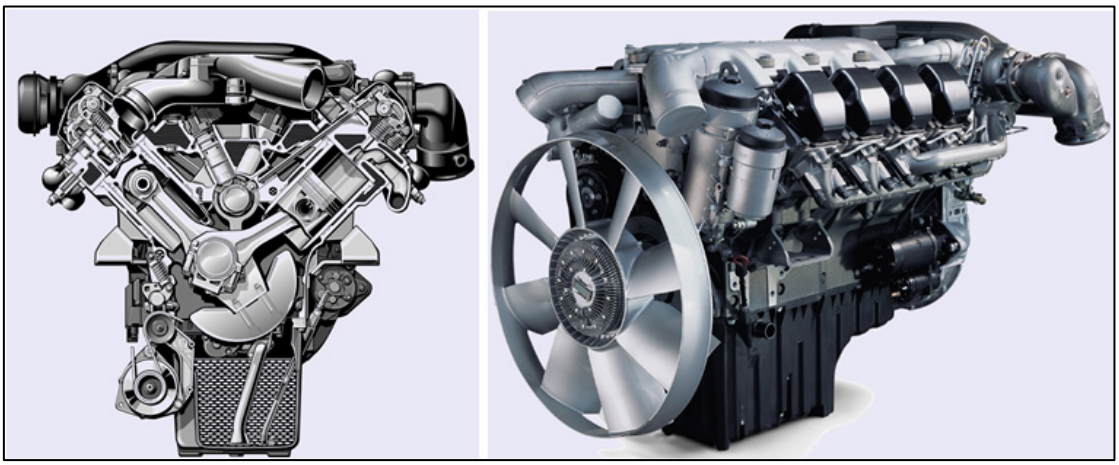


Fig. 4—Mercedes-Benz OM 502 engine.

With EU Stage II, the NOx and particulates are not as strict as the latest standard, but there is a requirement for after treatment (Selective Catalytic Reduction) to meet Stage IIIB. With reference to Figure 5, the EU Stage II is a significant reduction in emissions from the EU Stage I emission outputs of the current MTU 12V2000 and this reduction is achieved while maintaining simple engine architecture to deliver the most fuel efficient and clean burning variant.

Additionally the Mercedes-Benz OM502 is expected to deliver a lower radiated noise compared with the existing MTU 12V2000 engine.

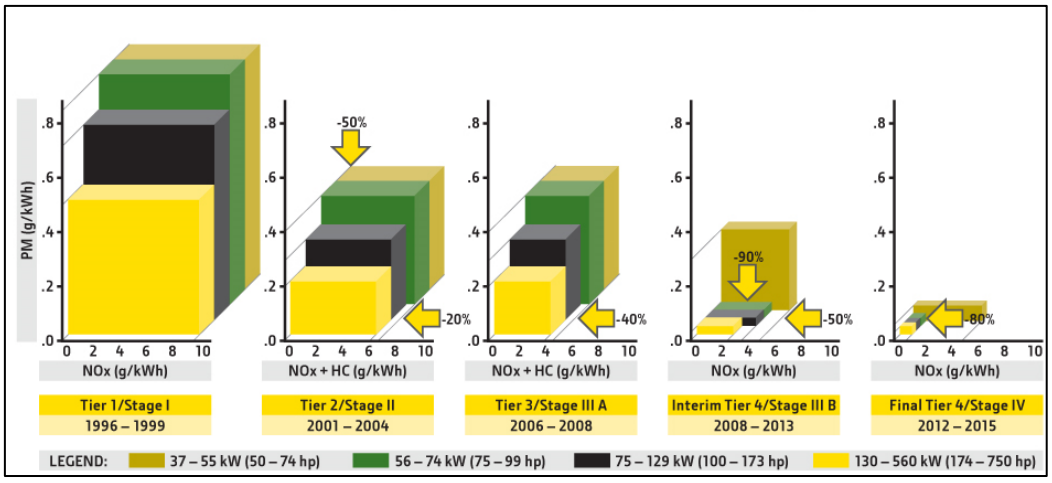


Fig. 5—EPA and EU non-road emissions regulations 37–560kW. (John Deere, 2016).

The Mercedes-Benz OM502 engine has been specified at the factory level specifically for the 40 tonne locomotive duty and locally modified for the application with the following changes:

- 140A high mount alternator.
- Torsional drive for fan motor.
- Rear mounted Power Take Off (PTO) for reversing box lubrication and clutch pump.
- SAE1 flywheel housing with Allison flywheel.
- Top mounted charge-air connections.
- Poly-Vee serpentine belt drive for water pump, Freon compressor and alternator drive.
- Twin cylinder air compressor with gear drive.
- Low profile sump.
- Locally mounted Freon compressor specifically chosen for this application.

The B10 service life of this engine is expected to be in the order of 20 000 hours in this application.

Transmission details

The transmission selected to upgrade the drivelines of the sugar industry 40 tonne locomotives was the Allison Series 4000 automatic transmission (Figure 6). This transmission is also widely used in industry throughout the world with tens of thousands being manufactured each year.

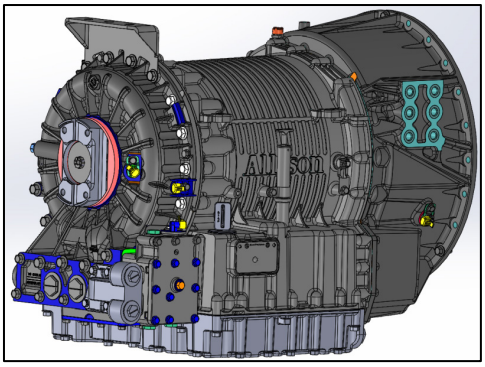


Fig. 6—Allison 4000 SPR automatic transmission.

The transmission is fully automatic with electronic controls and torque converter drive. There are several options available with a range of speeds, PTO drive provision, retarder provision and oil sump configurations (Allison On-Highway Transmissions 2008). The shift sequence option selected for this application was 1C,2C,2L,3L,4L,5L,6L (C = converter, L = lockup) with the close ratio gear set.

A flexplate drive is used to transfer engine torque to the transmission and the transmission is mounted directly to the engine. The torque converter, which forms a part of the transmission, consists of three primary elements being the pump, stator and turbine (Figure 7). The converter pump (connected to the engine) acts as a centrifugal pump as it absorbs power from the engine, forcing transmission fluid to circulate as a function of its rotational speed. The circulating fluid transmits power to the turbine causing it to rotate. As the fluid leaves the turbine, the stator redirects the fluid back to the pump in the same direction as the pump rotation, resulting in multiplication of the torque. Various torque multiplication options are available and the torque converter multiplication selected at stall for this application was 1.58.

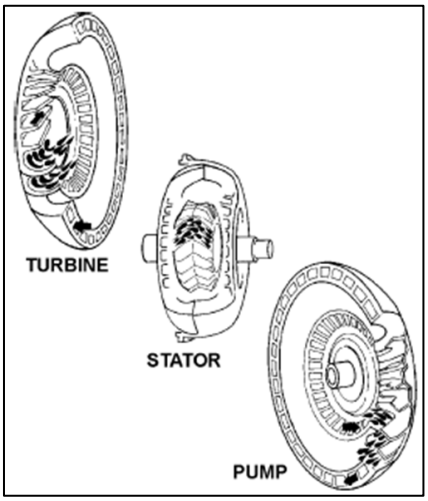


Fig. 7—Torque converter elements.
(Allison On-Highway Transmissions, 2006).

A standard feature of the transmission is the torque converter clutch which, when engaged, locks the torque converter pump and turbine together, enabling them to rotate in unison at engine speed.

This condition is commonly referred to as 'lock-up' and provides high efficiency, direct drive through to the transmission gearbox. Lock-up drive enhances fuel economy and engine braking. The torque converter clutch is regulated by the shift controls to engage and disengage automatically.

The transmission includes three planetary sets of constant mesh, helical gears and five clutches, which are used to direct flow of torque through the transmission. By engaging various combinations of clutches, the planetary sets act singularly or in combination to produce six forward gears, neutral and one reverse gear.

The control module, or transmission sump, provides the enclosure for the bottom of the transmission, houses the transmission control components, contains provisions for servicing the transmission fluid and filters, and acts as the reservoir for the transmission fluid.

The transmission uses a microprocessor-based control system that can operate with either 12 volt or 24 volt DC systems. A transmission mounted control valve body receives commands from the transmission control module (TCM) to control the hydraulic operation of the transmission.

The 4000 Series transmission is designed to be capable of being overhauled and it is expected that the transmission life will be similar to the engine life, at 20 000 hours.

Tractive effort calculations

Simulations were run using the web-based Allison iSCAAN tool, which predicts locomotive speed on grade, engine speed, tractive effort, drawbar pull, wheel power, gradeability and transmission heat rejection.

The Allison iSCAAN program is an application tool that calculates and analyses vehicle performance. The program considers the following factors:

- Engine.
- Converter.
- Engine/converter match.
- Transmission characteristics.
- Vehicle and environmental factors.

Once the user inputs the required data e.g. vehicle mass and dimensions, weight on drive wheels, engine torque curve (user defined variables, such as peak torque, peak power and governed speed) driveline details (drop box ratio, final drive ratio, wheel diameter), iSCAAN generates reports describing various vehicle operational and design parameters.

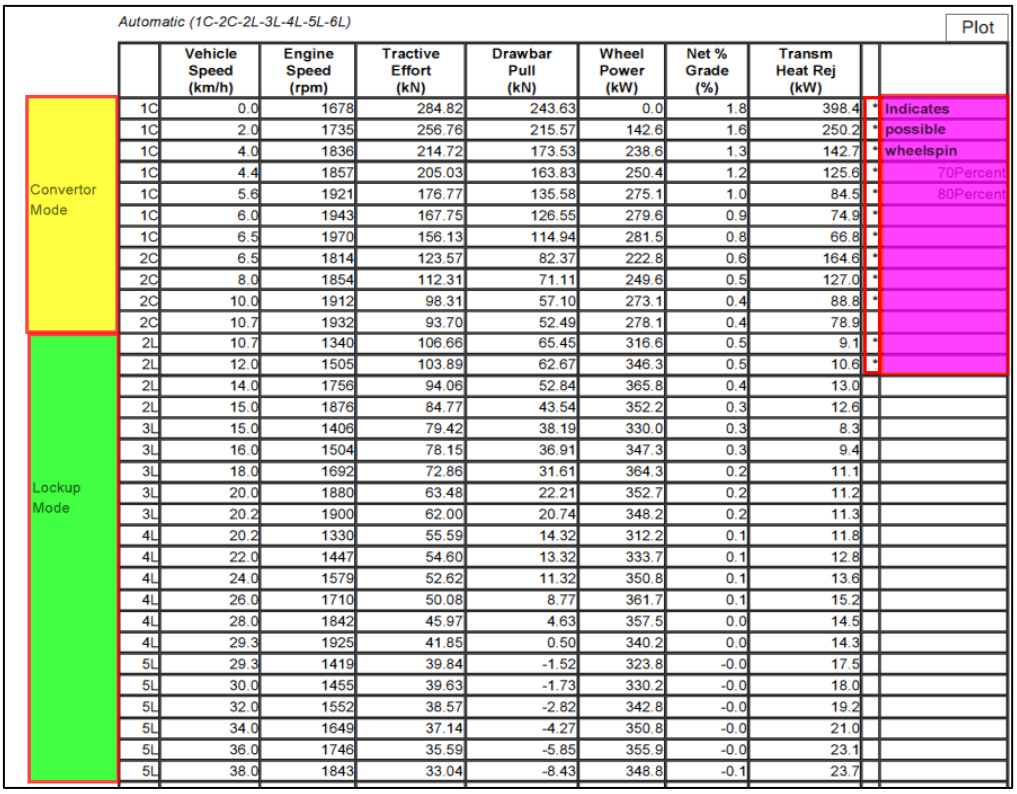
Utilising the user-entered data, iSCAAN predicts the vehicle's launch gradeability, low speed gradeability and also a maximum speed on grade. Launch gradeability is defined as the vehicles ability to start from rest on an upgrade and low speed gradeability is the vehicles ability to maintain speed on an upgrade when under full power and locked up in first gear. Additionally, the vehicle full throttle performance in each range is predicted.

During full throttle performance, iSCAAN determines vehicle speed compared with engine speed, tractive effort, wheel power and drawbar pull in kN. Transmission heat rejection is calculated at nominated operation points as well as 70% and 80% converter efficiency points, which are the standard slip conditions used to determine cooling package requirements.

Wheel slip is predicted on preloaded traction coefficients (0.4 for steel on steel) to ensure that the vehicle can break traction and so minimise the probability of the occurrence of a transmission overheat caused by a continuing stall condition.

The Allison iSCAAN program is designed for on-highway vehicles. This tool is also suitable for the locomotive market, though friction resistance around curves is not considered in the performance calculations.

However, factors such as wind resistance and rolling resistance are accounted for. The Allison iSCAAN also allows for parasitic losses (fan power, implement drives) and driveline efficiencies (user defined or in-built percentages) to ensure the prediction is reasonably close to reality. A typical output from the Allison iSCAAN tool for this duty is shown in Figure 8.



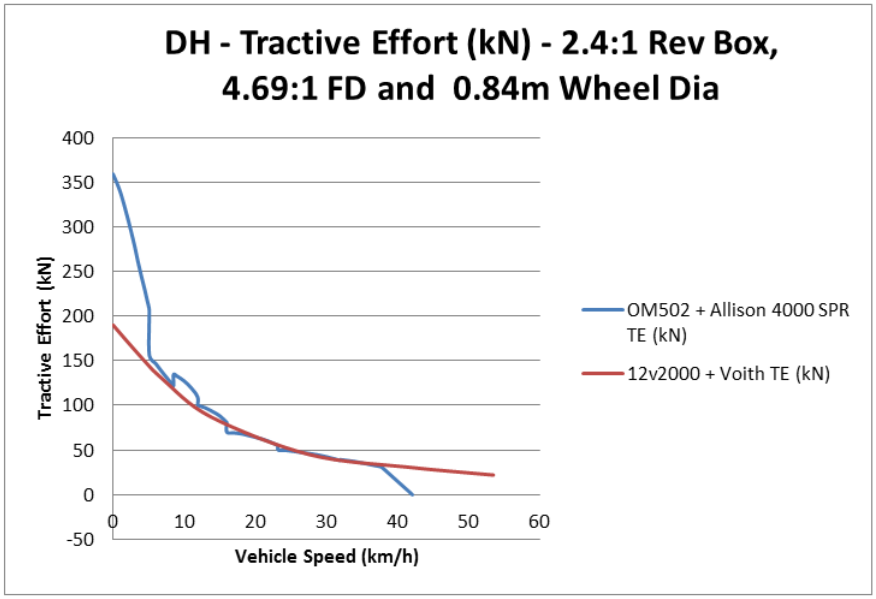
Automatic (1C-2C-2L-3L-4L-5L-6L)

Plot

		Vehicle Speed (km/h)	Engine Speed (rpm)	Tractive Effort (kN)	Drawbar Pull (kN)	Wheel Power (kW)	Net % Grade (%)	Transm Heat Rej (kW)		
Convertor Mode	1C	0.0	1678	284.82	243.63	0.0	1.8	398.4	*	Indicates
	1C	2.0	1735	256.76	215.57	142.6	1.6	250.2	*	possible
	1C	4.0	1836	214.72	173.53	238.6	1.3	142.7	*	wheelspin
	1C	4.4	1857	205.03	163.83	250.4	1.2	125.6	*	70Percent
	1C	5.6	1921	176.77	135.58	275.1	1.0	84.5	*	80Percent
	1C	6.0	1943	167.75	126.55	279.6	0.9	74.9	*	
	1C	6.5	1970	156.13	114.94	281.5	0.8	66.8	*	
	2C	6.5	1814	123.57	82.37	222.8	0.6	164.6	*	
	2C	8.0	1854	112.31	71.11	249.6	0.5	127.0	*	
	2C	10.0	1912	98.31	57.10	273.1	0.4	88.8	*	
	2C	10.7	1932	93.70	52.49	278.1	0.4	78.9	*	
Lockup Mode	2L	10.7	1340	106.66	65.45	316.6	0.5	9.1	*	
	2L	12.0	1505	103.89	62.67	346.3	0.5	10.6	*	
	2L	14.0	1756	94.06	52.84	365.8	0.4	13.0		
	2L	15.0	1876	84.77	43.54	352.2	0.3	12.6		
	3L	15.0	1406	79.42	38.19	330.0	0.3	8.3		
	3L	16.0	1504	78.15	36.91	347.3	0.3	9.4		
	3L	18.0	1692	72.86	31.61	364.3	0.2	11.1		
	3L	20.0	1880	63.48	22.21	352.7	0.2	11.2		
	3L	20.2	1900	62.00	20.74	348.2	0.2	11.3		
	4L	20.2	1330	55.59	14.32	312.2	0.1	11.8		
	4L	22.0	1447	54.60	13.32	333.7	0.1	12.8		
	4L	24.0	1579	52.62	11.32	350.8	0.1	13.6		
	4L	26.0	1710	50.08	8.77	361.7	0.1	15.2		
	4L	28.0	1842	45.97	4.63	357.5	0.0	14.5		
	4L	29.3	1925	41.85	0.50	340.2	0.0	14.3		
	5L	29.3	1419	39.84	-1.52	323.8	-0.0	17.5		
	5L	30.0	1455	39.63	-1.73	330.2	-0.0	18.0		
	5L	32.0	1552	38.57	-2.82	342.8	-0.0	19.2		
	5L	34.0	1649	37.14	-4.27	350.8	-0.0	21.0		
	5L	36.0	1746	35.59	-5.85	355.9	-0.0	23.1		
	5L	38.0	1843	33.04	-8.43	348.8	-0.1	23.7		

Fig. 8—Allison iSCAAN Tool Prediction (typical output).

The tractive effort curves for the DH with a 4.69:1 final drive ratio and 73 Class with a 4.34:1 final drive ratio were calculated using the iSCAAN tool. This includes running simulations with new 840 mm diameter wheels and worn 760 mm diameter wheels. Small variations in the tractive effort (TE) curve can be seen in Figure 9, as the transmission changes gears.



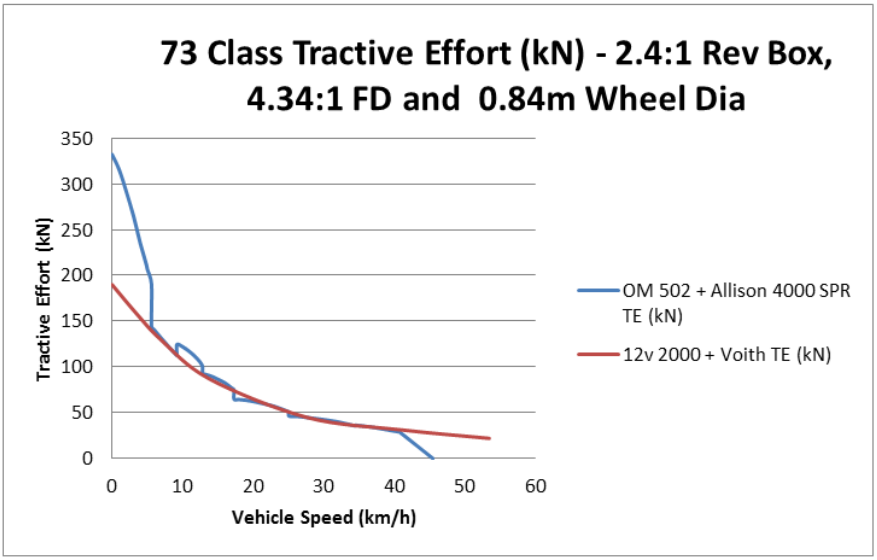


Fig. 9—Tractive effort predictions.

A point of note is that the tractive effort for the OM502–Allison 4000 combination is superior to the existing 12V2000–Voith driveline below 5km/h.

For reference, the gear ratios for the Allison 4000 transmission are shown in Table 1.

Table 1–4000 SPR transmission gear ratios.
(Torque converter multiplication of 1.58 at stall).

Gear range	Close ration (CR) models
Forward	
First	3.51
Second	1.91
Third	1.43
Fourth	1.00
Fifth	0.74
Sixth	0.64

The final assembly of the Mercedes-Benz OM502 to the Allison 4000 SPR is shown in Figure 10.

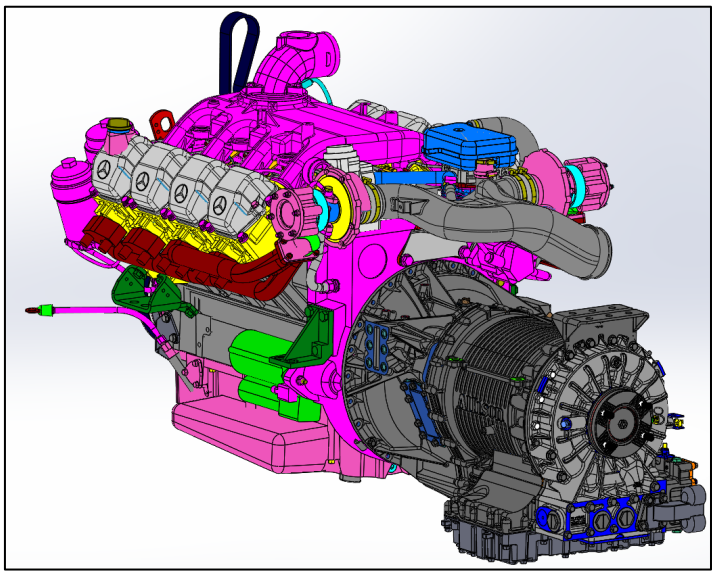


Fig. 10—Mercedes-Benz OM502 to Allison 4000 SPR assembly.

Reversing gearbox design process

A Swiss company, Renk-Maag GmbH, was engaged to design and build the reversing box, which was specified to bolt in to the original Voith transmission mount position (both DH and 73 Class via the use of spacer plates) and retain the existing driveshaft connections. iSCAAN was used to optimise the gear ratio for the reversing box so as one transmission could fulfil the 73 Class requirements and the DH requirements with both new and worn wheels. The optimising process delivered a ratio requirement of 2.4:1 reduction.

The agreed functional requirements given to Renk-Maag were:

- Allison 4000 torque and speed output.
- Typical cane field locomotive duty cycle.
- 30 000 hour bearing life, 90 000 hour design gear life.
- Bearings and seals–off the shelf with no special tools required for maintenance or disassembly.
- Internal oil reservoir.
- Connection points concentrated on the top of the gearbox for ease of access.

In general the process of establishing a new designed gearbox concept is a complex process and requires several iterations at each stage of the design process. The final target is that all parts work together in harmony to deliver an optimised operation in accordance with the agreed list of requirements.

Elements that require consideration early on in the iterative design process are material selection, the choice of lubricant and the selection of bearings. For the cane field locomotive duty, simple and robust is best, provided that all other design objectives have been satisfied and size and mass are liberal constraints. With material selection, the choice of the same material for the through-hardened shafts and gears was made to simplify the manufacturing process and reduce cost, although the larger components resulted in more mass than required. The external gearbox envelope had sufficient allowance to accommodate this decision.

The lubrication oil selected was ISO VG 220 mineral oil for its high load carrying capacity at low speed and high load, on gear mesh and roller bearing and for its compatibility with seals. The roller bearings were selected based on their capability at low speed, high load and high speed with minor power loss and less lubrication required than for journal bearings.

Renk-Maag used a consultative design approach of seeking input on design choices to arrive at a transmission that would be the most suitable for the duty. From these requirements, Renk-Maag drafted several design proposals for consideration with three different methods of engaging reverse.

One of the more significant preliminary design directions was the choice of forward/reverse direction activation in the transmission. The traditional shifting method (sliding gear) was offered in the event that a conservative design approach was required (Figure 11).

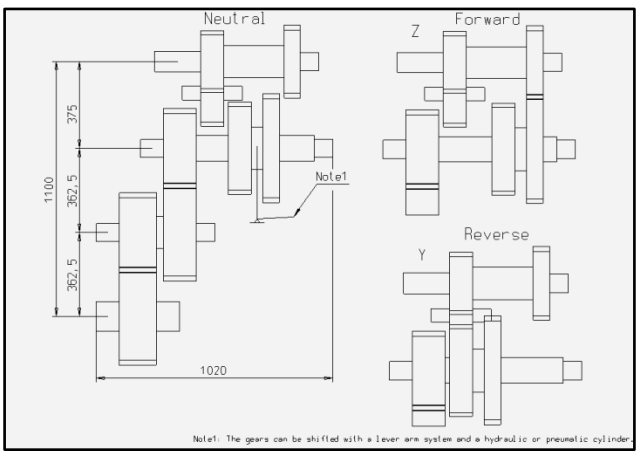


Fig. 11—Direction change sliding gear shifting mechanism.

The industry understands that the traditional shifting mechanism has limitations and therefore two alternative options with fewer compromises, that have been made more acceptable with improvements in technology, were offered.

An option proposed by Renk-Maag that has previously been used in similar applications was the tilt concept where the gear carrier is tilted sideways to engage either forward or reverse gear (Figure 12).

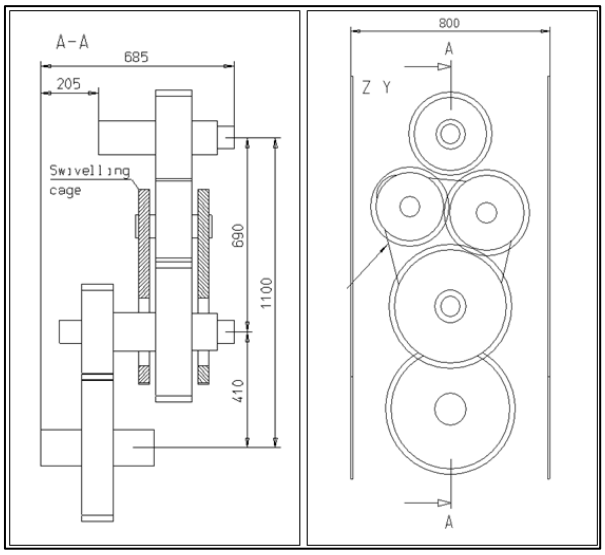


Fig. 12—Direction change using tilt concept.

The final option proposed by Renk-Maag that has previously been used in similar applications was the hydraulic shifting with constant mesh concept, where the forward and reverse gears are always engaged but a clutch is actuated to engage either forward or reverse gear (Figure 13).

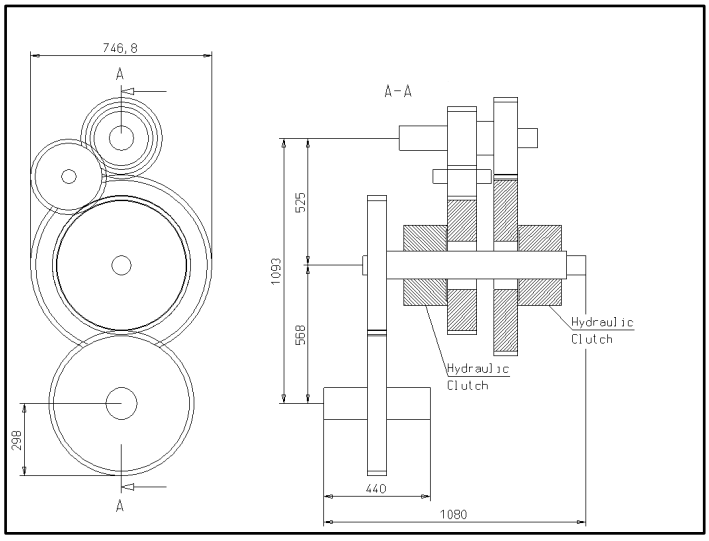


Fig. 13—Direction change using hydraulic shifting with constant mesh.

The final selection for the application was the option of constant mesh gearing with hydraulic friction clutch actuation for forward and reverse selection. This option is attractive for remote shunting unit (RSU) operations and with established improvements in hydraulic clutch technology was considered the most appropriate compromise for the duty, as it has a simple shifting process with no synchronising issue and has a high level of operational flexibility.

Other design directions that were made to realise the final design included: the lubrication was to be pressure fed by an engine driven pump; clutch oil pressure was to be provided by an engine driven pump; and scavenging of the lower gearing section was to be by the engine driven pump.

The next step in crystallising the design was to define the gearbox load profile (Table 2) based on sugar industry experience with engine load histograms of 40 tonne locomotives. The load profile was used to size the gears, shafts and bearings.

Table 2—Reversing box load spectrum.

Share (%)	Speed (r/min)	Torque (Nm)	Fx input (N)	Fx output (N)
5.0	100	4000	11 564	18 373
0.5	200	11 070	32 003	50 847
10.0	313	4500	13 009	20 669
4.5	313	8600	24 862	39 502
15.0	781	3500	10 118	16 076
10.0	781	6500	18 791	29 856
25.0	1563	3500	10 118	16 076
15.0	2344	2000	5782	9186
14.5	3125	1500	4336	6890
0.5	3440	1000	2891	4593

The thrust loads on the input and output shafts due to the slip joint movement were calculated with the following equations:

$$F_{x,IN} = \frac{T_{IN} * 2000}{d_{spline,IN}} * \mu$$

$$F_{x,OUT} = \frac{T_{IN} * 2000 * i_{DP}}{d_{spline,OUT}} * \mu$$

$$\mu = 0.11$$
$$d_{spline,OUT} = 115 \text{ mm}$$
$$d_{spline,IN} = 76.1 \text{ mm}$$

Where:

$F_{x,IN}$ is the thrust load on the input shaft

$F_{x,OUT}$ is the thrust load on the output shaft

T_{IN} is input torque

$D_{spline,IN}$ is the diameter of spline on input shaft

i_{DP} is the ratio of the drop box

$D_{spline,OUT}$ is the diameter of spline on output shaft

μ is the coefficient of friction at the spline interface

The magnitude of $F_{x,IN}$ and $F_{x,OUT}$ was then used to select the appropriate bearings for the shafts.

Further steps in the design process were:

- Develop a detailed list of requirements.
- Dimensioning of gears and shafts.
- Arrangement of gears and shafts.
- Dimensioning and arrangement of bearings.
- Casing concept.
- Hydraulic system concept development.

By this stage a final concept design was able to be produced and the final concept design was then able to be reviewed. At this step the gearbox scavenging system concept was changed from utilising a level sensor to utilising a weir system and an engine driven pump rather than a gearbox driven scavenge pump.

Steps in detailing the final concept were:

- Complete the 3D gearbox model with all parts inclusive of casing.
- Detailing all gearbox parts.
- Defining and optimising purchasing of parts in collaboration with suppliers.
- Exact strength and durability calculations for the gearbox parts.

The final review of the detailed gearbox then followed and the finished design drawings and documents were then reviewed to confirm that all outstanding issues were addressed. A significant consideration during the design process was to design-in transmission efficiency.

One consideration to achieve this efficiency was to remove the ability of the clutches to 'shift-on-the-fly' from forward to reverse direction. The pre-set condition of detecting a physical stop prior to direction change (achieved by PLC logic) had the ability to increase overall transmission efficiency.

This implies that the clutches are designed for shifting at standstill. The final general arrangement of the transmission is shown in Figure 14.

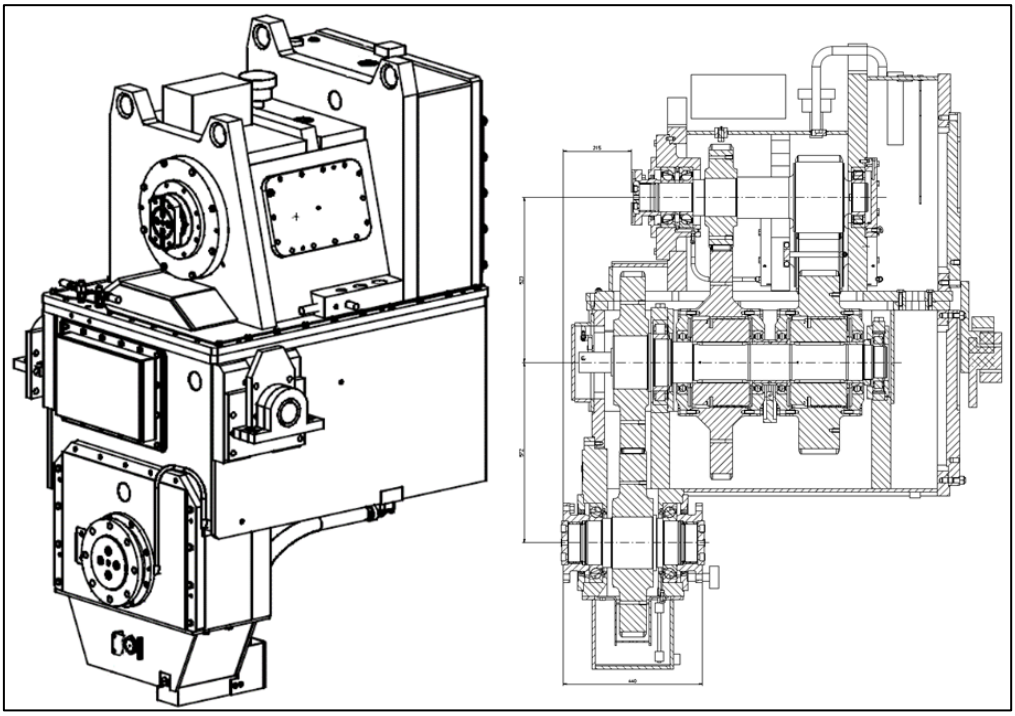


Fig. 14—Renk-Maag reversing box.

The efficiency and power loss for the final gearbox design are as shown in Figure 15 with a projected maximum power loss of 16.5 kW and minimum efficiency of 96.3% which accounts for the drag torque of the disengaged clutch.

The prototype reversing box will be load tested at the factory prior to shipping. The arrangement of components for the load testing program is shown in Figure 16. The dynamometer load testing program that includes a no load and a full load test has been defined and will reduce the probability of defective operation when the transmission is installed for duty.

On completion of full power and speed testing the reversing box will be disassembled for inspection of gear tooth contact pattern and bearing integrity.

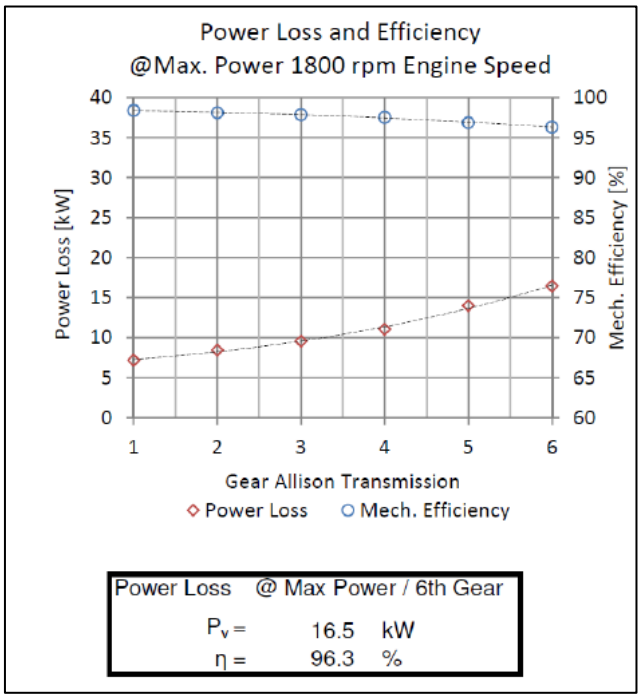


Fig. 15—Renk-Maag reversing box power loss and efficiency.

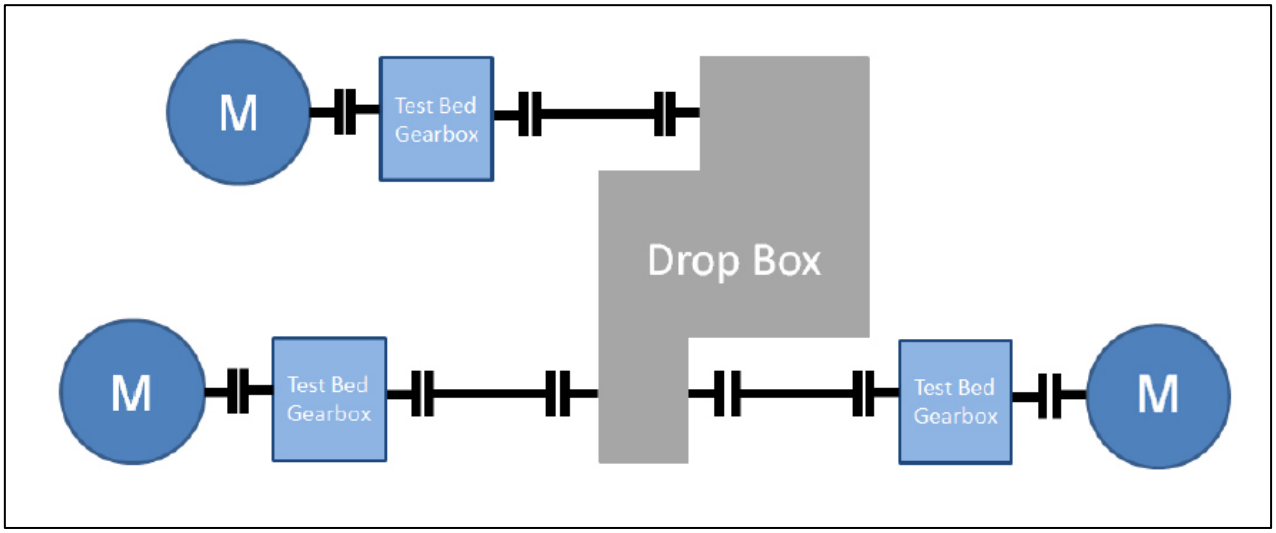


Fig. 16—Renk-Maag reversing box test concept.

Fuel economy benefits

The 40 tonne locomotive repower is an extension of the work completed with the smaller 18 tonne, 24 tonne and 32 tonne locomotives, where the industrial torque converters have been replaced by an Allison automatic transmission with lock up capability to improve fuel economy. In this application the Allison/Renk-Maag reversing box will replace either the Voith L4r4U2 or the L420 rU2 (neither of which is equipped with a lock up clutch).

There are several areas where the Allison transmission contributes to improved fuel economy and these are:

a) **Lock up clutch**. The standard Allison shift logic is 1C,2C,2L,3L,4L,5L,6L (C = converter, L = Lockup). This means in second gear the Allison shift logic actuates the lock up clutch (when the transmission pump and turbine in the torque converter approach coupling point) to cause the pump and turbine to rotate together so there is no slip. This results in no heat generation in the converter, hence improved efficiency.

The 2C to 2L shift will occur at about 10 km/h in the 40 tonne locomotive, so between 10 km/h and 40 km/h, the losses in the Allison transmission are limited to gearing and pumping losses (no converter losses). Referring to a iSCAAN prediction in 2C at 9.7 km/h, the transmission heat rejection is 78.9 kW (engine speed 1 930 r/min), in 2L at the same 9.7 km/h, the transmission heat rejection is 9.1 kW (engine speed 1 340 r/min). This is a saving of 69.8 kW of heat rejection.

b) **Gearing and engine map**. The other obvious advantage of the Allison transmission is that the ability to change gear ratio makes it possible to keep the engine running in the optimum speed range. Shift logic is set up to ensure the shift points are such that at each shift the engine speed will not drop below the engines peak torque at full throttle. Under part-throttle shifting, it is possible to shift below peak torque, because full power is not being requested. By maintaining the engine operating speed between 1 330 r/min and 1 850 r/min, fuel consumption is kept in the most efficient area of the curve for all locomotive operating speeds (Figure 17).

The Allison iSCAAN program also matches the engine torque curve to the correct converter option as there are six converter options for the Allison 4000 series transmission. This ensures that the converter slip is minimised and converter stall occurs close to peak power to maximise performance.

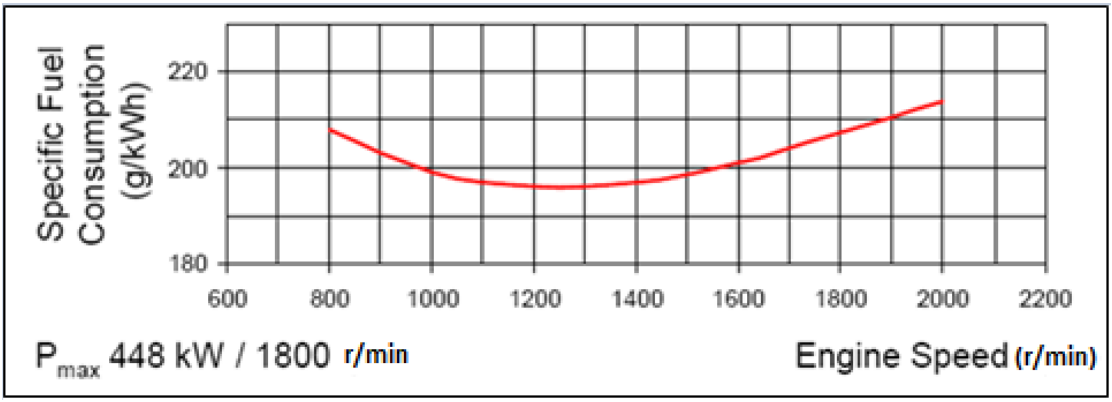


Fig. 17—Specific fuel consumption for Mercedes-Benz OM502.

Cooling system

The cooling system has been designed as a complete 'drop on' cooling module that replaces the existing locomotive nose cone. The cooling module has been designed so that all fluid and electrical connections are made at the firewall between the engine room and the cooling module. The cooling module has eight circuits and has provision for:

- Engine jacket water.
- Engine charge air.
- Allison transmission oil to water plate heat exchanger (water circuit cooled by engine jacket water circuit).
- Allison sump cooler–air to oil.
- Renk-Maag reversing box–air to oil cooler.
- Compressed air cooler.
- Hydraulic fan drive cooler.
- Air conditioning condenser.

The design ambient capability of the cooling module is for a maximum ambient temperature of 45 °C. A variable speed hydraulic fan has been located in the roof of the cooling module and will draw air in from the sides and discharge vertically.

The control of fan speed will be by the engine management system. The cooling system contributes to the total package fuel efficiency in that the parasitic fan engine load for cooling has been reduced by one half of the original power by considering the total cooling system engineering design.

Locomotive control system

The driveline has been designed to be controlled by the locomotive-based PLC and the electronic engine and electronic transmission control interfaces utilise a J1939 data communication network. The fan speed control is provided by the engine Pulse Width Modulated (PWM) functionality. The driveline has been integrated into the RSU architecture and is therefore RSU-compliant.

Driving and retardation torque is regulated by the operator primarily by regulation of the throttle and the percent retarder controls. This driveline torque regulation philosophy is intended to substantially reduce wheel brake application and it is expected that brake and wheel wear will decrease substantially as a consequence.

Conclusions

The 40 tonne locomotive driveline upgrade reduces the engine capacity in the 40 tonne locomotive from a V12 567 kW (23.9 litre) to a V8 448 kW (15.9 litre),–a 33% reduction in engine capacity. This was made possible with the higher transmission efficiency of the Allison and Renk-Maag reversing transmissions.

Understanding that the 40 tonne locomotives perform over one half of the locomotive cane haul duty for the Queensland sugar industry, this solution, when fully implemented, will translate into an indicative 15% improvement in fuel efficiency for the Queensland sugar industry locomotives.

The Renk-Maag reversing box has been designed specifically for 40 tonne locomotive application in cane field duty and utilises hydraulically actuated clutch packs for forward and reverse engagement. The new engine and transmission package has been modelled for tractive and braking effort, confirming that it meets or exceeds the previous system benchmarks.

Additional expected benefits of the 40 tonne locomotive driveline upgrade are:

- Improved locomotive tractive effort from 0–5 km/h.
- Improved reliability, that has been achieved by the minimum number of high quality easily maintainable parts.
- Readily available spare parts.
- Improved emissions output, complying with EU Stage II and lower radiated noise output.

In summary, the expectation is to realise a driveline with a reduced total life-cycle cost to deliver a more cost-effective asset for cane transport operations in Queensland.

The locomotive ‘Piralko’ at Invicta Mill is scheduled to be the first 40 tonne locomotive to receive the repower package with this work scheduled to commence at the end of 2016 crushing season. Commissioning and testing of the ‘Piralko’ is to be completed in readiness for commencement of 2017 crushing season.

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