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RESEARCH ON THE POSITION OF THE AUXILIARY DECELERATION LANE ON LONG DOWNHILL ROAD BASED ON ENGINE AND HYDRAULIC RETARDER

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Although the safe haven has solved the problem of the long downhill runaway of the vehicle to a certain extent, the vehicles entering the safe haven cannot drive out, which increases the rescue cost. The auxiliary deceleration lane is set in parallel with the main line. After the vehicle is in danger, it enters the auxiliary avoidance lane to decelerate, and the vehicle speed is controlled and then self-driving. How to reasonably determine the auxiliary hedging lane is related to the use efficiency of the auxiliary hedging lane and the design economy. In this paper, by studying the working characteristics of engine brake and hydraulic retarder combined braking, combined with the brake temperature rise model, the vehicle downhill stall model is constructed to determine the downhill safety distance, which makes the position setting of the auxiliary deceleration lane more reasonable.

Keywords: Engine braking; Hydraulic retarder braking; Combined braking; Auxiliary deceleration lane

1. Introduction

When the vehicle is driving on a long downhill section, the service brakes frequently work, and the brake temperature is continuously increased, resulting in a decrease in braking capacity and causing traffic accidents [1-3]. By setting the auxiliary deceleration lane, the driving resistance of the vehicle is increased, the number of use of the service brake is reduced, and the safety of the vehicle is improved.

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Based on the data obtained from the test, the joint braking model of engine brake and hydraulic retarder is established. The relationship between downhill slope and vehicle speed is obtained under different gears, which provides a basis for assisting the slope range of the deceleration lane. Combined with the temperature rise model of the brake drum driving on the ramp, the driving conditions of the slope in the safe temperature threshold of the brake drums of different gears are studied, and the relationship between the vehicle speed and the slope of the road and the downhill distance is analyzed, and then derived. Auxiliary deceleration lane setting position.

2. Research on combined braking characteristics of engine brake and hydraulic retarder [4]

2.1 Modeling of the combined action of engine brake and hydraulic retarder

The horizontal road surface with good road attachment conditions was selected to test the engine braking and hydraulic retarder braking flat road of heavy-duty vehicles respectively. The test data was obtained and the relationship between engine braking torque and speed was derived.

$$\begin{aligned} T_{b\_motor} &= A_{b\_motor} n^2 + B_{b\_motor} n^2 + C_{b\_motor} \quad \left(800rpm \le n \le 2100rpm\right) \\ &= -3.303608 \times 10^{-5} n^2 + 1.995282 \times 10^{-1} n + 3.476678 \times 10^2 \end{aligned} \tag{1}$$

In the formula (1), $A_{b\_motor}, B_{b\_motor}, C_{b\_motor}$ are the polynomial coefficients, which are obtained by fitting multiple sets of test data, n is the engine speed.

Relationship between braking torque of each retaining force retarder and transmission shaft speed.

$$\begin{aligned} T_{b\_hyd\_2} &= -6.2283 \times 10^{-4} {n_{zhou}}^2 + 1.4266 n_{zhou} - 3.5869 \times 10^2 \\ T_{b\_hyd\_3} &= -1.3440 \times 10^{-3} {n_{zhou}}^2 + 3.4498 n_{zhou} - 2.8289 \times 10^2 \\ T_{b\_hyd\_4} &= -2.0652 \times 10^{-3} {n_{zhou}}^2 + 5.4730 n_{zhou} - 9.2446 \times 10^2 \\ T_{b\_hyd\_5} &= -4.3352 \times 10^{-3} {n_{zhou}}^2 + 1.0723 \times 10^1 n_{zhou} - 3.0015 \times 10^3 \end{aligned} \tag{2}$$

In the formula (2), $T_{b\_hyd\_i}$ is the hydraulic retarder braking torque with different gears, i=2, 3, 4, 5. n_{zhou} is the speed of transmission shaft.

Simplify the formula (2) into the general form (3):

$$T_{b\_hyd\_i} = A_{b\_hyd\_i} n_{zhou}^2 + \mathrm{B}_{b\_hyd\_i} n_{zhou} + C_{b\_hyd\_i} \tag{3}$$

In the formula (3), $A_{b\_hyd\_i}$, $\mathrm{B}_{b\_hyd\_i}$, $C_{b\_hyd\_i}$ are the fitting coefficients of the relationship between the braking torque of hydraulic retarder and the speed of transmission shaft.

The relationship between the braking torque and the speed of the transmission shaft when the engine brake and the hydraulic retarder are combined by the equations (1) and (3) is as follows:

$$\begin{aligned} T_{b\_motor+hyd} &= T_{b\_motor} i_g + T_{b\_hyd\_i} \\ &= i_g \eta_T \left[A_{b\_motor} (n_{zhou} i_g)^2 + B_{b\_motor} (n_{zhou} i_g) + C_{b\_motor} \right] \\ &\quad + (A_{b\_hyd\_i} n_{zhou}^2 + \mathrm{B}_{b\_hyd\_i} n_{zhou} + C_{b\_hyd\_i}) \\ &= A \cdot n_{zhou}^2 + B \cdot n_{zhou} + C \end{aligned} \tag{4}$$

In the formula (4), $T_{b\_motor+hyd}$ is the sum of engine braking torque and the retarder braking torque. i_g is the ratio of transmission. η_T is the efficiency of driveline.

The relationship between the sum of the air resistance and the rolling resistance obtained by the off-slip test and the vehicle speed.

$$F_f + F_w = 0.2960565 u_a{}^2 + 13.86905 u_a + 3128.490 \tag{5}$$

In the formula (4), F_f is the rolling resistance, F_w is the air resistance. u_a is the speed of vehicle.

After calculation, the vehicle braking torque provided by the engine brake and the hydraulic retarder is a function of the vehicle speed.

$$\begin{aligned} T_{brake\_motor+hyd\_i} &= T_{b\_motor+hyd\_i} + \left(F_f + F_w\right) r \\ &= D_1 \cdot u_a^2 + E_1 \cdot u_a + F_1 \end{aligned}$$

$$\begin{cases} D_1 = \left(A_{b\_motor} i_g^3 \eta_T{}' + A_{b\_hyd\_i} \eta_T \right) \dfrac{i_0^3}{0.377^2 r^2} + 0.2960565 r \\ E_1 = \left(B_{b\_motor} i_g^2 \eta_T{}' + B_{b\_hyd\_i} \eta_T \right) \dfrac{i_0^2}{0.377 r} + 13.86905 r \\ F_1 = \left(C_{b\_motor} i_g \eta_T{}' + C_{b\_hyd\_i} \eta_T \right) i_0 + 3128.490 r \end{cases} \tag{6}$$

In the formula (6), D_1, E_1, and F_1 are the braking torque coefficients of the fitting relationship between the vehicle braking torque and vehicle speed under the combined braking of the engine brake and the hydraulic retarder. See Table 1, 2 for the coefficient table for details.

Table 1

Engine Brake and Hydraulic Retarder 3rd Gear Combined Brake Speed -Wheel Brake Torque Coefficient Table

Transmission gear	9 block	10 block
D_1	-2.057481E+00	-1.951501E+00
E_1	2.982457E+02	2.833035E+02
F_1	1.913137E+03	1.454989E+03

Table 2

Engine Brake and Hydraulic Retarder 4-speed Combined Brake Speed - Wheel Brake Torque Coefficient Table

Transmission gear	9 block	10 block
D_1	-3.099066E+00	-2.993086E+00
E_1	4.483841E+02	4.334419E+02
F_1	-5.331697E+02	-9.913169E+02

2.2 Research on Downhill Capacity of Engine Brake and Hydraulic Retarder

The car runs downhill on the slope with the slope i. According to the longitudinal dynamic equation of the car, the slope range of the car can be obtained when the slope component of the downward weight of the vehicle is balanced with the total continuous braking force of the vehicle.

$$m \cdot g \cdot \frac{i}{\sqrt{1+i^2}} = F_f + F_w + \frac{T_{b\_motor+hyd\_i}}{r} \tag{7}$$

Convert formula (7) to formula (8):

$$i = \frac{\dfrac{\dfrac{T_{b\_motor+hyd\_i}}{r} + F_f + F_w}{m \cdot g}}{\sqrt{1 - \left(\dfrac{\dfrac{T_{b\_motor+hyd\_i}}{r} + F_f + F_w}{m \cdot g} \right)^2}} \tag{8}$$

It can be found from equation (8) that the vehicle can maintain the maximum gradient at constant speed when the engine brake and the hydraulic retarder are combined to brake. When the average slope of the long down slope is greater than the slope, the service brake is involved.

3. Brake temperature rise model research [5]

3.1 Establishment of the flat road brake heating model

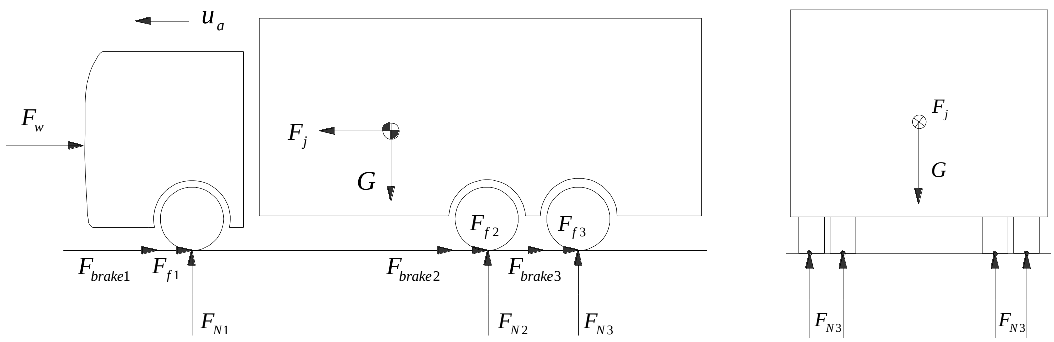

Fig. 1. Figure of the road braking process

The braking force of the flat road braking process is shown in Fig. 1. There are two tires on the steering shaft, and four tires on the second and third axes respectively. In the Fig.1, G is the gravity of the vehicle; F_w is the air resistance; F_j is the inertia force of vehicle; F_{Ni} is the normal force of the ground to the wheel; F_{fi} is the rolling resistance; $F_{brake\ i}$ is the braking force (i =1,2,3). When the car brakes, the vehicle speed decreases and the kinetic energy decreases. The kinetic energy consumed by the car brake process is converted into the energy consumed by the service brake, the energy consumed by the rolling resistance and the air resistance, and the sum of the energy consumed by the continuous braking force.

$$\frac{1}{2}mu_0{}^2 = F_b \cdot s + F_f \cdot s + F_w \cdot s + F_{b\_con} \cdot s \tag{9}$$

According to the principle of brake energy distribution, the energy distributed by each brake is equal, and its sum is equal to the energy consumed by the brake of the whole vehicle. Among them, n is the number of brakes, F_{bh} is the friction produced by a single brake, S_{bh} is the relative rotation displacement of brake friction parts.

$$F_b \cdot \mathrm{s} = \mathrm{n} \cdot F_{bh} \cdot S_{bh} \tag{10}$$

Convert formula (9) to formula (10):

$$\mathrm{n} \cdot F_{bh} \cdot S_{bh} = \frac{1}{2}\mathrm{m}u_0^2 \text{-} F_{\mathrm{f}} \cdot \mathrm{s} \text{-} F_{\mathrm{w}} \cdot \mathrm{s} \text{-} F_{\mathrm{b\_con}} \cdot \mathrm{s} \tag{11}$$

In the process of braking, there will be friction between the brake lining and the brake drum, which will cause the temperature of the brake to rise. The temperature change rate of a single brake can be expressed as follows. F_{bh} is the friction between the brake drum and the friction lining in the process of braking; V_{bh} is the relative speed between the brake drum and the friction lining.

$$P'_{bh} = F_{bh} \cdot V_{bh} \tag{12}$$

In the process of driving, the wheel and brake drum have the same angular speed. u is Vehicle speed; R_{g1} is brake drum radius; R_{t} is wheel radius. Then:

$$\frac{V_{\mathrm{bh}}}{R_{\mathrm{g1}}} = \frac{\mathrm{u}}{R_{\mathrm{t}}} \tag{13}$$

The friction on a single brake can be expressed as:

$$F_{bh} = \frac{\frac{1}{2}\mathrm{m}u_0^2 \text{-} F_{\mathrm{f}} \cdot \mathrm{s} \text{-} F_{\mathrm{w}} \cdot \mathrm{s} \text{-} F_{\mathrm{b\_con}} \cdot \mathrm{s}}{\mathrm{n} \cdot S_{bh}} \tag{14}$$

According to the functional principle of the service brake, the energy consumed by the brake of the entire vehicle is distributed to the brakes, and 95% of the heat generated by the brake friction is absorbed by the brake drum.

$$
\begin{aligned}
P_{bh} &= 0.95 \cdot \varepsilon \cdot F_{bh} \cdot V_{bh} \\
&= 0.95 \cdot \varepsilon \cdot \frac{\frac{1}{2} m u_0^2 - F_f \cdot s - F_w \cdot s - F_{b\_con} \cdot s}{n \cdot S_{bh}} \cdot V_{bh}
\end{aligned} \tag{15}
$$

Where F_{b_con} is the continuous braking force, F_{bh} is the friction generated by the single brake; S_{bh} is the relative rotational displacement of the brake friction member. V_{bh} is the relative speed between the friction plate and the brake drum; P_{bh} brake drum heat absorption rate; is the correction factor.

During the flat road braking process, the braking distance and time are short, so the brake heat dissipation factor of the braking process is not considered. The equation is established according to the conservation of the warming energy of the brake drum.

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$$
m_g \cdot c_g \cdot \Delta T = P_{bh} \cdot \Delta t \tag{16}
$$

Where m_g is the mass of the brake drum and c_g is the specific heat capacity of the brake drum.

From equations (10) and (11), it can be concluded that:

$$
\Delta T = \frac{0.95 \cdot \varepsilon}{m_g \cdot c_g} \cdot \frac{\frac{1}{2} m u_0^2 - F_f \cdot s - F_w \cdot s - F_{b\_con} \cdot s}{n \cdot S_{bh}} \cdot V_{bh} \cdot \Delta t \tag{17}
$$

Since $S = \int u dt$, the brake heating model for flat braking is

$$
\begin{aligned}
T_+ &= T_0 + \int_0^t \frac{0.95 \cdot \varepsilon}{m_g \cdot c_g} \cdot \frac{\frac{1}{2} m u_0^2 - F_f \cdot s - F_w \cdot s - F_{b\_con} \cdot s}{n \cdot S_{bh}} \cdot V_{bh} \cdot dt \\
&= T_0 + \frac{0.95 \cdot \varepsilon}{m_g \cdot c_g \cdot n} \left(\frac{1}{2} m u_0^2 - F_f \cdot s - F_w \cdot s - F_{b\_con} \cdot s\right)
\end{aligned} \tag{18}
$$

The heat dissipation methods of the brake drum mainly include heat conduction, heat convection and heat radiation. Therefore, when the cooling model is established, the influence of thermal convection on the brake temperature is mainly studied, and the heat conduction and heat radiation heat dissipation are ignored.

According to the Newtonian cooling formula, when the brake drum is cooled by the heat dissipation of the surrounding air, the heat flow of the convective heat transfer is:

$$P_d = h_r \cdot A_2 \cdot (T - T_a) \tag{19}$$

Where, the convective heat transfer strength coefficient between h_r brake drum and air; T brake drum temperature; T_a average temperature around the brake drum; A_2 brake drum outer surface area.

According to the test data of heavy-duty vehicles, the function of the convection heat coefficient of the brake drum [6-7] is:

$$h_r^{'} = 5.224 + 1.5525 \cdot u_a \cdot e^{-0.0027785 u_a} \tag{20}$$

Because only the heat convection is considered, the heat dissipation of the brake drum is the heat exchange between the brake drum and the air, which can be expressed as:

$$P_{-} = h_r^{'} \cdot A_2 \cdot \left(T_{-} - T_{\mathrm{a}}\right) \tag{21}$$

According to the law of conservation of energy, we can get:

$$\mathrm{m}_g \cdot c_g \cdot \Delta T = -P_{-} \cdot \Delta t \tag{22}$$

According to the cooling drum energy conservation energy conservation equation, the mathematical model of the brake drum cooling is obtained, namely:

$$\begin{aligned} T_{-} &= (T_0 - T_a) e^{-At} + T_a \\ A &= \frac{h_r^{'} \cdot A_2}{m_g \cdot c_g} \\ &= \frac{(5.224 + 1.5525 \cdot u_a \cdot e^{-0.0027785 u_a}) \cdot A_2}{m_g \cdot c_g} \end{aligned} \tag{23}$$

3.2 Establishment of temperature increasing model for slope brake

According to the mathematical model of horizontal road heating, the thermal energy of the brake comes from the conversion of kinetic energy. However, during the ramp driving, the energy of the brake heating is due to the joint transformation of the gravitational potential energy and the kinetic energy. Combined with the cooling model, the ramp heating model of the ramp can be obtained as follows:

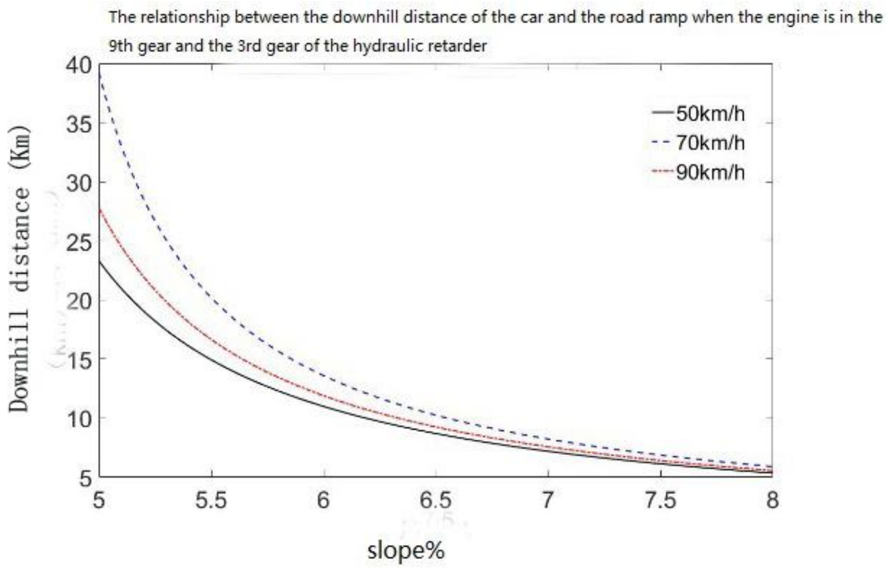
$$\begin{aligned} T &= T_0 + T_+ - T_- \\ &= 2T_0 + \frac{0.95\cdot\varepsilon}{m_g\cdot c_g\cdot n}\left(\frac{1}{2}m(u_0^2-u_0^2)+mgs\frac{i}{\sqrt{1+i^2}}-\right. \\ &\left.\frac{D_1\cdot {u_a}^2+E_1\cdot u_a+F_1}{r}\cdot s\right)-(T_0-T_a)e^{-At}-T_a \end{aligned} \tag{24}$$

4. Auxiliary deceleration lane setting position analysis model

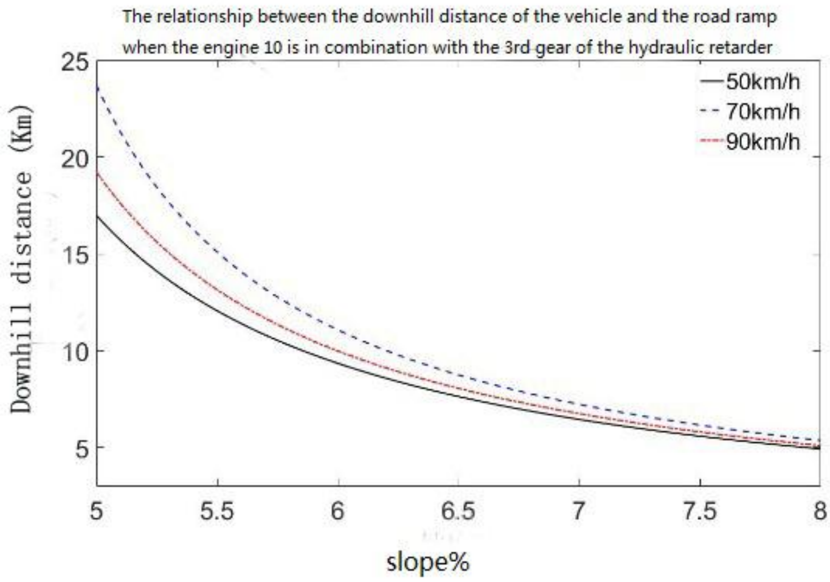
Converting equation (17) to equation (18), it can be determined that when the engine brake, hydraulic retarder and service brake are combined, the brake drum temperature is within the safe temperature threshold, the vehicle travel distance s, the road gradient i and The relationship between the speed u_a.

$$s = \frac{[T+(T_0-T_a)e^{-At}+T_a-2T_0]\cdot m_g\cdot c_g\cdot n/(0.95\cdot\varepsilon)-\frac{1}{2}m({u_t}^2-u_0^2)}{mg\frac{i}{\sqrt{1+i^2}}-\frac{D_1\cdot u_a^2+E_1\cdot u_a+F_1}{r}} \tag{25}$$

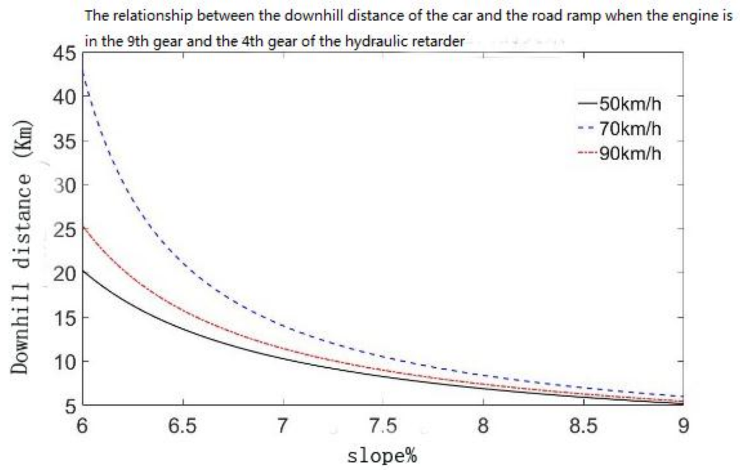
Related studies have shown that [8], when the temperature of the brake lining of the resin material reaches around 250 °C, the braking performance of the brake drops sharply. Therefore, this study will have T=250 °C, T_0= 250 °C, T_a=250 °C. The gears commonly used for downhill slopes of large trucks are selected for simulation. According to formula (8), the average minimum slope range of the service brakes is determined, and the corresponding safe downhill distance for different speeds can be obtained.


The relationship between the downhill distance of the car and the road ramp when the engine is in the 9th gear and the 3rd gear of the hydraulic retarder
Downhill distance (Km)
slope%
50km/h
70km/h
90km/h
40
35
30
25
20
15
10
5
5.5
6
6.5
7
7.5
8

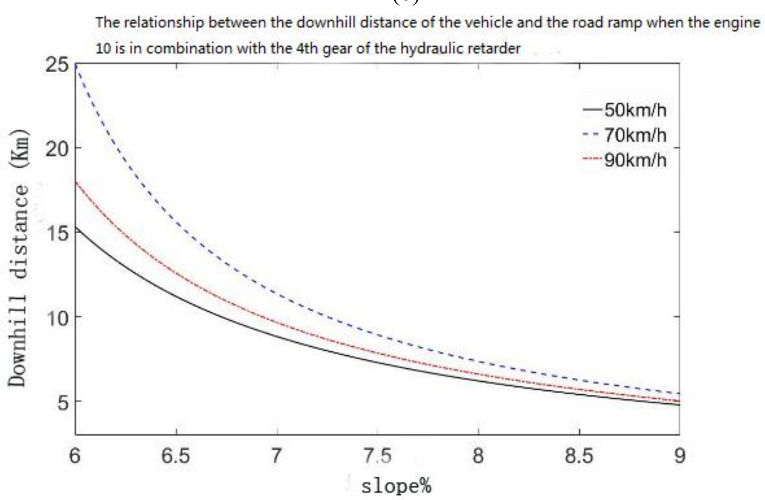
(a)


The relationship between the downhill distance of the vehicle and the road ramp when the engine 10 is in combination with the 3rd gear of the hydraulic retarder
Downhill distance (Km)
slope%
50km/h
70km/h
90km/h
25
20
15
10
5
5.5
6
6.5
7
7.5
8

(b)



(c)



(d)

Fig. 2. Relationship between downhill distance and slope of combined braking of engine and hydraulic retarder

Fig. 2(a), (b), (c) and (d) are the downhill safety distance and slope under the condition that the transmissions 9 and 10 are combined with the hydraulic retarder 3 and 4 gears respectively at different speeds relationship. Guide design through downhill safe distance

The setting position of the auxiliary deceleration lane. In Fig. 2(a), the average slope of the road is 6%, the truck is downhill at an ideal average speed of 70km/h, and the auxiliary deceleration lane should be set at 14km from the top of the slope.

5. Conclusion

In this paper, the design of the auxiliary deceleration lane is studied for the problem of the brake failure of the vehicle in the long downhill section. Firstly, by establishing the combined braking model of engine brake and hydraulic retarder, the downhill capacity is studied, and the downhill gradient range of the vehicle is obtained when the vehicle is under load balance, and the minimum average slope of the service brake is obtained. Then, by establishing the brake temperature rise model, the relationship between the brake temperature, the vehicle speed and the road gradient is obtained, and the distance that the car can travel under the safe temperature threshold is obtained, and the optimal position of the auxiliary deceleration lane setting under different roads is determined.

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REFERENCES

[1]. *L. Chen, M. Xiao, P. Wang,* "Research on the Setting of Mountain Road Hazard Lanes", Journal of Hunan City University (Natural Science Edition), 2014-02.

[2]. *B. Su, S. Fang, J. Wang,* "Research on slope length limit of mountain highway based on braking performance of large trucks", Journal of Chongqing Jiaotong University (Natural Science Edition), **vol. 2**, 2009.

[3]. *E. Yan, B. Du,* "Temperature Model of Long Downhill Truck Brakes", Journal of Highway and Transportation Sciences, **vol. 28**, no. 2, 2011, pp. 133-136.

[4]. *Y. Guo, C. Wang,* "Temperature Rise Model of Drum Brake", 2013 International Conference on Mechanics and Mechatron-ics, 2013.

[5]. *Z. Wang, Q. Yu, P. Shi,* "Research on continuous braking characteristics of heavy-duty trucks driving downhill", Industrial Instrumentation and Automation. no. 5, 2017.

[6]. *P. Shi, Q. Yu, M. Yu, et al.* "Research on Heating Model of Heavy Downhill Brake for Heavy Commercial Vehicles", Journal of Highway and Transportation, 2016, **vol. 33**, no. 1, pp. 147-152.

[7]. *D. Li,* "Accident prevention research based on braking performance of long downhill vehicles", Chang'an University, 2010.

[8]. B. Du, "Modeling of Freight Car Brake Temperature and Safety Study of Long Downhill", 2010