POLITECNICO DI MILANO





XVII Scuola Nazionale di Dottorato – Bertinoro, 8-13 luglio, 2013

Longitudinal dynamics: introduction and problem setting

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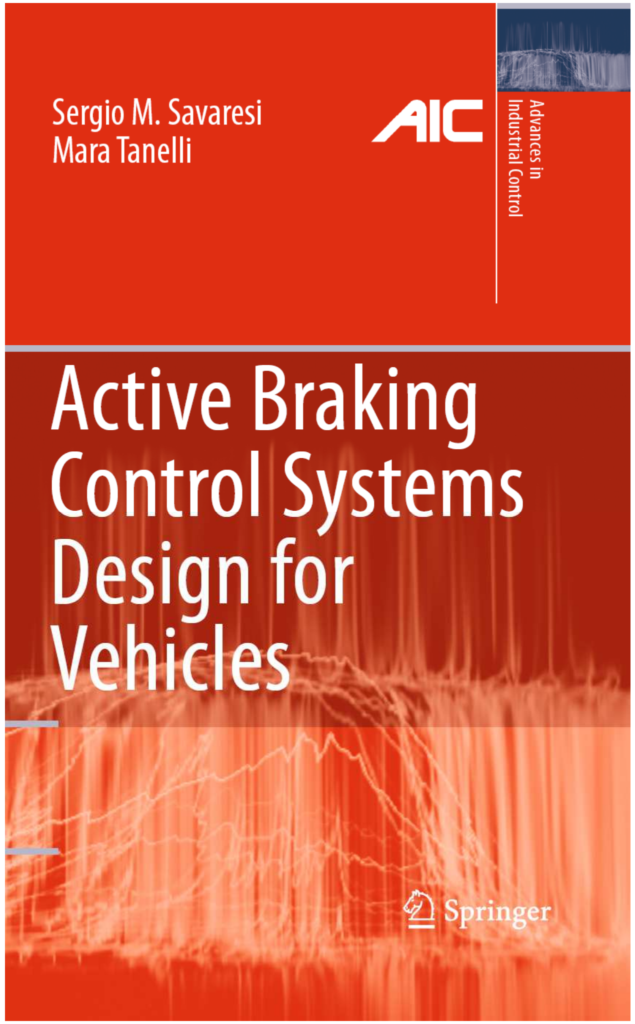
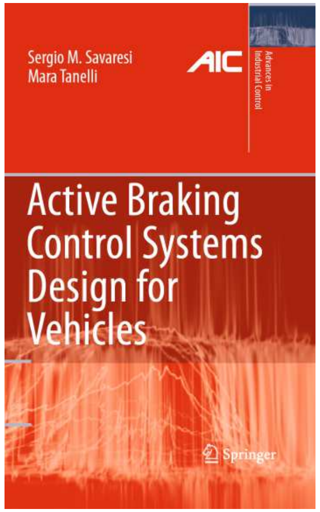
Main references:

TA Johansen, I. Petersen, Kalkkuhl J., J. Lüdemann (2003). "Gain-Scheduled Wheel Slip Control in Automotive Brake Systems." IEEE Transactions on Control Systems Technology, Vol 11, No. 6, pp.799-811.

Savaresi s.m., M. Tanelli, C. Cantoni (2007). Mixed slip-deceleration control in automotive braking systems. ASME Transactions: Journal of Dynamic Systems, Measurement and Control, Vol.129, No. 1, pp.20-31 [*2008 ASME Dynamic Systems and Control Rudolf Kalman Best Paper Award*][Patent]



References - Book



The following slides are adapted from those used in the course ''Automazione nei mezzi di trasporto''

(M.Sc. course at the Politecnico di Milano, Prof. Sergio M. Savaresi)

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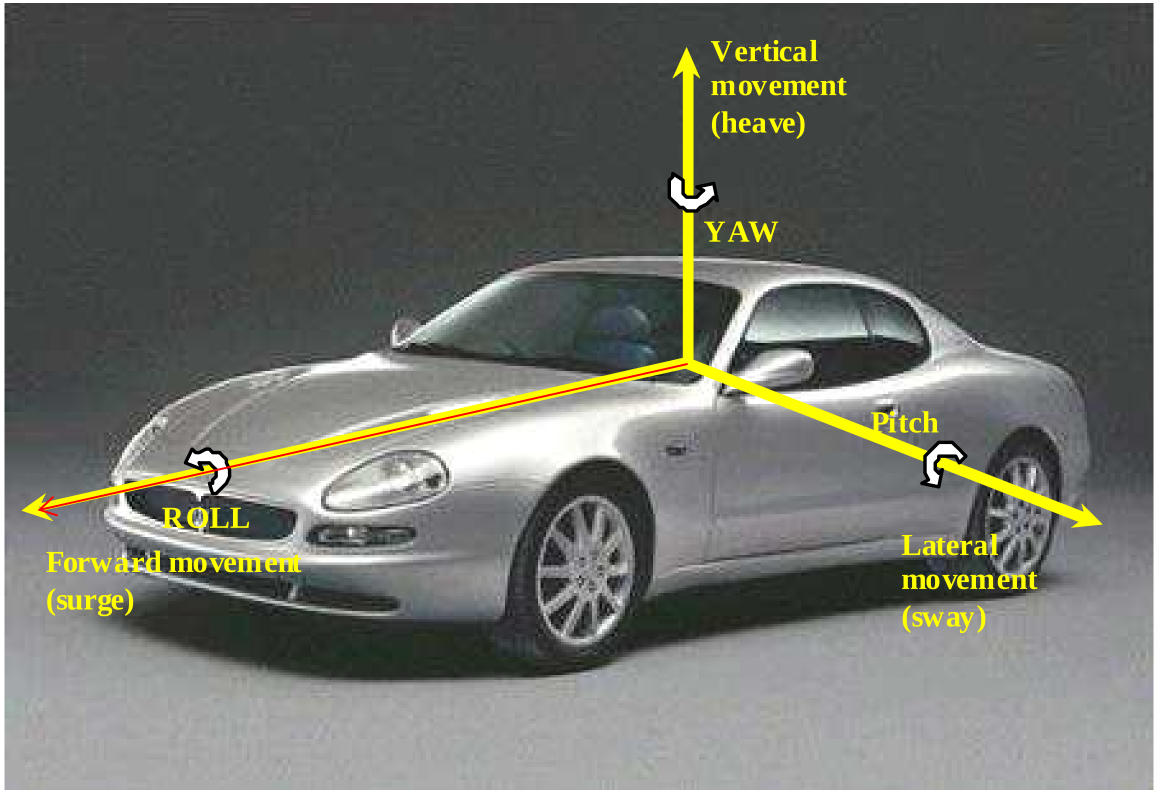


Braking&traction-control = longitudinal (forward) movement

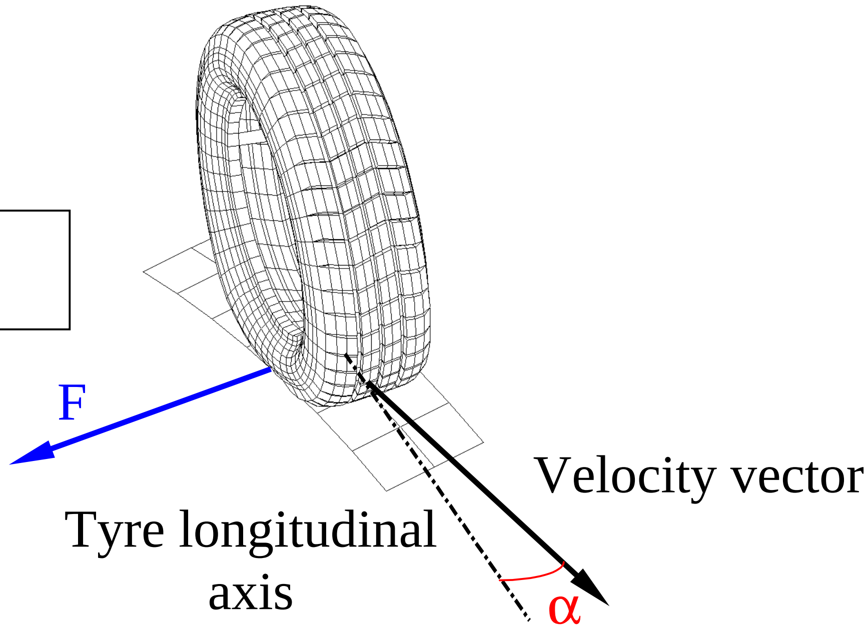
Tire-road contact forces: introduction

Separable (along the road surface) into two orthogonal components: longitudinal (F_l or F_x) and lateral/side (F_s or F_y)

$$F_l(\lambda, \alpha, F_z, \gamma) \qquad \mu_l := \frac{F_l}{F_z}$$

$$F_s(\lambda, \alpha, F_z, \gamma) \qquad \mu_s := \frac{F_s}{F_z}$$

μ : friction coefficient



λ= longitudinal slip
α= Slip angle
γ= Camber angle
Fz= Vertical load

Road-tire contact forces: the definition of (longitudinal) slip

$$\lambda = \frac{v - \omega r}{\max\{v, \omega r\}}$$

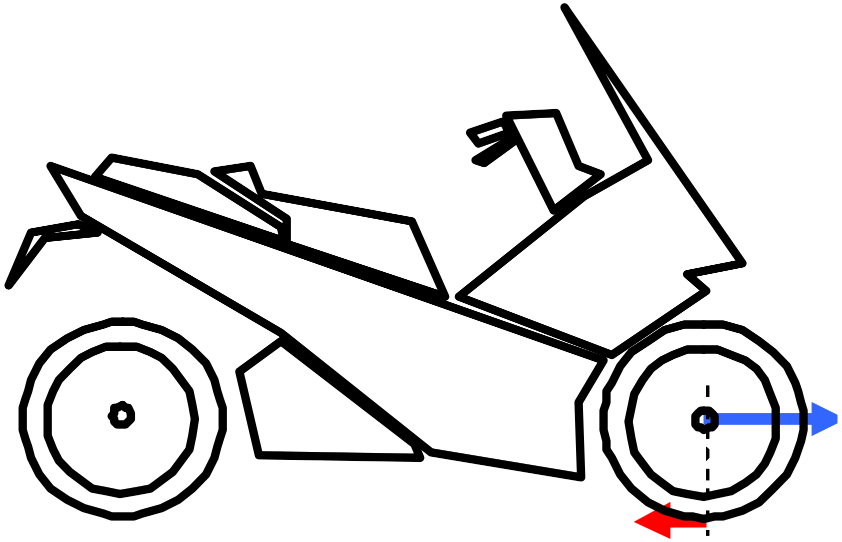
If brake:

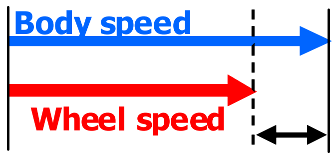
$$\lambda := \frac{v - \omega r}{v}$$

If traction:

$$\lambda = \frac{-(v - \omega r)}{\omega r}$$

v: body speed
r: rolling radius
ω: angular wheel speed





$$\text{"SLIP"} \quad \lambda = \frac{\text{Body speed} - \text{Wheel speed}}{\text{Body speed}}$$

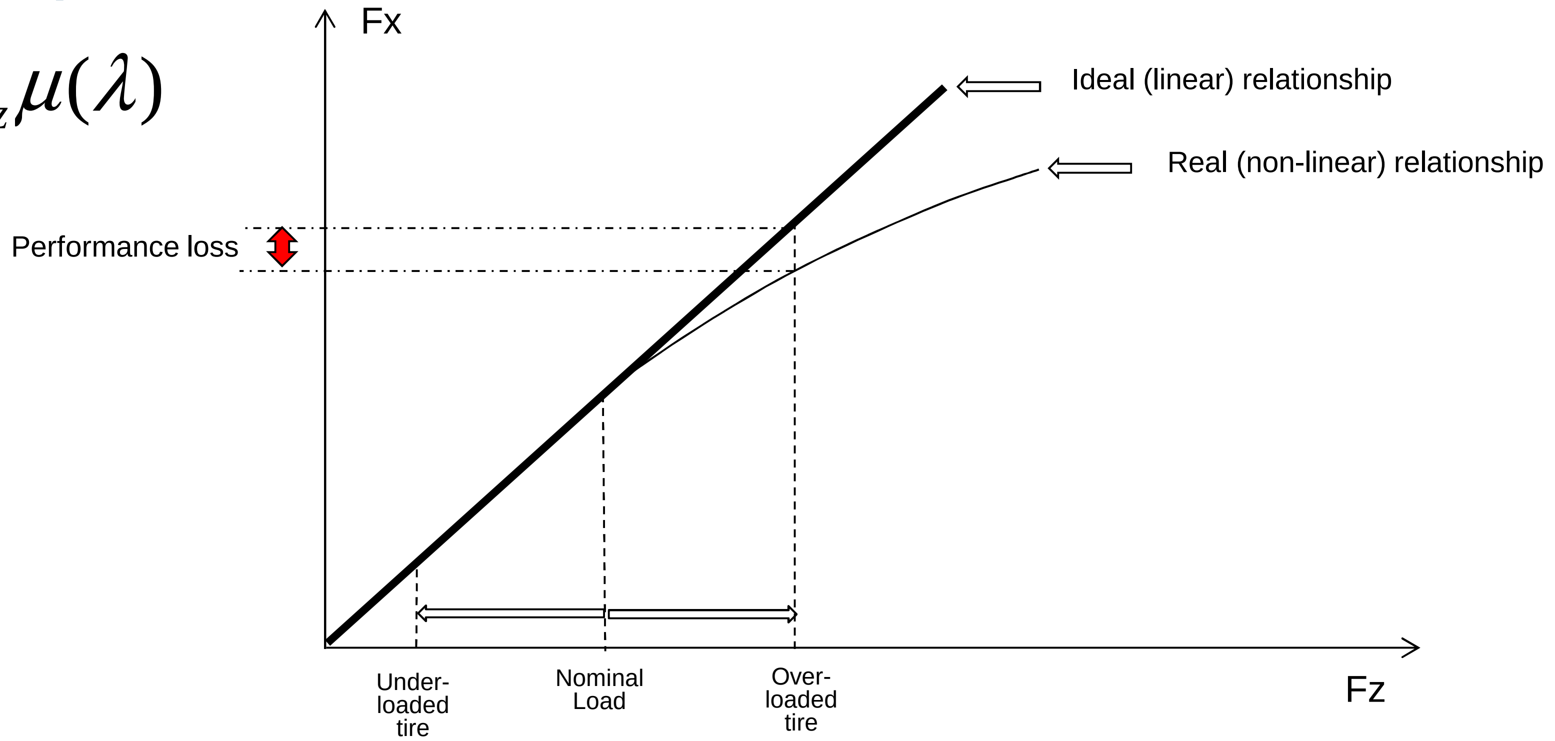
Linearity with respect to Fz - REMARK

Cosider the simplified relationship (alpha=0; gamma=0)

$$F_x(F_z,\lambda)$$

Usually it is approximated by assuming a LINEAR relationship:

$$F_x = F_z\mu(\lambda)$$



Tire-road contact forces: dynamics

$$F_l(\lambda,\alpha,F_z,\gamma)$$
$$F_s(\lambda,\alpha,F_z,\gamma)$$

are algebraic relations

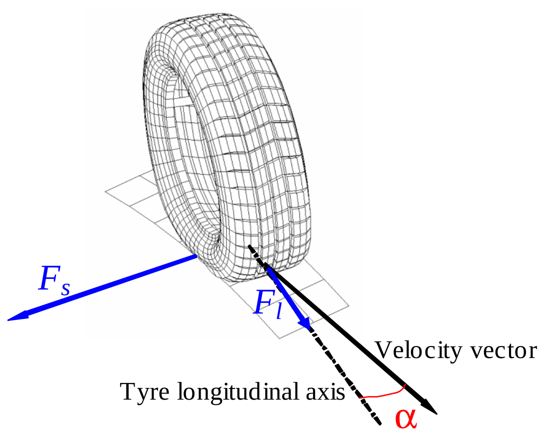
The real behavior is dynamic, due to the transients in tire deformation.

Dynamics (simplified) model:

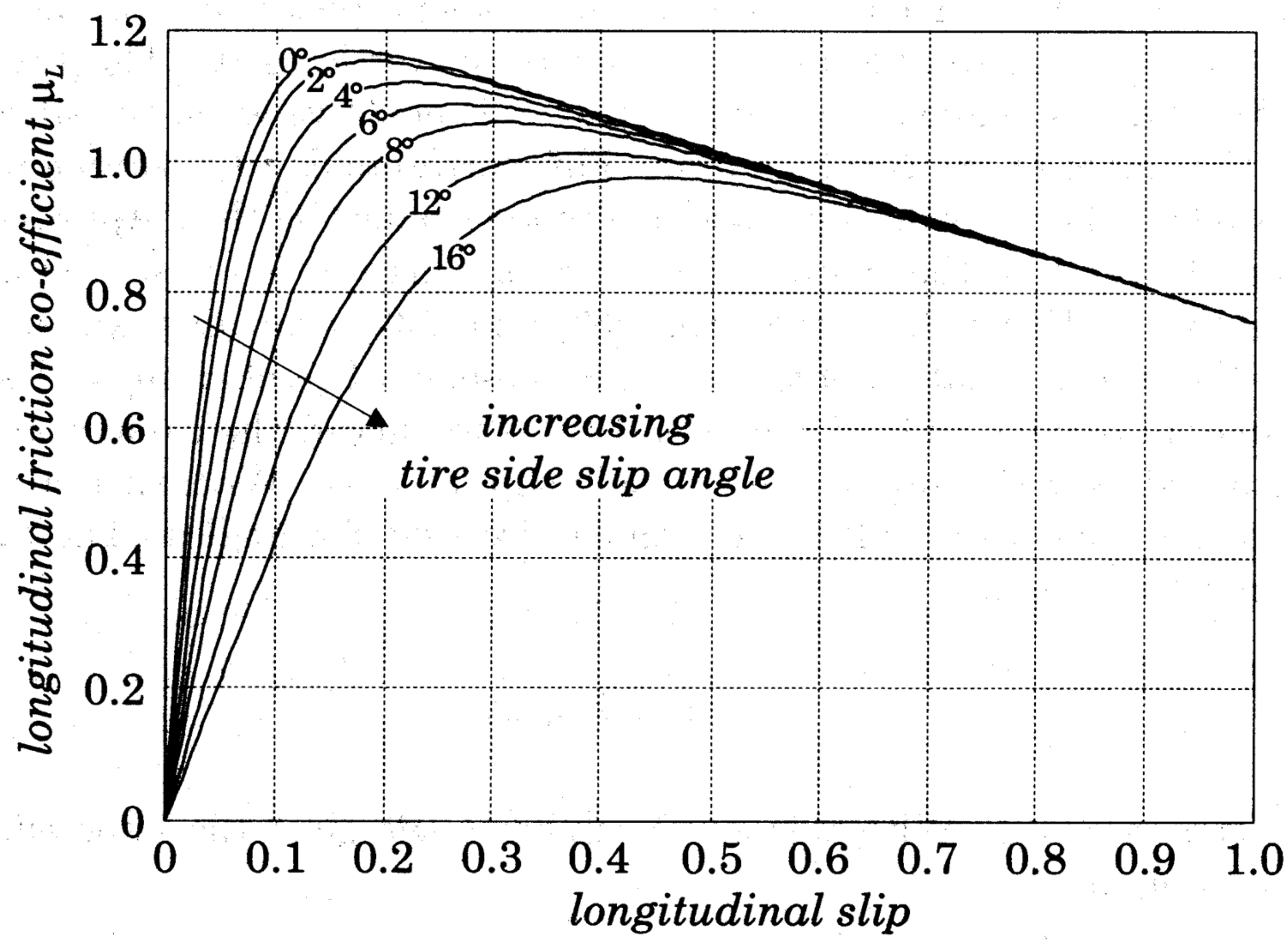
$$\left[\frac{\left({}^{v_w}/_{S_{0l}}\right)}{s+\left({}^{v_w}/_{S_{0l}}\right)}\right]$$

wheel speed

relaxation length (about 1/2 circumference)



Contact forces: longitudinal



Dry surface
Camber: null

if there is no slip the longitudinal force is zero

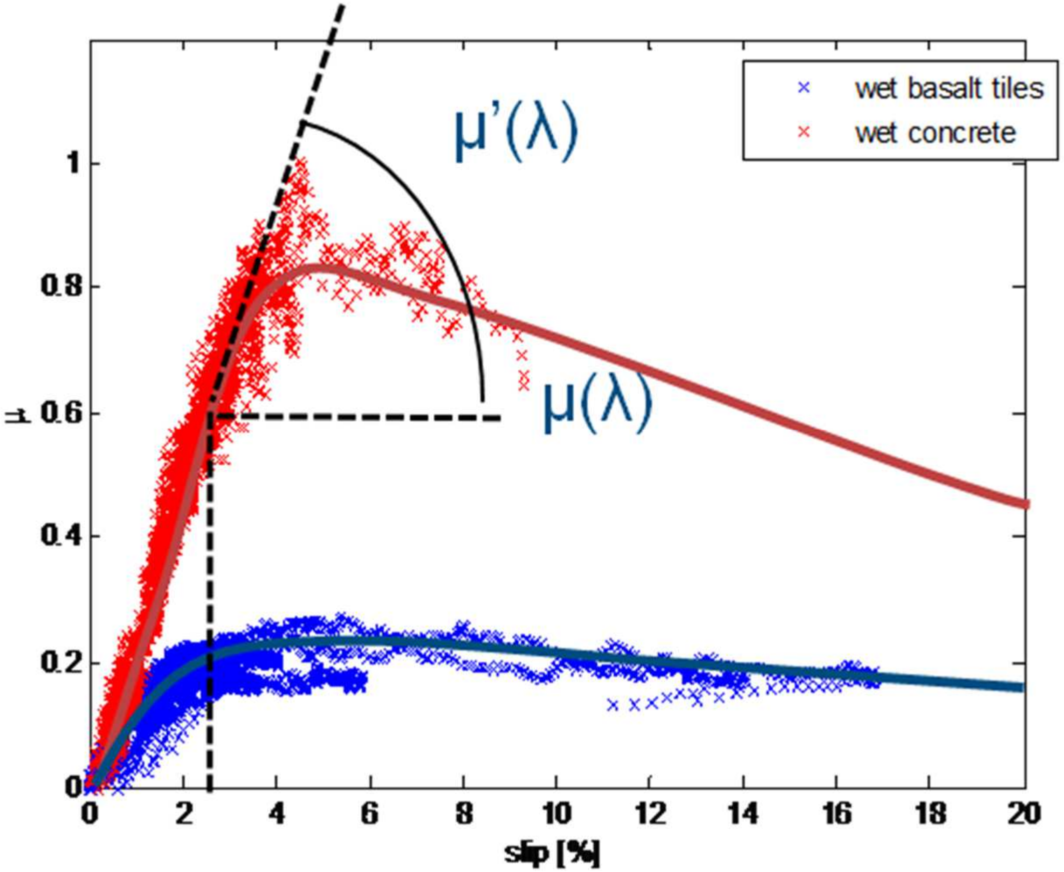
force increases linearly up to values of λ of about 0.1-0.15

beyond the maximum: slope sign changes

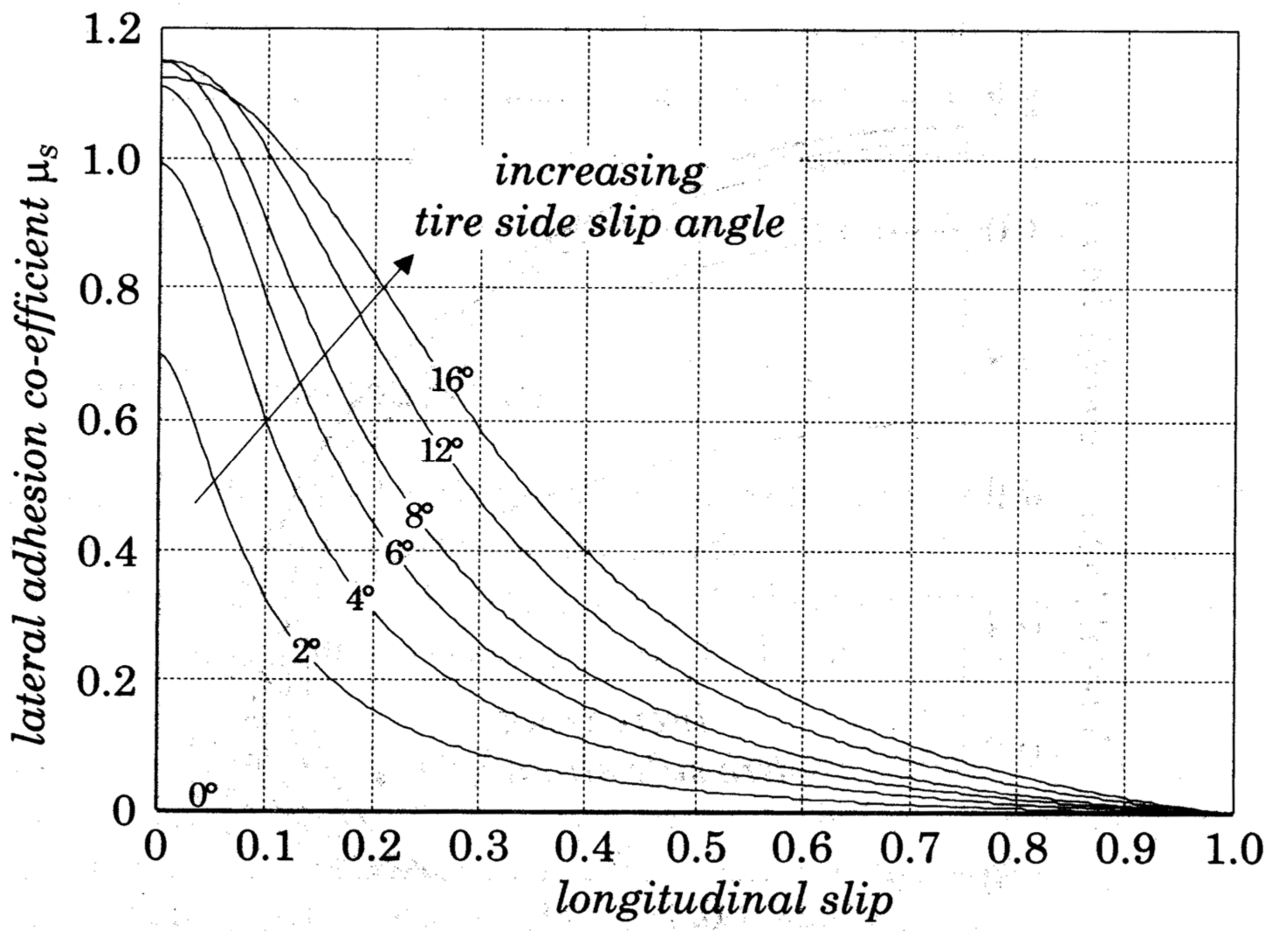
Increasing side-slip angle:

- Reduction of longitudinal force
- The maximum point moves forward

Experimental results



Contact forces: lateral



Dry surface

Camber: null

if there is no side-slip angle: lateral force is zero

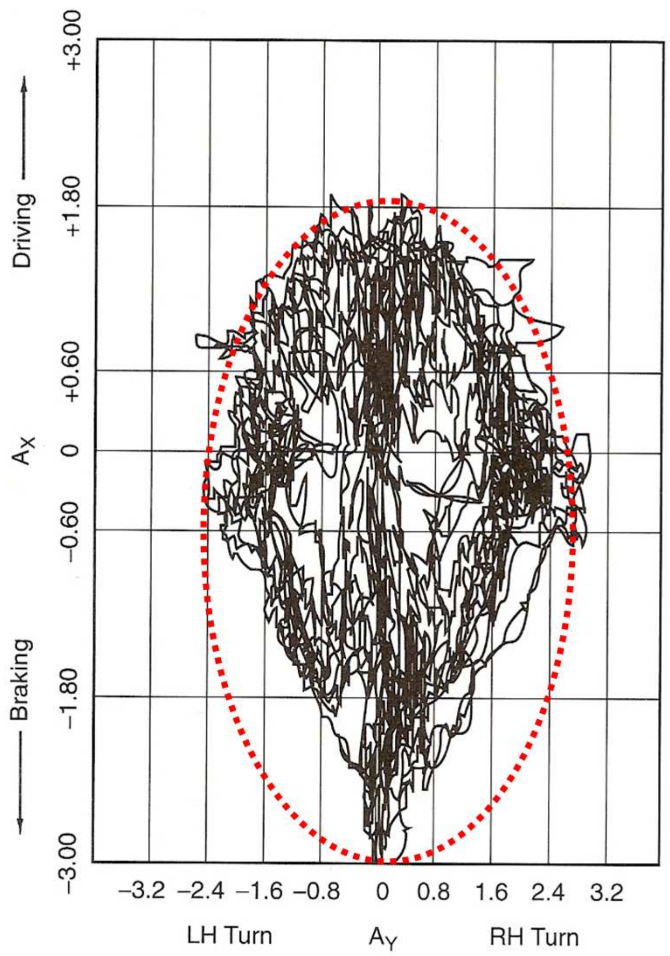
Force increases with increasing side-slip angle

Force reaches its maximum at λ= 0

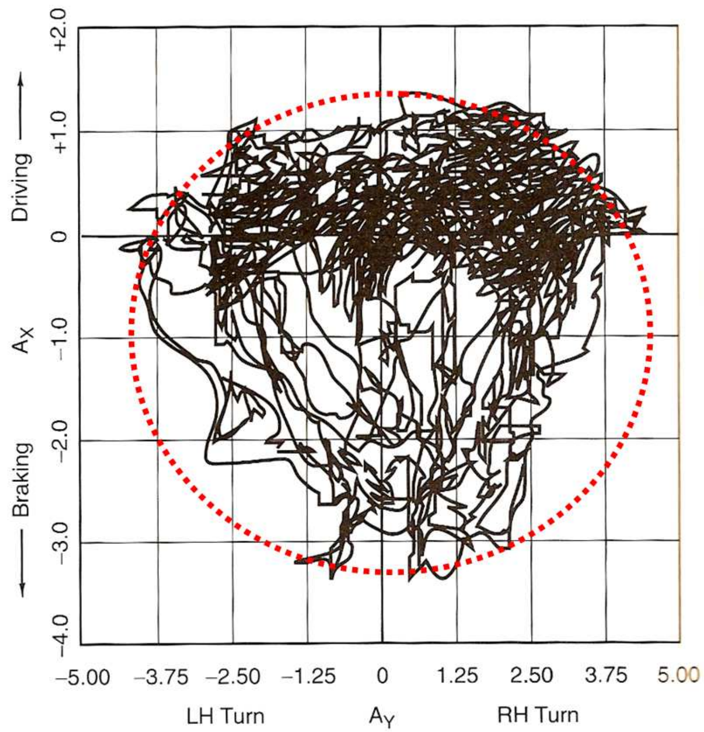
If wheels locked (λ= 1) lateral force vanishes (total loss of directionality)

Note: for fixed α, varying λ transfers force to/from longitudinal to/from lateral force

g-g plots

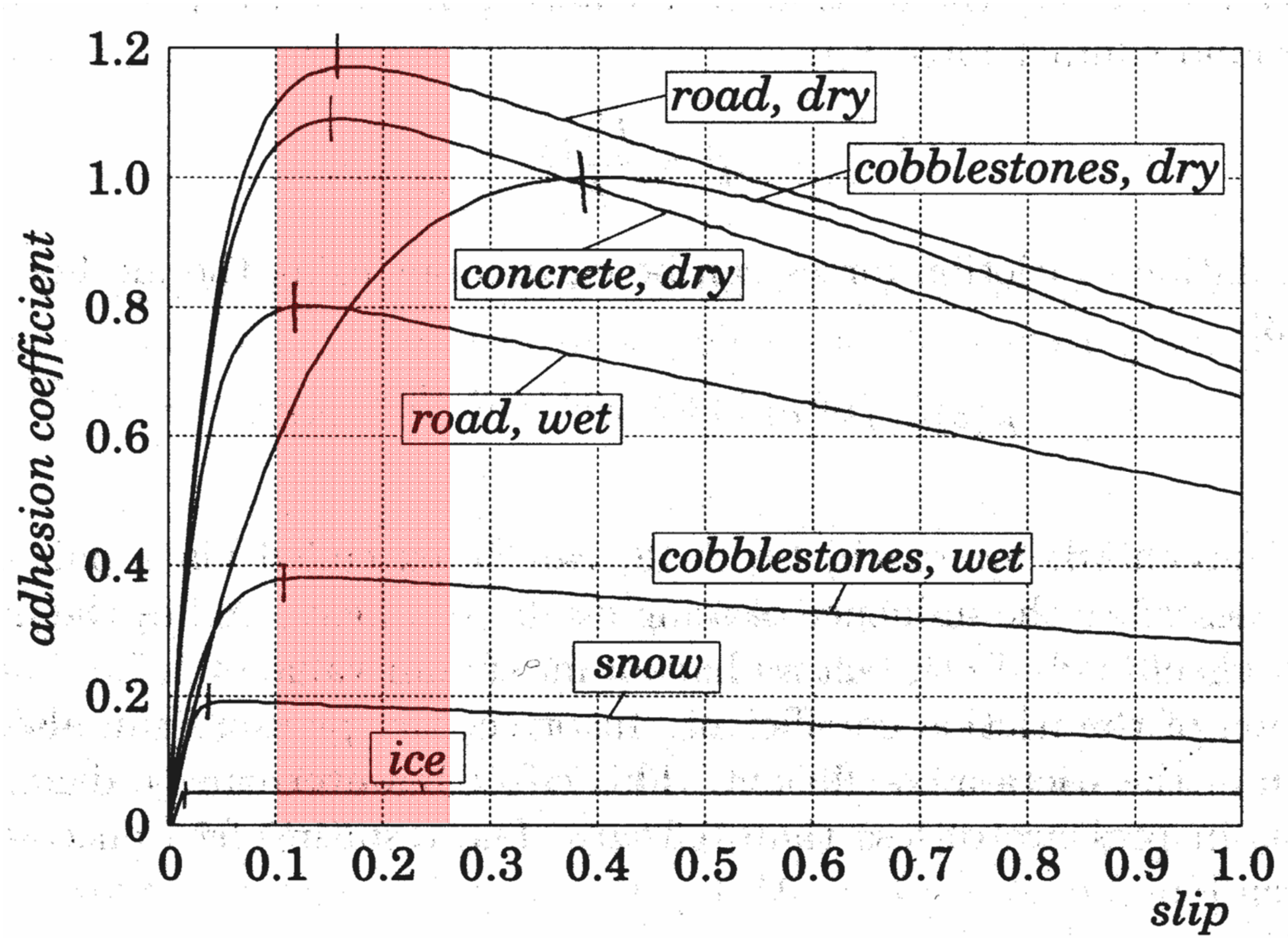


Senna, '87, Adelaide



Herbert, 93, Silverstone

Dependence of the contact forces from the road surface (longitudinal force)



Variation of the shape

Variation of the peak

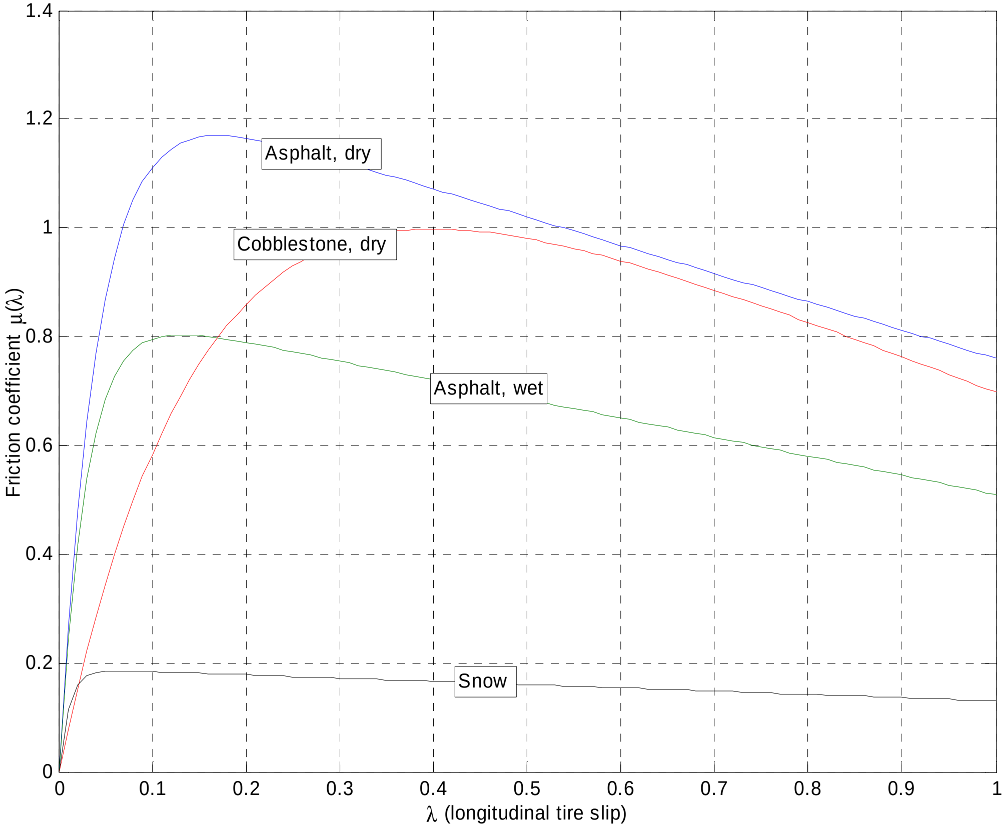
Formulas for modeling tire-road contact forces

Year	Model Name	Model Properties	Features
	Piecewise Linear Model	Empirical	1. Cannot accurately fit curves 2. Easy to identify
1993	Burckhardt Model	Semi-Empirical	1. Can accurately fit curves 2. Has some revised formula
1994	Rill Model	Semi-Empirical	Easy to identify
1987	Magic Formula	Semi-Empirical	1. Can accurately fit curves 2. Has lost of revised formula 3. Can employ different factors
1977	Dahl Model	Analytical	1. Can describe Coulomb friction. 2. Can produces smooth transition around zero velocity
1991	Bliman-Sorine Model	Analytical	Can capture the Stribeck effect in addition to Dahl model
1995	LuGre Model	Analytical	Can combine pre-sliding & sliding in addition to Bliman-Sorine Model

Integrated longitudinal and lateral tire / road friction modeling and monitoring for vehicle motion control, Li Li, Fei-Yue Wang; Qunzhi Zhou, IEEE Transactions on Intelligent Transportation Systems, Volume 7, Issue 1, March 2006 Page (s): 1-19

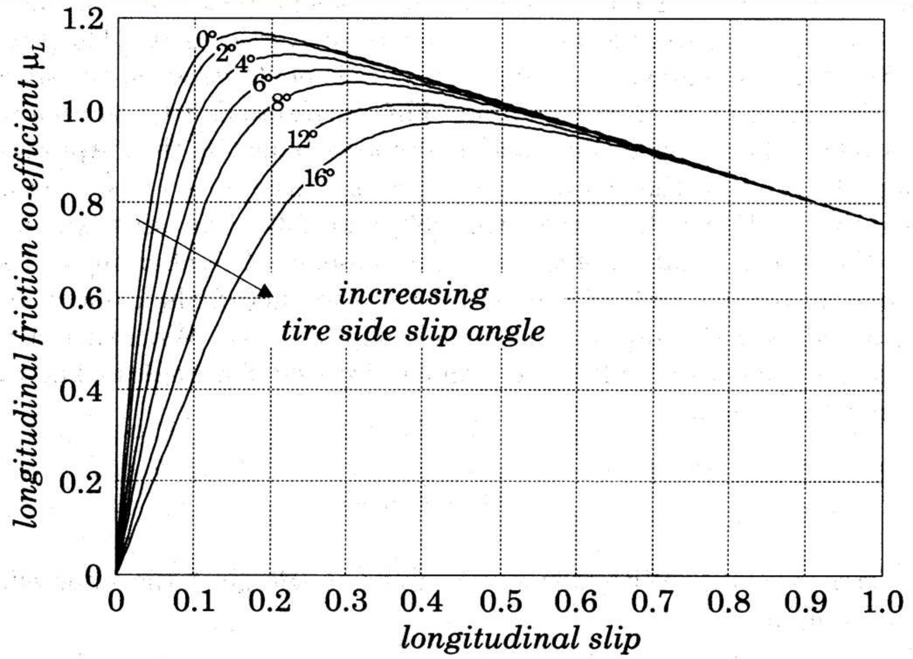
Burckhardt formula for longitudinal force

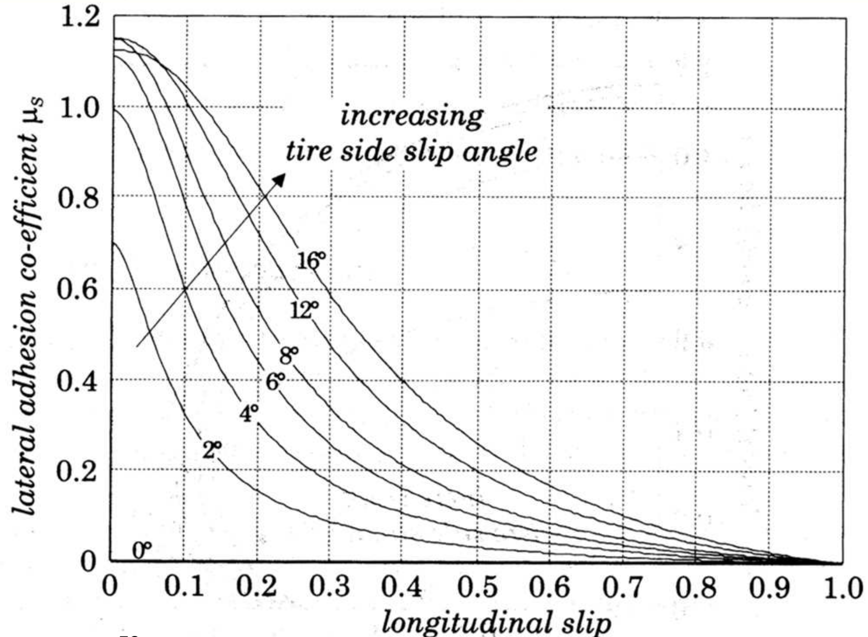
$$\boxed{\mu(\lambda;\theta_r) = \theta_{r1}\big(1-\exp(-\lambda\theta_{r2})\big) - \lambda\theta_{r3}}$$



	θ_{r1}	θ_{r2}	θ_{r3}
Asphalt, dry	28.01	23.99	00:52
Asphalt, wet	0.86	33.82	12:35
Cobblestones, dry	1:37	6:46	0.67
Snow	00:19	94.13	12:06

Problem of brake/traction control: summary





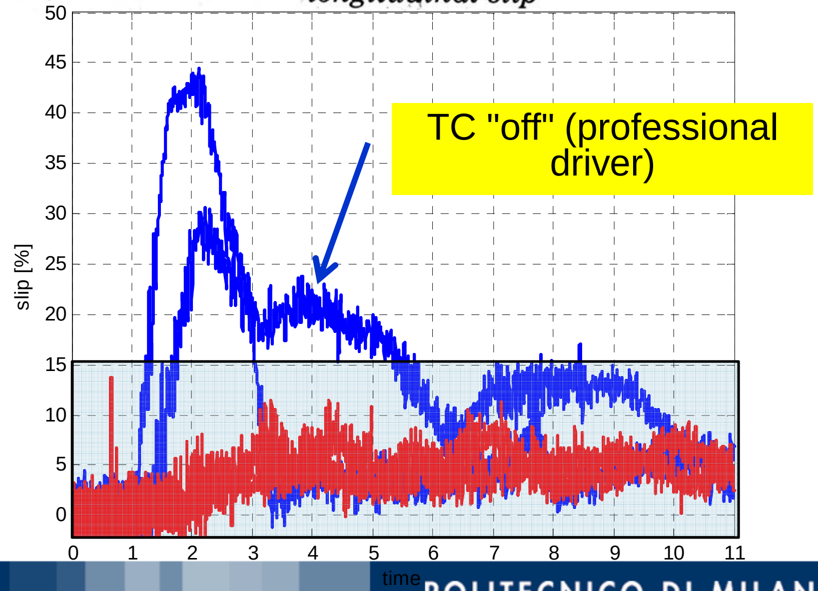
Straight brake / acceleration:
ideal λ is around the peak.

Three issues (ABS/TC motivations):

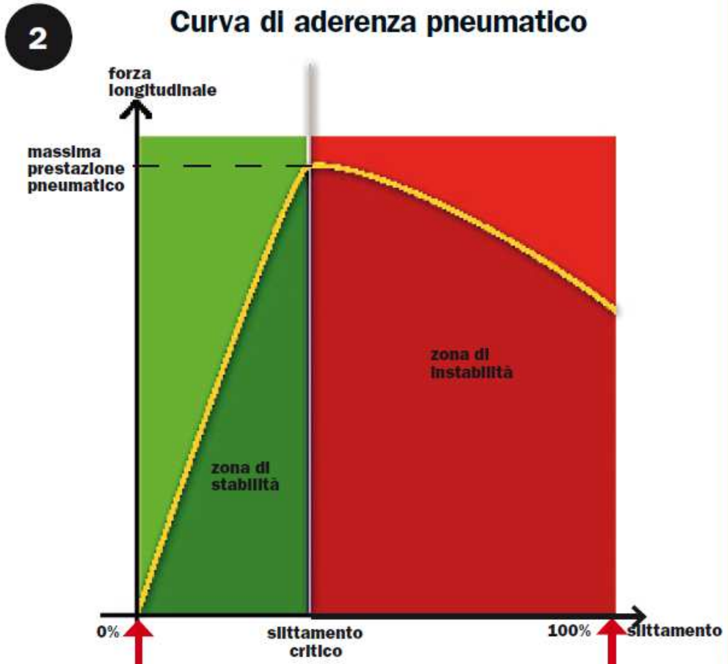
λ=1: loss of brake/traction (- 30%)

λ=1: NO lateral forces

After the peak (λ> 0.15): open-loop instability!

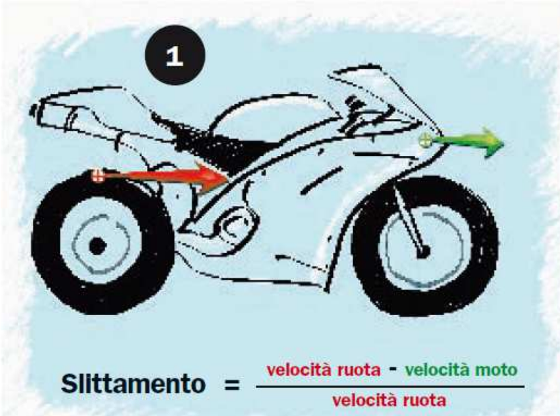


The problem of “TC”: why useful?



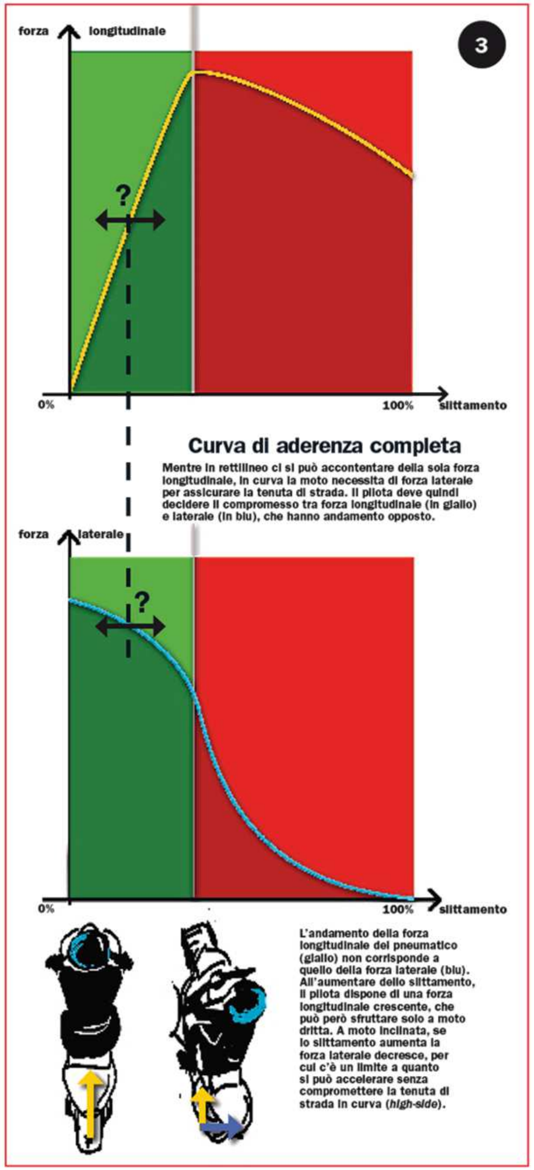
Nella marcia a velocità costante e bassa il pneumatico lavora con slittamento quasi nullo.

Durante un burn-out lo slittamento è al 100%: la gomma gira a moto ferma. La forza a terra è limitata.



Un po' di slittamento non fa mal male
La definizione di slittamento è legata al fatto che la deformazione del pneumatico, necessaria a produrre l'aderenza a terra, tende a tradursi in una velocità periferica superiore a quella di effettivo avanzamento del veicolo. Solo se le due velocità coincidono, lo slittamento è pari a zero; all'estremo opposto, quando la ruota si muove a moto ferma lo slittamento è totale (100%). In uscita di curva (sopra), la situazione è intermedia.

The dilemma of the driver when cornering ...



ABS: History ...

One of the first patents "An Improved Safety Device for Preventing the Jamming of the Running Wheels of Automobiles When Braking ') was filed in Europe in 1932, while a similar result (' Apparatus for Preventing Wheel Sliding ') was obtained in the United States in 1936.

In the railway industry: first application around 1943

The aeronautical field: 1947, B-47 bombers to avoid an excessive increase in braking distance on icy surfaces and damage to the tires.

In 1968: airplanes 'Thunderbirds'.

Ford, Chrysler and Cadillac offered the ABS on a limited number of vehicles. Pneumatic Actuators: very slow and increased stopping distances.

Germany: a joint venture formed by Telefunken and Bendix tried to market a system called ABS Tekline. But the electronic design and manufacturing technologies were too young and unreliable.

1978: Bosch announces "Anti-Blockier System" (hence the acronym ABS).

In the same year, Mercedes Benz offered for the first time the ABS as an option for its high-end (S-class) cars.

Association of European Car Manufacturers (ACEA): equipment required for all new cars registered from 1 July 2004.

ABS will becoma mandatory for motorcycles in Europe since 2017.

1936
A Robert Bosch viene riconosciuto il brevetto numero 671925 per un dispositivo definito «anti incollaggio dei freni».

1965
Viene avviato lo sviluppo dell'impianto Abs regolato elettronicamente.

1978
Comincia la produzione per il montaggio sulla Mercedes «classe S» e successivamente sulla BMW «serie 7». È un optional molto costoso.

1986
Viene fornito il milionesimo impianto.

1987
Al dispositivo viene aggiunta la funzione Asr, ossia l'antipattinamento in accelerazione.

1991
La Bosch presenta l'«Abs/Asr 5», più leggero, compatto e meno costoso, quindi adatto anche alle vetture medie.

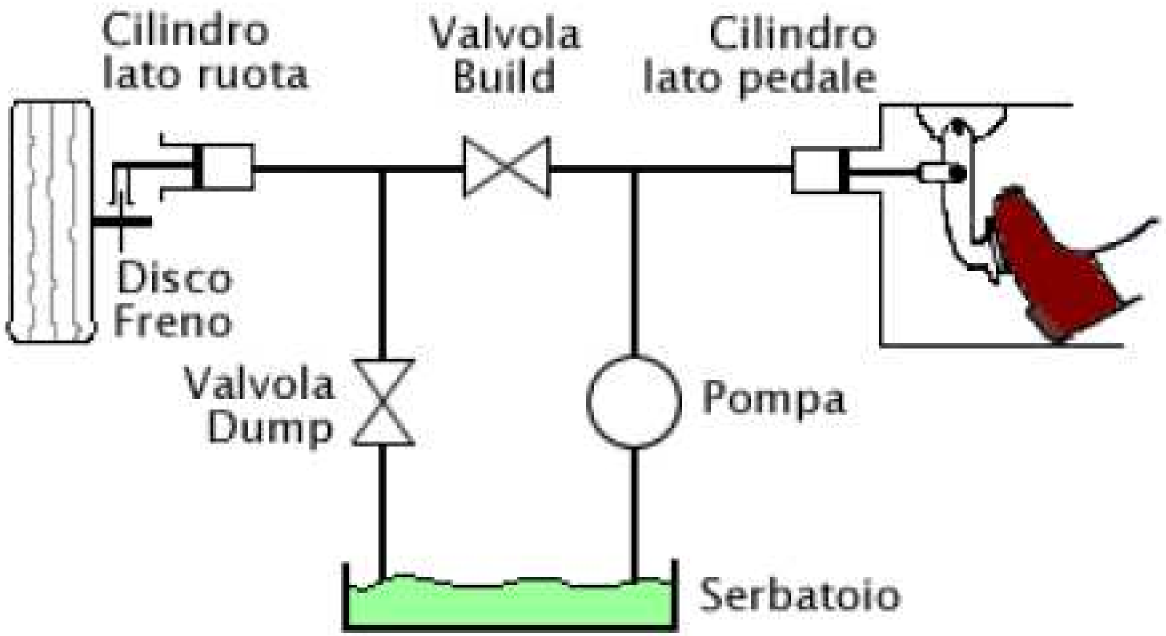
1995
Si raggiungono i 20 milioni d'impianti prodotti.

2003
La produzione Bosch tocca la soglia dei 100 milioni di pezzi.

LUGLIO 2004
Entro questa data tutte le auto nuove vendute in Europa dovranno avere di serie l'Abs, come deciso volontariamente dai Costruttori Acea.

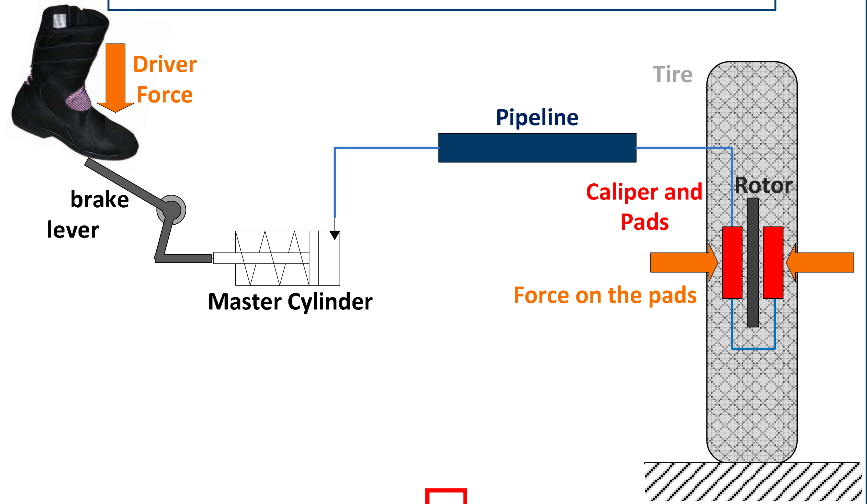
Actuators (traditional layout)

HAB: Hydraulic Actuated Brakes (Increase, hold, decrease)

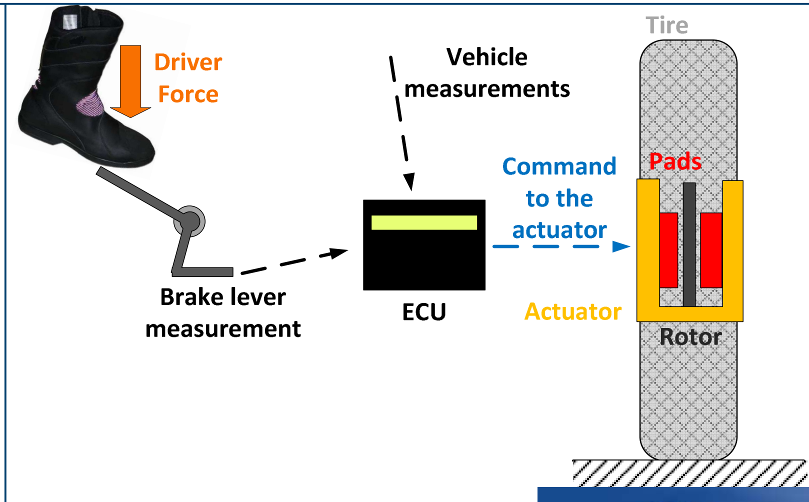


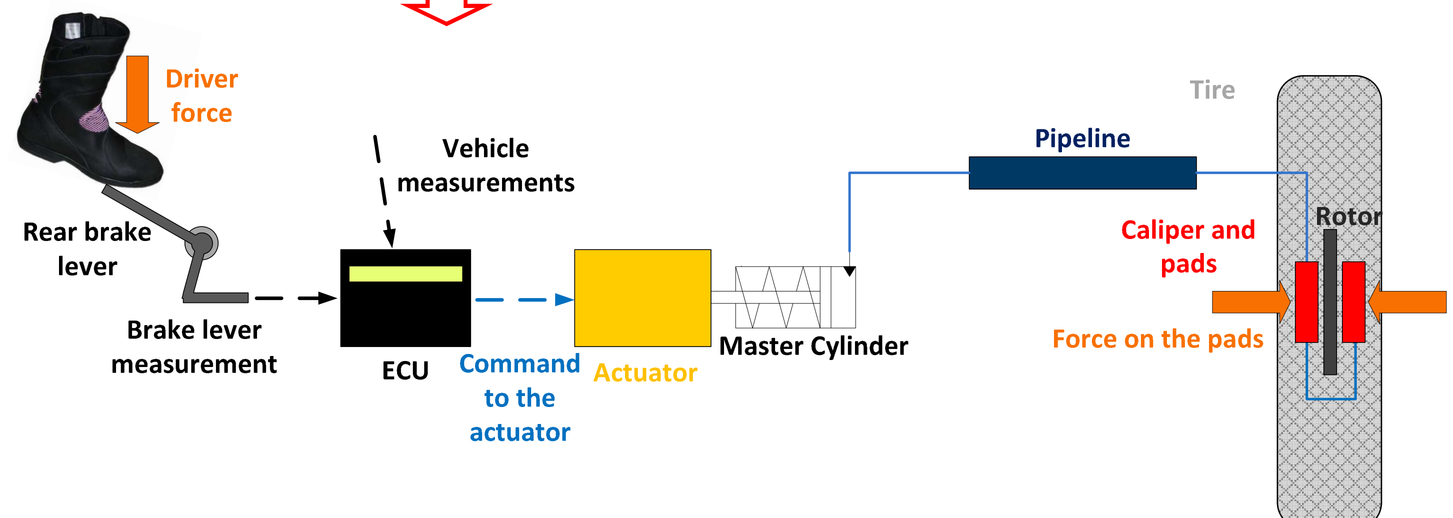
Actuation layout in brake-by-wire architectures

Traditional architecture



Pure brake-by-wire EM architecture





brake-by-wire hybrid architecture

The traditional hydraulic structure is preserved

Measurement problem: slip estimation

Control Problem: slip tracking

- with advanced BBW architecture
- with a traditional on-off-valve architecture

Problems:

- which is the best target slip?
- knowledge of friction conditions?

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1) Simplified (control-oriented) model
2) Dynamic analysis
3) Controller design

"Single-corner" model

$$\boxed{\begin{cases} J\dot{\omega} = rF_x - T_b \\ m\dot{v} = -F_x \end{cases}}$$

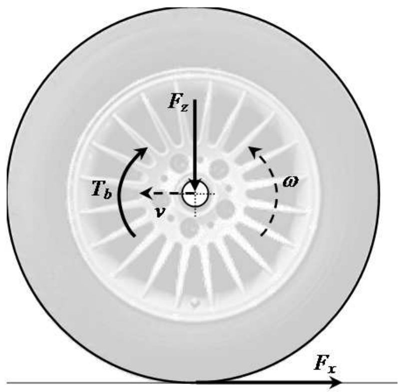
$$F_x = F_z\mu(\lambda, \cancel{\alpha;\theta_r})$$

$$\lambda = (v - \omega r)/v = 1 - r\omega/v$$

$$\begin{cases} J\dot{\omega} = rF_z\mu\left(\dfrac{v-\omega r}{v}\right) - T_b \\ m\dot{v} = -F_z\mu\left(\dfrac{v-\omega r}{v}\right) \end{cases}$$

- ω : angular speed of the wheel ([rad/s]; $\omega > 0$ is assumed);
- v : longitudinal speed of the vehicle body;
- T_b : braking torque (control/input variable);
- F_x: longitudinal road-tire contact force;
- F_z : vertical road-tire contact force;
- J, m and r are the momentum of inertia of the wheel, the quarter-car mass, and the wheel radius

(e.g. $J = 1Kg\ m^2$, $m = 225Kg$, $r = 0.28m$).



SISO system, 2nd order (n= 2), non-linear, time-invariant