Evaluation of Dynamical Behaviour of Long Heavy Vehicles Using Performance Based Characterstics

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EVALUATION OF DYNAMICAL BEHAVIOUR OF LONG HEAVY VEHICLES USING PERFORMANCE BASED CHARACTERSTICS

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ABSTRACT– Long Heavy Vehicle Combinations (LHVCs) are an attractive alternative to Conventional Heavy Vehicle Combinations (CHVCs) in goods transportation because of reduction in fuel consumption, reduced costs and emissions. One major issue concerning LHVCs is their potential impacts on traffic safety that is the most controversial issue of LHVCs. Road safety performance of LHVCs depends on their technical features such as power train and braking systems capability, lateral dynamical stability, manoeuvrability and etc. By introducing LHVCs as a part of future transportation, there is a need to ensure that they are performing within specific boundary conditions. Defining proper technical characteristics, denoted as Performance Based Characteristics (PBCs) for LHVCs would assist to create operational requirements by which LHVCs will be allowed to operate in the road network with less negative road safety impacts. A comprehensive list of safety related PBCs is defined in (1).

In this paper the test conditions and methods for each characteristic are specified and a set of heavy vehicle combinations (including European vehicle combinations, modular vehicle combinations and perspective modular vehicle combinations) are assessed and compared by using computer simulations. In addition, it is shown that by design parameter changes it is possible to improve the performance of a truck with double centre-axle trailers to perform almost similar as existing European conventional vehicle combinations like a truck and centre-axle trailer combination.

1 INTRODUCTION

Freight transport has dramatically grown during the past decades and accounts for approximately 45% of total transports (tonnes-km) within EU. The freight transport is expected to have an increase by 55% between years 2000 and 2020, (2). In another forecast about future transport performed by the European Commission, inland transport (truck, rail and inland waterways) is foreseen to grow by almost 53% (tonnes-km) from 2005 to 2030 within EU27, with a growth rate of 43%, 79% and 39% for truck, rail and waterways transport, respectively, (3).

The rapid growth in the road freight transport led to an increased demand for heavy vehicle transportation which results in major concern on environmental effects, road freight traffic and increased infrastructure usage. Increasing cost of fuel, congestions and gas emissions make Long Heavy Vehicle Combinations (LHVCs) an attractive alternative to Conventional Heavy Vehicle Combinations (CHVCs) in goods transportation. Longer length of LHVCs results in larger volume compared to CHVCs that enables more loading capability.

Several research projects carried out to analyse the potential benefits of longer and heavier combinations. In a Swedish study, it has shown that by using longer combinations, there will be 15% reduction in fuel consumption and emissions. Moreover, the cost savings will be about 23% and the number of trips will be decreased by 32%, (4). In another study performed by Woodrooffe et al in Canada and Alberta, it is concluded that the use of LHVCs represents a reduction of 44% in heavy vehicle-kilometres travelled, an saving of about 29% for operational costs for users of truck transportation services, around 32% reduction in fuel consumption and greenhouse gas emissions and 40% reduction in pavement wear, (5). The obtained results from different studies showed economical benefits in terms of cost, energy and environmental effects.

Another major issue concerning LHVCs is their potential impacts on traffic safety, that is the most controversial issue of LHVCs and there have been several conflicting opinions and conclusions about LHVCs safety issues. The group who are not agreed with having longer combinations mentioning that LHVCs are unsafe. They are arguing that LHVCs take more space, their acceleration can be slower and they might take more space than a

lane width or go into lanes of opposite direction, (6). In addition, it is stated that LHVCs cause longer overtaking time for other road users and increased visibility problems for both combination driver and other road users which results in additional road safety risks, (7). On the other hand, the opponent groups are arguing that using LHVCs instead of CHVCs to transport the same amount of goods results in fewer number of vehicles on the road that positively influences the traffic flow and therefore decreases the probability of traffic accidents, (6,8,9).

In a study performed by SAFER (10), the effect of increased length of vehicle combination on the severe crash rates by travelled vehicle-kilometers was investigated. They classified vehicle combinations in three groups based on the length, short (<= 12 m), medium (12.01-18.75 m) and long (18.75-25.25 m). The results of this study showed that the crash rate for short, medium and long vehicle combinations was 137, 56, and 44 severe accidents or fatalities per billion vehicle kilometer travelled. This indicates that short vehicle combinations were involved in severe accidents or fatalities per billion vehicle-kilometers travelled about three times more than long combinations. Therefore, no evidence was found showing that longer combinations would be less safe than shorter ones.

Traffic safety depends on various factors such as the technical features and type of vehicles on the road, road infrastructure design and drivers behaviour. The road infrastructure are designed for typical conventional heavy vehicles, therefore using LHVCs instead of CHVCs might increase safety risks. LHVCs performances can be improved to perform safer by using new technical solutions such as design parameters improvements, using advanced braking adding propelled/steered axles and etc, (6). Road safety performance of LHVCs depends on their technical features such as power train and braking systems capability, lateral dynamical stability, manoeuvrability and etc.

Traditional safety regulations of heavy vehicle combinations are specifying prescriptive design requirements and limits that need to be fulfilled to achieve acceptable safety levels. These regulations are defined as a series of design rules and prescriptive vehicle limits which applying them put some restrictions on vehicle design but does not directly address the performance of the vehicle combination. By introducing Performance Based Characteristics (PBCs), the vehicle performance as the way that it interacts with the road network is determinant factor whether a vehicle is allowed on the road or not. PBCs are stating desired levels of safety that must be achieved. On the other hand, this gives more freedom and opportunities to develop technologies and design of vehicle combinations.

2 PERFORMANCE BASED CHARACTERSTICS

To evaluate performance of Heavy Vehicle Combinations (HVCs), a set of PBCs have been defined in (1). PBCs contain fourteen safety related characteristics that can be divided into two groups, longitudinal and lateral characteristics. The PBCs are listed as follows, (1):

- Longitudinal PBCs: startability, gradeability, acceleration capability, stopping distance, and down-grade holding capability
- Lateral PBCs: Rearward Amplification (RWA), Swept Path Width (SPW), High Speed Transient offtracking (HSTO), High Speed Steady-state Offtracking (HSSO), Yaw Damping Coefficient (YDC), Straight Line Offtracking (SLO), Lateral Clearance Time (LCT), Steady-state Rollover Threshold (SRT), and deceleration capability in a turn

Startability and gradeability characteristics indicate the ability of the vehicle combination to start from rest on an up-grade and maintain speed on an up-grade, respectively. Acceleration capability reflects the vehicle's ability to clear intersections and rail crossings etc. These three characteristics are power train- and tyre- related characteristics. Stopping distance and down-grade holding capability characteristics concern braking system, (1).

The steady-state rollover threshold, yaw damping ratio and deceleration capability in a turn are vehicle combination's characteristics reflecting the vehicle lateral stability. Rearward amplification, high speed offtracking and straight line offtracking indicate the trailers dynamic characteristics. Swept path width is concerning the vehicle combination manoeuvrability and insuring that the vehicle safely manoeuvres around corners. Lateral clearance time is another HVCs' lateral characteristics indicating the influence of combination's length in clearing the lane change manoeuvres, (1).

Compared to Australian PBSs, definitions of some of those common characteristics are modified and some new characteristics are added such as stopping distance, down-grade holding capability, lateral clearance time, high speed steady-state offtracking and deceleration capability in a turn. In this work a bicycle model of vehicle combination has been used. Hence two characteristics, steady-state rollover threshold and deceleration

capability in a turn, are not covered in this study; a two-track model of the vehicle combination is needed for these two characteristics.

3 SELECTIONS OF HEAVY VEHICLE COMBINATIONS CANDIDATES

A set of HVCs, shown in Table 1, is selected to evaluate the vehicle combinations' performance on the road. The first three combinations are *'European Vehicle Combinations'* for international traffic based on directive 96/53 EC with Gross Combination Mass (GCM) of 40 tonnes. According to Directive 96/53/EC, the maximum total weight allowed for the vehicle with the length up to 18.75 m is 40 tonnes.

The European Modular System (EMS) as an innovative approach for transportation tasks have been practiced in Sweden and Finland since 1997 which allows longer combinations built of the existing vehicle units (modules) to be used in the transport system. The maximum permitted length and weight of these combinations is 25.25 m and 60 tonnes, respectively. These vehicle combinations are classified in the table as *'Modular Vehicle Combinations'* in the Nordic countries. These combinations consist of either a truck or trailer as the lead unit and one short (7.82 m) and one long (13.6 m) modular units as towed units which depending on how they are connected together, various combinations can be defined, (2).

The five last combinations in the table are possible longer and heavier combinations named as *'Prospective Modular Vehicle Combinations'*. These modular combinations are made based on EMS concept by connecting different modules in different ways. Note that the rear axles of the motor vehicles have double tyres, shown with red colour, all other axles have single tyres, shown with green colour, except VCMC11 combination which only its first axle has single tyres and other axles have double tyres.

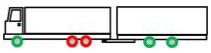
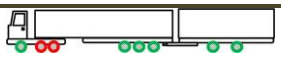
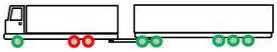
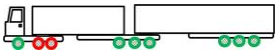
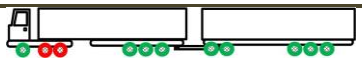
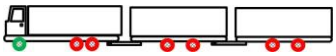
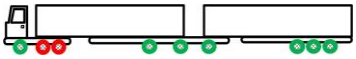
Type	Heavy vehicle combinations	Assigned name	Combination Scheme	Length/GCM [meters]/[tonnes]
European vehicle combination	Tractor and semitrailer	VCMC2		16.5/40
	Truck and centre-axle trailer	VCMC3		18.75/40
	Truck and full trailer	VCMC4		18.75/40
Modular vehicle combinations	Tractor, semitrailer, centre-axle trailer	VCMC5		25.25/60
	Truck, dolly and semitrailer	VCMC7		25.25/60
	B-double (7.82 m+13.6 m semitrailers)	VCMC9		25.25/60
Prospective modular vehicle combinations	A-double	VCMC10		31.5/80
	Truck and double centre-axle trailers	VCMC11		27.3/66
	B-double (13.6 m+13.6 m semitrailers)	VCMC13		30.9/80
	Truck and B-double(7.82 m+13.6 m semitrailers)	VCMC15		33.8/90
	B-triple	VCMC16		33.8/90

Table 1: Heavy vehicle combination classification

3 VEHCILE DYNAMICS MODEL

The method used for the investigation is system simulation. The system model, vehicle model and driver model, is simulated during different manoeuvres in order to determine the system performance. The vehicle model used in analysing PBCs is a bicycle model originated from the model introduced in (12) which has three degrees (deg) of freedom in longitudinal, lateral and yaw direction. Equations of motion of the combination' units are obtained from following equations, for each unit *j*:

$$\dot{V}_{x_j} = \left(m_j V_{y_j}\dot{\varphi}_j + \sum_{i=1}^{n_j} F_{x_{ji}} \cos(sc_{ji}\theta_j) - \sum_{i=1}^{n_j} F_{y_{ji}} \sin(sc_{ji}\theta_j) + F_{cx_{(j-1)}} \cos(\theta_j) - F_{cy_{(j-1)}} \sin(\theta_j) - F_{cx_j} - F_{res_j}\right) \Big/ m_j$$

$\dot{V}_{y_j} = (-m_j V_{x_j}\dot{\varphi}_j + \sum_{i=1}^{n_j} F_{x_{ji}} sin(sc_{ji}\theta_j) + \sum_{i=1}^{n_j} F_{y_{ji}} cos(sc_{ji}\theta_j) + F_{cx_{(j-1)}} sin(\theta_j) + F_{cy_{(j-1)}} cos(\theta_j) - F_{cy_j})/m_j$

$\ddot{\varphi}_j = (\{\sum_{i=1}^{n_j} F_{x_{ji}} sin(sc_{ji}\theta_j) + \sum_{i=1}^{n_j} F_{y_{ji}} cos(sc_{ji}\theta_j)\} l_{ji} + \{F_{cx_{(j-1)}} sin(\theta_j) + F_{cy_{(j-1)}} cos(\theta_j)\}a_j - F_{cy_j} b_j + M_{cz_{(j-1)}} - M_{cz_j})/I_j$ (1)

where $i=1:n_a$ and n_a is the number of axles per unit, $j=1:n$ and n is the number of units. Parameters m_j and I_j are total mass and yaw moment of inertia of unit j, respectively. Here, F_{cx0}, F_{cy0}, M_{cz0}, F_{cxn}, F_{cyn} and M_{czn} are zero and $[\theta_1\ \theta_2\ \theta_3 \ldots]=[\delta_f\ \ (\varphi_1-\varphi_2)\ (\varphi_2-\varphi_3)\ (\varphi_3-\varphi_4) \ldots]$ where δ_f is the front axle steering angle, φ_j is yaw angle of unit j. Parameter sc_{ji} is a coefficient assigned to the i_{th} axle of the j_{th} unit which is zero for unsteered axles and nonzero for steered axles. Vx_j, Vy_j and $\dot{\varphi}_j$ are longitudinal velocity, lateral velocity and yaw velocity of unit j, respectively. lji is the distance between CoG (centre of gravity) and the centre of the ith axles of the jth unit. aj is the distance between CoG and the centre of the front coupling point of the jth unit while bj is the distance between CoG and the centre of the rear coupling point of the jth unit. Note that a_1 and b_n are zero.

Coupling forces and moments, $[(F_{cx1}, F_{cy1}, M_{cz1}),\ldots,(F_{cxn}, F_{cyn}, M_{czn})]$, for articulation joints between units are obtained from the following equations:

$F_{cx_j} = c_j\Delta x_j + d_j\Delta V_{x_j} \quad , F_{cy_j} = c_j\Delta y_j + d_j\Delta V_{y_j} \quad and\ \ M_{cz_j} = d_{z_j}\dot{\theta}_j$ (2)

where lateral and longitudinal velocity differences, $[(\Delta V_{x_1}, \Delta V_{y_1}),\ldots,(\Delta V_{x_{(n-1)}}, \Delta V_{y_{(n-1)}})]$, on each coupling can be calculated as follow:

$\Delta V_{x_j} = V_{x_j} - V_{x_{(j+1)}} cos(\theta_{(j+1)}) - (V_{y_{(j+1)}} + a_{(j+1)}\dot{\varphi}_{(j+1)}) sin(\theta_{(j+1)})$

$\Delta V_{y_j} = (V_{y_j} + b_j\dot{\varphi}_j) + V_{x_{(j+1)}} sin(\theta_{(j+1)}) - (V_{y_{(j+1)}} + a_{(j+1)}\dot{\varphi}_{(j+1)}) cos(\theta_{(j+1)}$ (3)

Lateral and longitudinal slip angles of units' axles are presented in following equation:

$\alpha_{y_{ji}} = sc_{ji}\theta_j - (V_{y_j} + \dot{\varphi}_j l_{ji})/V_{x_j} \quad and \quad \alpha_{x_{ji}} = (R_w\omega_{ji} - V_{x_j})/min(R_w\omega_{ji}, V_{x_j}, \varepsilon)$ (4)

The Pacejka tyre model for combined longitudinal and lateral slip is used for calculating tyre lateral and longitudinal forces, F_y and F_x given in (13).

The driver model considered in this work is based on a Proportional, Integral and Derivative (PID) control design to follow a prescribed path by applying proper steering angle. The steering angle can be obtained using the following PID control:

$\delta_f = K_p e_p + K_i \int e_p + K_d \frac{d}{dt} e_p$ (5)

where e_p is the difference between the error signal between the reference path and a point; this point is either the path of the centre of front axle of the first unit in a single lane change manoeuvre or the path of the right outer corner of the first unit in a turning manoeuvre. K_p, K_i and K_d are proportional, integral and derivative gains of the controller, respectively.

The major resistive forces acting on the vehicle are aerodynamics driving resistance force, rolling resistance forces acting on tires and gravitational force due to the road slope which can be calculated using equations:

$F_{res_j} = F_{roll_j} + F_{air_j} + F_{g_j}$

$F_{roll_j} = C_r m_j g\, cos(\beta) \quad , F_{air_1} = 0.5\rho C A {V_{x_1}}^2 \quad and \quad F_{g_j} = m_j g\, sin(\beta)$ (6)

where β is the road slope and ρ is the mass density of the air, C the coefficient of the aerodynamic resistance, A the frontal area of the first unit, g the gravitational coefficient and C_r the coefficient of the rolling resistance. It is assumed that the values of F_{air_j} are zero except F_{air_1} which is calculated in the above equation.

The rotational acceleration of tires on each axle is calculated as follows:

$\dot{\omega}_{ji} = (T_{drive_{ji}} - T_{brake_{ji}} - R_w F_{x_{ji}} - R_w F_{roll_{ji}})/(J_w\, nn_{ji})$ (7)

where $T_{drive_{ji}}$ and $T_{brake_{ji}}$ are driven torque and braking torque on axles where the second and third axles of first units are driven axles and driven torque for other axles is considered to be zero. The parameter R_w is the tyre radius, J_w the inertia of the tire and nn_{ji} is the number on tyres on the i_{th} axle of the j_{th} unit. $F_{roll_{ji}}$ and $F_{x_{ji}}$ are rolling resistance and longitudinal force acting on tyres of the i_{th} axle of the j_{th} unit.

To ensure different units of a combination contribute with similar amount of braking effort in a heavy vehicle combination there should balanced distribution on each axle of each unit and also compatibility balanced between units in the combination. Assume that T_{brake_total} is required to have a desired rate of deceleration. To have compatibility balance between units, each unit braking torque will be obtained proportional to unit weights and axle's actual vertical loads as follows:

$T_{bj} = \frac{m_j}{m_{total}} T_{brake\_total}$ (8)

where T_{bj} and m_j are the braking torque and the mass of the unit j, respectively. m_{total} is the total Gross Combination Mass(GCM). Balanced braking between axles means each axle provides braking force proportional to the weight on that axle. Braking force on each is calculated as following:

$$T_{brake_{ji}} = min\left(\frac{w_{ji}}{m_{j1}}\, T_{bj}, \mu\, F_{zmaxji}\, R_w\right) \quad and \quad T_{brake_{11}} = min\left(\frac{w_{11}}{m_1}\, T_{b1}, \mu\, F_{zmax11}\, R_w, 25\ kNm\right) \tag{9}$$

where w_{ji} and F_{zmaxji} are actual the vertical load and the maximum road grip on the i_{th} axle of the j_{th} unit, respectively. $T_{brake_{ji}}$ is the braking force on the i_{th} axle of the j_{th} unit except the first axle of the first unit. The parameter μ is the road-tyre friction coefficient. $T_{brake_{11}}$ is the braking force on the first axle of the motor vehicle combination which is also limited by *25 kNm*.

In (11) typical data of axle loads, yaw moment of inertia for various units and tyre properties can be found. It is assumed that the motor vehicles in all vehicle combinations are equipped with identical front tyres and identical rear tyres and all trailers have identical tyres.

5 PERFORMANCE BASED CHARACTERSTICS ANALYSIS

Results of longitudinal and lateral PBCs assessed for HVCs, listed in Table 1, are shown in Tables 2 and 3. For these tests, the road surface considered to be uniformly hard and dry with a friction level of higher than 0.8. The same engine size, 750 hp, is considered for all combinations.

5.1 LONGITUDINAL PERFORMANCE BASED CHARACTERSTICS

Startability is the maximum upgrade, in units of percentage, that the combination is able to maintain steady forward motion when the vehicle speed is increasing. Moreover, the combination should be able to reach velocity of 10 km/h in 20 s. The aim of this characteristic is to ensure that the vehicle will not be stuck and has a proper startability capability on uphill. Another power train ability related characteristics is gradeability which is the maximum upgrade, in units of percentage, that the combination is able to maintain steady forward motion while driving by a speed of 80 km/h. Regarding acceleration capability characteristics, the performance level for assessed combinations is calculated as the time taken to accelerate from rest and travel 100 m distance while being fully laden. The objective of acceleration capability criterion is to assess the vehicle ability in clearing intersections, traffic lights and etc.

Startability, gradeability and acceleration capability highly depend on tyre/road friction levels, engine specifications (torque output versus engine speed), drive train specifications (gear and final ratios), vertical load on driven axles and GCM, (14). For instance, more load on driven axles result in more traction capability which means higher road grip.

HVCs / Longitudinal PBCs	VCMC2	VCMC3	VCMC4	VCMC5	VCMC7	VCMC9	VCMC10	VCMC11	VCMC13	VCMC15	VCMC16
Startability [%]	24	32	32	15	25	17	18	22	18	15	15
Gradeability [%]	4.1	5	4.9	3	3	3	2	2.6	2	1.7	1.7
Acceleration Capability [s]	10.5	10.2	10.2	12.4	11.7	12.2	12.3	12.1	13	13.6	13.6
Stopping distance [m]	39.7	39.7	39.6	37.9	37.7	40.4	35.8	38.7	41.8	45.8	40.9
Down-grade holding Capability [%]	8.7	8.7	8.7	6.1	6.1	6.1	4.9	5.7	4.9	4.5	4.5

Table 2: Longitudinal PBCs assessment

The stopping distance performance level for a fully laden vehicle combination during straight line full braking is calculated as the distance travelled by the vehicle to stop from an initial velocity of 80 km/h with a deceleration of 0.7 g. This characteristic is an evaluation of service braking system of HVCs while down-grade holding characteristics is assessing the engine braking system and retarders ability of HVCs. The driver reaction time is not included in stopping distance calculations.

Downgrade holding capability is the maximum downgrade, in units of percentage, which the combination is able to maintain steady forward motion using auxiliary braking while driving by a speed of 80 km/h. The engine braking can work in its full power for 20 s and after that it works with half of the power. This characteristic is assessed here considering that the auxiliary braking contains full power engine braking plus compact retarder for a time period of less than 20 s.

The combinations showed acceptable longitudinal PBCs on a dry road test condition. But it might be problematic while performing on a low friction road surface.

5.2 LATERAL PERFORMANCE BASED CHARACTERSTICS

Rearward amplification (RWA) is defined as the ratio of the maximum value of the motion variable of interest (e.g. yaw rate or lateral acceleration) of the worst excited following vehicle unit which is usually the last unit to that of the first vehicle unit during a specified manoeuvre. The rearward amplification can be determined in different manoeuvres using either a path-following single lane change or a single sine-wave steer input.

In this performance measure, the single lane change manoeuvre specified in ISO 14791 is used. The vehicle is driven with a velocity of 80 km/h in a single lane change manoeuvre that has a maximum lateral acceleration of 0.2 g and a steer frequency equal to 0.4 Hz. This metric highly depends on the number of articulation joints, combination type, length of wheelbase, steered axles, mass and inertia of units and position of coupling point.

HVCs / Lateral PBCs	VCMC2	VCMC3	VCMC4	VCMC5	VCMC7	VCMC9	VCMC10	VCMC11	VCMC13	VCMC15	VCMC16
Rearward amplification (RWA) of Lateral acceleration in CoG [$\frac{m}{s^2}/\frac{m}{s^2}$]	0.96	1.69	2.1	1.77	2.03	1.08	1.90	2.04	1.31	1.8	1.49
Rearward amplification (RWA) of Yaw rate [$\frac{rad}{s}/\frac{rad}{s}$]	1.03	1.53	2.37	1.92	1.90	1.09	1.81	1.61	1.37	1.87	1.49
Swept path width (SPW) [m]	5.86	5.51	5.27	6.86	6.55	7.80	8.21	6.93	8.92	8.72	9.42
High speed transient offtracking (HSTO)[m]	0.19	0.55	0.61	0.79	0.55	0.38	0.91	0.82	0.65	1.08	0.95
High speed steady-state offtracking (HSSO)[m]	0.31	0.34	0.43	0.5	0.51	0.45	0.64	0.43	0.62	0.7	0.67
Yaw damping coefficient (YDC)[-]	0.61	0.28	0.39	0.25	0.52	0.52	0.33	0.22	0.34	0.28	0.24
Straight line offtracking (SLO)[m]	0.10	0.11	0.13	0.17	0.16	0.17	0.22	0.15	0.23	0.24	0.25
Lateral clearance time (LCT) [s]	4.49	7.53	6.36	9.25	5.90	6.20	7.52	10.5	7.97	10.4	11.8

Table 3: Lateral PBCs assessment

Most often RWA of yaw rate and RWA of lateral acceleration are similar. However, there are some cases that the use of RWA of lateral acceleration as a stability criterion is very misleading. One such case is where friction between tyres and road is reduced (e.g. due to bad tyre or road condition) RWA of yaw rate increases significantly, but this increase cannot be seen in the RWA of lateral acceleration and instead it decreases. It is well known that the use of steered (especially self-steered) trailer axles has a very negative effect on the RWA of yaw rate such that this value increases significantly but RWA of lateral acceleration is not affected so much. Another issue considering RWA is practical difficulties to position lateral acceleration sensor at CoG, thus yaw rate measurement is more reliable because it will be consistent in the complete unit.

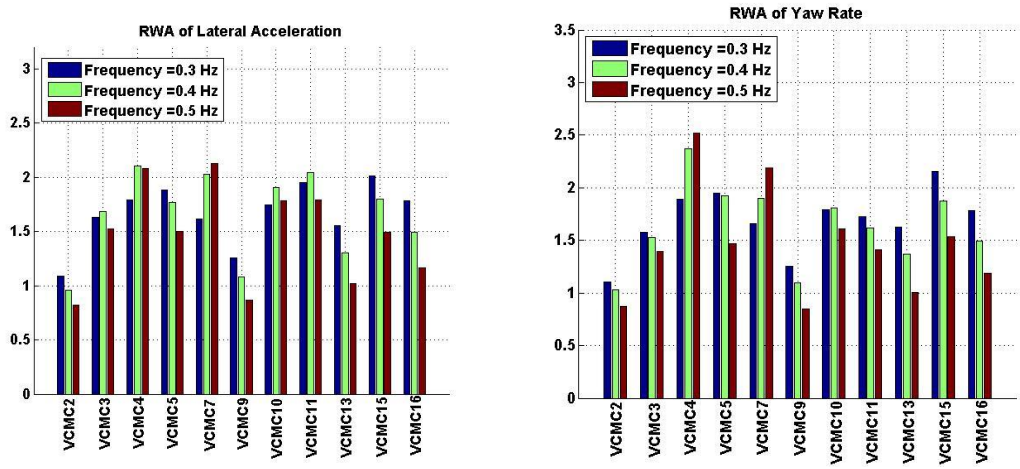


Figure 2: RWA of various combinations for three different steer frequencies and constant velocity of 80km/h (a) RWA of lateral acceleration (b) RWA of yaw rate

A significant factor influencing RWA is the type of combination. For instance, using full trailers result in larger RWA as can be seen in VCMC4 which has the highest RWA among the combination in Table 3. It is expected that by increasing the number of articulation joints get higher RWA, but RWA of European truck-full trailer, VCMC4, with one articulation joint is among highest values. As mentioned before, this is due to the type of units that are also influencing lateral stability of combinations.

Each combination might have its maximum RWA at different frequency of excitation as shown in Figure 2. In addition, RWA increases while the velocity increases. However, RWA results are highly influenced by velocity and frequency, in order to be able to compare combinations in Table 3, the frequency and velocity for this test considered to be 0.4 Hz and 80 km/h, respectively.

Swept Path Width (SPW) or Low Speed Swept Path (LSSP) is defined as the maximum width of the swept path between outer most and inner most points of the vehicle combination in a low speed turn. For this PBC test, it is assumed that the maximum full laden combination negotiates 90 deg, 12.5 m radius turn at a velocity of 5 km/h. For SPW characteristic, the radius of 12.5 m is considered based on directive 96/53/EC where it is mentioned that any vehicle combination must be able to turn within a swept circle having an outer radius of 12.5 m. The SPW level is influenced for instance by the trailers wheelbase and increases as the wheelbase increases. SPW can be decreased by using some techniques such as shortening units' wheelbase, increasing the number of articulation joints or using steered axles.

Any motor vehicle or vehicle combination must be able to negotiate with a turn with outer radius of 12.5 m and inner radius of 5.3 m according to directive 96/53/EC and swept path width shouldn't be larger than 7.2 m. This criterion cannot be applied for modular vehicle combinations used in Sweden and Finland because these combinations might not be able to fulfil this criterion due to their longer length and need more space to turn. Therefore, Scandinavian countries have used another legislation for required swept path width which states that modular vehicle combinations must be able to negotiate with a 360 deg turn of outer radius of 12.5m and an inner radius of 2 m, (2). Thus, the maximum permitted swept path width for modular vehicle combinations is 10.5 m according to Swedish regulations. According to Table 3, VCMC11 represents better SPW performance compared to other combinations in perspective modular vehicle combinations and VCMC16 which is one of the longest combinations in the list shown the worst performance.

As shown in Figure 3, all combinations can handle a 90 and 180 deg turn with a radius of 12.5 m. However, SPW levels of the 180 deg turn in the radius of 12.5 m for some combinations are very high. The combinations VCMC9, VMC10, VCMC13, VCMC15 and VCMC16 cannot negotiate a 360 deg turn with a radius of 12.5 m. On the other hand, all combinations can negotiate 90, 180 and 360 deg turn on a 20 m radius turn as shown in the figure. Larger turning angles results in higher low speed swept path width. Moreover, it is obvious that SPW is larger for either turns with smaller radius or turns with larger angles. For instance, SPW values are larger for 360 deg turn than 90 deg turn but how large this difference can be depends on the combination type.

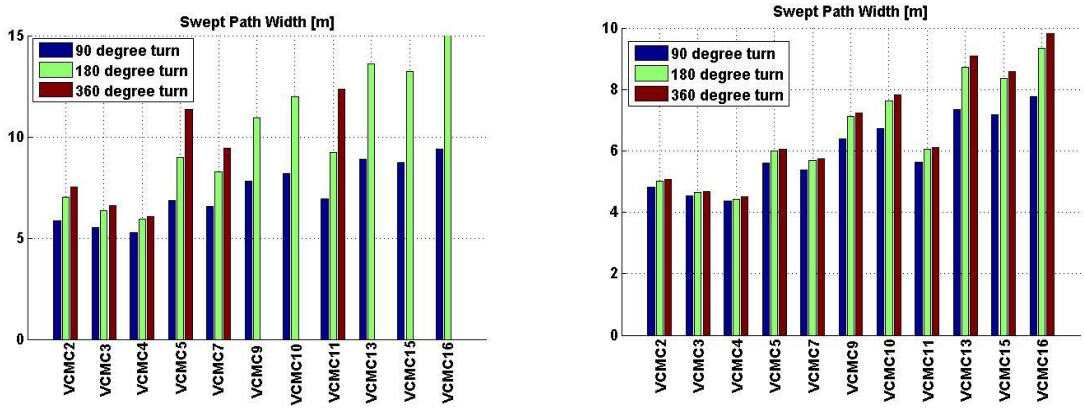


Figure 3: SPW for 90/180/360 deg turns for various combinations and velocity of 5km/h (a) 12.5m radius turn (b) 20m radius turn

Another characteristics considered here is High Speed Transient Offtracking (HSTO) which is defined as an overshoot in the lateral distance between the paths of the centre of the front axle and the centre of the most severely offtracking axle of any unit. The vehicle is driven with a velocity of 90 km/h in a single lane change manoeuvre that has a maximum lateral acceleration of 0.2 g and a steer frequency equal to 0.4 Hz which the results are shown in Table 3.The combination VCMC15 has shown the worst HSTO characteristics among the tested combinations.

High Speed Steady-state Offtracking (HSSO) is defined as the lateral offset between the paths of the centre of the front axle and the centre of the most severely offtracking axle of any unit in a steady turn. This performance measure is evaluated for a constant-radius curve of radius 319 m negotiated at a speed of 90 km/h with a lateral acceleration of about 0.2 g. The VCMC15 HSSO performance level is the worst among studied combinations.

Yaw damping coefficient is the damping ratio of the least damped articulation joint's angle of the vehicle combination during free oscillations actuated by a single sine-wave steering input with frequency of 0.4 Hz. The turn is performed at a speed of 80 km/h. Higher values of YDC are desired which VCMC11 showed the least value of YDC performance here by YDC value of 0.22 which means higher oscillations in articulations angle. Considering the performance level of 0.15 for YDC according to Australian PBS, all combinations show acceptable YDC values higher than 0.15.

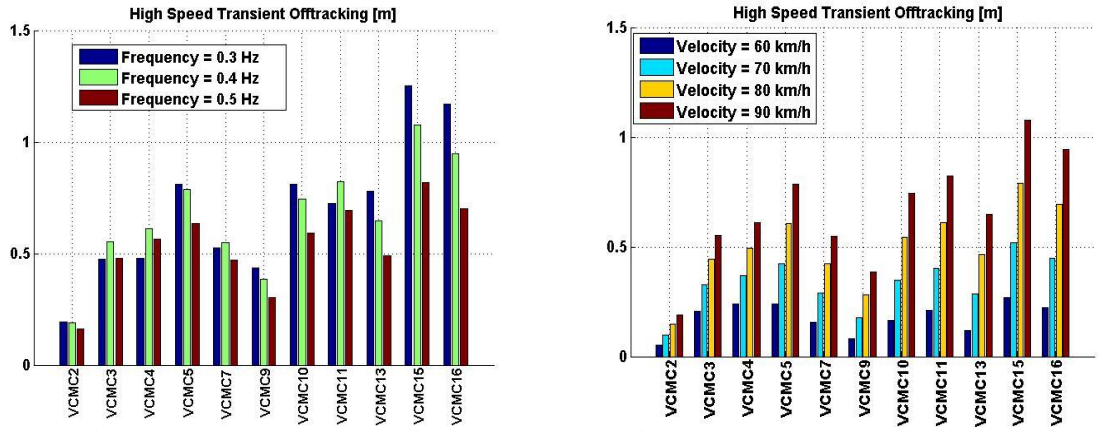


Figure 4: HSTO for various combinations (a) Various frequencies and velocity of 90km/h (b) Various velocities and frequency of 0.4Hz

Straight Line Offtracking (SLO) is defined as the maximum offtracking between the paths of the centre of the front axle and the centre of the most severely offtracking axle of any unit while traveling straight on a banked road. The vehicle being assessed is assumed to be driving at a speed of 90 km/h while following a straight path with a cross fall angle of 5%. This characteristic assessment results for various combinations is shown in Table 3.

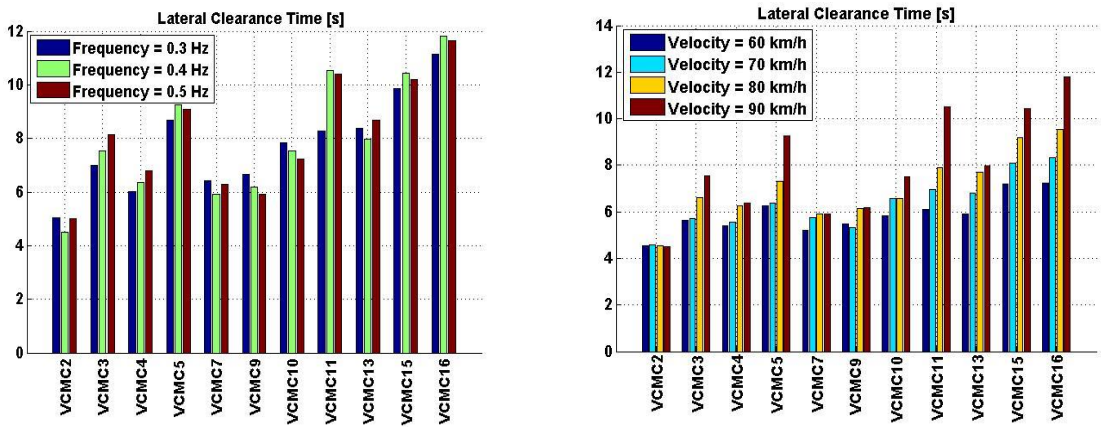


Figure 5: LCT for Various Combinations (a) Various frequencies and velocity of 90km/h (b) Various velocities and frequency of 0.4Hz

Lateral Clearance Time (LCT) is defined as the time taken by a combination to clear a certain lateral distance and to have the paths of the centre of the front axle and the centre of all units in the same line. Likewise HSTO, the vehicle is driven with a velocity of 90 km/h in a single lane change manoeuvre that has a maximum lateral acceleration of 0.2 g and a steer frequency equal to 0.4 Hz. This performance characteristic depends on both yaw damping characteristics of combination and the length of combination. As shown in Table 3, LCT level of VCMC16 is highest among other combinations. Figure 5 depicts LCT characteristic dependency on changes in longitudinal velocity and frequency of the lane change manoeuvring.

Lateral characteristics are highly dependent on combinations design parameters which some of them are combination and units type, number of articulation joints, wheelbase length of different units, axle distances, drawbar length, coupling position, type of tyres, steered axles and the centre of gravity of different units. These

factors influence different characteristics in different degree. For instance, increased in the number of articulation joints increases rearward amplification in general but decreases swept path width.

In the following section, lateral performance characteristics of one of the studied combination, VCMC11, will be investigated in both prescriptive regulations and PBCs scheme.

6 PERFORMANCE BASED CHARACTERSTICS VS PRESCRIPTIVE REGULATIONS

For this section, VCMC11 has been considered to be studied and see how the lateral PBCs can be influenced by changing some of design parameters. This modular combination consists of a rigid truck and two centre-axle trailers which carries three 7.82 m loading modules. The total GCM of the combination is 66 tonnes and its total length is 27.3 m, which are, more than permitted maximum weight of 60 tonnes and maximum length of 25.25 m in Sweden and Finland. But specified axles load are within the allowed range of axle loads.

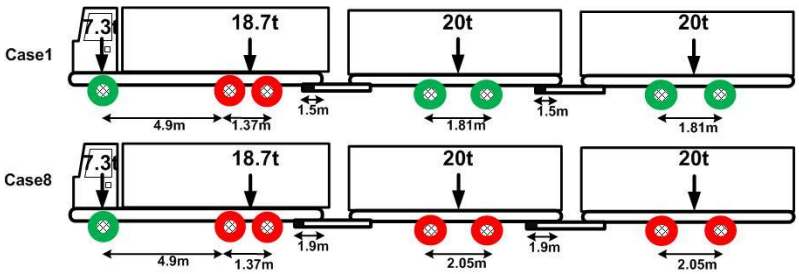


Figure 6: Truck-double centre axle trailers combination (VCMC11), Case 1 and Case 8

Figure 6 shows configuration of two different cases 1 and 8 which are listed in Table 4. The coupling distances for Case 1 and 8 are 1.5 m and 1.9 m, respectively. The distances between axles for both trailers are 1.81 m for Case1 and 2.05 m for Case8. Axles shown with red colour have double mounted tyres and axles shown in green colour have single tire. Trailers' tyre stiffness in Case 8 is 25% higher than Case1.

Different studied cases related to parameter changes compared to Case1 are listed in Table 4. For instance in Case 2, the trailers axles distances are increased by 0.24 m and in Case 3, twin tyres are used in the trailers axles. In Case 5, the coupling distances for both trailers are increased to 1.9 m and the length of drawbars were added by 0.4 m to get the same length of combinations as the combination in Case 1.

The cases in Table 4 are listed in order of improving RWA, YDC and LCT values. Moving coupling distance forward shows larger impact on RWA and YDC characteristics compared to changes in Cases 2, 3 and 4. For SPW it is opposite and by moving the coupling distance forward, the SPW becomes larger. By increasing the trailers tyre stiffness, HSSO, HSTO and SLO value becomes smaller compared to changes in Cases 2, 3 and 5.

Cases	**Changes**	**RWA** $[\frac{m}{s^2}/\frac{m}{s^2}]$	**YDC**	**SPW [m]**	**HSSO [m]**	**HSTO [m]**	**SLO [m]**	**LCT [sec]**
Case 1	VCMC11: Configuration shown in Figure 6	2.58	0.05	6.55	0.64	1.40	0.19	21.46
Case 2	VCMC11: Increased trailers axle distance by 0.24 m	2.55	0.06	6.55	0.64	1.39	0.19	20.46
Case 3	VCMC11: Using double tyres for trailer axles	2.53	0.08	6.55	0.57	1.28	0.17	19.61
Case 4	VCMC11: Increased trailers tyres stiffness by 25%	2.49	0.10	6.55	0.50	1.12	0.16	17.87
Case 5	VCMC11: Moving forward coupling points and increasing drawbar length by 0.4 m	2.25	0.13	6.92	0.58	1.16	0.19	12.88
Case 6	VCMC11: Case 2 and 5	2.23	0.13	6.92	0.59	1.16	0.19	12.9
Case 7	VCMC11: Case 2 , 4 and 5	2.13	0.18	6.93	0.48	0.92	0.16	10.99
Case 8	VCMC11: Case 2 ,3 , 4 and 5 (baseline configuration used in the section 5)	2.04	0.22	6.93	0.43	0.83	0.15	10.52
Case 9	VCMC4: Truck and full trailer	2.1	0.39	5.27	0.43	0.61	0.13	6.39

Table 4: Lateral PBCs assessment Results

Case 9 in the table indicates the lateral PBCs results for a truck-full trailer combination, which is a European vehicle combination with maximum length of 18.75 m and maximum total weight of 40 tonnes. Comparing this case with Case 8, which is VCMC11 with maximum length of 27.3 m and total mass of 66 tonnes, it can be found that VCMC11 in Case 8 has less RWA than VCMC4 and in other characteristic acceptable results comparing to VCMC4.

As shown, increasing the trailers wheelbase length, increasing drawbar length, forward coupling position, using double tyre on trailer axles and increasing tyres stiffness have a stabilizing effect one by one individually such that decreasing RWA, HSTO, HSSO, LCT while increasing YDC. On the other hand, only moving coupling position forward and increasing length of drawbar worsens SPW and in other study cases this change influences various characteristics in a higher extent.

As seen in the above example, by regulating HVCs using PBCs instead of existing prescriptive regulation, restrictions on maximum total weight and length can be released in some extents and by changing design scheme can have longer and heavier combinations on the transport system.

7 CONCLUSION

The main purpose of PBCs is to objectively evaluate different heavy vehicle combinations. The PBCs aim at capturing performance in low speed manoeuvring as well as high speed stability and longitudinal characteristics. A wide range of PBCs have been developed in many researches for the evaluation of HVCs' performance, for instance Australian performance based standards.

Many of these PBCs have already been introduced as industrial standards within ISO such as rearward amplification, offtracking, yaw damping and etc. In this paper, couples of new ones are defined such as down-grade holding capability, lateral clearance time, straight line offtracking and high speed steady-state offtracking.

A set of HVCs have been chosen and their dynamical performances have been compared based on defined lateral and longitudinal PBCs. Evaluation of longitudinal PBCs for these combinations showed acceptable performance level on the dry road test condition. Lateral PBCs results show that HVCs performance level depends on design parameters that can be improved in the design stage.

The introduced PBCs allow innovation in the vehicle design by exceeding the existing legal prescriptive mass and dimension limits under condition that the proposed vehicle's design meets the PBC rules. This results in significant improvement in LHVCs dynamical performances and achieving higher productivity and road safety.

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