Improving transmission efficiency via a novel radial piston cam lobe motor architecture

Pierre Bernard *, Vincent Langlois, Maxime Maurer

Poclain Hydraulics Industrie, pierre.bernard@poclain.com

Abstract

This paper presents the energy efficiency optimization of a closed loop hydrostatic transmission, using the example of a heavy tandem roller compactor application. An innovative concept of radial piston cam lobe motor is proposed, which allows a paradigm shift in hydrostatic transmissions sizing, essentially moving away from breakout torque as the entry point to sizing and refocusing the transmission sweet spot on its most used conditions. The comparative simulation of a standard machine and the same machine equipped with the new concept of hydraulic motor is realised using a 1D simulation software. The energy consumption of both machines is simulated over a reference usage profile derived to represent long-term machine use. On this usage profile, simulations demonstrate a fuel consumption reduction of 7.7% using the updated design of hydraulic motor.

Keywords: efficient transmission, radial pistons motor, cam-lobe, energy efficiency optimization, hydraulic evolution.

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1 Introduction

Climate change and energy management are at the heart of preoccupations and innovations in all sectors. The road construction sector accounts for approximately 5% of global GHGs emissions in Europe [1], and off-road machinery in Europe is subject to the European Commission's *Green Deal* [2], which targets net zero emissions of GHGs by 2050. Road compaction equipment participates to the CO2 emissions and global GHGs of the construction sector [3]. To the knowledge of the authors, no study has investigated the energy efficiency of compaction equipment and specifically heavy tandem rollers. However, as will be shown in the coming sections, their operation is quite repeatable, which makes them a good candidate for a study of fuel consumption over a representative mission profile.

The concept of hydraulic motor proposed in this paper is designed to refocus the sizing of the transmission towards efficiency, treating the maximum breakout torque requirement of the machine as a separate issue.

After giving an overview of the application and its usage profile, a standard system currently in use on this type of machine will be used as a reference case for the energy efficiency study. Then, the novel radial piston cam lobe motor architecture will be introduced, and the sizing philosophy it allows will be described. In simulation studies, this upgraded motor and its optimized transmission will be substituted for the reference machine transmission, and the energy efficiency evolution will be evaluated.

2 Heavy tandem roller compactors

2.1 Function, purpose

A heavy compaction tandem roller is a compaction machine for road creation or refurbishment. These machines are used to compress the asphalt's support and top layers in order for the pavement to carry the traffic loads and to resist to weather and temperature induced stresses. Several compaction methods and machine exist, each dedicated to specific conditions and types of soil. The machine studied in the present work is a vibratory heavy articulated tandem roller, meaning that a vibration of the drums is induced by an off-centred mass, driven in rotation by a high speed hydraulic motor. As explained in [4], the vibration of the drums drastically increases compaction efficiency by reducing internal friction in the mix.

The choice of the type of compactor to use is based on several parameters. Vibratory tandem rollers are adapted to the base and sub-base layers, as well as to the top asphalt layer (see [5]).

2.2 Hydraulic system

The hydrostatic transmission commonly used on such machines is a standard closed loop circuit with two hydraulic motors fed in parallel by a variable displacement pump, usually an axial piston type. Each drum is equipped with one fixed-displacement motor, which is responsible for the forward and reverse motion of the machine. An auxiliary closed loop circuit is used for the vibration function. The targeted machine speed is defined by the operator, and regulated by the on-board electronics, which also manages the acceleration and deceleration ramps.

2.3 Standard use

The machine is brought on a trailer from the contractor's parking site to the work site. Then, asphalt compaction is achieved in several passes, the first of which is made just after the mix has been laid-down. The standard lay-down temperature is approximately 130 to $160°C$, and the first compaction pass is made right away, in order to finish it with mix temperatures still higher than $115°C$, a temperature above which bitumen is more fluid and compaction using vibration is efficient. Subsequent compaction passes (generally between 2 and 4) are then made before the mix cools down to temperatures lower than $60°C$ and becomes stiffer and harder to compact.

For the reasons described above, a standard compaction pass, a description of which is given in Figure 1, is made in a specific pattern on a 30 to 60 m long strip behind the paver (the machine depositing the asphalt mix), with a slight overlap of 10 to 20 cm between each strip. To preserve compaction homogeneity, during the start and stop phases of the compaction, smooth acceleration and decelerations are mandatory, and a constant compaction speed during the passes is required. Suitable compaction speeds range between 2 and 6 km/h depending on the mix composition and layer thickness. On a worksite, the compaction machine also has to travel from one point to another, be it from its parking site to the location of the compaction strip start, or between road sections. These travels are made at higher speeds, up to 12 km/h, and the rolling resistance is much less than during compaction passes.

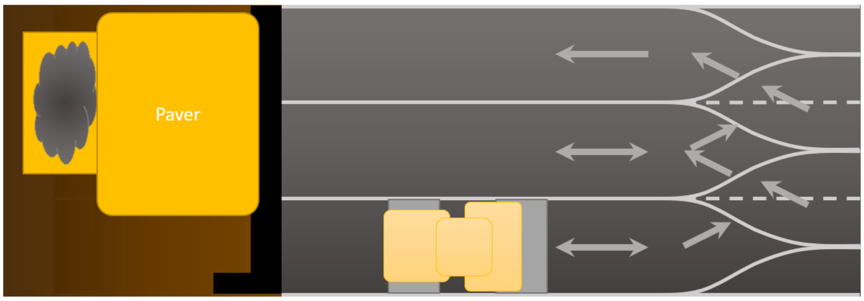


Figure 1 Illustration of the optimised pattern of a compaction pass.

2.4 Simplified reference duty-cycle

In order to estimate the energy efficiency of a heavy tandem roller, a simplified drive cycle is elaborated to represent the use of the machine on a worksite, including travel and work periods. The reference duty-cycle includes portions of flat, uphill and downhill travel for both transfer and work periods. While the gradability of vibratory heavy tandem rollers is usually between 30 and 40% (see for example [6], [7]), it is rare to find roads with this kind of gradient, and smaller compaction machines are used for mountain roads. As a reference, the European Road Safety Observatory [8] defines motorways as a road with, amongst other characteristics, maximum gradients of 4 to 5%. Therefore, an arbitrary value of uphill/downhill slope of 7.5% is chosen for the reference mission profile to represent an average value of the slopes encountered by the heavy tandem roller.

A real worksite in southern France was used to consolidate the different time ratios between work and travel modes, as well as between flat, uphill, and downhill sections. A proportion of approximately 10% of the time is allocated as “7.5% slope” on our reference mission profile.

Figure 2 shows the speed, slope (both on left ordinate axis) and rolling resistance (right ordinate axis) in function of time for the heavy tandem roller duty-cycle. The reference profile is nearly 20 minutes long, which is due to the importance of the acceleration and deceleration phases. Indeed, measurements on a machine have shown acceleration ramps defined to reach the targeted speed in 2 to 3 seconds, whether for working speeds of 5 kph, or travelling speeds of 12 kph. As a result, acceleration and deceleration times of 2.5 seconds have been set in the reference mission profile.

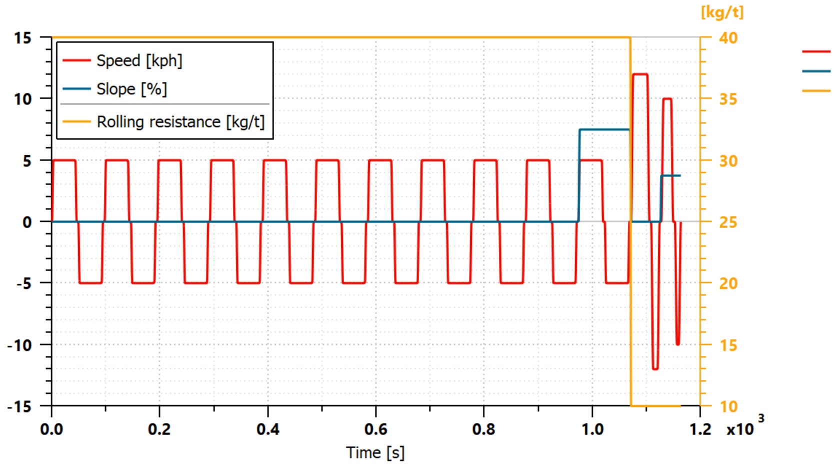


Figure 2 Reference mission profile for heavy compaction energy efficiency determination.

2.4.1 A note on rolling resistance

Sun et al. in [9], give several definitions of rolling resistance, from an energy and a force standpoint. In the context of a compaction roller rolling on fresh, not yet compacted, asphalt, the asymmetry in force distribution at the contact line can be attributed to the asphalt mix deformation. This deformation is closely linked to the apparent viscosity of the asphalt mix, a characteristic that varies with mix temperature, and thus with time. Different situations of the compactor roller are shown, a low rolling resistance travel situation in Figure 3 (a) ; what could be a first compaction pass with hot, low viscosity, asphalt in Figure 3 (b), leading to a high rolling resistance ; and what could be a finishing, higher viscosity, pass in Figure 3 (c), with intermediate rolling resistance.

These different conditions mean that the rolling resistance would evolve with time during compaction, and especially in the different compaction passes. To simplify the mission profile, only two values are used for the coefficient of rolling resistance C_{rr}: 40 kg/t for compaction mode, and 10 kg/t for travel mode. The 40 kg/t work value of rolling resistance is taken as a representative value for all compounded compaction passes. Separating compaction phases would lead to values between 50 kg/t for the first compaction pass, and 35 kg/t for finishing compaction passes.

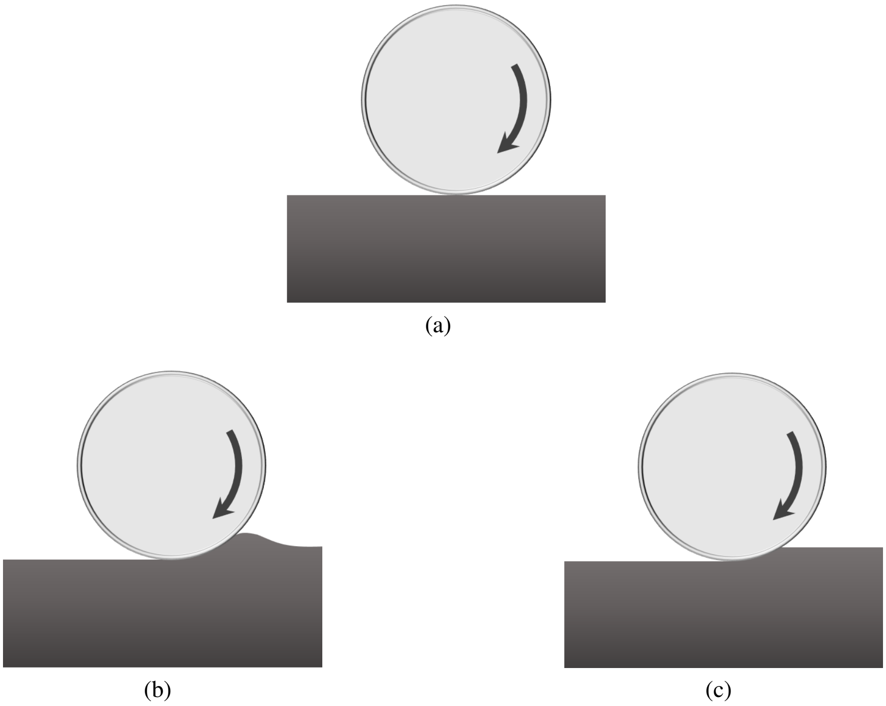


Figure 3 Roller rolling on road (a), on hot asphalt mix (b), on “cold” mix (c).

3 Current situation

3.1 Transmission sizing

As exposed in the previous sections, the compactor’s transmission and mission profile are rather simple, and for these reasons, a heavy tandem roller’s performance level assessment is unlikely to be based on the sole hydrostatic transmission capability. When evaluating a compaction machine, the focus is on compaction efficiency, which encompasses speed stability, compaction ability, as well as vibration frequencies and amplitudes.

3.1.1 Machine gradability

The main characteristic used for sizing the hydraulic motors of the compactor is the gradability of the machine, which means that the maximum slope to be climbed defines the torque requirement on the motors. From this, the displacement D_M of the motor is obtained by the standard formula: $D_M = \frac{T}{\Delta P}$, with T the torque and ΔP the differential pressure at the motor’s ports. Using this displacement, and knowing the maximum required speed, drum rolling circumference, and internal combustion engine (ICE) speed, one is able to define the required pump displacement D_P. As mentioned in §2.4, the gradability of heavy tandem rollers is in the 30–40% range. Due to the mass of the machine, and the incline of such a slope, this max climbable slope is driving the maximum torque requirement on the hydraulic motors installed in the drums. The torque T required for each of the two drum motors in order to climb a slope s is given by:

$$T = \frac{1}{2} M \cdot g \cdot R_{Drum} \cdot (\sin(\arctan(s)) + C_{rr} \cdot \cos(\arctan(s))) \quad (1)$$

where M is the mass of the machine, and R_{Drum} is the radius of the drums.

3.1.2 Application

Applying the previously described method, without considering efficiencies, to a real case with the machine parameters listed in Table 1, representative of a heavy tandem roller available for test, gives a required torque on the motor $T_M \approx 17000 N.m.$

Table 1 Machine parameters.

M	[kg]	13300	Compactor mass
s_{max}	[%]	39	Gradability
C_{rr}	[kg/t]	10	Rolling resistance coefficient on max slope
R_{Drum}	[m]	0.7	Drums radius
p_{max}	[bar]	400	Max system pressure
p_{charge}	[bar]	20	System charge pressure

With the available differential pressure in the system (differential pressure $\Delta P = p_{max} - p_{charge}$), this gives a displacement of $D_M = 2800 cc/rev$ for each of the two motors in the system.

3.2 Fuel consumption over reference cycle

Simulation of this machine over the reference mission profile described in §2.4 is performed using the AMESim 1D-simulation software.

3.2.1 Model description

The model, the sketch of which is shown in Figure 4, makes use of the capabilities of AMESim to implement vehicle, ICE, and hydraulic circuitry and components in the same model. The vehicle model calculates the longitudinal acceleration, velocity and displacement of the machine based on road profile, wheels (in our case, drums) torque. The implemented model accounts for rolling friction and road slope. The machine speed is controlled by way of

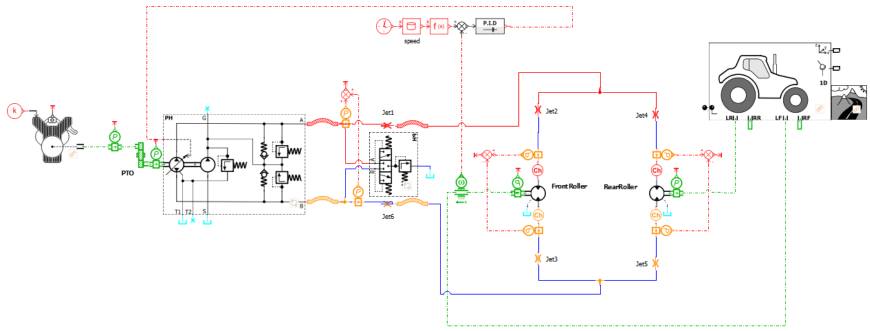


Figure 4 Model for reference machine energy efficiency simulation.

a PID controller on the pump displacement, the internal combustion engine speed target being fixed at 2300 rpm. The engine model manages acceleration ramps of the engine to avoid stalling, and returns fuel consumption based on tabulated values function of engine speed and torque. The hydraulic system is modelled using physical models of the pump and motors, including tabulated data for hydro-mechanical and volumetric losses, and pipes including pressure drops and compressibility. The pump and motor losses models are based on elementary measurements and advanced simulation results, aggregated in simplified models for energy analysis.

3.2.2 Reference machine energy efficiency analysis

The simulation of the AMESim model of the machine described in the previous paragraph over the reference mission profile described in §2.4 gives several results, of which the fuel consumption of the machine over the complete cycle is the metric chosen in this study. Figure 5 shows the mechanical energy in Wh on the pump's shaft and the differential hydraulic energy on the pump's ports, as well as the fuel consumption in L over the nearly 20 minutes reference mission profile described previously.

The total fuel consumption for the machine is $2.5L$ over this duty-cycle. The energy and fuel consumption are plotted over time, and it can be seen that the instantaneous energy consumption is greater near the end of the mission profile, during phases of compaction in slope, and travel.

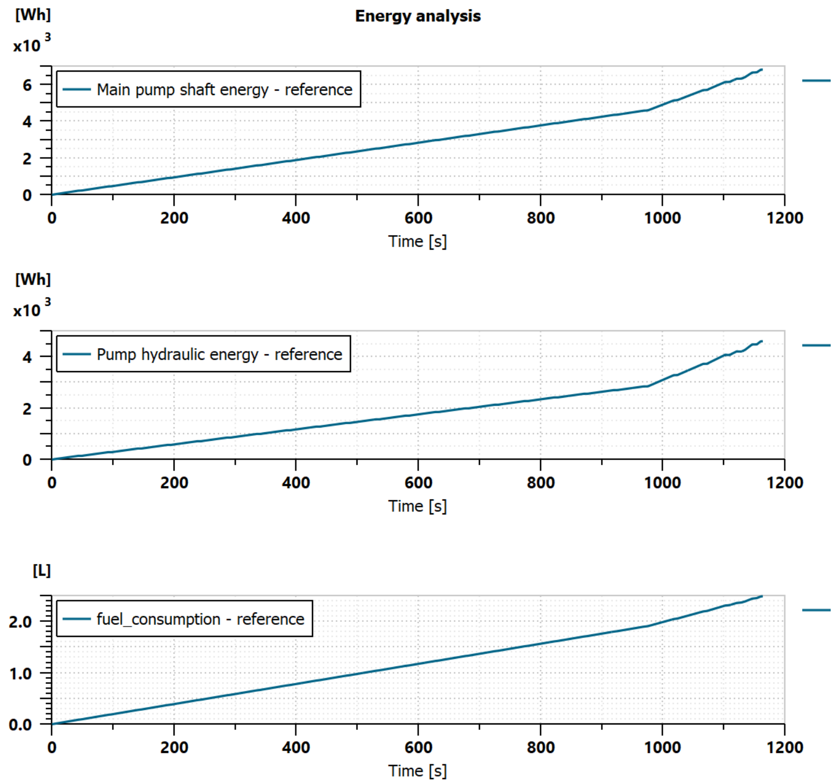


Figure 5 Energy analysis of reference solution over reference mission profile.

4 Proposed system improvement

As described in §3.1.1, the current standard situation for the hydrostatic transmission of a heavy tandem roller, is to size the hydraulic motors for the maximum duty point. This can largely be generalized to other hydrostatic transmissions, and the maximum duty point can be referred to as the maximum breakout torque, which in the case of the heavy tandem roller, has been shown to be the torque required to climb the maximum gradable slope of the machine. The sizing of the hydrostatic transmission is therefore made while disregarding the number of occurrences of, or time spent at, this maximum breakout torque condition, thus ignoring energy efficiency.

4.1 Sizing philosophy

In order to bring energy efficiency into focus, a new "sizing philosophy" is required. Sizing a hydrostatic transmission for maximising energy efficiency entails to size the hydraulic motors to try minimising system losses for the most frequent working conditions of the machine. In the case of the heavy tandem roller, the most time is spent in compaction mode, at low speed, on the flat, or on moderate slopes. Using Equation 1, it is possible to determine the motor torque required for stabilised conditions, but torque is also required to accelerate the machine up to the stabilised speeds depicted in Figure 2. This acceleration torque is added to Equation 1 to yield:

$$T = \frac{M \cdot R_{Drum}}{2} \cdot \left(g \cdot (\sin(\arctan(s)) + C_{rr} \cdot \cos(\arctan(s))) + \frac{\mathrm{d}V}{\mathrm{d}t} \right) \tag{2}$$

From this, and the maximum available differential pressure in the system, the minimum hydraulic motor displacement required at all times can be computed. Figure 6 displays a normalised time distribution of this required displacement, with the discrete distribution in the thin blue stair plot, and an approximated distribution in thick blue line. It shows that the required displacement very seldom exceeds half of the displacement determined in §3.1.1. In fact, 90% of the time, a displacement of 1370 cc/rev is sufficient to realise the mission of the machine.

By setting the motor displacement to 1370 cc/rev, the flow in the hydraulic circuit is reduced, and the pressure is increased, which brings hydraulic components into more efficient operating conditions zones. However, this efficiency increase is at the cost of the unattainable operating points that the lower motor displacement forbids.

4.2 Characteristics of the new radial pistons cam-lobe motor

To solve the problem of these heavy duty operating points, a new radial pistons cam-lobe motor (RPCLM) is designed, which is able to provide additional torque for limited periods of time.

4.2.1 RPCLM operating principle

A radial pistons cam-lobe motor is a kind of fixed displacement hydraulic motor, on which pistons are arranged radially in a cylinder-block which is acting as the rotor. The stator is placed around the rotor, and has a specific internal profile designed to force a reciprocating movement of the pistons in

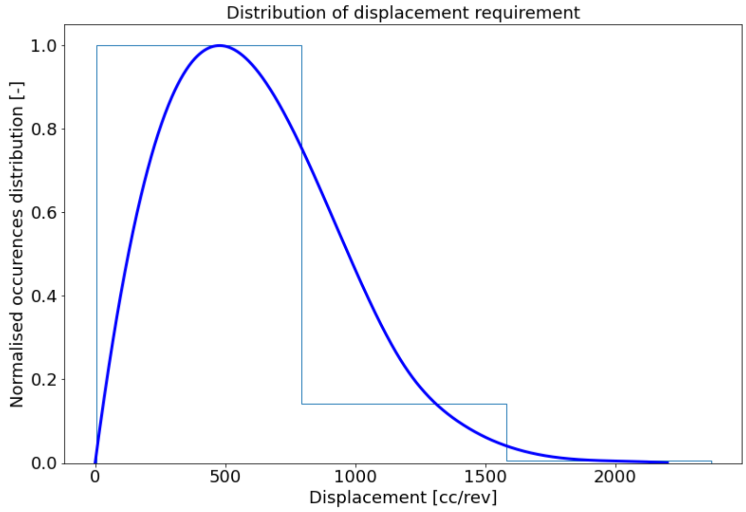


Figure 6 Normalised distribution of displacement requirement over the reference mission profile.

their chambers several times per rotation of the rotor. Isaksson in [10], and Giavarini in [11], provide descriptions of the operating principle of examples of such hydraulic motors. Figure 7 shows a cut-view schematics of the central part of the hydraulic motor. On this illustration, the cam (stator) is coloured to represent the zones on which pistons are submitted to high (dark grey) and low (light grey) pressure. In the position depicted, the rotor is rotating in the counter-clockwise direction, the high pressure zones corresponding to a piston moving out of its chamber.

A RPCLM is a fixed displacement motor, but it is possible to make use of internal symmetries to create several "sub-motors": by creating an internal bypass loop, the motor can work with a reduced apparent displacement. It is frequent to have RPCLMs with two discrete displacements, and some commercially available RPCLMs have up to four distinct displacements. As an example, the RPCLM shown on Figure 7 is able to work with a second displacement by bypassing three of the six cam lobes.

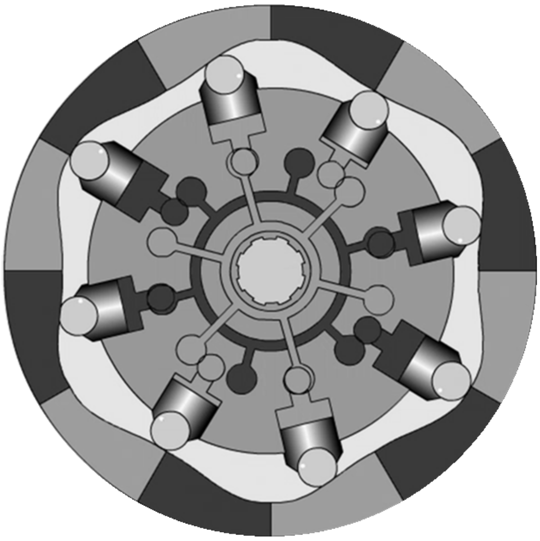


Figure 7 Illustration of the RPCLM "torque module" in cut view.

4.2.2 Particularities of the new RPCLM

The proposed new RPCLM design is a fixed one-displacement motor of the type described in the previous paragraph. The torque provided by a hydraulic motor is:

$$T = \Delta P_M \cdot D_M \tag{3}$$

From Equation 3, two ways of modifying the provided torque appear:

1. Changing the displacement of the motor D_M.
2. Changing the differential pressure at the motors ports ΔP_M.

While the usual way of dealing with torque requirements is to increase the displacement of the motor, the approach taken in the new RPCLM design is to act on the differential pressure. The maximum pressure in a system being enforced by several parameters (all components, be it the pump, the pipes and hoses, or the valves, have a maximum sustainable pressure value), an internal pressure modifier is used to locally change the pressure in the hydraulic motor without affecting other components of the hydrostatic transmission. This pressure modifier is used to amplify ΔP_M when – and only when – the differential pressure in the system is insufficient to provide the required torque.

The pressure modifier is entirely integrated to the RPCLM, and its use is transparent to the system. Figure 8 shows a schematics view of the new

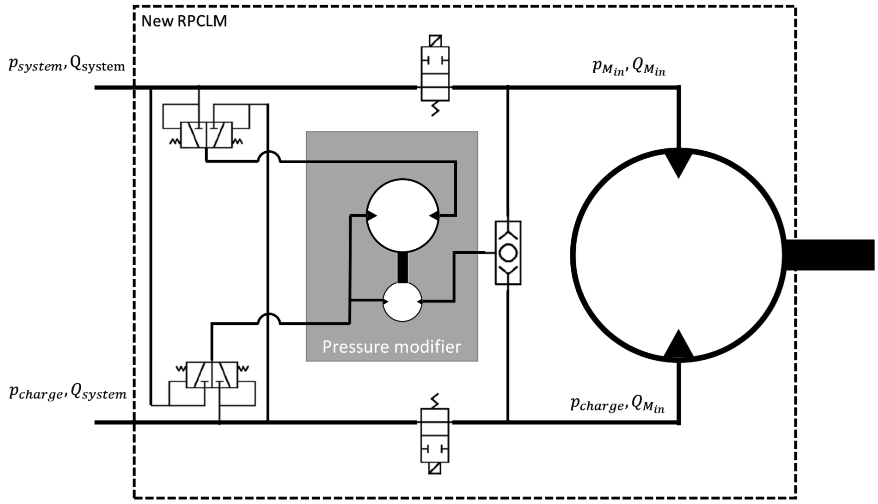


Figure 8 Schematics of the new RPCLM principle.

component, with the dashed line depicting the bounding box of the motor. A control block is included inside the motor, which commands the pressure amplifier's activation automatically when a pressure higher than can be supplied by the hydraulic circuit is required. The details of the exact design being currently in a patenting process, it is not detailed further in this paper.

When the amplifier is inactive, the pressure and flow at the motor's inlet $(p_{M_{in}}, Q_{M_{in}})$, are equal to the pressure and flow in the system (p_{system}, Q_{system}) and the same is true for the outlet: $(p_{M_{out}}, Q_{M_{out}}) = (p_{charge}, Q_{system})$. Because the RPCLM design ensures that $p_{M_{out}}$ is always equal to p_{charge}, and $Q_{M_{out}}$ is always equal to $Q_{M_{in}}$, both variables have been directly substituted on Figure 8.

When the pressure modifier is activated, there is a conversion of flow and pressure, where:

$$p_{M_{in}} = p_{system} \times r_{amp} \tag{4}$$

$$Q_{M_{in}} = Q_{system} \times \frac{1}{r_{amp}} \tag{5}$$

Due to the law of power conservation, the pressure modifier transforms a low pressure and high flow rate power into a high pressure and low flow rate power: $p_{M_{in}} \cdot Q_{M_{in}} = p_{system} \cdot Q_{system}$. In addition to this reduced flow, the amplification of pressure comes at the cost of the transformation efficiency η_{amp}. However, there is no impact of the pressure modifier when it is inactive, only the control block incurs some small pressure losses, which means that the motor efficiency is preserved during standard use phases.

4.3 Integrating the new RPCLM in the system

A version of the new "boosted radial pistons cam-lobe motor" (bRPCLM) is designed to fit the heavy tandem roller application described in §3.1.2 and this bRPCLM is substituted to the motors in the system shown in Figure 4. According to the required displacement derived in §4.1 based on the normalised time distribution of required displacement shown in Figure 6, the bRPCLM displacement is set to 1370 cc/rev, and an effective amplification ratio $r_{eff} = 2.1$ is targetted. The model described in §3.2.1 is otherwise unchanged. The bRPCLM model replacing the RPCLM model of the reference system includes the pressure modifier model, itself including physical models for hydro-mechanical and volumetric losses, as well as the control block and pressure losses incurred by its valves.

4.4 Modified machine energy efficiency analysis

The modified model of the machine is made to run over the reference mission profile of §2.4, and the same results as for the reference simulation of §3.2.2 are extracted.

The total fuel consumption for the machine equipped with two bRPCLM motors is $2.3L$ over this duty-cycle.

5 Comparison and discussion

Both the standard solution described in §3.2.1, and the proposed concept of bRPCLM introduced in §4.3 have been simulated over the reference mission profile of §2.4.

5.1 Fuel consumption reduction

As the results of §3.2.2 and §4.4 show, the fuel consumption of the proposed solution is reduced from 2.5 to $2.3L$ over the reference mission profile. Figure 9 shows the two systems fuel consumption in function of the time. The integration of the two bRPCLM instead of standard motors allows a 7.7% reduction of the fuel consumption of the heavy tandem roller over its reference mission profile.

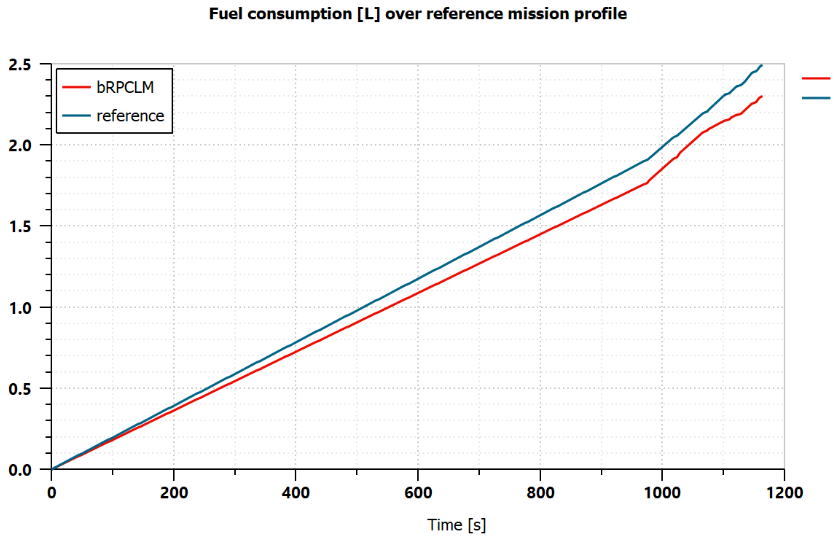


Figure 9 Comparative fuel consumption of reference and improved solutions.

5.2 Efficiency gain

The fuel consumption improvement brought about by the proposed bRPCLM concept can be attributed to the principles introduced in §4.2. The re-focusing of the hydraulic motors efficiency towards the work conditions of the machine bring efficiency improvements on the component itself, and the reduced displacement reduces losses in the hydraulic system. During the compaction work at stabilised speed, the bRPCLM efficiency is improved by 3.3% over the standard hydraulic motors of the reference solution.

Looking at the hydraulic energy at the outlet of the pump shows the total efficiency improvement allowed by the modified system. The bottom part of Figure 10 shows the differential hydraulic energy at the pump's ports in Wh in function of the time for the reference and modified solutions. At the end of the mission profile, the proposed solution has required 22.4% less hydraulic energy than the reference solution, for the same energy at the machine's drums. The top part of Figure 10 shows the energy at the pump's shaft, for which the bRPCLM equipped machine has required 17.3% less energy at the end of the cycle. Further gains in fuel consumption are therefore possible, by reducing the pump's and the thermal engine's displacements to fully convert the 22.4% hydraulic energy improvement into fuel consumption reduction.

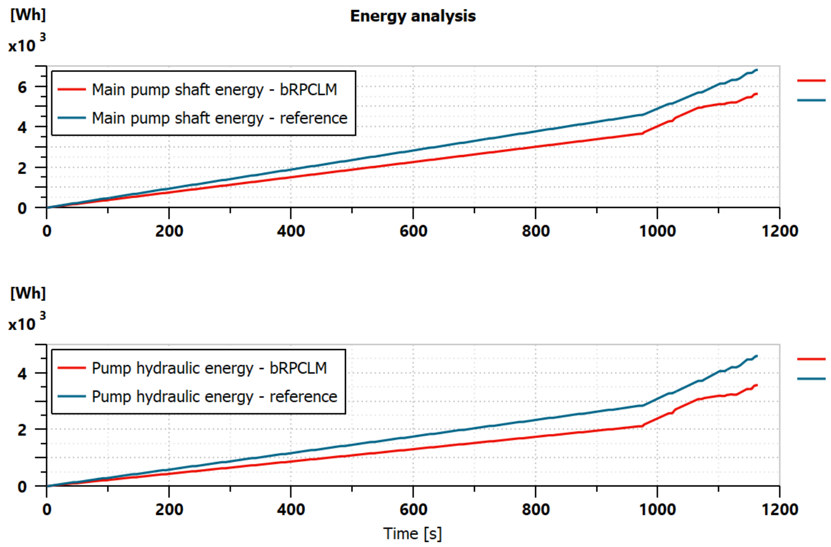


Figure 10 Comparative energy analysis of reference and improved solutions.

5.3 Comparative strengths and weaknesses of the sizing approaches

Radial piston cam-lobe motors are fixed displacement motors, with the possibility to create sub-displacements. Alternative technologies, comprising an infinitely variable axial piston motor coupled to a reducer (APMGB) also allow the reduction of the motors displacement to best fit the operating point. In the case of the bRPCLM, the displacement is smaller, but pressure is amplified when additional torque is required.

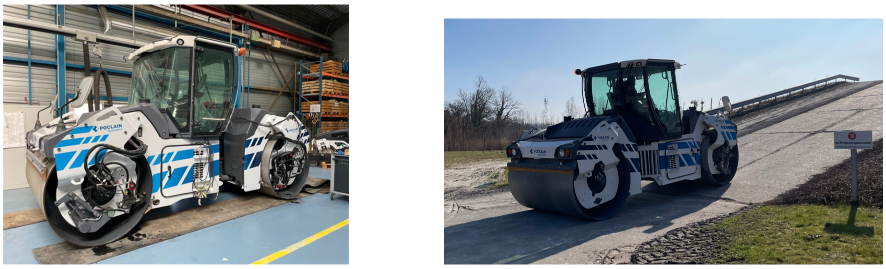
The standard sizing methodology described in §3.1 applies to both RPCLM and APMGB, while the bRPCLM is designed to apply the sizing methodology introduced in §4.1. As highlighted by the mission profile of §2.4, most of the time is spent at lower duty points, making the efficiency of the transmission on these operating points an important part of the energy efficiency of the machine. This is the purpose of the bRPCLM, which targets maximum efficiency during work mode, when lower torque levels are required, and tries to moves any inefficiencies to the maximum duty points, when the pressure amplifier is activated.

The amplification ratio of the pressure modifier implies a limitation of the rotation speed of the motor in high torque mode. For example, with an amplification ratio r_{amp} of 2.5, the transformation efficiency will mean the flow available for motor rotation $Q_{M_{in}}$ is less than $\frac{Q_{system}}{2.5}$, and the pressure available to create torque is less than $2.5 p_{system}$. With an efficiency of 80% for the transformation, this means an amplification of pressure by a factor of 2, while the rotation speed of the motor is divided by 3.125. This limitation of the proposed concept seems acceptable for the heavy tandem roller application, as operating points requiring high torque are low speed points. For other applications, it could lead to lower speeds than would ideally be wished for, a concession that is to be put on the balance in front of the expected energy efficiency gains.

The pressure modifier being autonomous in its activation, it brings simplicity into the system's control when compared to a multi-displacement RPCLM or an APMGB, both of which require some degree of control algorithms and valves to control the displacement value.

6 Experimental validation

The machine modelled and simulated in this study has been selected because of its availability to be tested with the two hydraulic system configurations. As no road is available to be compacted, the testing and correlations are done on a simplified use cycle, using flat asphalted ground and a concrete ramp with a 20 % slope. Figure 11 shows the heavy tandem roller equipped for testing.



(a) (b)

Figure 11 Test machine parked (a), and in a 20 % slope (b).

These tests have confirmed the fuel consumption improvements witnessed in the simulation results. They were also used to fine-tune the machine characteristics for the simulation (head loss in the pipes for example), and test the hypotheses concerning rolling resistance values discussed in §2.4.1.

Fuel consumption was measured at 0.11 L in average for a 50 m run on the flat asphalted test road at 5.5 km/h. This result was obtained by averaging a number of runs of the same cycle. The machine equipped with bRPCLM showed improvements in the fuel consumption over the same duty-cycle, with an average of 0.07 L of diesel used over the 50 m. This improvement, more consequent than what the model predicts over the mission profile used in this study (§5), is partly due to a reduced engine speed, from 2300 to 1800 rev/min for the bRPCLM test, a behaviour the model is able to reproduce, and an avenue of improvement that will be part of the next steps in the study. Adjusting for the ICE speed using the model, a fuel consumption reduction of 6% can be inputted to the bRPCLM, the rest of the improvement coming from the lower speed permitted by the smaller displacement of the bRPCLM.

The tests available at the moment of writing were performed with a prototype version of the bRPCLM, the efficiency of which is not up to par with the design used in this study, which might explain the slightly lower gain in testing (6% in testing vs 7.7% in this study). Testing this improved bRPCLM design is the next step in the project.

7 Conclusion

In this paper, a new radial pistons cam lobe motor integrating a pressure amplifier and a control block is proposed. This bRPCLM aims at refocusing the sizing of the hydraulic transmission towards its most frequent usage points. The traditional sizing point of the transmission for required breakout torque is addressed by the inclusion of the pressure amplifier, which allows a punctual increase of the output torque of the motor by an increase of the internal pressure in the component, pressure rise which is invisible to the system itself, allowing the use of standard components (e.g. pump, valves, pipes).

The demonstration of the energy efficiency gains obtainable through this new hydraulic motor concept is made via the simulation of the energy efficiency of a heavy tandem roller over a reference duty-cycle constructed for the study. The comparative simulations show a 7.7% reduction of fuel consumption on the machine over the twenty minutes reference cycle. The simulation results also indicate that further fuel economies could potentially be made by adapting the pump and diesel engine, as the mechanical energy required on the pump's shaft is reduced by 17.3% by the use of the bRPCLM.

Appendix

Table 2 Full machine parameters.

M	[kg]	13300	Compactor mass
s_{max}	[%]	39	Gradability
C_{rr}	[kg/t]	10	Rolling resistance coefficient on max slope
R_{Drum}	[m]	0.7	Drums radius
d_{drums}	[m]	3.9	Wheel base
p_{max}	[bar]	400	Max system pressure
p_{charge}	[bar]	20	System charge pressure
$\mathcal{D}_{pump}$	[cc/rev]	90	Pump max displacement
$\mathcal{D}_{RPCLM}$	[cc/rev]	2812	Initial motor displacement
$\mathcal{D}_{bRPCLM}$	[cc/rev]	1370	Improved motor displacement

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