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NUMERICAL MODELING AND TESTING FOR HILL START ASSIST IN HEAVY LOAD VEHICLES

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Abstract

Vehicle Hill-start assist (HSA) system is designed specifically with intention of developing safety in the heavy load automobiles during the uphill. In the anti-lock braking system (ABS), HSA plays an integral role in avoiding accidents and ensuring the optimum brake pressure is supplied to the system to have braking and hold. The paper discusses about the dynamic analysis and real time validation of the HSA proto model developed. Performance test and the leakage test are performed to evaluate the actual performance of the system. The results of the tests were promising and inline to the expected standards. The vehicle testing was done on the truck with various combinations like boost pressure, high grade boost pressure and mild grade with boost pressure. The results were compared to check whether, the HSA performance varies as per the response expected.

Keywords: ABS, dynamics, HSA, results, tests, validation

1. INTRODUCTION

Hill start assist system is a value-added component to ensure the safety of any vehicle. The most important parameter part is to ensure the safety that the vehicle does not roll back and have enough boost pressure while climbing up-down the hill. Pneumatic systems are used for the braking in heavy load vehicles. These are basically of two types ABS (APAC and NA)-EBS (European region). The braking components include air generating unit (compressors), air processing unit, multi circuit protection valves, hand and foot braking components, various sensor valves, and actuators [1] [2]. The motions that usually are the reason for the accidents in heavy load vehicle are lateral motion, jack knifing or combination of both. The disturbance cause vehicle losing stability resulting to accident [3]. The accidents are majorly classified a preventable, potentially preventable, non-preventable and preventable unknown based on the statical analysis. The research concluded that around 50% of the accidents could be avoided if it would have some warning or safety alarm device that would keep driver alert [4].

2. LITERTATURE REVIEW

With the ministry of road and transportation focusing on the safety, it was announced in 2014 that the new vehicles will have a mandatory requirement of the ABS. The main purpose of the ABS is to control the vehicle during sudden barking when the vehicle is at high speed by locking-unlocking at very high speed by controlling pressure to wheel. A report suggested that India tops the list of the accidents as in 2012. In overall accidents, trucks, trailer, buses are involved for more than 30% [5]. Vehicles are mainly segregated two categories as cars and truck-bus/trailer. With the awareness and raise in safety, the heavy load vehicles with hill start assist system is expected to be lead in demand. The market includes Asia Pacific, Europe, North America, and the Middle East and Africa. Growth is expected in these areas. The growing market and strong automotive industry help this growth. China, Japan, and India stand out in the Asia Pacific region due to support from government norms [6]. It was established that mass plays a major role in the vehicle stability. Hence it was necessary to determine these parameters accurately. To determine the road grade and vehicle load, McIntyre and Ghotikar attempted to create a two-stage Lyapunov-Based Estimator in 2009. The least squares estimation method was used to estimate the vehicle mass and road gradient. The clutch was disengaged once the results were confirmed for both braking phases [7]. Dozio and Mandrioli (2009) investigated a controlled system for the hill start assistance in heavy load vehicles via torque detection and vehicle movement detection prior to EPB release. Three stages of the study were conducted: the first covered vehicle behavior, the second covered the detection system, and the third covered the control method [8]. Balasubramanian, in 2015 published a research paper which discusses about the sensor-less hill assist, which an electric light-duty vehicle's traction control could accomplish. The PID controller was conceived and constructed to ascertain the holding torque required for the vehicles. Through a controlled loop, vehicle load and slope data were sent into the PID as inputs. Throughout the investigation, mathematical representations of the rollover force were also developed, accounting for variations in the mass factor [9]. Carlos and Nuss de Souza proposed a solution which could work by communicating with ABS system to provide the brake force on wheels exactly as requested without driver holding brake pedal. The solution makes it independent of slopes, configuration of vehicle and temperature of brake [10]. In 2012, there was research was done focusing on the heavy truck with the AMT's hill start control. A new feature requires the driver to press a push button as soon as the truck begins to climb a hill. By using pneumatic actuation, the control strategy would switch the gear to the appropriate one. As soon as the driver releases the hand brake, the air is ready to pass through ABS valve which is controlled by ECU. [11] To avoid the use of sensors and harmonization the vehicle type, Kolte and Kurup did an investigation to develop an anti-roll back prevention system. They used Ratchet and Pawl mechanism. The ratchet is fitted in the rear drive shaft and the Pawl is fitted with the frame. If the vehicle is on the slope, the lever makes pawl touch the ratchet [12].

3. NUMERICAL MODELING AND DYNAMICANALYSIS OF THE VEHICLE

3.1 MATHEMATICAL MODEL of EPB

Parking brake consoles, solenoids, electrical relay valves, air tank, and other sensing devices builds up the EPB model. With the EPB system, an electronic switch takes the role of the parking manual control valve. To improve control, the EPB system is equipped with an acceleration and air pressure sensor. The air pressure in the braking chamber is immediately detected by the air pressure sensor at the control port of the relay valve. [13]. When the FBV is pressed and the vehicle begins to move uphill, ECU receives a signal. The requirements are verified by EPB. When it determines that the conditions for release have been satisfied, the parking brake system automatically releases the parking brake force. Compressed air from the air compressor enters the relay valve's control port and opens its output port when the ECU triggers the solenoid valve. The solenoid valve of the EPB system acts as an on/off valve when the HSA system is in operation, and it is used to open and close the parking brake chamber to regulate the pressure inside. Dynamic characteristics of solenoid valves include mechanical, magnetic, and circuit component models. [14]

Solenoid valves electromagnetic model can be written as

$$U = IR + N * \frac{d\emptyset}{dt} \quad (1)$$

$$IN = \frac{\sigma\emptyset}{\mu Ac} \quad (2)$$

I - coil's operating current (A)
U - solenoid valve's operating voltage (V)
R - coil's internal resistance (Ω)
N - coil's total number of turns
t – time, sec
$\emptyset$ - coil's magnetic flux

The mechanical component's dynamic model can be expressed as below,

$$\ddot{x} = \frac{1}{me}[\emptyset^2/2\mu_e A_c - P_e A_e - K_e(x+x_p) - c\dot{x}] \quad (3)$$

m_e - mass of valve core
x - movement distance of valve core
c - valve core movement's equivalent viscous damping.
K_e - solenoid valve's return spring stiffness.
P_e - control port's air pressure.
x_p - spring preload
A_e - valve core's cross-sectional area
A_c - gaps cross-sectional area
μ_e - air permeability

Based on the standard parameters and components of braking components, table 1 shows the critical parameters from a Knorr Bremse brake component.

Table 1: Critical variables of pneumatic EPB system

Variables	Unit	Value
Solenoid Voltage	V	24DC
Resistance of coil	Ω	41
Cross section area of Air Gap	m^3	$4.72\text{x}10^{-5}$
Spool quality	kg	0.02
Solenoid valve hole diameter	cm	0.22
Coil wounds	-	600
Air Gap Length	m	0.001
Spool cross sectional area	m^2	$5.03\text{x}10^{-5}$
Piston mass of Relay Valve	kg	0.33
Piston diameter of Relay Valve	mm	80
Relay Valve diameter	mm	24
Piston return spring stiffness	N/mm	0.77
Valve return spring stiffness	N/mm	5.39
Spring preload of return Piston	N	10
Preload force of Valve return spring	N	140
Piston diameter of Brake Chamber	mm	120
Piston rod diameter of Brake Chamber	mm	25
Volume of Brake Chamber	m^3	0.00265

The parking brake chamber's dynamic equation, excluding heat transmission, is as follows.

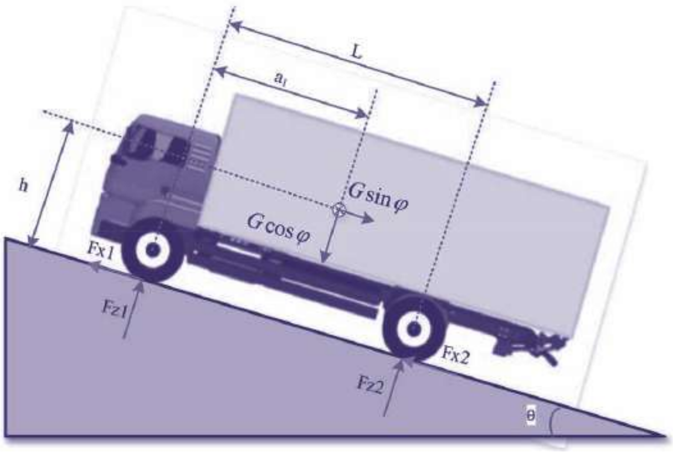
$$\left.\begin{array}{l} \frac{dp}{dt} = C_f C_h x \frac{P_{us}}{\sqrt{T}} \left(\frac{2\gamma}{R_{gs}(\gamma - 1)} \left(\frac{P_{us}}{P_{ds}}\right)^{\frac{2}{\gamma}} - \left(\frac{P_{us}}{P_{ds}}^{\frac{\gamma+1}{\gamma}}\right)\right)^{\frac{1}{2}} \\ Pus/Pds < 0.528 \\ Pus/Pds \geq 0.528 \\ 0.405\, C_f C_h x \frac{P_{us}}{\sqrt{T}} \end{array}\right\} \quad (4)$$

C_f - flow coefficient
C_h - valve hole's flow coefficient
T - Air temperature
R_{gs} - gas constant
γ - adiabatic index.
P_{us} - constant during inflation
P_{ds} - brake chamber pressure

4. VEHICLE DYNAMICS

The vehicle's real motion state is the result of a combination of inputs from numerous components, and the controller is the main component. By combining the signals, the controller executes the signals and processes the control signals to brake actuator based on the state of vehicle. The vehicle moves downward when it starts on a slope because of the downward resistance created by gravity. Through gear control, the driver directs the engine to produce an upward driving force. Before the driving force overcomes the resistance, the direction of the braking force travels up along the slope to help the driving force preserve the vehicle's static state and stop it from rolling back [15]. Fig 1 below shows various forces acting on vehicle.

Figure 1: Free Body Diagram Showing Forces Acting on Vehicle



$$\left.\begin{aligned} &M_a = F_{x1} + F_{x2} - \frac{1}{2}\, C_d \rho_a S V^2 + M_g(\sin\theta + \mu_a \cos\theta) \\ &F_{z1} + F_{z2} - M_g \cos\theta = 0 \\ &-F_{z1a1} + F_{z2}(L - a_1) - (F_{x1} + F_{x2}) * h = 0 \end{aligned}\right\} \quad (5)$$

The normal forces F_{z1} and F_{z2} which are based on vehicle dynamics can be obtained by removing F_{x1} and F_{x2},

$$F_{z1} = \frac{M.g.\cos\theta(L - a_1 - \mu_a h) - M.g.\sin\theta.h - M_a - \frac{1}{2}\, C_d\rho_a S V^2}{L}$$

$$F_{z2} = \frac{M.g.\sin\theta.h + M.g.\cos\theta(a_1 + \mu_a h) + \frac{1}{2}\, C_d\rho_a S V^2 + M_a}{} \quad (6)$$

Demand Torque of motor is given as

$$T_{refi} = T_{tot}\, \frac{F_{zi}}{2\sum_{i=1}^{2} F_{zi}} \quad (7)$$

Where,
M is total mass of the vehicle
g – gravitational acceleration
Rω - wheel effective radius
θ - road slope
V - vehicle speed
Cd - drag coefficient of air
Pa - the air density
S - effective windward area
F_{x1} - road driving force of the front axles
F_{x2} - road driving force of the rear axles
F_{z1} - normal force of front axles
F_{z2} - normal force of rear axles
T_{tot} - total moment required
μ_a - rolling friction coefficient
h - height of the centroid
a_1 - distance from the centre of mass to the front axis
L - separation between wheel-base and the center of mass.
a - vehicle self-acceleration

The vehicle's longitudinal dynamic equation, in which the driving torque acting on the wheels and the vehicle's ultimate effective driving torque are represented by $\sum_{i=1}^{4} T_{di}$ and $\sum_{i=1}^{4} T_{acti}$

$$\frac{\sum_{i=1}^{4} T_{acti} + T_{brk} - \sum_{i=1}^{4} T_{di}}{R_\omega} = \frac{1}{2}\, C_d\rho_a S V^2 + M_g(\sin\theta + \mu_a\cos\theta) \quad (8)$$

$$M\dot{V} = \frac{\sum_{i=1}^{4} T_{di}}{R_\omega}$$

T_{brk} is the single-wheel braking torque produced by the tire's static friction and the pneumatic brake system, and it satisfies

$$T_{brk} = min\{\mu_0 M_g R_\omega \cos\theta, \epsilon P_{act}\} \quad (9)$$
$$P_{act} \leq P_{max}$$

where,
µ0 -static friction coefficient of the tire
ϵ -pressure conversion coefficient
P_{act} - pneumatic parking brake systems actual braking pressure
P_{max} - pneumatic parking brake systems maximum braking pressure

5. ANALYSIS OF HILL START ASSIST SYSTEM

The EPB system gradually lowers the parking brake force Fbr k as the vehicle total driving force Ft increases as the vehicle starts to drive down the ramp. (Ft) = Fx1 + Fx2, is vehicle's total driving force at time t1 equalling slope resistance (Fξ).
Ideally, the parking brake force should be completely released when the driving force just surpasses the slope resistance.
Some wheels may slip as the vehicle begins to ascend because of uneven roads or low adhesion coefficients.
The actual total driving force of the vehicle F*t = F* x1 + Fx2 is less than the nominal total driving force of the vehicle F_t , that is F*t < Ft .
The vehicle's nominal total driving force F_t, is greater than its actual total driving force, F*t = F* x1 + Fx2, meaning that F*t < F_t.
Vehicle driving force must be changed to enhance hill start control and backward rolling. As the adjusted total driving force rises, the parking control unit progressively reduces the braking force until the parking brake hand or pedal force is completely eliminated.
Force applied due to EPB F_{brk} is written as,

$$F_{brk} = F_{bmax} - \frac{F_{bmax}}{P_{max}}\, P_{act} \quad (10)$$

$$F_{brk} = F_\xi - \sum_{i=1}^{4} F_{ti} \quad (11)$$

Where,
F^*_t -actual driving force of vehicle
F_t - driving force of vehicle
F_{bmax} - maximum force for braking during parking,
P_{max} - pressure at which the parking brake force is not actuating
F_ξ -slope resistance

Equations can be used to determine the actual driving force and slope resistance. (12) and (13)

$$\sum_{i=1}^{4} F_{ti} = \frac{\sum_{i-=1}^{4} T_{acti}}{R_\omega} \tag{12}$$

Slope resistance

$$F_\xi = C_d \rho_a SV + M_g \left(\sin\theta + \mu_a \cos\theta\right) \tag{13}$$

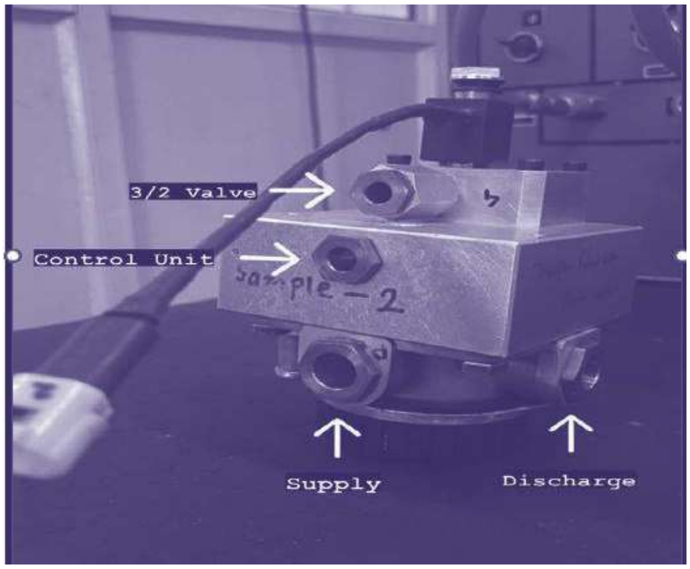
Combining above equations, (5), (8), (10) and (12) we get demand pressure

$$P_{dem} = \left(F_{bmax} + \frac{\sum_{i-=1}^{4} T_{acti}}{R_\omega} - C_d \rho_a SV - M_g \left(\sin\theta + \mu_a \cos\theta\right)\right) * \frac{P_{max}}{F_{bmax}} \tag{14}$$

6. TEST SETUP AND RESULTS

As the part is to be fitted in the truck, as per the regulations and norms various specific tests needs to be performed to prove the performance as well as durability of the product. It must be robust from not only the performance, but also from the design prospect. The setup was done in the local lab by simulating actual truck conditions. The pressure input from the FBV is supplied and a continues 8 bar pressure is maintained in the pressure line. 3 samples were developed inhouse and were tested accordingly as per the standard specifications of the braking norms.

Figure 2: Samples to test HSA



A. Leakage Test: Leakage test is one of the most important tests in the braking system. It is to be ensured that there are no leakages as its directly proportional to the braking performance. The nominal pressure that is in the truck pneumatic system is 8 bar. Tests are done in stringent environment, carried out at 10 bar.

Figure 3: Leakage Test on universal test bench

No. of samples: 03 Nos. (S1, S2, S3)
Supply pressure: P1:10 Bar, P1-1=10 bar & P4: 8 bar

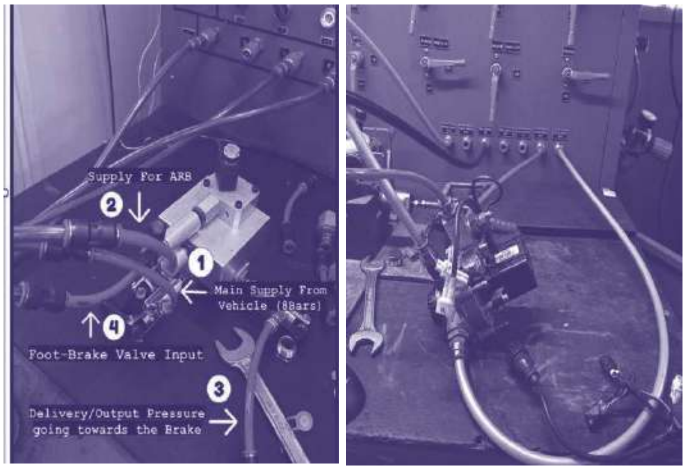
Table 1: Leakage Test

Characteristics	Unit	Specs	Actual values		
			S1	S2	S3
Leakage @ Braking	cc/mm	≤ 20	3.6	3.1	2.8
Leakage @ Release	cc/mm	≤ 20	2.1	1.8	1.5

Result/Observations: Leakage rate is ≤ 20, hence samples met the requirements.

B. Performance Test: It is important to ensure that the performance of the HSAS is as per standard requirements. The setup is arranged to simulate the inlet pressure from Foot brake valve and the main supply from the vehicle which is constant at 8 bar. The delivery/output pressure is going towards the brake.

Figure 4: Performance test with the components



No. of sample: 03 Nos. (S1, S2, S3)

Supply pressure: P1=10 Bar, P1-1=10 bar & P4=8 bar

Table 2: Performance Test

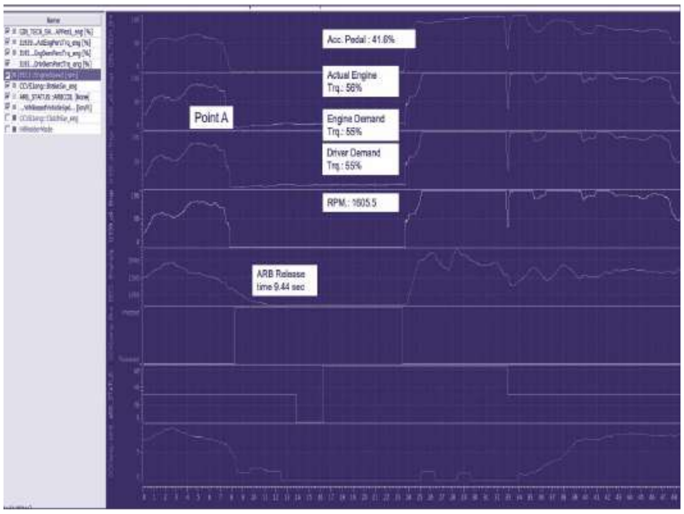
Characteristics	Unit	Specs	Actual values		
			S1	S2	S3
Threshold (Beginning of Pressurisation output Port)	bar	≤ 0.25 bar	0.20	0.23	0.22
Response P2 with P4 =2 bar	bar	1.8±0.2 bar	1.95	1.98	1.93
Response P2 with P4 =5 bar	bar	5±0.2 bar	4.96	5.01	5.03
Response P2 with P4 =8 bar	bar	8.2±0.2 bar	8.02	8.06	8.03
Hysteresis @ 1 bar	bar	≤ 0.25 bar	0.19	0.23	0.21
Hysteresis @ 5 bar	bar	≤ 0.25 bar	0.20	0.24	0.21

Result/Observations: Samples did meet the performance test.

7. VALIDATION ON TRUCKSIM SOFTWARE

7° gradient with test vehicle GVW of 10560 kg considering 15% overload overrated load of 9150 kg (on 14.2% gradient). Logic **1: Base Logic- Boost Pressure (Figure 5)**

Figure 5: Boost Pressure

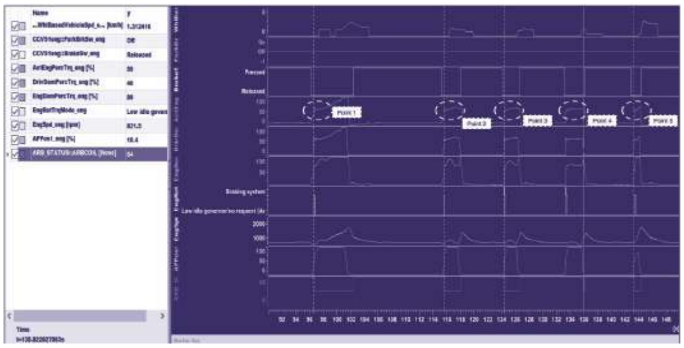


Observations and Conclusions:

- Based on the demand for the acceleration to march–off the vehicle the engine torque was suddenly rising and getting to the peak torque / saturation due to which the HSA was not able to calculate the exact point of release.
- To explain if we refer the highlighted point "A" in which the engine characteristic has quick response to the smallest driver demand resulting in miss-out of the data and calculation time for the release of HSA.
- Eventually this resulted in the delay in releasing of the brake intermediately.

Logic 2: Higher Grade with Boost Pressure (Figure 6)

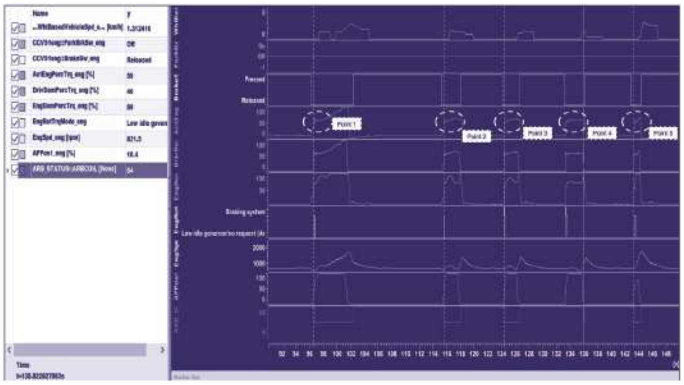
Figure 6: Higher Grade with Boost Pressure



- With the revised logic it is clearly observed that the vehicle is not having any brake grabbing and march off is smooth.
- In one of the instances, i.e., at point 4 as marked in graph, it was observed that there was no substantial drop in engine torque and hence ARB was not able to detect this event, however this drop was evident at other point as highlighted
- Further fine tuning was requested to address such one odd event.

Logic 3: Mild Grade with Boost Pressure (Figure 7)

Figure 7: Mild Grade with Boost Pressure



- Same trials were repeated on a mild grade (approx. 8%) and we could see ARB release as intended.
- In one of the instances, i.e., at point 1 as marked in graph, it was observed that there was no substantial drop in engine torque and hence ARB was not able to detect this event, however this drop was evident at other point as highlighted (Point 2, Point 3)
- Further fine tuning was requested to address such one odd event.

8. CONCLUSION

The Hill Start Assist Help Framework (HSA) is a valuable innovation that assists drivers with performing uphill begins more effectively and securely. Temporarily pressing the brakes stops the vehicle from rolling backwards while the driver moves foot to the accelerator from the brake.

The venture to execute HSA in a vehicle can be viewed as a triumph on the off chance that it meets the following standards:

- Effectiveness: Even with a heavy load or in bad weather, the HSAS should prevent the vehicle from rolling backward on a hill.
- Safety: The HSAS shouldn't affect how the brakes normally work, and it shouldn't cause accidental acceleration or other safety problems.
- Driver experience: Drivers should not be confused or frustrated by the HSA because it should be simple to use and intuitive.
- Cost-effectiveness: Based on the benefits it provides, the cost of implementing HSAS ought to be reasonable and justifiable.

The HSA can be a useful addition to a vehicle if all these conditions are met, providing drivers with greater safety and peace of mind.

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