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Hybridization of a Mobile Work Machine

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Abstract

Interest to hybridize mobile work machines has increased substantially during ongoing decade. Reasons for increasing interest are mainly tightening emission regulations and trend of rising fuel prices. To get better understanding for the benefits of hybridization, Helsinki University of Technology (TKK) has started 5 years project to research different aspects, how to improve fuel economy in mobile work machines. For case work machine is chosen an underground mining loader which will be first researched as conventional version and then it will be converted to a hybrid version. The tests for conventional version were done in the early 2009 and for hybridized version till the end of 2010. The results will be then compared to each other.

Keywords: electric drive, PHEV (plug in hybrid electric drive), off-road, series hybrid, power management

1 Introduction

This paper introduces a work machine hybridization project (HybLab) coordinated by Helsinki University of Technology (TKK). TKK has longtime experience to work with Finnish work machine industry. During the ongoing decade there has be several hybrid technology related co-projects between TKK and Finnish work machine industry. Duration of the HybLab project is 5 years started from the beginning of 2008. Results will be published.

2 Scope of research

The scope of the research is to study different aspects how to improve fuel economy and productivity with hybrid technology. Goal is to achieve 50% fuel savings on chosen duty cycle.

The project includes 3 main phases:

1. On the first phase was researched efficiencies and performance of a conventional work machine during the duty-cycle. Tests were implemented in real mine environment. These results will be used as reference when making comparison for the hybridized version.

2. On the second phase the loader will be hybridized and it will go through same tests in same conditions as the conventional version. The hybridization phase will start on 2009 and first tests will be done till the end of 2010.

3. After the hybridization phase the project will continue with optimization of control software which is assumed to be the most time consuming phase. Also some minor hardware updates will be done to achieve better efficiency and performance. At this

point it's anyway too early to address what those updates could be.

3 Basic data

The origin of the basic data is from technical specification of the case machine and from the measurements done during the first phase of the project.

3.1 The case work machine

The case work machine is a frame steered underground mining loader. Basic data of the loader is listed on the table 1:

Table 1: Tech. spec. of the conventional case machine

Technical Specification	
Engine power	85kW
Total weight	18t
Pay load	4t
Max. speed (on flat)	10km/h
Transmission	Hydrostatic (a motor on both axles)
Implement (boom)	Hydraulic
Steering (frame steering)	Hydraulic

The performance data on the table 1 is based on the measured data.

The case loader represents small size loaders. Mid size loaders are approx. 2 times bigger and big loaders are approx. 3-4 times bigger than the case loader. All loaders has anyway pretty much similar construction so results can be adapted to bigger loaders quite reliably by using just a appropriate scaling factor.

3.2 Duty cycle

Typically underground mining loaders gets mined rock from ends of mine caverns and brings it to main tunnel where the load will be dumped to a dumper-truck or conveyor. Driving distance for loaders is typically 200-400m and slopes in caverns are typically between 0-20%. The duty cycle chosen for the project is described in the picture below.

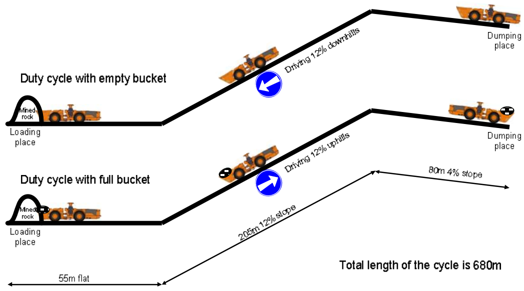

Figure 1: The chosen duty cycle for the loader

The duty cycle is based on the real test route in the mine where the conventional loader was tested.

Driving speeds in mines are generally limited to 10, 20 or 30km/h. The case machine achieves max speed 10km/h on flat and 5km/h on up-hill. Total duty-cycle time is 7 minutes.

4 Results so far

To get needed values for calculations and simulation models the loader was tested and measured in real mine environment. To collecting needed data the machine was instrumented extensively. Number of sensors was 46 and rest of the signals were CAN-signals from the loaders CAN network. Total number of measured signals was 74 and measuring frequency was 5ms.

4.1 Data acquisition

From driveline hydraulics and from implement hydraulics was measured flow, pressure and temperature values. These measurements were done over all pumps and motors.

From hydraulic cylinders were measured forces and speeds from pivot points of bucket and steering mechanisms. Also pressure over cylinders were measured

From cooling system were measured flow, pressure and temperature values to get understanding about thermal behaviour of the power pack during continuous repeated duty cycle.

Drive speed and distance was measured with ground speed radar and slope angles were measured with inclinometer. With these measurements was possible to define the real driving performance of the loader and also the profile of the duty cycle route.

4.2 Measured performance

On time of writing this paper the analysis of the measured data was still in process. Only measured data presented here is driving data. Loading and dumping data needs substantially more time to analyze, so that data is out of scope in this paper. In fuel saving context the driving data has anyway very significant importance because most of the duty cycle time goes for driving.

In the table below are listed different actions during the cycle and how long time each action takes. The gray coloured actions are out of scope in this paper.

Table 2: Time vs. Action during the duty cycle

Time [sec]	Action
36	Driving 4% up-hill (empty bucket)
78	Driving 12% down-hill (empty bucket)
25	Driving on flat (empty bucket)
35	Loading the bucket
25	Driving on flat (full bucket)
148	Driving 12% upp-hill (full bucket)
36	Driving 4% down-hill (full bucket)
35	Dumping the bucket
418	**TOTAL time**

In the figure 2 is presented measured power values from ICE flywheel and summarized mechanical output power from hydraulic traction motors. The values are mean values during each driving action. For the 12% down-hill drive is given value zero even thought that's not absolute true. On the down-hill the hydraulic system transfer power "backwards" so that the hydraulic pump tries to rotate the ICE. So basically ICE's fuel consumption is zero but it's a bit unclear can the system utilize excessive power somehow. This issue will be studied more carefully later but in this paper we assume that the power on the 12% down-hill is zero.

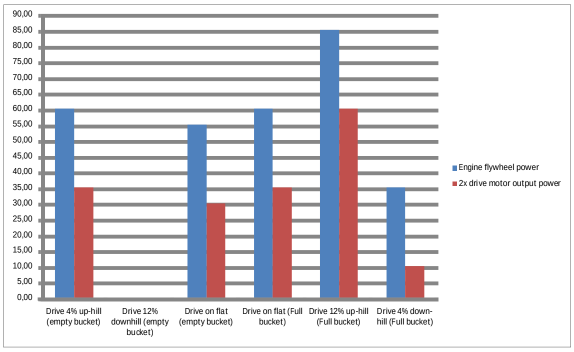

Figure 2: Measured power [kW] values

Difference between the ICE and drive motor values give power value which is same as losses in the hydraulic drive system. The lost power is about 25kW. That value gives good estimate about how much only the driveline efficiency could be improved between ICE power source and mechanical gear in axles.

In the figure 3 is presented energy consumption values from engine flywheel and summarized mechanical output energy from traction motors. The values are mean values during each driving action. The values are calculated from the power values presented in the figure 2 and times of each action presented in the table 2.

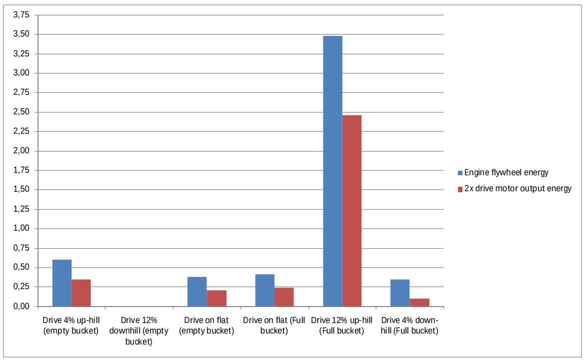

Figure 3: Measured energy consumption [kWh] values

As expected the 12% up-hill is the most energy consuming action during the duty cycle. That gives valuable information for the hybrid version's energy storage dimensioning. Decision must be done between acceptable fuel consumption and acceptable payback time for battery/supercapacitor system.

5 Performance calculations for the Hybrid version

For the hybrid version is done preliminary performance calculations. The calculations are based partly on measurements done for the conventional machine and partly for educated guesses. Goal is that the hybrid versions nominal performance will be at least same as conventional machines performance.

In the figures 4 and 5 are shown drive-line power needs during the duty cycle and also how the power is shared between gen-set and energy storages. Energy storages are used also for loading and dumping actions which are not shown in the figures. Duration of each action is listed in the table 2.

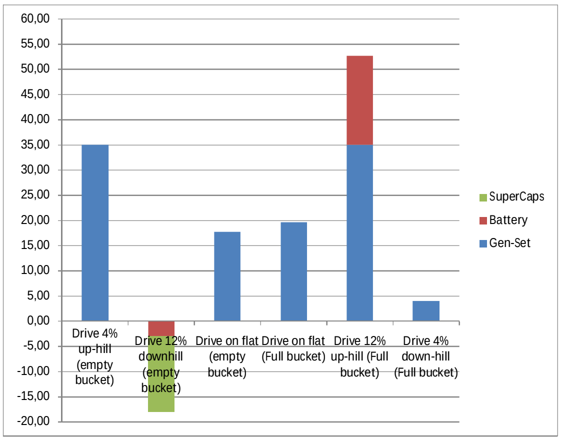

Figure 4: Power need [kW] in drive line during the duty cycle

The gen-set power in this calculation is limited to 35kW which is found to be also approximately mean power for the whole duty cycle.

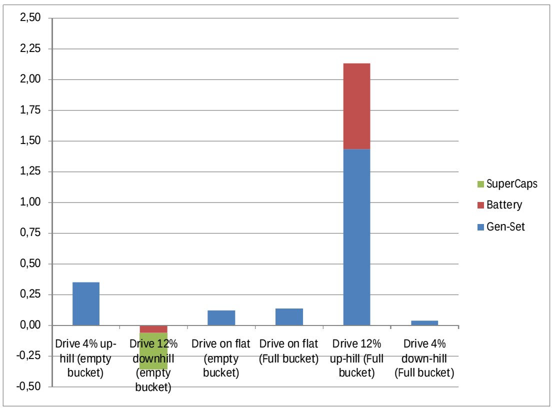

Figure 5: Energy consumption [kWh] in drive line during the duty cycle

6 Hybridization concept and dimensioning basis

The hybridization concept will be based on series hybrid construction with gen-set and two energy buffers. The work and steering hydraulics will be based on electrically driven pumps. The backbone of the hybrid system is 650V DC-link. Auxiliary systems, like cooling devices, are connected to low voltage network 24V which takes power from high voltage network via DC-DC converter. System lay-out is described on the picture below.

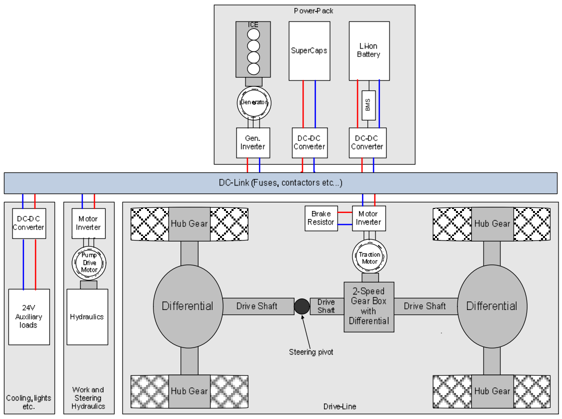

Figure 6: System lay-out drawing

6.1 Power pack

Dimensioning of the power-pack is based on idea that the gen-set produces mean power which is calculated to be approx. 35kW and additional power needed on uphill, approx. 50kW, is taken mainly from battery. Super capacitor-packet is used primarily for power peak shaving during loading and dumping but it can be used also parallel with battery e.g. on uphill if energy in super-cap packet happens to be available.

Energy storage usage during the duty-cycle is based on idea that the energy buffers are charged only with regenerative energy. Regenerate energy can be captured mainly on downhill. Charging strategy on downhill is such that the super-cap packet will be charged up to 100% and rest of energy will be charged to battery. This will lead to situation that charge-level in battery will decrease during each duty cycle. For the demonstrator loader batteries are dimensioned so that it lasts about 13-14 cycles which means a bit over one hour use.

For production loader the battery dimensioning should be doubled if dimensioning basis will be time between driver's breaks (e.g. coffee, lunch and change of shift breaks). In this strategy the battery pack will be charged up during breaks. Charging power can be taken from mains if available or from gen-set by running it on optimal energy efficient speed. Gen-set charging should be anyway avoided because fuel energy is more expensive than from mains supplied electric energy and also because power conversions in inverter and DC/DC converter lose energy more than charging directly from mains via charger. Mines usually has good electric network so charging from mains should not be a problem.

6.1.1 Gen-set

The basic requirements for the gen-set are specified but the device is not ordered yet. Power class for the ICE (diesel engine) should be around 40-50kW and its best efficiency should be between 30-40kW. The generator's nominal power should be around 30-40kW and max power about same as the ICE's power. Other requirements are electric controls in ICE and generators compatibility with 650V system. Number of commercially available and suitable packets is quite limited.

6.1.2 Battery system

The battery system is based on Kokam 40Ah lithium polymer cells. The cells are packet in to 7xcell hard-pack modules. The battery system consist of 14-modules which gives nominal voltage 362,6V. Discharge current is 200A cont., 400A max and charge current is 80A [A].

Figure 7: Kokam 7S1P battery module with 40Ah cells [A]

6.1.3 Supercapacitor system

The supercapacitor system is based on Maxwell 390V module. The module's useful capacity is between 100-390V where the available energy is about 0,36kWh. Discharge/Charge currents are 150A cont and 950A max [B].

Figure 8: Maxwell Boostcap HTM Power Series 390v ultracapacitor module [B].

The 390V module is not anymore available in markets.

6.2 Drive-line

Drive line construction is based on single traction motor which will be placed between axles. To achieve comparable performance level with the conventional version the traction motor dimensioning is based on the engine power of the conventional loader which is 90kW. The motor will be connected to a two-speed gearbox which has also lockable differential gear. Two-speed gearbox is needed to achieve high enough torque during loading the bucket and on the other hand to achieve approx. 25-30km/h top speed which is general top speed level for loaders. Differential gear is needed to avoid power/energy losses during turnings and lockability is needed during loading when full traction force is needed from both axles.

6.3 Work and steering hydraulics

Work and steering hydraulics will be realized by driving hydraulic pump with electric motor. At this point the final concept is still a bit open. Under consideration are two options which are centralized hydraulic pump and distributed pumps. Centralized concept will mean that all hydraulic functions will be driven by one pump and distributed concept will mean that each function would have own separate pump. It's possible that the first version will be based centralized concept but this will be defined later.

7 Auxiliary systems

Auxiliary systems in mobile work machines are traditionally considered as minor loads and they are ignored while improving the energy efficiency of the machine. When ICE, drive line and work hydraulics efficiency has been continuously improved, auxiliary systems open up a potential for energy savings and better total efficiency of the machine.

Many studies has proven, that electrification of auxiliary system components, such as water and oil pumps, cooling fans and air compressors, have increased the energy efficiency and decreased emissions in heavy vehicles [C, D, E, F].

7.1 Cooling System in the conventional machine

The cooling demand of mobile work machine is large and cooling system is often the most energy consuming of auxiliary systems.

The case machine's maximum operating temperature is 50 °C. Heat rejection for 90 kW ICE is 39 kW for engine cooling and 22 kW for charge air at rated power [G]. Heat rejection in hydraulics is 18 kW.

The cooling radiator is a combination of water, hydraulic oil and charge air cooling. Cooling fan is directly coupled to the ICE crankshaft and rotates continuously with engine shaft producing the air flow for all cooling circuits at the same time. The fan and the radiators are dimensioned based on maximum heat rejection of system.

The ICE coolant circulation is realized by mechanical driven coolant pump. The pump delivers 216 LPM at rated speed, which is the actual coolant flow all the time because the engine runs continuously at rated speed.

The Hydraulic fluid cooling circulation is realized by mechanical driven brake pump, which also takes care of parking brake pressure. Cooling and parking brake pressure are controlled by a set of valves.

The biggest problem with this type of cooling arrangement is that the energy efficiency is poor. Cooling fan and pumps are running all the time whether there is need for cooling or not.

7.2 Cooling system for the Hybrid version

Advanced cooling arrangements makes possible to control fan and pump speed independently. This solution has a lot of potential of saving energy when the cooling power can be adapted for the actual cooling need. Usually these systems use hydraulic or electrical driven fans and electrical driven pumps.

The hybrid version will need two separate water cooling circuits. The first is for ICE cooling and the other one is for power electronics and electric motors. The reason for this is that power electronics needs to operate in lower temperature than ICE. In all cooling circuits, the fluid will be circulated by a electrical pump. Also all fans will be electrically driven. Fans will be connected directly onto single layer radiator. Many smaller fans gives flexibility to adapt cooling power and thereby is more efficient solution than one big fan [H].

In hybrid machines the most efficient way to use cooling system would be high voltage powered fans and pumps, but so far there have been problems with availability. When fans are pumps are 24V devices this usually means also need to improve power capacity of the 24V network.

Because in hybrid version the engine is down sized the heat rejection is much less than in conventional machine. It could be estimated that the average heat rejection of the engine to be a half of the original system – 20 kW for engine cooling and 10 for charge air. This can be managed with relatively small cooling unit. Fig.9. show s one commercial solution to do that.

Fig.10. shows that the cooling unit in Fig. 9. is powerful enough in ambient temperatures under 40 °C. In higher temperatures either higher coolant temperature or bigger units are needed. It is also possible to install multiple units in regions because of modular structure of the unit.

Figure 9: Electrically driven fan-radiator unit for water cooling [I]

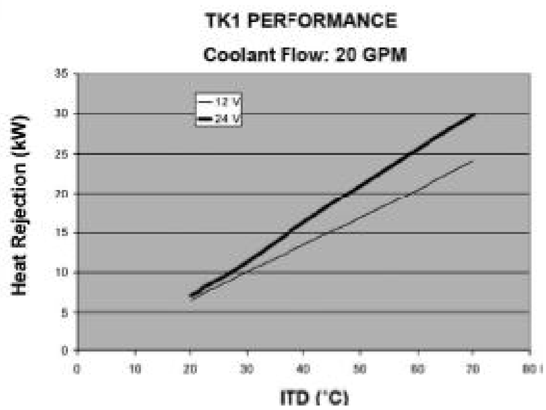

Figure 10: Performance curve for fan-radiator unit presented in Fig.9. (temperature ITD = coolant temperature – ambient temperature) [I]

Cooling of power electronics and electric motors can be arranged in same way than engine cooling. Heat rejection in this part of the cooling system is about the same size than with the engine. The main challenge is the low coolant temperature in this cooling circuit. The maximum temperature of inlet coolant for power electronics and electric motors is usually 55-65 °C. This means that the full cooling power of 2 to 4 fan-radiator unit is needed - or one bigger one.

Hydraulic fluid does not necessarily need a forced cooling if the loading frequency in work cycle is low. In case machine the fluid would be cool down by a radiative transition into air, if there is enough time between loading and dumping. Also the electro hydraulic pump unit can be air cooled instead of water cooled.

Fig. 3. presents one possible solution for cooling arrangement in hybrid version. ICE and brake resistors are connected to the same cooling circuit. They use a common electric driven pump and cooling fan-radiator unit. There are also two controllable valves, which are used to control coolant flow between the systems. The valve in ICE outlet side is used to control engine warm-up. The valve after the coolant pump is used to control the flow to cool brake resistors when they are activated in case of over voltage in high voltage electric bus. This setup saves some components and it's possible, because engine and brake resistors are never used at the time.

In power electronics cooling circuit all equipments share a common cooling fan units. Each subsystem has own pump, which makes it possible to control coolant flow individually in subsystems based on actual heat rejection of systems. Power electronics are placed at the cooler side of the cooling circuit, because they usually are more sensitive.

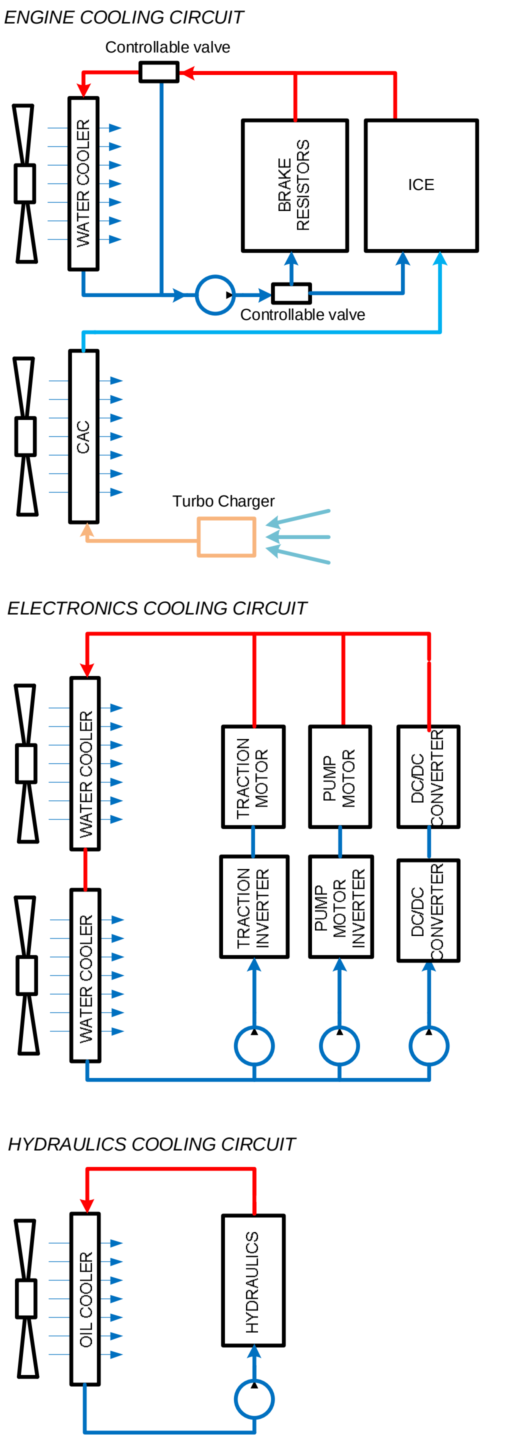

Figure 11: Simplified cooling system layout of the hybrid version

7.3 24V DC system in the conventional machine

The currently installed system in the case machine is a 80 A alternator and a 92 Ah lead acid battery. The main loads in the system are lightning are control logic. The average output power is 600 W.

7.4 24V DC system the Hybrid version

The Hybrid versions average power demand from 24V system is expected to be 3-4 kW or higher than in the conventional machine. A normally operated alternator has an average efficiency of 50% [C]. In hybrid machine the 24 V power can be converted from high voltage system with efficiency over 90% with a power electronics device and the efficiency will rise dramatically. In the hybrid version energy saving of 40% can be reached when using DC/DC converter instead of alternator. The effect in total power consumption in work cycle is 7% in hybrid version. Power is available at any engine speed.

Fig.12. presents an example of DC/DC converter for hybrid vehicles. The output power is 300 A at 27,8V DC. The 24 V DC system and high voltage system is isolated. The device is water cooled which enables compact structure.

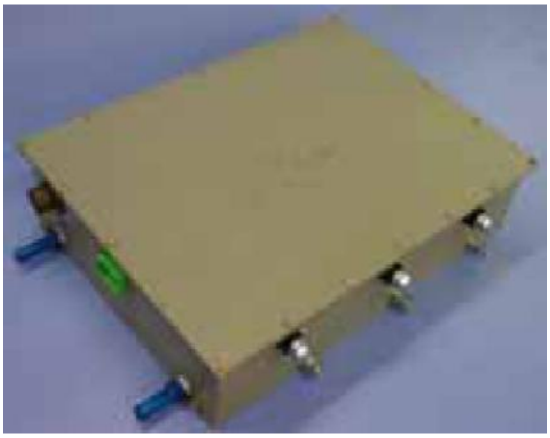

Figure 12: DC/DC converter for hybrid vehicles [J]

8 Conclusion

The HybLab project’s goal is to achieve 50% fuel savings for the case machine on the chosen duty cycle. The results so far look promising. There seems to be several areas where energy consumption can be decreased significantly, even 50%. One of the biggest savings is expected to come when the hydrostatic drive line will be replaced by electro mechanical drive-line. Results indicates that hydraulic drive-line wastes about 20-

25kW during driving. Also downsizing the ICE, adding electric energy storages and building electrically controlled cooling system are expected to improve energy efficiency remarkably.

Acknowledgments

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References

[A] Kokam, http://kokam.com/english, accessed on 2009-04-15

[B] Maxwell, http://www.maxwell.com/, accessed on 2009-04-15

[C] C. Andersson, *On auxiliary systems in commercial vehicles*, Doctoral thesis, ISBN 91-88934-31-4, Department of Industrial Electrical Engineering and automation, Lund University, Sweden, 2004

[D] B. Surampudi et. Al., *Electrification and integration of accessories on a class-8 tractor*, SAE Technical paper series 2005-01-0016, SAE World Congress 2005, Detroit, USA, April 11-14 2005

[E] M. Lasecki, J. Cousineau, *Controllable electric oil pumps in heavy duty diesel enginess*, SAE Technical paper series 2003-01-3421

[F] W. Hnatczuk et Al., *Controllable electric oil pumps in heavy duty diesel enginess*, SAE Technical paper series 2003-01-3421

[G] Mercedes-Benz, *Technical datasheet: OM 904 LA*, Daimler AG 2007.

[H] N. Staunton, *Reduction of parasitic losses in heavy-duty diesel engine cooling systems*, Engine Expo 2008, Stuttgart, Germany, May 6-8 2008

[I] EMP, http://www.emp-corp.com, accessed on 2009-04-15

[J] US Hybrid, http://www.ushybrid.com, accessed on 2009-04-15

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