Design of a New Anti-Lock Braking System for Motorcycles

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Abstract: A new anti-lock braking system (ABS) for motorcycles is proposed in this study. It functions by processing the speed of the motorcycle tire in contact with the ground relative to the speed of the motorcycle itself and by calculating the slip ratio (λ) of the tire slipping on the ground while braking, and reducing it to a minimum which leads to increased controllability of the motorcycle and reduction of the stopping distance, especially when the ground is slippery. The design of the new ABS and its pilot model comprises mechanical parts, hydraulics, and an electrical circuit. The pilot model providing the testing facility for the brake system functioning in a fixed place is in fact a simulation of the movement of a motorcycle on the ground. When the electric motor is turned on, and its flywheel reaches the desired speed, a controllable load is applied to the flywheel by the motorcycle tire which is a modelling of the weight of the motorcycle and the rider. Then, by turning the electric motor off, the throttle is released and the brake is activated. In this state, without the tire being locked, it will stop within a shorter time and distance than the non-ABS, because the new system keeps the tire in the threshold state of slipping relative to the ground, which is the maximum friction coefficient and the maximum brake force. The results show that the stoppage time for the new ABS is about 40% less than that is the non-ABS type.

Keywords: Motorcycle, New ABS, Pilot model, Slip Ratio

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1 INTRODUCTION

Motorcycles have had and continue to have an accident rate which is disproportionately high for their number of vehicle miles. At present, they account for only about two per cent of vehicle miles in Britain but produce twenty per cent of the casualties caused in vehicle accidents [1].

If the wheel of a motorcycle is overbraked, lock-up and vehicle instability rapidly occurs. Capsize with vehicle damage and rider injury is likely to follow. Anti-lock braking has been available for many years on more stable vehicles such as cars and trucks, but nothing for motorcycles even though transport and road research laboratory (TRRL) demonstrated its worth in 1964 [2].

In the past few decades, many types of anti-lock braking systems (ABS) have been installed in different kinds of vehicles, but not in lightweight motorcycles, used frequently in big cities. When riders of motorcycles without ABS perform emergency braking maneuvers, they are frequently thrown off the vehicle. This is particularly apparent in wet conditions. As a result, non-ABS light motorcycles cause many casualties. Most of the ABS installed on four-wheel vehicles adopt an additional hydraulic pump and valves to regulate the brake pressure. But both size and cost constraints prevent their installation on light motorcycles [3]. The new braking system processes the speed of the contact of the motorcycle tire with the ground relative to the speed of the motorcycle, and calculates the amount of tire slippage coefficient on the ground at the time of braking; minimizing it as much as possible resulting in an increase in the controllability of the motorcycle and in the reduction of the stopping distance especially on slippery roads. The new ABS and its pilot model comprise mechanical and hydraulic parts as well as electrical circuits (Fig. 1).

Fig. 1 The new ABS and its pilot model

The hydraulic parts of the system include three cylinders and pistons, one acting as an accumulator; two solenoid valves; two check valves and related hoses, fittings and connectors. The solenoid valves are controlled by an electric circuit. The pilot model comprises a chassis, a structure, an electric motor, a flywheel, a belt, an axis, a pedal, pulleys, bearings, motorcycle tires and accessories, and circular discs and related sensors for measuring the speed. The pilot model which provides the conditions for the functioning of the braking system in a fixed place is in fact a type of modelling device for the mobility of the motorcycle on the ground.

When the electric motor is switched on and the flywheel reaches the desired speed, a controllable load is exerted on the flywheel by the motorcycle tire which is a very near imitation of the motorcycle weight and its riders. Then, by switching the electric motor off i.e., the release of the accelerator of the motorcycle while braking, the process of braking starts. Without the brakes being locked, the motor stops within a shorter time and distance compared to when the ABS system is not active. The reason is the designed system keeps the tire in the threshold of slippage (maximum friction coefficient) relative to the ground and the brake force with its most power reduces the stoppage time. Because cost is one of the constraining and affecting factors for its use and development, in the design of the new system, economic constraints are taken into consideration in a way that the mass production of this product and its installation on conventional motorcycles is economical.

2 GOAL AND LIMITATIONS OF THE RESEARCH

This design is intended to increase the ability of the motorcycle rider in keeping his stability while braking and to increase his safety during hard braking. Hydraulic equipment and two solenoid valves are used in this system to control the brake pressure and to direct it into an optimal braking situation. The faster the reaction of these valves, the better the system is controlled and directed. Since solenoid valves with fast reaction are not available on the market, as a last resort, we had to use valves with more reacting time than expected with reduced accuracy for controlling the brakes system (the minimum response time for the valves used in this circuit is 0.04 seconds while the desireable time in the design was 0.02). It seems that weak functioning of motorcycle brakes is one of the causes for high casualties and accidents.

3 DYNAMICS OF THE TIRES ON THE GROUND WHEN BRAKING

When the vehicle is running, ground forces acting on the tire are shown in Fig. 2. They include the normal force F_z, the longitudinal force F_x and the lateral force F_y. F_z comes from the weight and load of the vehicle. Its magnitude also varies due to the load transfer effect during braking.

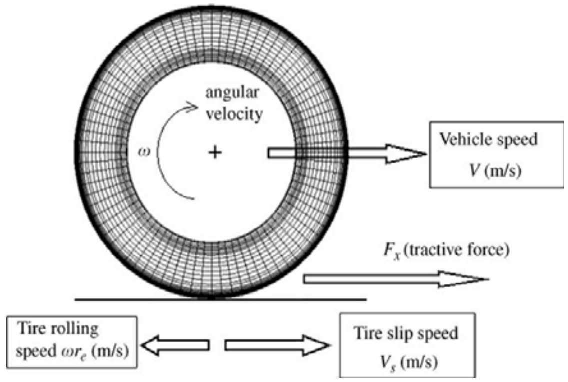


Fig. 2 Free body diagram of a single tire

The longitudinal force F_x is produced during driving and braking. The lateral force F_y helps the vehicle in turning and maneuvering, or in some cases, in resisting a side disturbance force such as the wind. The reduction in braking distance depends on whether F_x can be maintained at its maximum value. During braking, the contact patch between tire and ground begins to slip. This substantially affects both the longitudinal and lateral friction coefficients. The well known parameter to represent slippage is the slip ratio. The tire slip ratio is defined as:

$$\lambda = V_s/V = (V - \omega r_e)/V \qquad (1)$$

where V is the absolute vehicle speed, V_s is the tire slip speed, r_e is the effective radius of the tire, and ω is the angular velocity of the tire. From the definition, one can conclude that when λ=0, the tire is in perfect rolling motion without slipping. On the other hand, when λ=1, the tire is locked and pure sliding happens. The slip ratio greatly affects the tractive force (F_x) [4-5].

On the basis of the experiments conducted, and referring to Fig. 3, the approximate maximum value of the diagram i.e. the maximum value of the friction force between the tire and the surface of the road is about λ=20%. Also, before the value of λ reaches this limit, the value of the friction coefficient (μ) is within the stable limit. This means that the stable range of braking is approximately from=λ 0% to λ=20% is, i.e. the value of μ and accordingly the friction force is increased by an increase in the force of the pedal, and is reduced by decrease in the pedal force, but when λ exceeds this critical boundary (about 20%), it is within the unstable limit, i.e. an increase in λ from this limit is no longer dependent on the increase in the unit force exerted on the brake pedal. So, when λ exceeds this limit, it suddenly and quickly reaches its maximum value, i.e. λ=100%; and in a very short time, the tire comes to a complete slip or skidding state on the road. The appropriate distribution of the braking force, its coordination with the conditions on the surface of the road and the tire in a way that λ between the tire and the road is within the permitted limit necessitates the use of automatic systems to control tire slip [5-6].

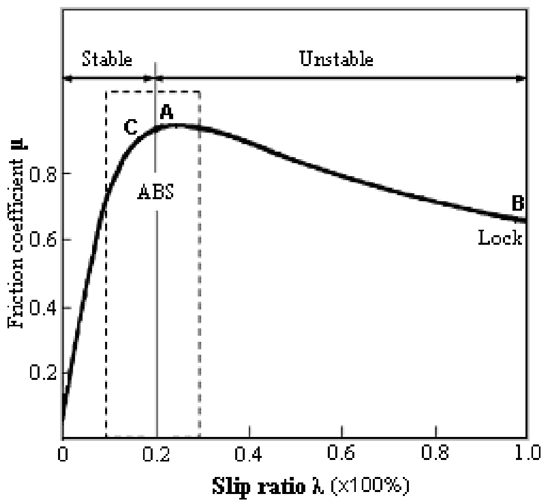


Fig. 3 Variation of friction coefficient against the slip ratio

4 SLIP CONTROL PROCESS

The pressure of the brake is increased during braking and according to Fig. 3, at its maximum state, it reaches the boundary between the stable and unstable limit. At this point, the brake force does not increase ($F_B=\mu N$). With an increase in the brake pressure or an increase in the torque exerted on the tire, we witness the drop of the frication coefficient and the stopping force. The increase in the torque exerted on the tire results in the stoppage of the tire within a very short time accompanied by a rather great reduction in the negative acceleration of the tire (reduction of brake force). Also, in this state, the tire on the road is very unstable. Change in the speed of the tire is measured by the electrical control unit and when it reaches the critical value, the brake pressure is cut off from behind the tire. If the negative acceleration of the tire exceeds the threshold limit (point A) and the tire moves toward locking, the tire is ultimately at the maximum state of slip or at a complete stoppage, (point B). Before the tire reaches this state, the pressure behind the tire brake shoes is reduced up to point C and is kept there. At this stage, the speed of the tire reaches the speed of the vehicle from zero (or near zero) and the λ slip coefficient enters into the stable limit. In the next stage, the pressure of the brake is suddenly increased and the speed and negative acceleration of the tire is controlled again; and if required, the electronic control system takes action and transmits the required command to the solenoid valves.

5 SUGGESTED ABS SYSTEM AND COMPONENTS DESIGN

This system is composed of a hydraulic component, an electric component, and two speed sensors. The sensors are of the infrared types in front of which a holed disc passes. For every hole passing the sensor, a pulse is sent to the electric circuit. The time lapse of two successive pulses transmitted to the electronic section is calculated and the speed of the movement of the tire is determined. The speed of the motorcycle is determined by the sensor on the front tire and that of the tire with the brake is determined by the rear tire sensor. The hydraulic circuit of the device is depicted in Fig. 4.

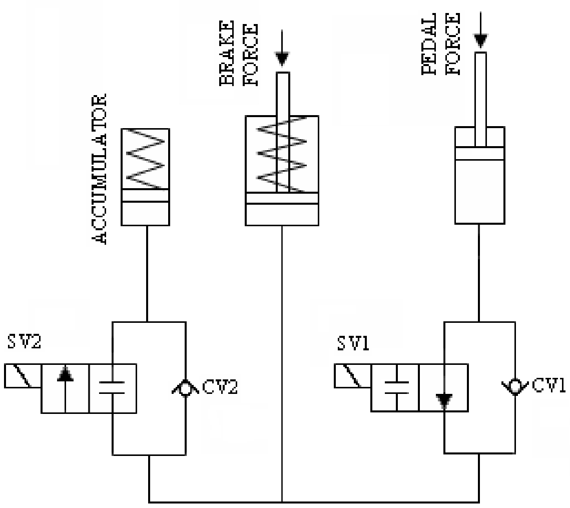


Fig. 4 The diagram of motorcycle new ABS hydraulic circuit

By pressing cylinder 1 through the pedal, the motorcyclist applies the brake force to cylinder 2 and then to the end of the brake by a certain increasingly changing ratio. The solenoid valve 1 (SV_1) connects and disconnects the oil flow between cylinders 1 and 2, and in this way when necessary, it prevents the increase in the force on the brake shoes by cutting the path between the cylinders. Solenoid valve 2 (SV_2) is responsible for the reduction of the pressure in the hydraulic system and reduction of the force on the brake shoes by rapid discharge of oil from the main circuit into the oil tank under pressure (accumulator). At the end of braking and by releasing the brake pedal, the oil enters the accumulator from the CV_2 path and is injected into the main path. The CV_1 path is related to the return of oil from cylinder 1 to cylinder 2 when SV_1 is closed, and CV_2 is related to the return of oil from accumulator into the circuit and cylinder 1. This hydraulic system is designed for pressures under 10 bars.

6 THE FLYWHEEL RADIUS AND MASS DETERMINATION

By considering a μ_g mean friction coefficient between the tire and the surface of the ground and μ_f between the tire of the motorcycle and the flywheel, the movement of the tire on the ground and its movement on the flywheel is modelled. This modelling is based on equal frictional force in the real model and real state in terms of equal force F_{load} on the tire axis, as shown in Fig. 5.

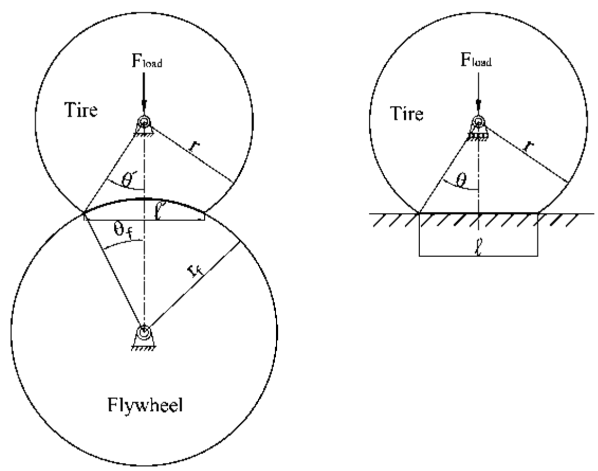


Fig. 5 Motorcycle tire and its movement on the flywheel

r = The radius of the tire

r_f = The radius of the flywheel

m= The mass of the tire

m_f = The mass of the flywheel

I= The inertia moment of the tire

I_f = The inertia moment of the flywheel

ℓ = The length of the contact of the tire on the ground

ℓ' = The length of the contact of the tire on the flywheel

If the increase in the pressure inside the tire on the ground is equal to the increase in air pressure when placed on the flywheel, or the deformed surfaces of the tire are equal in the two states, the radius of flywheel is obtained through Eq. (2) to Eq. (5):

$$\theta max = \arcsin(l/2r) \quad (2)$$

$$\theta' max = \arcsin(l'/2r) \quad (3)$$

$$l' = \sqrt{(2rf2\ (l - \cos 2\theta fmax))} \quad (4)$$

$$r(\theta max - \sin\theta max) = rf(\theta fmax - \sin\theta fmax) + r(\theta' max - \sin\theta' max) \quad (5)$$

In the actual state, the motorcycle with the mass of m and velocity of v is travelling on the road with a linear momentum of m_v. This momentum should also be simulated in an experimental state (when the tire is on the flywheel) in a way that the angular momentum of the tire around a contact point with the ground is equal to the angular momentum of the flywheel and the rotating equipment of the system around the same point expressed in the form [7]:

$$mr^2\omega = m_f r_f^2 \omega_f \quad (6)$$

Then m_f is obtained as

$$m_f = m(r/r_f)^2 \quad (7)$$

7 CONTROL ANALYSIS OF THE HYDRAULIC SYSTEM

The time and the state of equilibrium of the system are calculated in this section. In the hydraulic system of the device as shown in Fig. 4 and Fig. 6, when parameter (λ) exceeds the defined critical value (20%), the solenoid valve 2 opens and unlocks the oil path from the main cylinder (1) to cylinder (2). Cylinder (1) is connected to the brake lever and according to the test conducted on the brake system and its shoes, the brake shoes can be considered as a spring with a rather constant hardness coefficient of about $K_1 = 2.05\times10^5$ (N/m) in view of the relation between force with a replacement. Therefore, the value of the pressure inside cylinder (1) in terms of displacement from the base state (X_1) is equal to

$$P1 = (K1/A1)\ X1 \quad (8)$$

Cylinder (2) is used as a pressure reducer in the system which also contains a spring with K_2 hardness where

$$P2 = (K2/A2)\ X2 \quad (9)$$

With regard to the fact that the equilibrium of this system (from the time of the SV_2 opening) is obtained when the pressure in the whole system is equal to P ($P_1=P_2=P$) and also $P_1>>P_2$, the changes of X_1 and X_2 are obtained in terms of time. Also, the time for the system equilibrium is calculated by diagram (T-X); where R is the resistance of the system against the passage of the fluid whose unit is in terms of ($kg/m^4.s$).

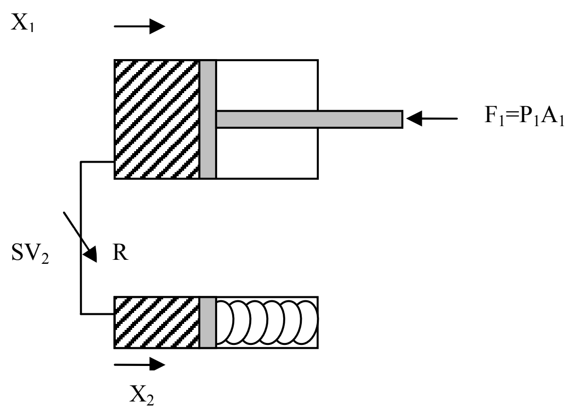


Fig. 6 The hydraulic system of the device

To calculate the approximate value of R, by using P-Q diagram (flow rate–pressure) presented for valve 2 in its catalogue, it can be calculated by

$$Q = P/R \quad (10)$$

By using control equations, the above system is a fluid system without the application of input signal and with the initial condition and when the SV2 is open, oil moves from cylinder (1) to cylinder (2). The above system is a two-order system.

The state equations of the hydraulic system are as follows:

$$(dX1/dt)A1 = -(P1-P2)/R \qquad (11)$$

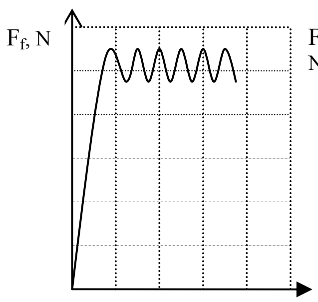
$$X1=-(K1/A12R)X1 + (K2/A1A2R)X2 \qquad (12)$$

$$(dX2/dt)A2 = (P1-P2)/R \qquad (13)$$

$$X2 = -(K1/A1A2R)X1 + (K2/A22R)X2 \qquad (14)$$

where, $K_2=10^4$ (N/m), $K_1=2.05\times10^5$ (N/m), $D_2=16$ mm, $D_1=40$ mm, $R=1.5\times10^9$ (kg/m^4.s).

The important point which has to be taken into consideration is that if the passing flow rate is in excess of time, it can be solved by increasing of the spring force through calculations as well as by increasing the rate of the path resistance by a needle valve through trial and error.



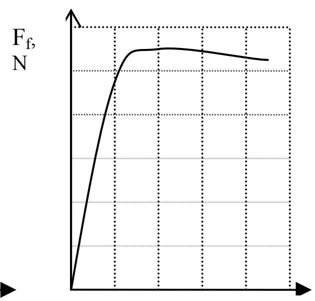


Fig. 7 Commom ABS **Fig. 8** Proposed ABS

The rate of the oil passing through the larger cylinder is reduced and if the flow rate is less in relation to time, it can be increased by reducing the spring hardness coefficient. The diagram for the location changes of cylinder 1 [X_1] and cylinder 2 [X_2] is drawn in terms of time. By the obtained state equation, the system equilibrium can be studied from the time when the valve reaches a fully open state to the fully closed state; and the system equilibrium is studied after that. Since time is very important in this time system and the rate of the displacement volume to attain the equilibrium state is not much and this displacement is done within a very short time (because of the difference between the high pressure of P_1 and P_2), the time and the general flow rate of the system cannot be ignored when the valve is fully open and fully closed. The calculation of the passing flow rate is done during the time when the valves are open and closed longer. Since the reaction time in this system is very short, small parameters effective on the speed and reaction time of the system should also be taken into consideration. One of the parameters is the time required for the solenoid valve to reach the open state from the closed state, or the time taken for it to be closed from the open state. In this device, these valves take 0.03 sec to open and 0.04 sec to close.

When the closed valve begins to open, the line resistance against fluid passing tends to R value from the infinite value. According to Eq. (10), the zero rate of the flow increases gradually so that the valve is opened completely. As it is not possible to apply R variations easily, in order to simplify the problem, it can be assumed that the R value during the time is constant and equal to R. But while the valve is open, the pressure variation increases linearly from zero to its final value which depends on the initial situation of the system. So, the system state equation will be as follows:

$$X1=[-(K1/A12R)X01+(K2/A1A2R)X02]\times(t/ted) \qquad (15)$$

$$X2=[-(K1/A1A2R)X01+(K2/A22R)X02]\times(t/ted) \qquad (16)$$

t_{ed} = the time for the valve to open or close

$0 \le t \le t_{ed}$

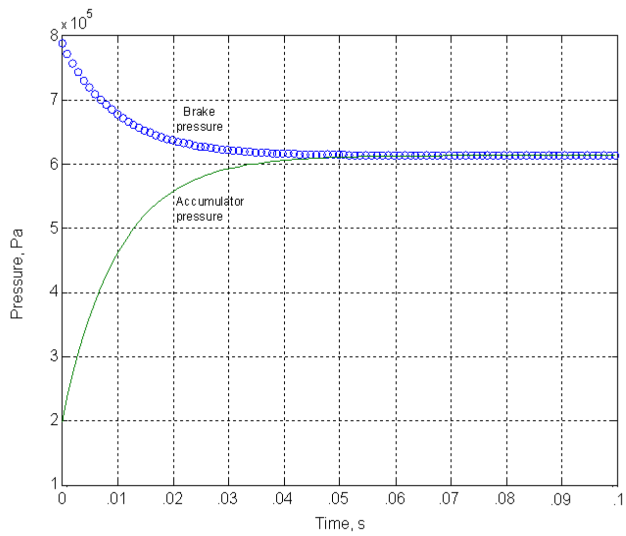


Fig. 9 The pressure variations of cylinder 1

The pressure variations diagram of cylinders 1 and 2 according to time (since the valve opens) has been shown in Fig. 9 and Fig. 10. These diagrams are investigative of the amount of pressure variations in cylinders 1 and 2 from the initial state (P_{01}, P_{02}) according to the time. Also, the displacement variations are calculable through these diagrams as:

$$(X=PA/K) \qquad (17)$$

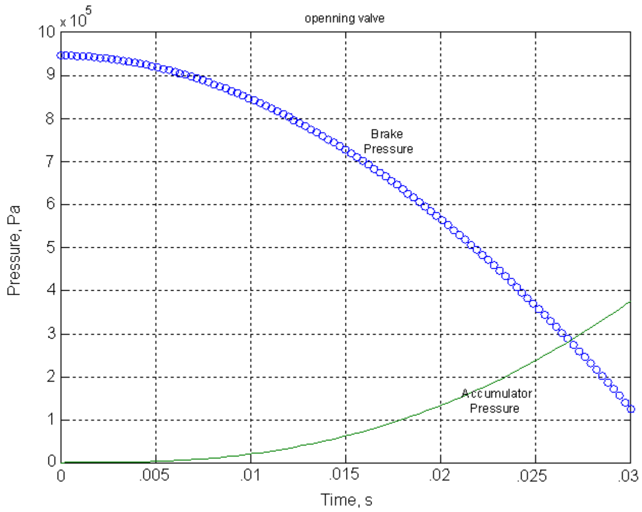


Fig. 10 The pressure variations of cylinder 2

8 METHOD OF PERFORMANCE

By measuring the tire speed and the vehicle speed, this device calculates the value of the λ coefficient and then the calculated λ is compared with the defined limits for λ and the limit in which λ is placed is controlled at any moment; and on the basis of each of the limits, a series of operations and commands are applied on the two solenoid valves. As a result of these operations, the value of λ is kept as approximate as possible to the optimal value of λ=20% (when required) at any moment within the permitted limit. If λ enters the unstable limit, the tire pressure is quickly reduced and, within the optimal limit, the pressure is kept constant and the pedal pressure is freely applied to the brake cylinder within the λ≤18% limit.

The limits defined in the software of the device are:

1- 0≤λ≤18% limit: It is the normal braking limit where there is no limit imposed by the solenoid valves on the applied pressure to the brake cylinder by the pedal.

2- 18≤λ≤20% limit: The value of the friction coefficient is very close to its maximum state within this limit and stoppage takes place within the shortest time. While the stability of the tire on the ground is within an acceptable limit, the pressure increase behind the brake cylinder is prevented.

3- λ≥20% limit: Entering the unstable limit where the cylinder pressure is separated from the system pressure and the pressure of the brake is reduced alternatively at great speed until λ enters the stable limit. If λ is in 18≤λ≤20% limit when it enters the stable limit, the pressure of the brake cylinder is kept at this value, and if it is within the limit of λ≤18%, the connection between the brake cylinder and the pedal is established and the cylinder pressure is increased again and the cycle continues in this manner until the vehicle is stopped or the driver releases the brake pedal. Therefore, by defining a λ coefficient independent of the conditions of the road and the weight, the friction coefficient can be kept at its maximum limit in critical conditions and the maximum attainable preventing force between the tire and the road is used.

The desirable operation of this system depends on the use of the solenoid valves with a fast response and reaction speed. The faster the valves operate, the more precise the operation of the system will be. Tests have shown that on very slippery surfaces, the shortest distance of stoppage occurs when the tire is at its locking state. The reason is when the tire slips on the surface of these roads in a locked state, a wedged surface is produced in front of the tire resulting from the drag of the tire on these surfaces which reduces the stoppage distance, but it should be taken into consideration that if the tire is to slip on the surface, the stability of the tire and the vehicle is lost. On these surfaces, the time frequency of the discharge valve should become longer so that the tire has more time for slipping. Therefore, it is desirable to define a state in the control system of the device separately for the driver driving on very slippery roads (completely icy roads) and a pressure sensor should also be provided in the system with a defined threshold pressure limit. Therefore, the controlling section controls the pressure and the value during braking in real time. If the pressure within the defined limit is on very slippery surface, the frequency changes the solenoid valves (reduces) and if the pressure in the brake system is not within this limit (increases), shorter frequencies are used. Also, because the transfer of λ from the stable and optimal limit to the unstable limit is done very fast on very slippery roads (the rate of road slippery is definable for the electronic system by the pressure sensor or pressure switch), an added limit should be defined before 18≤λ≤20% for the device on slippery roads (for example, 15≤λ≤18%) so that the determination of the rate of pressure increases, and the pressure transfer from cylinder 1 to cylinder 2 is carried out in a controllable manner within this limit. In this system, the 15≤λ≤18% of the main valve is responsible for the connection between the pedal with the brake shoes and is also connected and disconnect by a definite frequency and after each connection and disconnection. It controls the value and the growth rate of λ according to the defined limits and acts according to the mentioned limits. This brake control system is installed on the vehicle, and the braking system of the vehicle should operate on oil. For this purpose, two hydraulic cylinders of 32mm and 40mm in diameter, an oil tank which can compensate the oil shortage in the system and also an extra oil discharging container are required to make up for the reduction of the system pressure. A cylinder of 21 mm in diameter is used as the cylinder under the brake pedal and another cylinder of 40 mm in diameter as the cylinder exerting force on the lever of the brake shoes. It is worth mentioning that if this system is installed on motorcycles with disc brakes, there won't be any need for the pedal and caliber cylinders any longer. Another part of the hydraulic system consists of two solenoid valves, one with an orifice of 15mm and another with an orifice of 2.5mm. The larger valve which is for the main path of oil coming is represented by SV1 in Fig. 4. This valve closes the passage of the main oil from the pedal to the cylinder of the brake shoes. The second valve with an orifice of 2.5mm is the SV2 which reduces the pressure of oil behind the main cylinder of the brake. This valve closes and opens with a frequency of about 40Hz and reduces the oil pressure alternatively, and is suitable for the rate of reduction of the tire speed and the rate of tire slippage on the surface up to a limit proportional to the road conditions and the tire.

9 COMPARISON

Recently, the VSS (variable-structure system) design technique has successfully been applied to ABS control problems. The sliding mode ABS controller design, using pulse width modulation (PWM) and switching control have both obtained good experimental results on their test stand. [3]

As it was already mentioned, in the existing anti-lock braking systems, when a wheel is in the threshold of locking, i.e. the velocity of that wheel compared to other wheels is increasing a greater rate, the electronic controlling unit (ECU) device takes action and in proportion to the rate of the changes of the intended wheel velocity, sets the wheel free by a definite frequency alternatively and with the pressure change of the oil in the cylinder or the brake caliber related to that wheel (this frequency is proportional to the negative velocity changes of the wheel). With regard to the qualitative diagrams of Fig. 7 and Fig. 8 , the frictional force (F_f) changes in proportion to the time of the peak point related to the time of the tire at the threshold of slippage (it is known that the frictional coefficient is at the maximum state in this state). Parts of the curve immediately after the peak with sharp gradients are related to the state of the tire reaching complete slippage. The part of the diagram with decreased gradient is related to the time of brake pressure reduction simultaneous with the time when the tire is free of slippage, and has come to a complete rolling; and where the gradient of the curve is again positive is the time when the pressure behind the brake cylinder is increased and this stage is repeated. If we want to draw a diagram for the functioning of the proposed brake system approximately similar to the above, this device calculates the rate of the tire slippage on the surface of the ground at any point of braking process by calculating λ. On the basis of this assumption that the optimal state of the brake in which the maximum force of the brake between the tire and the ground is exercised, a percentage of slippage always happens within a definite limit ($18<\lambda<20\%$). The ECU circuit program is written by IC 8051 and set in a way that when entering this limit, λ is kept within this optimal limit and brings it as close to this limit as possible until sufficient force is applied on the brake pedal by the driver.

Therefore, contrary to the current anti-lock braking systems, there is no alternative decrease and increase in pressure in this process. But (when the brake pedal is sufficiently pressed hard), after reaching (or nearing) the optimal state, the frictional force of the road and the tire keep the tire within the limit near the slippage threshold, defined as the optimal limit, and the value of the pressure of the brake cylinder is kept at this optimal state until the braking process is ended (the motorcycle is stopped or the brake pedal is released). Therefore, the factors of the frictional force changes (preventive force) for this system are as follows:

One of the flaws of this mechanism is that since the brake shoes are continuously in contact with the brake disc during braking, the heat increase is high and during braking, there will be a reduction in the frictional coefficient resulting in a reduction in the applied torque by the tires, but since during braking speed is reduced and we know that the frictional coefficient is increased by reduction in speed, a part of the reduction of the frictional coefficient produced by an increase in the temperature of the brake shoes is compensated by an increase in the frictional coefficient resulting from a reduction in speed. On the whole, the negative gradient reduces the brake torque reduction. This negative gradient is more in drum brakes than is disc brakes. The more the reaction speed of the solenoid valves provided in the proposed system, the more the braking process approximates the optimal state (the height of the diagram is higher) and the more effectively the brake operates. One of the positive specifications of the proposed system is the omission of the minimum points of the diagram of the brake function which itself results in an increase in the surface under diagrams.

10 RESULTS AND DISCUSSIONS

An experiment was conducted on the tire at four different speeds for the braking systems at normal and ABS conditions and the time duration of stoppage was measured by a chronometer at each of the speeds and conditions. As it was mentioned, the reaction speed of the system's hydraulic valves were slow; therefore, the time duration for the stoppage of the tire at the ABS condition was considered on the basis of the first stoppage which included one error (Table 1).

Table 1 System's practical results

Stoppage Time New ABS	Stoppage Time non-ABS	Motorcycle Speed
1 s	1.66 s	50 km/h
1.13 s	1.82 s	60 km/h
1.22 s	2.06 s	70 km/h
1.43 s	2.47 s	80 km/h

The results show that the stoppage time for the new ABS is about 40% less than that in the non-ABS. This design does not have too many components to be installed on common motorcycles. Braking operation is performed electronically with more precision in all weather conditions including snowy, rainy, and normal conditions. In spite of the development made in motorcycle industry, most motorcycles are not equipped with desirable brakes and some are only equipped with disc brakes instead of brake shoes. It is clear that most of the accidents and casualties result from malfunctioning of the brakes installed on motorcycles which cannot control the tire slippage while braking. So, a relative solution to this problem can be an effective step toward safety of motorcycle rider and urban traffic. Naturally, the effects of this design are also clear from social and economic aspects.

12 CONCLUSION

1. The experimental results show that the new hydraulic anti-lock braking system based on slip ratio variations is applicable to motorcycles.

2. This new idea will result in a simple and cheap anti-lock braking system for motorcycles.

3. The results show that the stoppage time for the new ABS is about 40% less than the non-ABS.

4. These results are obtained based on available solenoid valves with 0.04 second response time while the desirable response time had been predicted 0.02 second.

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