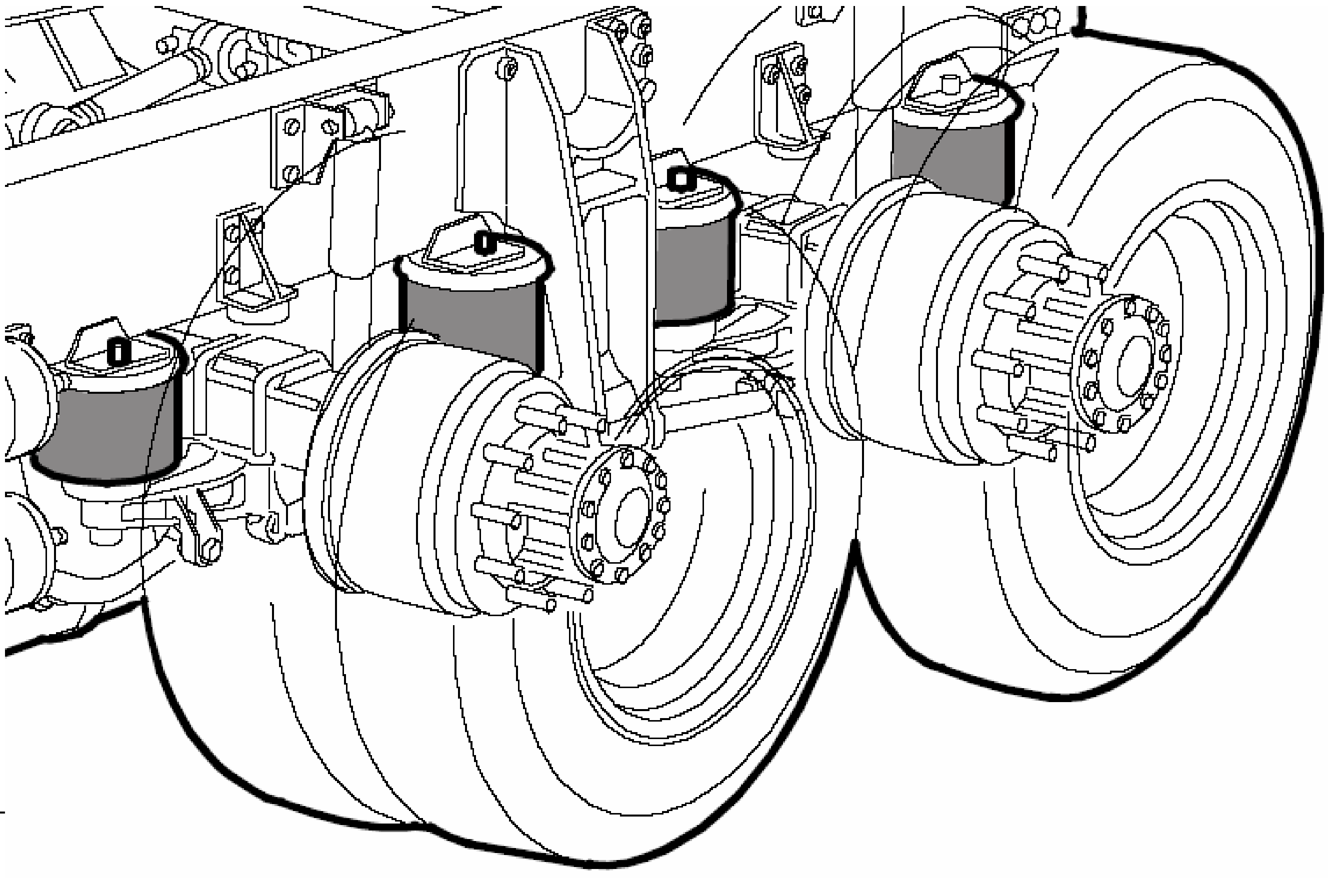
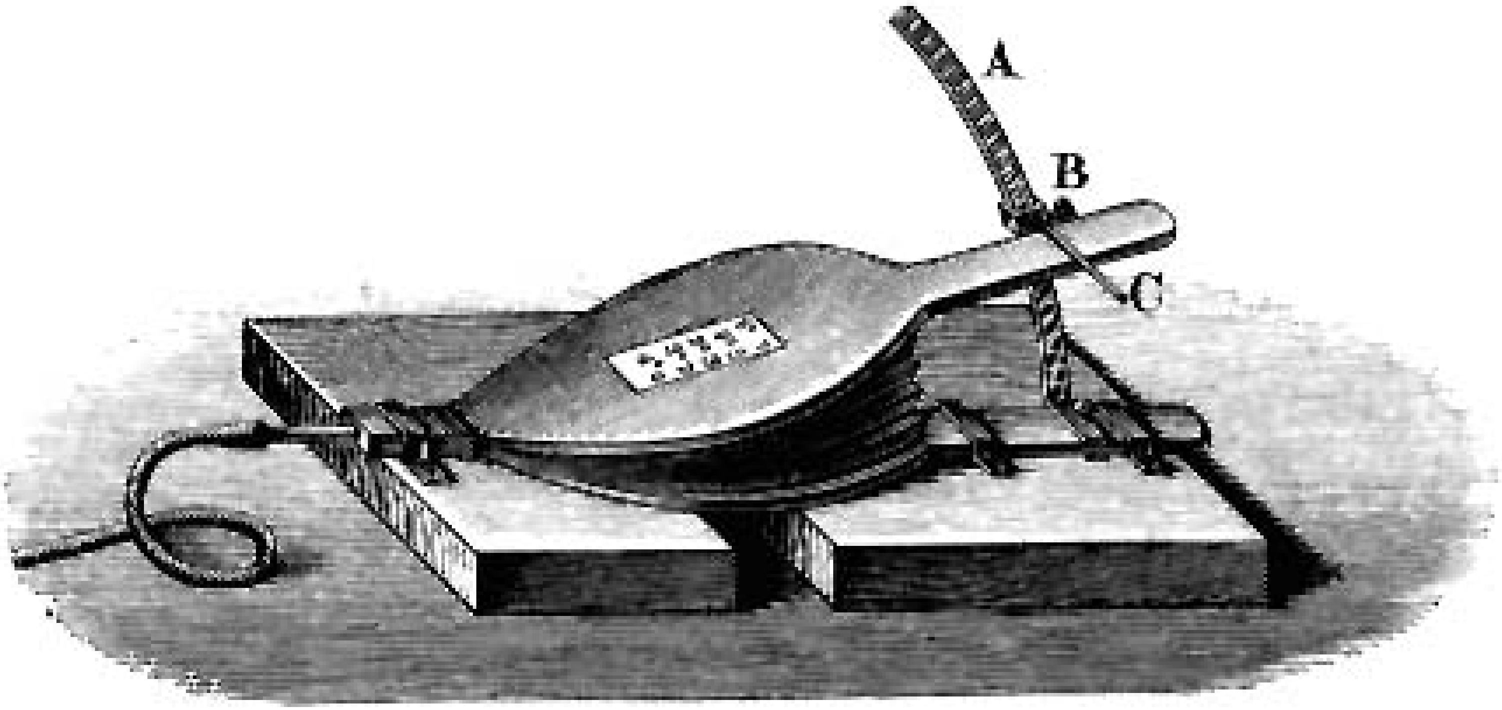
Dimensionieren und prinzipielles Verhalten von Luftfeder-/Dämpfermodulen

Univ.-Prof. Dr.-Ing. Peter Pelz

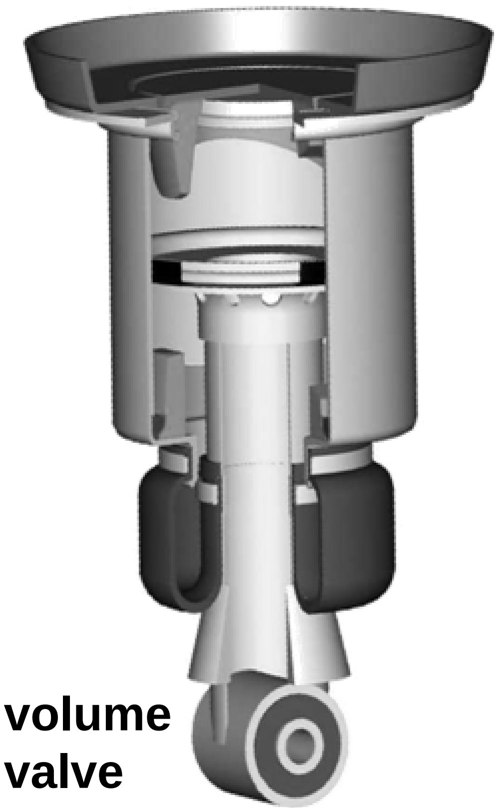


Evolution of air suspension system …



Gscheidlen 1876.
Physiologische Methodik
Braunschweig: Vieweg & Sohn

… to adaptive air spring damper



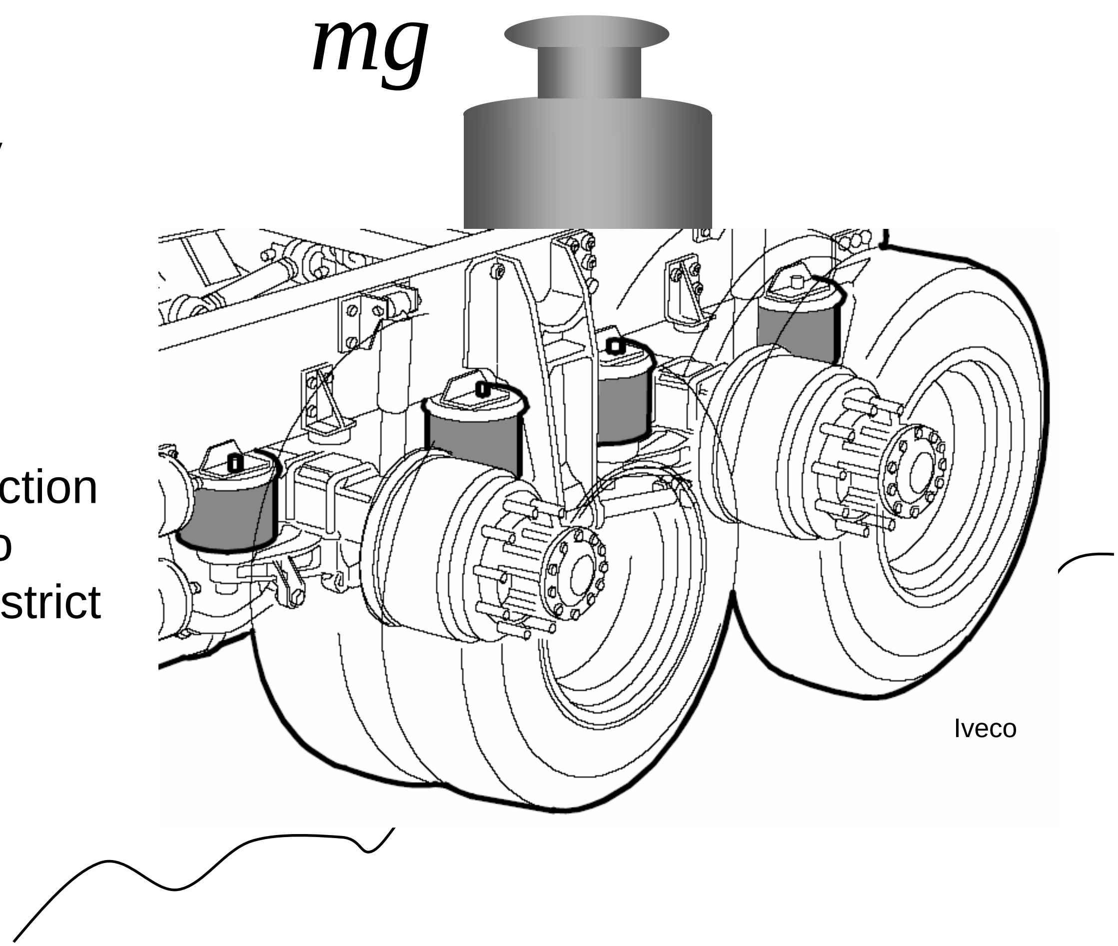
2 volume
1 valve

Vibracoustic

3 volume air spring damper
2 valve

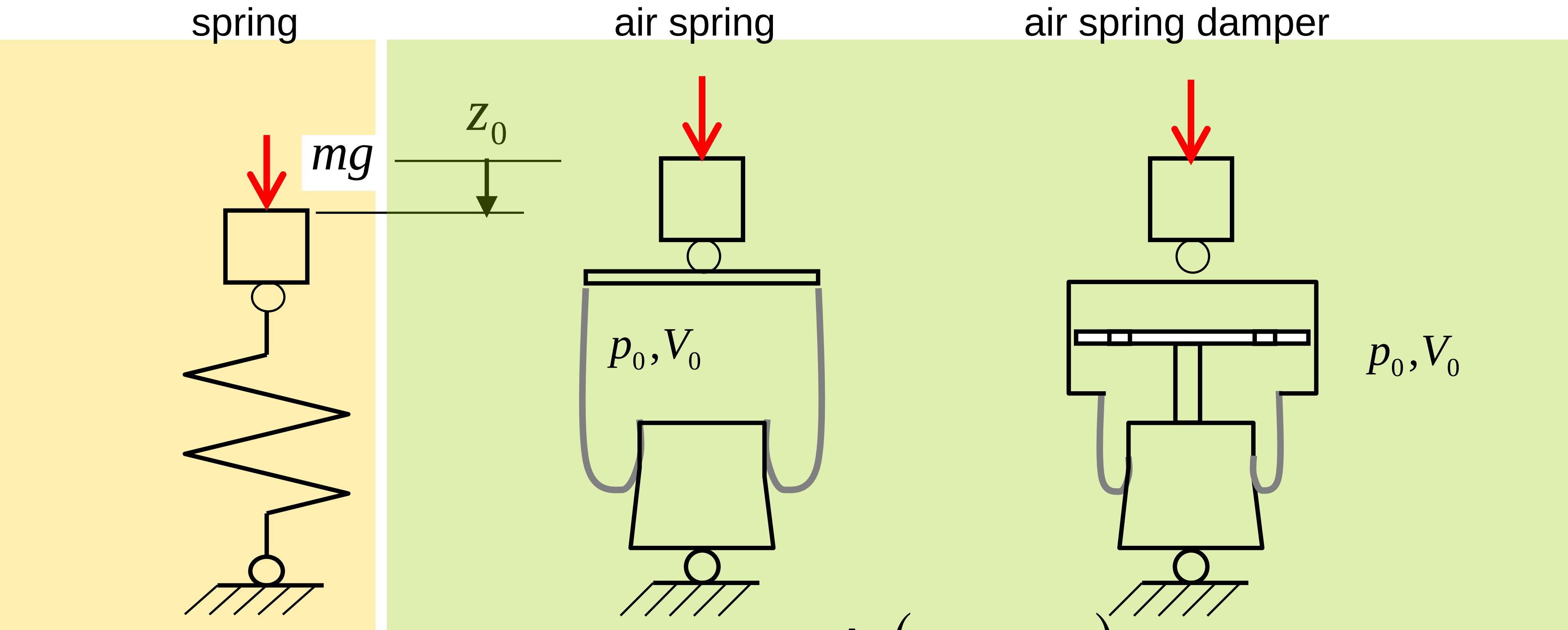
Vibracoustic

The tasks of any passive suspension system



- Carry the load!
- Store kinetic energy
  and leaving the eigenfrequency
  unchanged by loading!
- Dissipate kinetic energy
- Have as less as possible dry friction
  within the suspension system to
  reduce harshness and not to restrict
  the tuning possibilities
- Be robust, not to pricy,
  and in case easy to replace

Carry the load „the heavy mass”



$$mg = c_0 z_0$$

$$mg = A_T \left(p_0 - p_u\right)$$

no leveling

leveling

The tasks of any passive suspension system

- Carry the load!
- Store kinetic energy
  and leaving the eigenfrequency
  unchanged by loading!
- Dissipate kinetic energy
- Have as less as possible dry friction
  within the suspension system to
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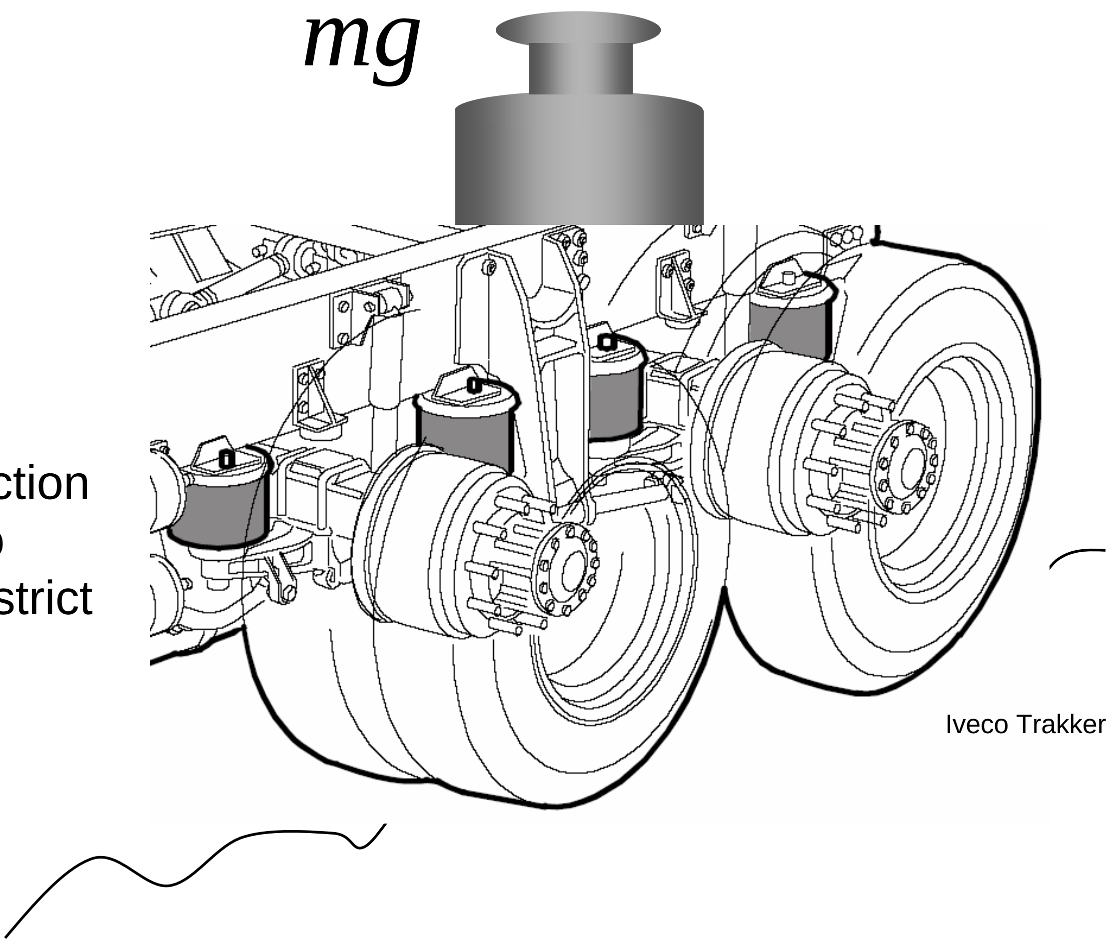

The Bridgman Postulat and the Buckingham Π Theorem

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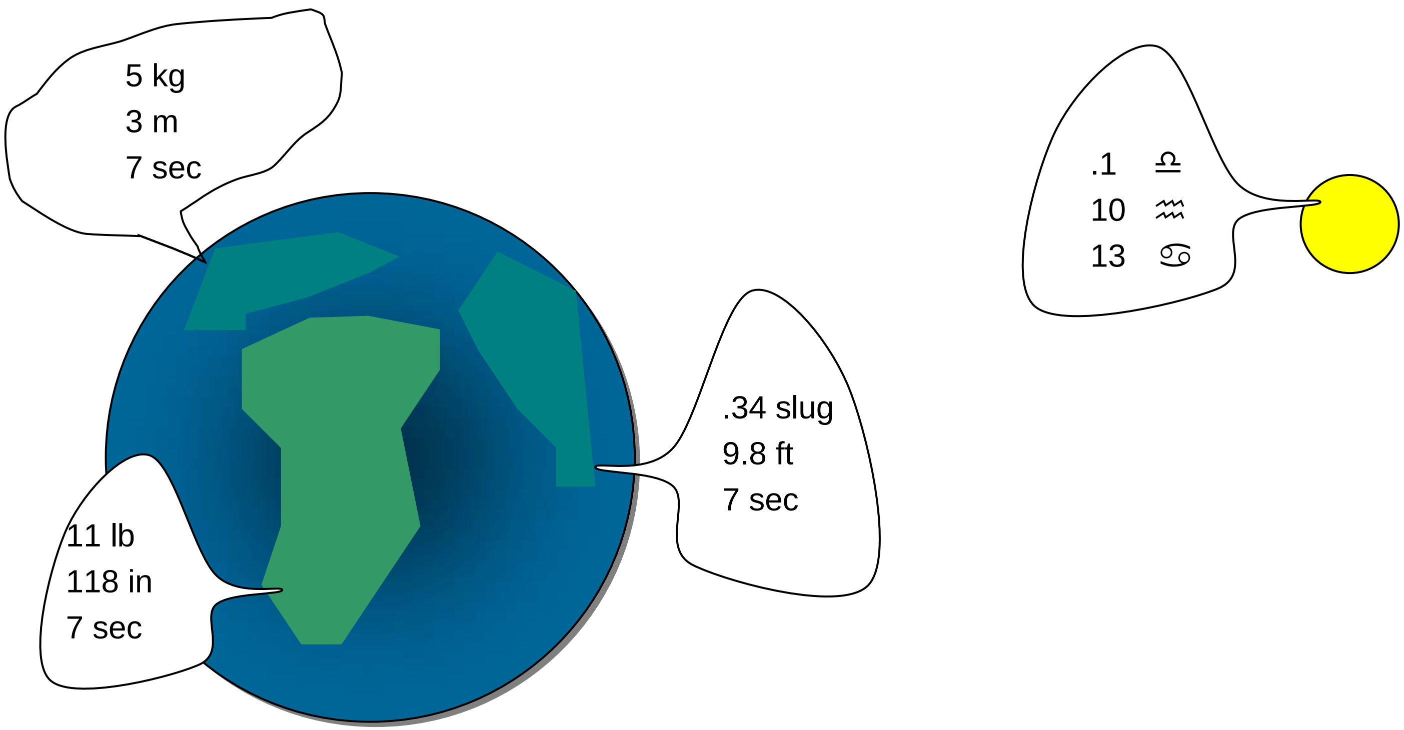
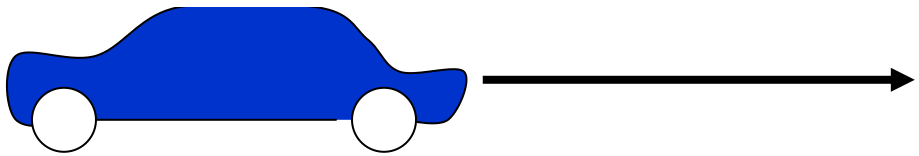
- Predefined basic physical quantities (e.q. MLT, FLT, FL, ...) and physical units (e.q. kg m s, kp m s, kp m, ...) restricts our technical-physical thinking in a sense, that one has always has to explain the definition of the quantities



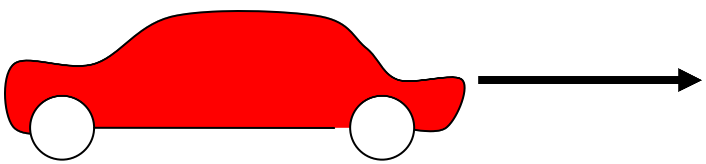
Only relativ quantities are absolut quantities (Bridgman Postulat)

- The blue car moves twice as fast as the red vehicle is a relative statement

$$72km/h = f(y_1, y_2, \ldots, y_r), \; 20m/\sec = f(c_1 y_1, c_2 y_2, \ldots, c_r y_r)$$

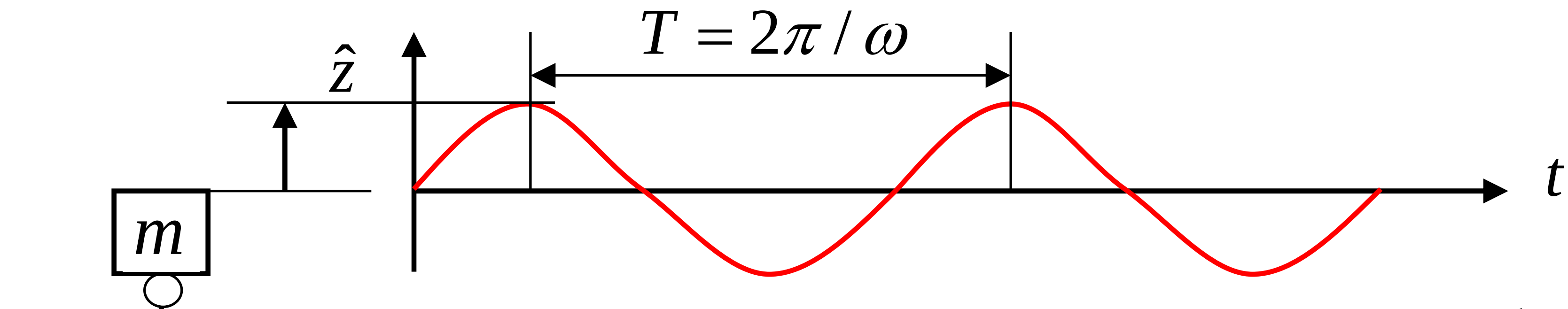


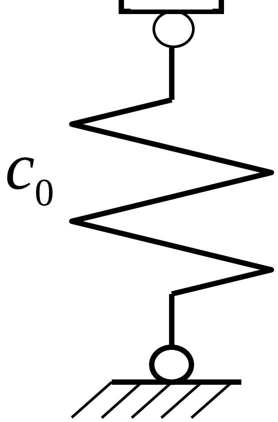
$$36km/h = f(x_1, x_2, \ldots, x_r), \; 10m/\sec = f(c_1 x_1, c_2 x_2, \ldots, c_r x_r)$$



$$\frac{x}{y} \overset{!}{=} \frac{x'}{y'} \longrightarrow \underline{\underline{x = x_1^{a_1} x_2^{a_1} \ldots x_r^{a_r}}}$$

Store kinetic energy „the inertia mass“





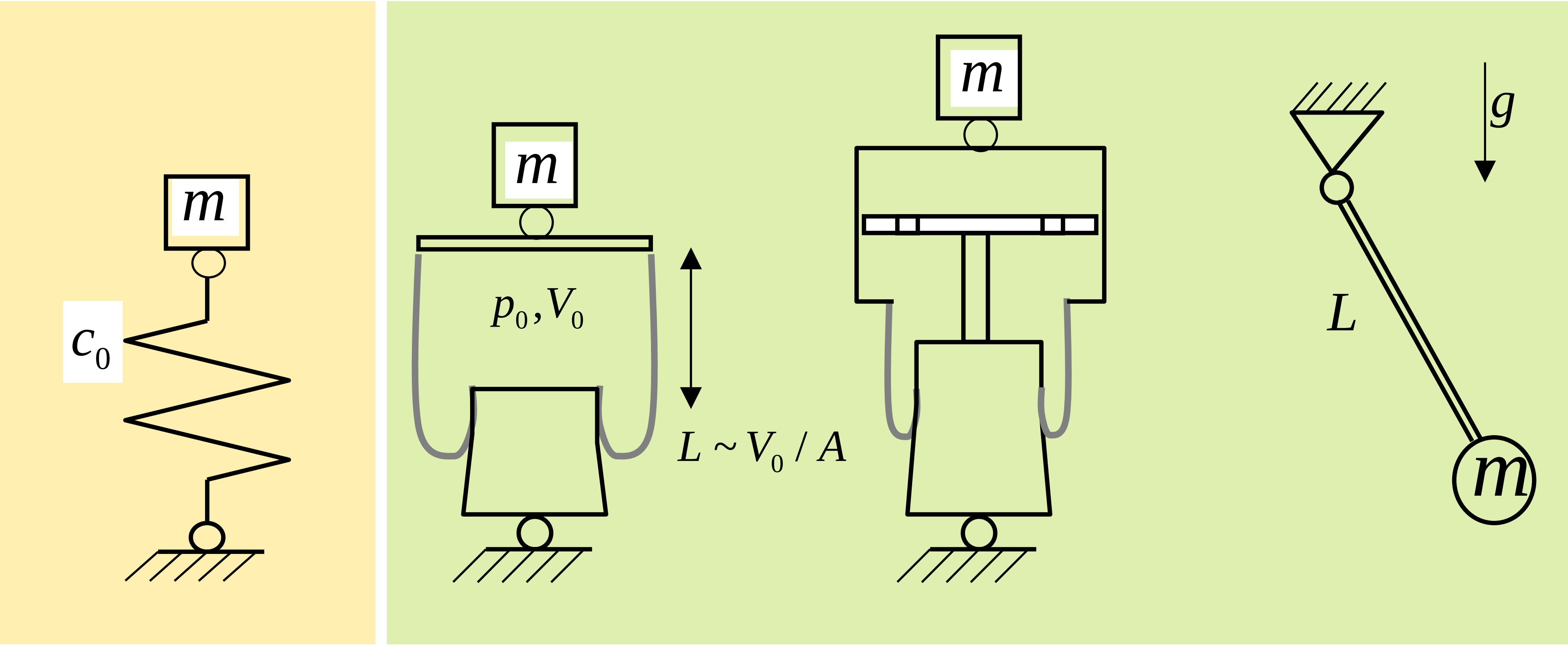
kinetic energy $K = \frac{1}{2} m \hat{z}^2 \omega^2 = 2\pi^2 m \left(\frac{\hat{z}}{T} \right)^2$

= stored potential energy $E = \frac{1}{2} c_0 \hat{z}^2$

oscillation period $T = 2\pi \sqrt{\frac{m}{c_0}}$

Store kinetic energy

spring | air spring | air spring damper | pendulum



$$T = fn(m, g, c_0)$$

$$T = fn(m, g, L, p_0(mg))$$

$$T = fn(m, g, L)$$

Store kinetic energy

spring

$$T = fn(m, g, c_0)$$

	T	m	g	c_0
L			1	
M		1		1
T	1		-2	-2

air spring

air spring damper

$$T = fn(m, g, L, p_0(mg))$$

	T	m	g	L
L			1	1
M		1		
T	1		-2	

Store kinetic energy

spring

$$\frac{T}{\sqrt{m/c_0}} = f(g, c_0) = const$$

	T	$\sqrt{\frac{m}{c_0}}$	g	c_0
L			1	
M				1
T	1	1	-2	-2

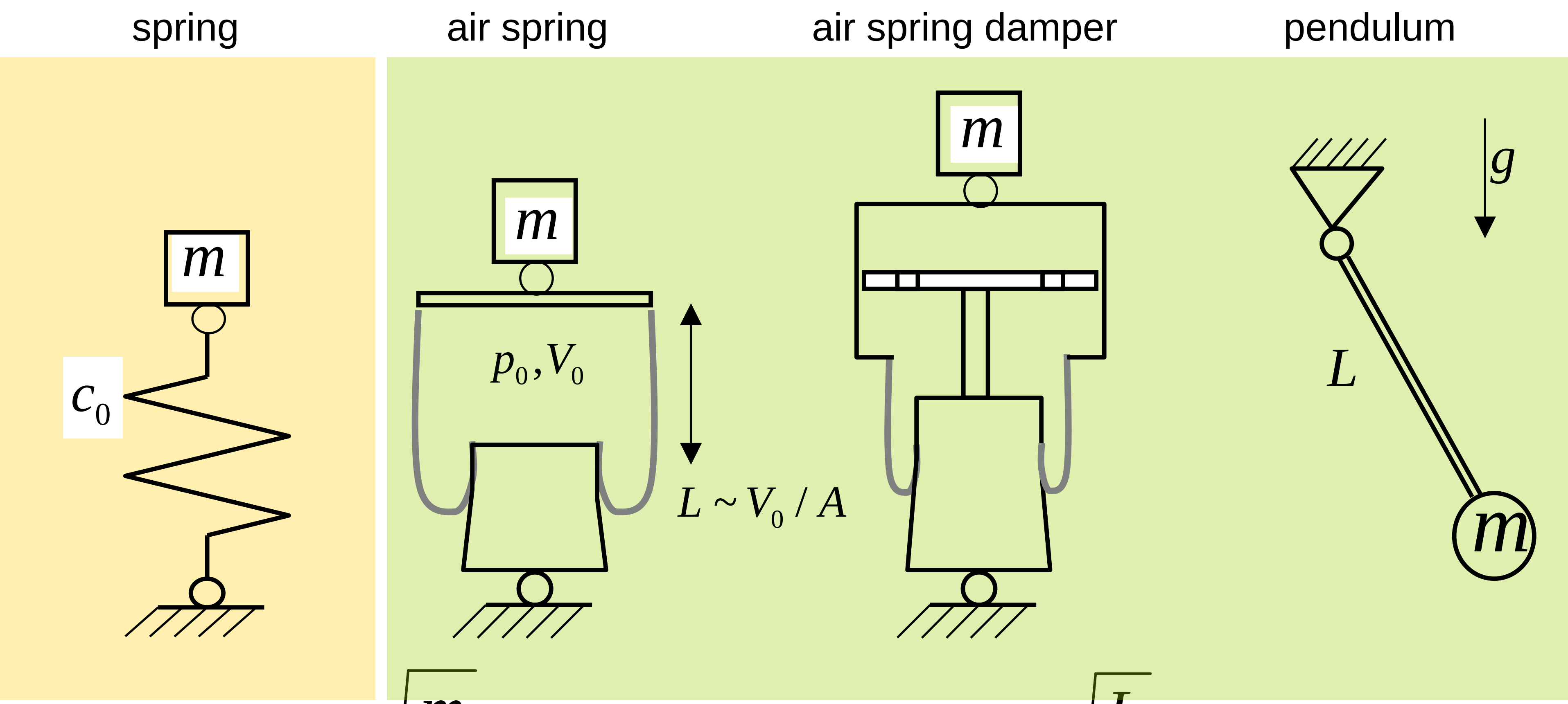
air spring

air spring damper

$$\frac{T}{\sqrt{L/g}} = f(m, L) = const$$

	T	m	$\sqrt{\frac{L}{g}}$	L
L				1
M		1		
T	1		1	

Store kinetic energy

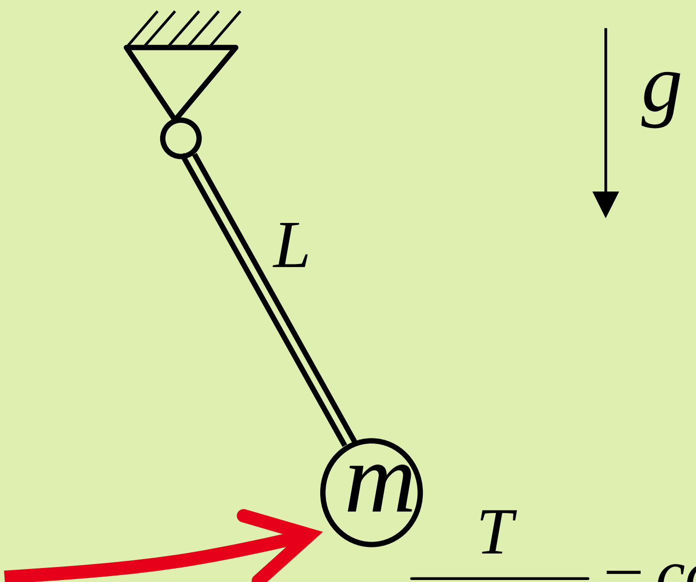


$$T = const \sqrt{\frac{m}{c_0}}$$

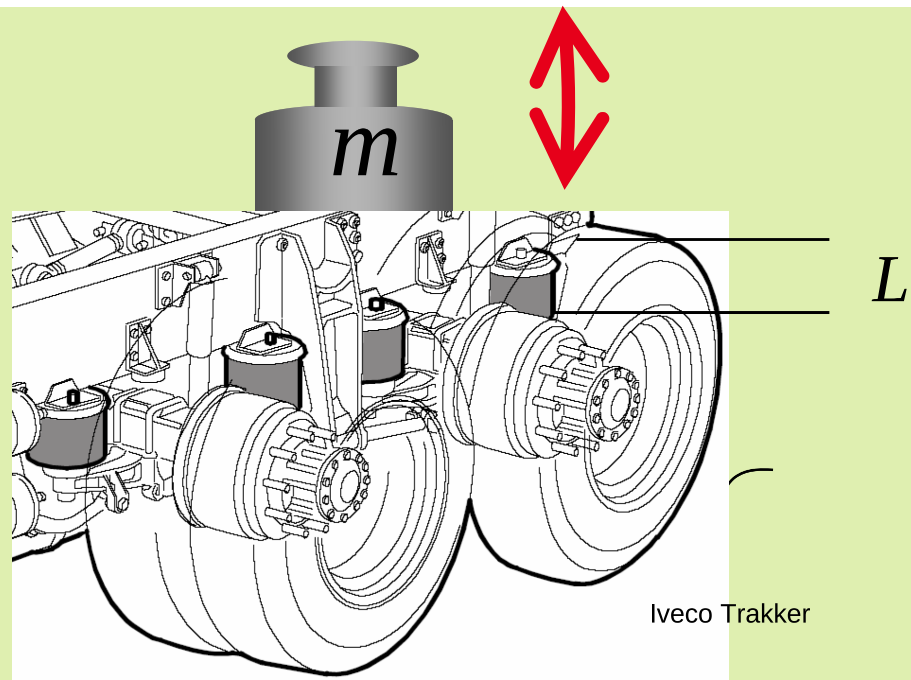
$$T = const \sqrt{\frac{L}{g}}$$

Store kinetic energy

- An air suspension system is equal to a pendulum
- Since the heavy mass is equal to the inertial mass (equivalence principle) the eigenfrequency is independent of the mass/ load.



$$\frac{T}{\sqrt{L/g}} = const$$



The typical length

the adiabatic stiffness of the air suspension system

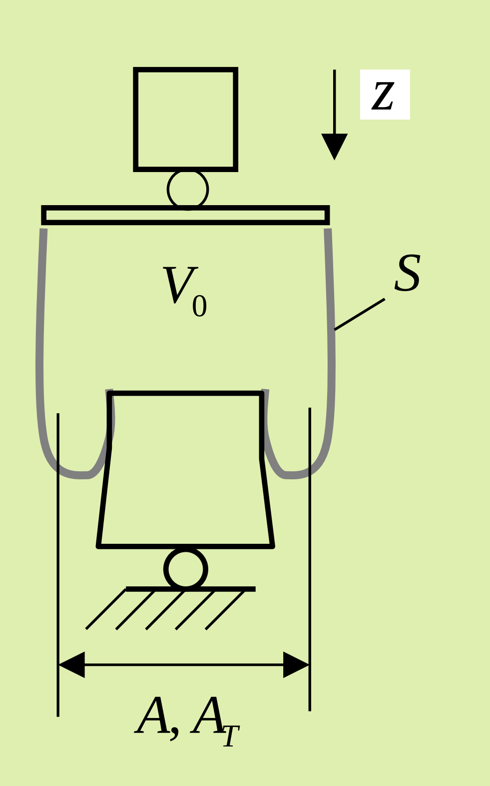
$$c := \frac{dF}{dz} = \frac{d}{dz}\left[(p-p_u)A_T\right] = \frac{dV}{dz}\frac{dp}{dV}A_T + (p-p_u)\frac{dA_T}{dz}$$

for $f >> f_\gamma \sim \left(\frac{S}{V_0}\right)^2 \frac{\lambda}{\rho_0 c_p} \sim 0.001...0.01 Hz$ $\quad p = p_0 (V_0/V)^\gamma$

for $f << f_\gamma$: $\quad p = p_0 (V_0/V)$

$$c_0 = -\gamma p_0 A_T \frac{1}{V_0}\frac{dV}{dz} + (p_0 - p_u)\frac{dA_T}{dz}$$

„volume part“ „piston part“



The typical length

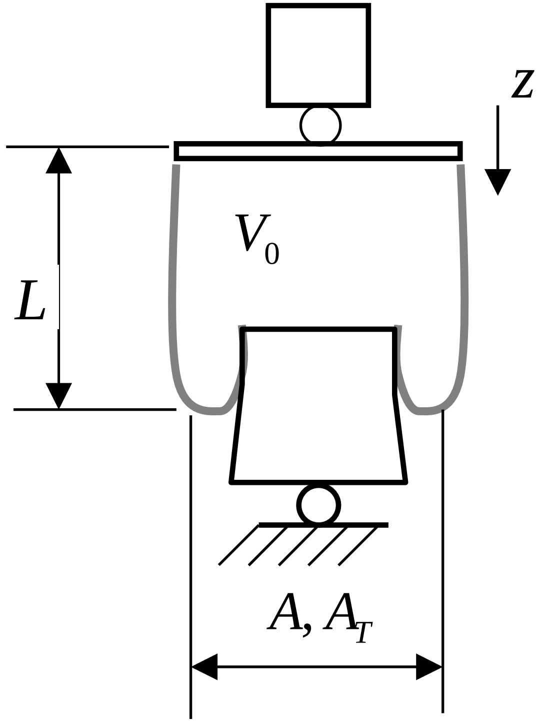
For $p_0 >> p_u$: $mg = (p_0 - p_u) A_T \approx p_0 A_T$

it follows

$$c_0 = \frac{mg}{L}\left(\gamma + \frac{1}{A_T}\frac{dA_T}{d(z/L)}\right)$$

$$L := -V_0 \frac{dz}{dV} = \frac{V_0}{A}$$

$$T = 2\pi \frac{\sqrt{L/g}}{\sqrt{\gamma + \frac{1}{A_T}\frac{dA_T}{d(z/L)}}}$$



The typical length for $p_0 >> p_u$ and $\frac{1}{A_T}\frac{dA_T}{d(z/L)} \equiv 0$

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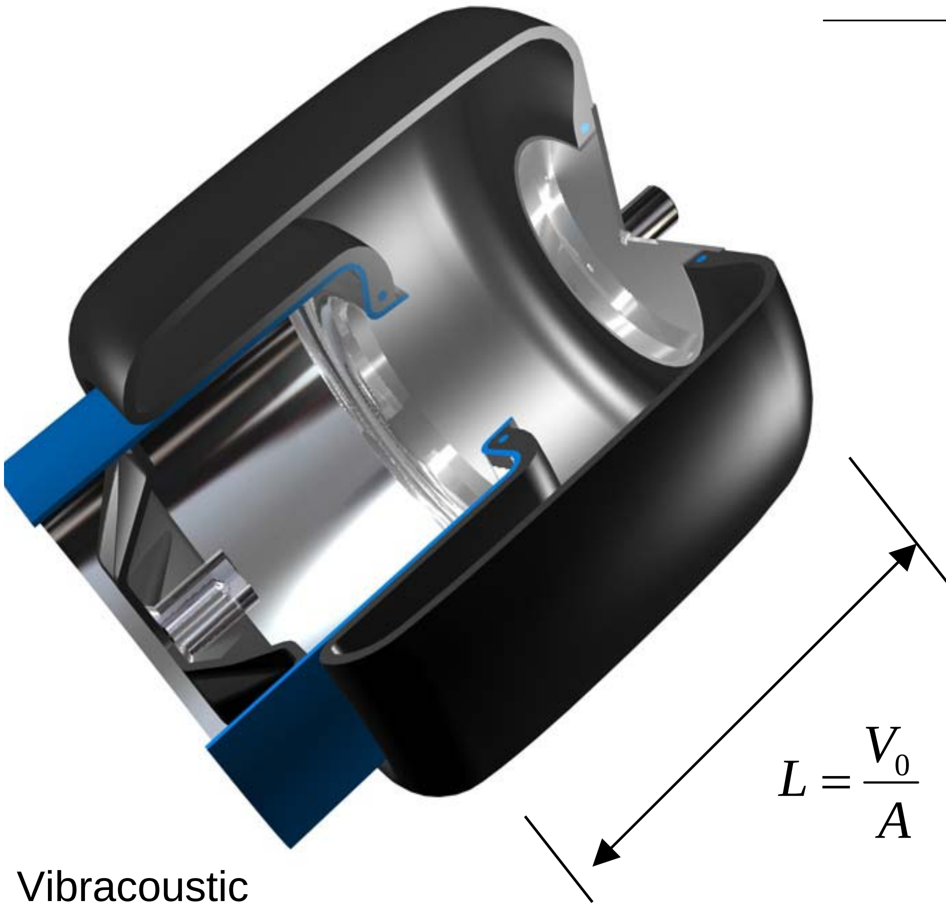
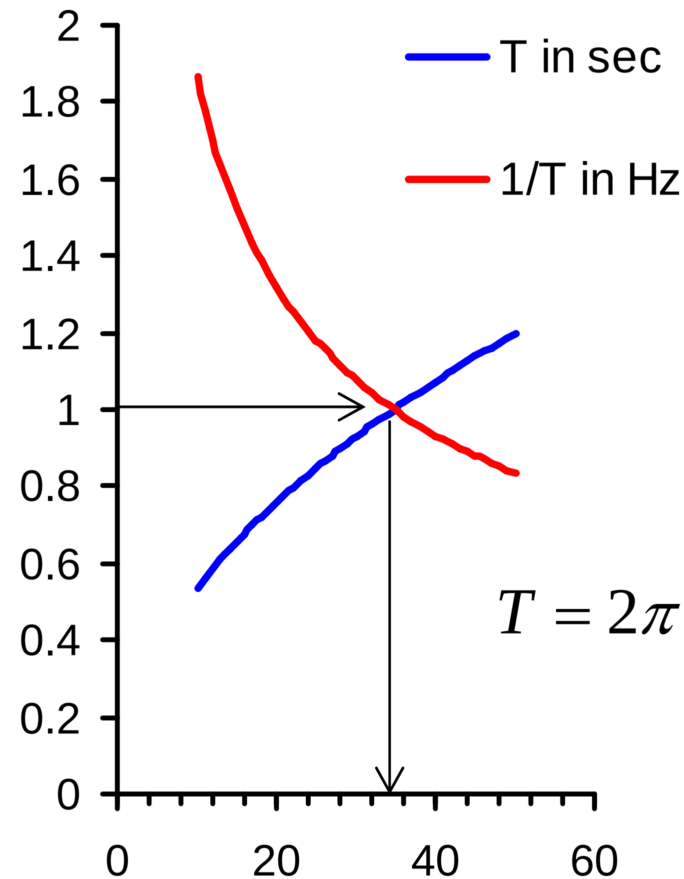


$$L = \frac{V_0}{A}$$

Vibracoustic



$$T = 2\pi\sqrt{\frac{L}{\gamma g}}$$

$L = \frac{V_0}{A}$ in cm

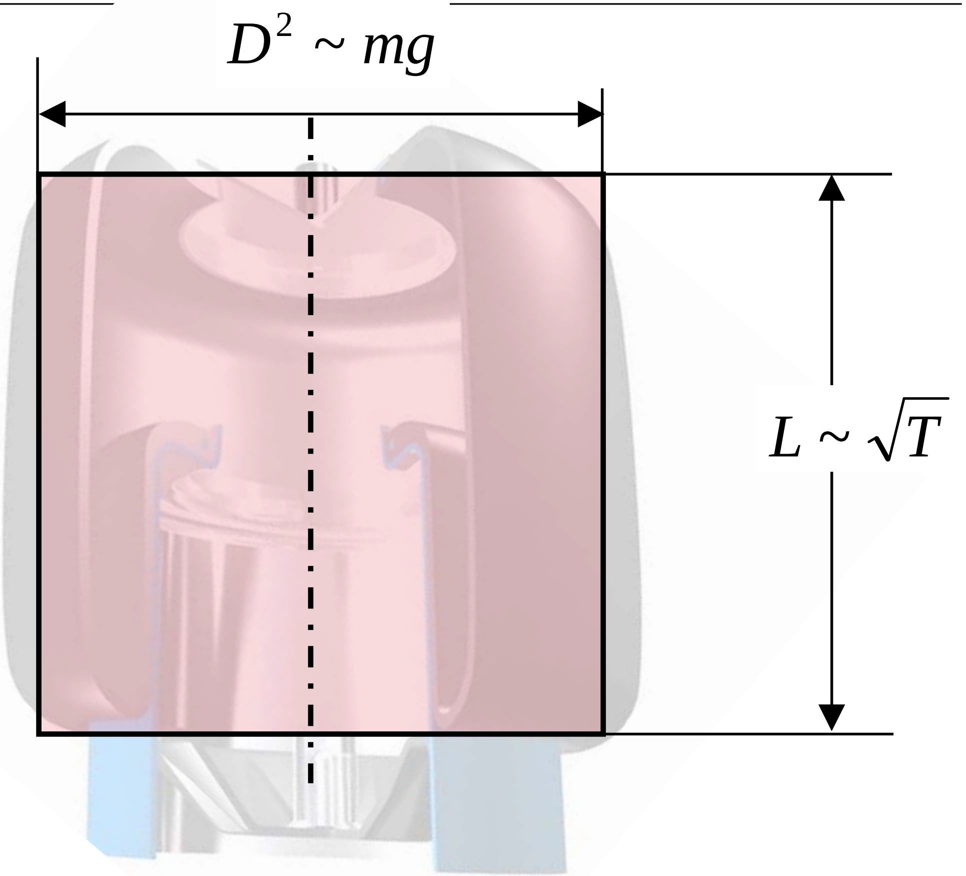
Advantage of an air suspension system: Seperation of function

- Carry the load at a given pressure

  ➔ diameter $D^2 \sim mg$

- Store the kinetic energy and leave the eigenfrequency unchanged

  ➔ height $L \sim \sqrt{T}$



for $p_0 >> p_u$ and $\frac{1}{A_T}\frac{dA_T}{d(z/L)} \equiv 0$

The tasks of any passive suspension system

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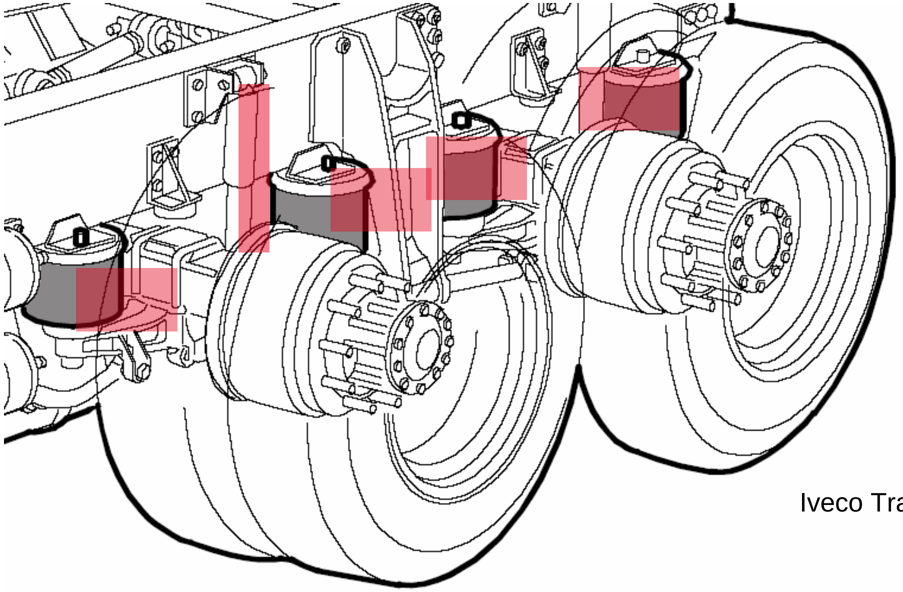
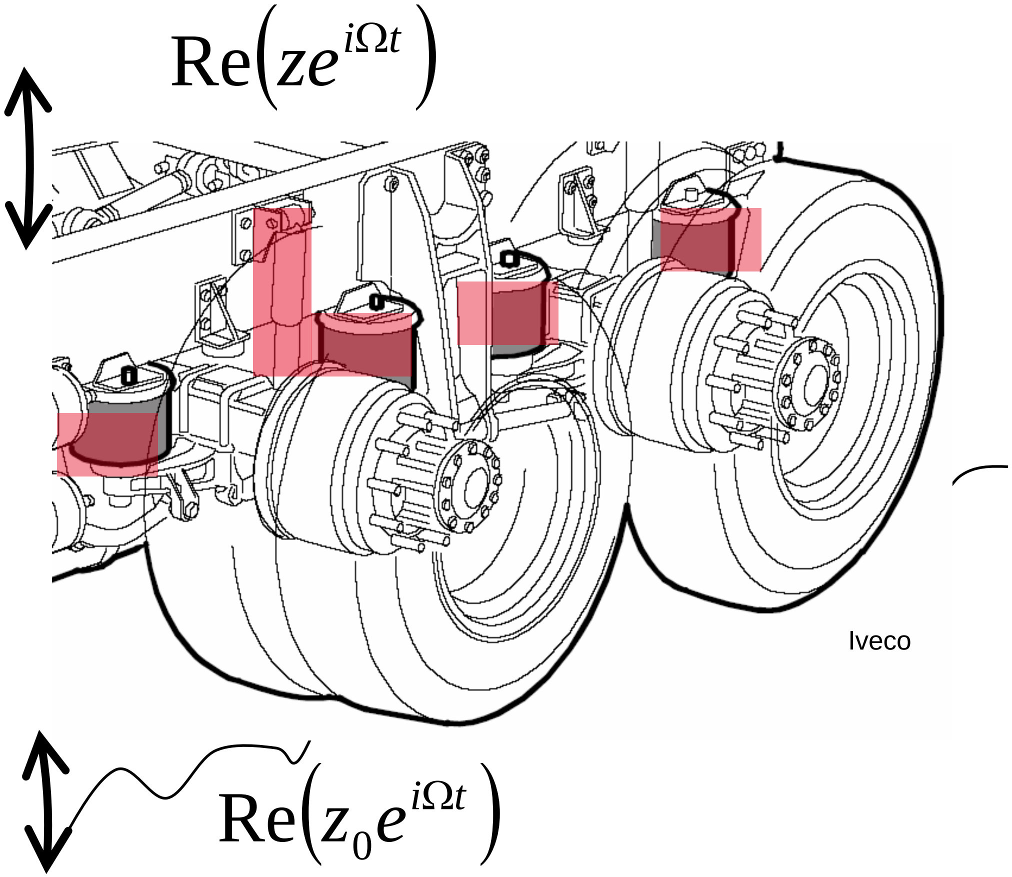
- Carry the load! ✓
- Store kinetic energy
  and leaving the eigenfrequency
  unchanged by loading! ✓
- Dissipate kinetic energy and keep the acceleration as low as possible no matter the load is!
- Have as less as possible dry friction within the suspension system to reduce harshness and not to restrict the tuning possibilities
- Be robust, not to pricy,
  and in case easy to replace

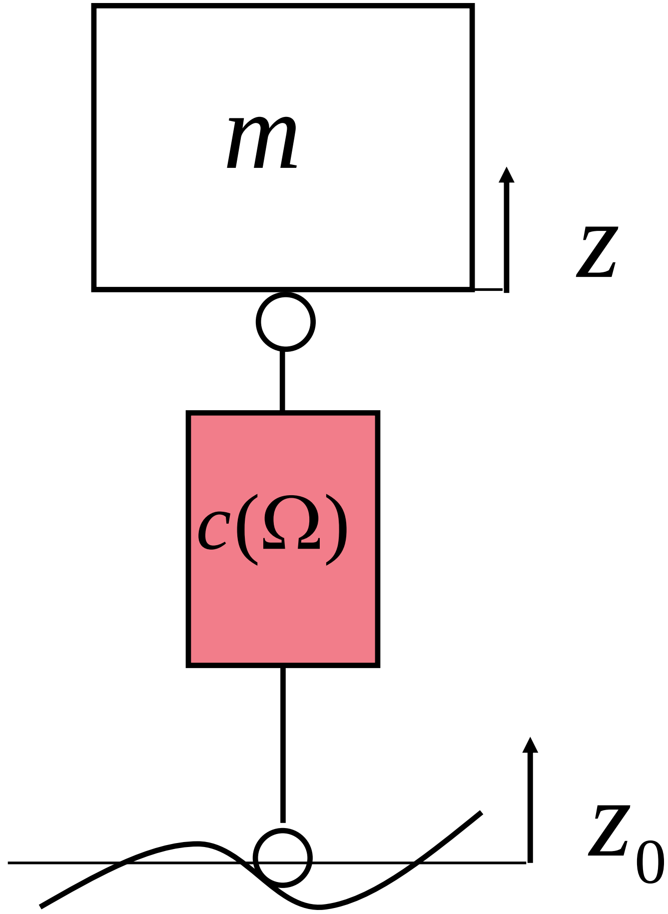


Iveco Trakker

FLUID SYSTEM TECHNIK

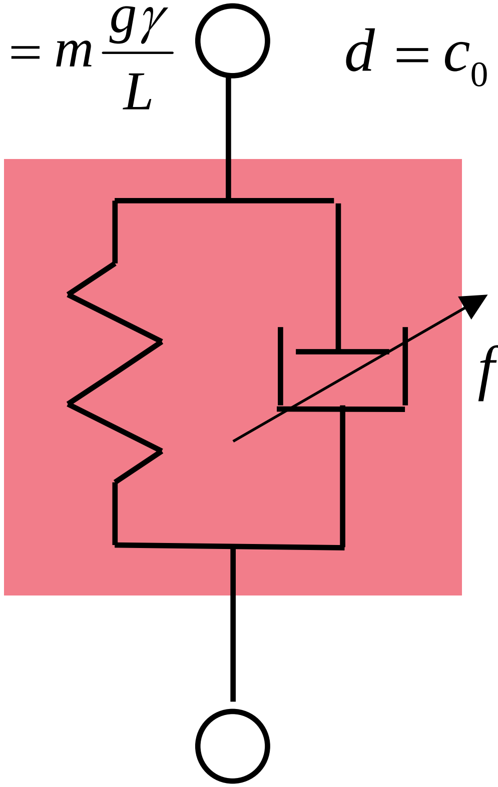
Damping


$\mathrm{Re}\left(z e^{i\Omega t}\right)$
Iveco
$\mathrm{Re}\left(z_0 e^{i\Omega t}\right)$

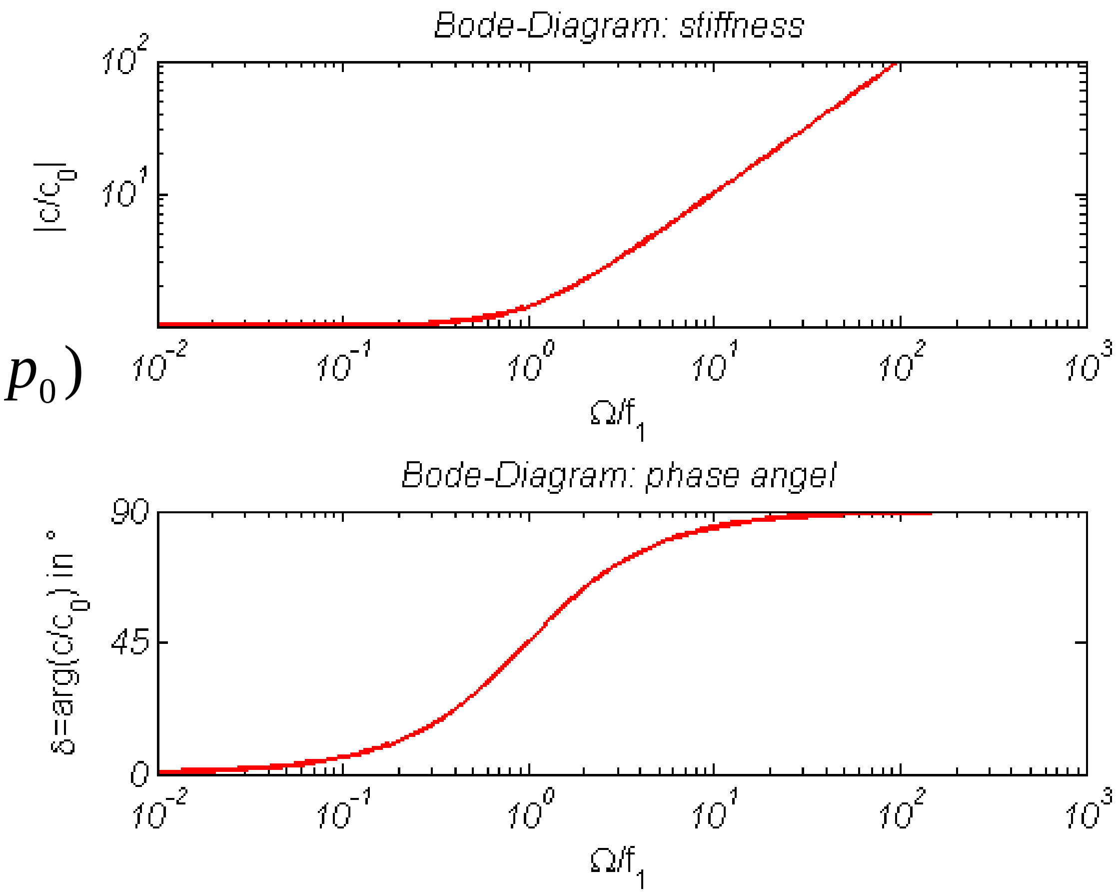

m
z
$c(\Omega)$
z_0

Bode diagram for (linear) hydraulic suspension system

$$c_0 = m\frac{g\gamma}{L} \qquad d = c_0 f_0$$



$$f_0 = fn(p_0)$$

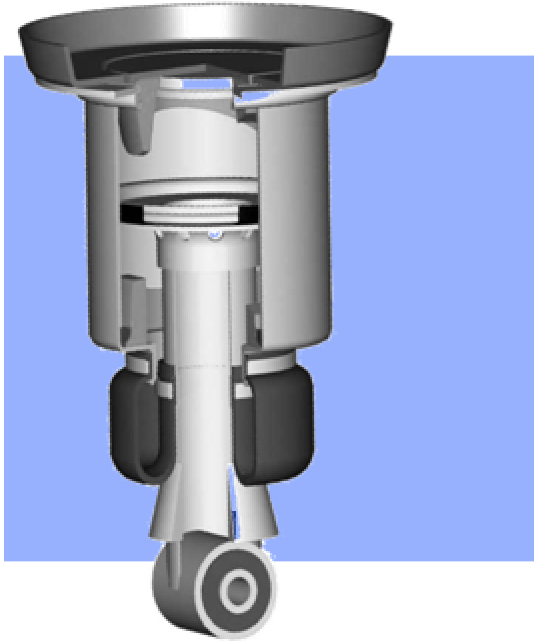


$$c(\Omega) = c_0 + id\Omega$$

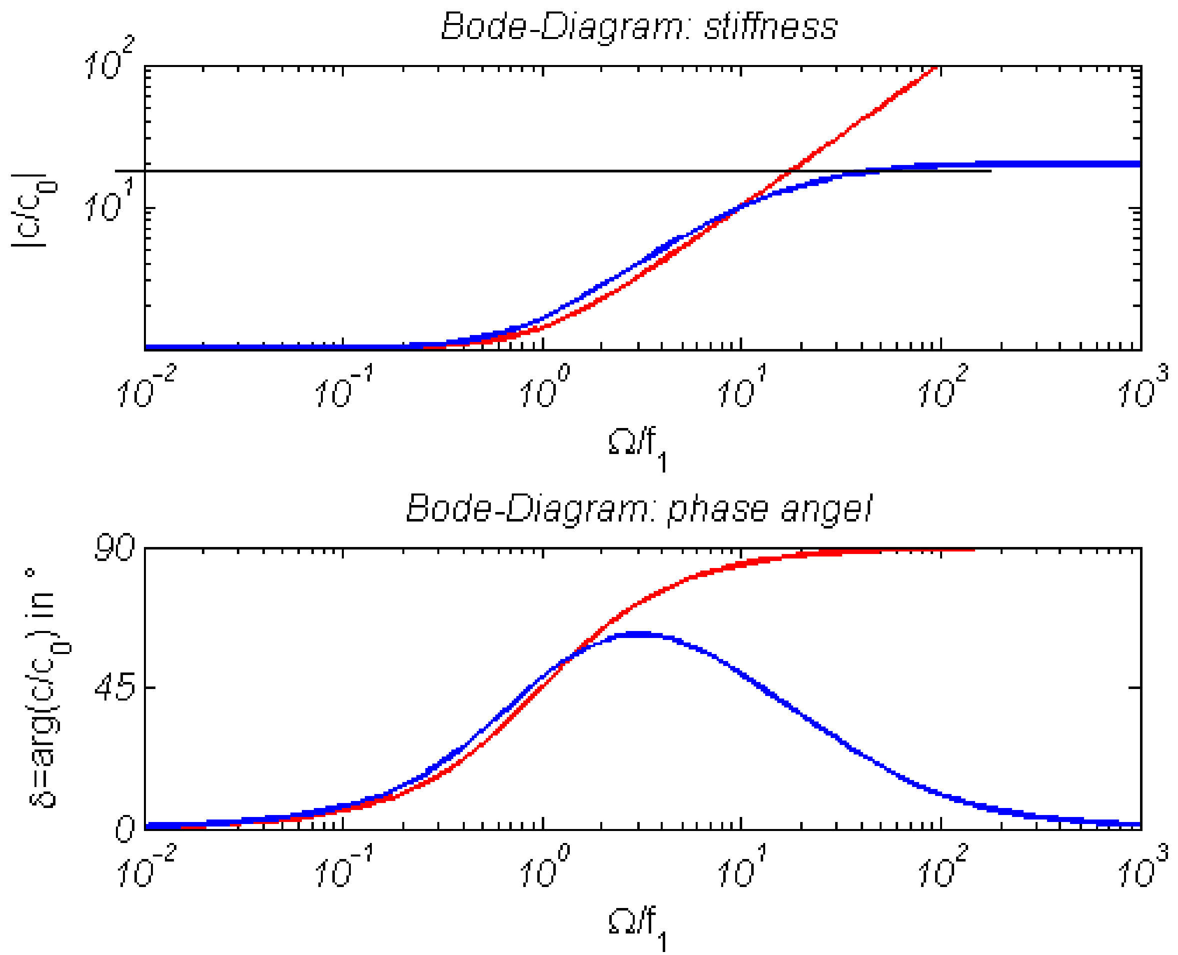
$$\bar{c}(\Omega) = 1 + i\Omega / f_0$$

tuning frequency = cut off frequency f_0

Bode diagram for (linear) air damping system



$$\bar{c} = \frac{1 + i\Omega / f_0}{1 + \frac{i\Omega}{f_0}\frac{1}{\bar{c}_\infty}}$$

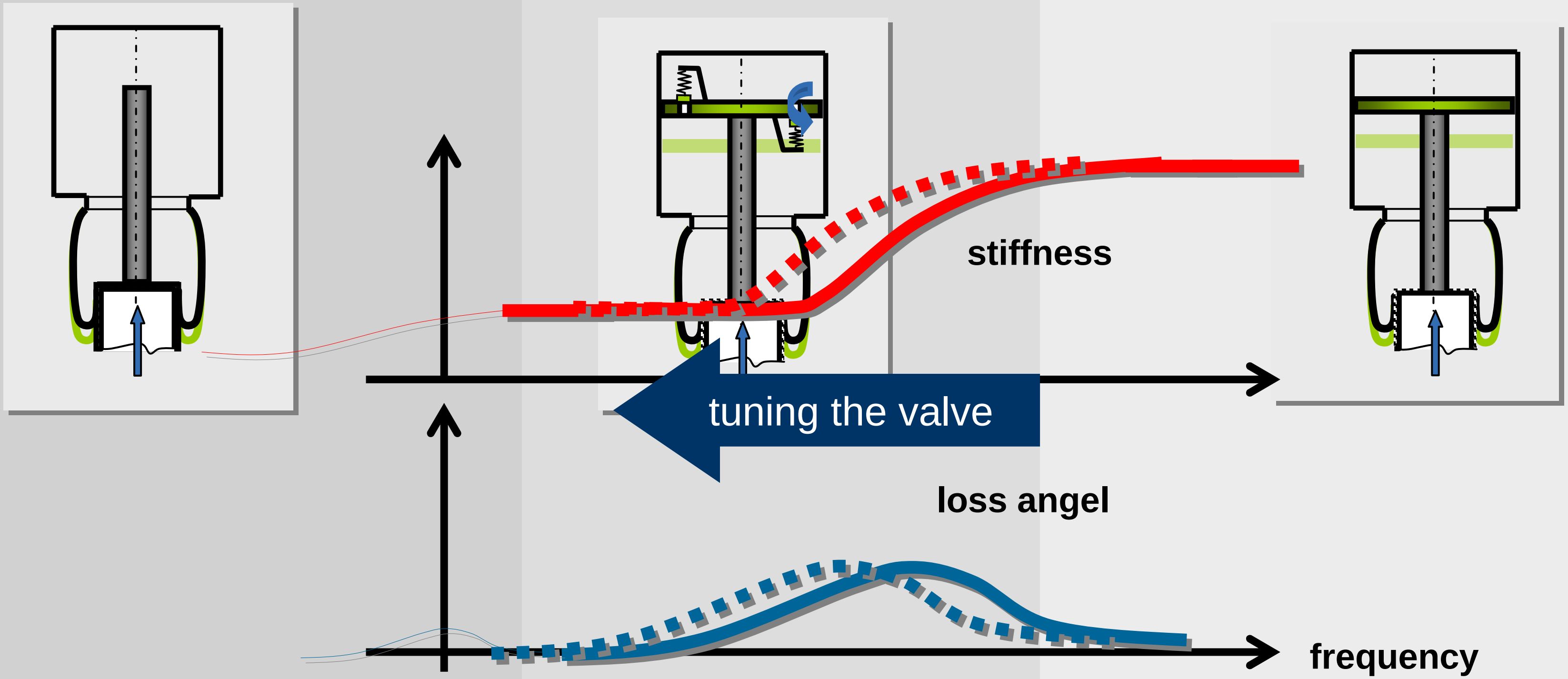


Air damping is easy to understand

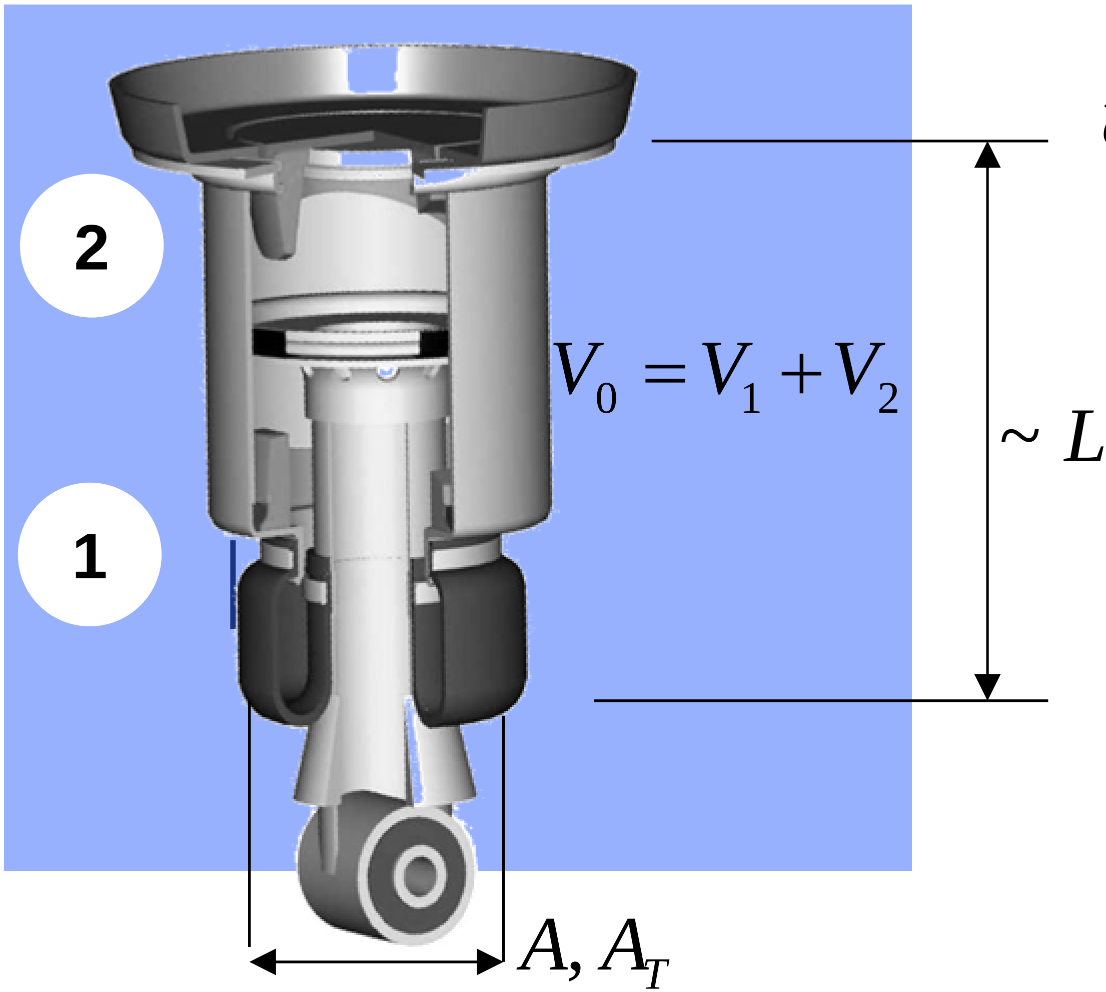
$f << f_0$
- No piston
- No dissipation
- Soft air spring
  V1 + V2

$f \sim f_0$
- dissipation

$f >> f_0$
- No valve
- Stiff air spring
  V1, V2 in parallel



The stiffness ratio is a question of geometry



$$\bar{c}_\infty := \frac{c_\infty}{c_0} = \frac{1-\nu(1-\alpha^2)}{\nu(1-\nu)(1-\alpha)^2} = $$

$$= fn(\kappa_i) = 4..20$$

$$\nu := \frac{V_2}{V_0}, \alpha := \frac{A_1}{A_2}$$

Why air damping?

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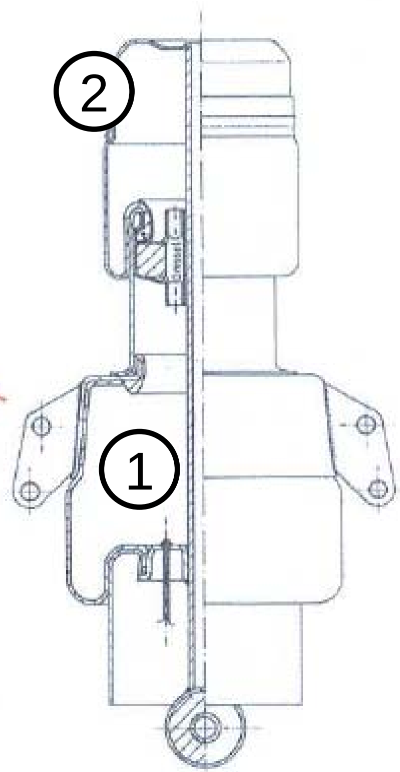
UNICAR IAA 1981



- Self adapting of stiffness and damping

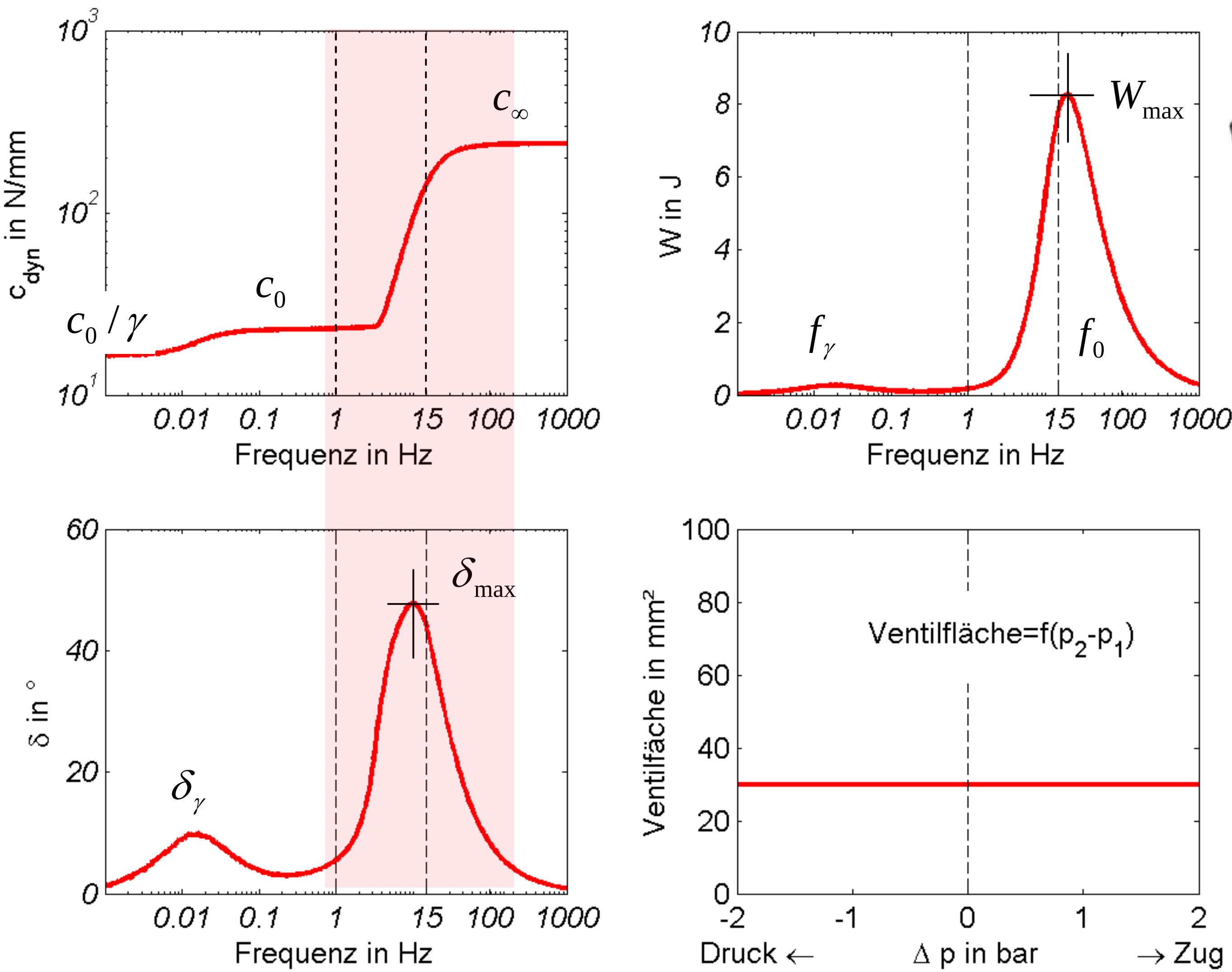
$$W \sim p_0 \sim mg / A_T + p_u$$

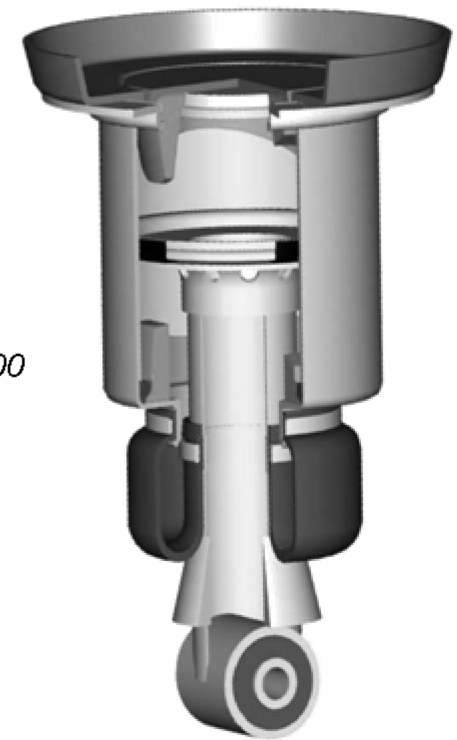
$$c \sim p_0 \sim mg / A_T + p_u$$

- Design solutions show small dry friction
- Frequency dependant damping

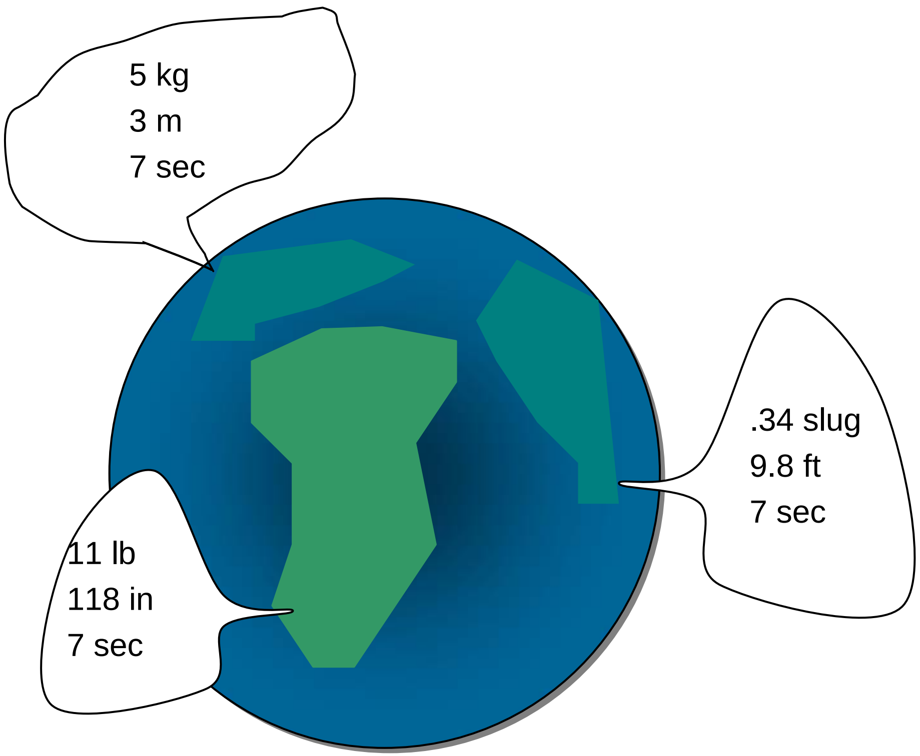
FLUID SYSTEM TECHNIK

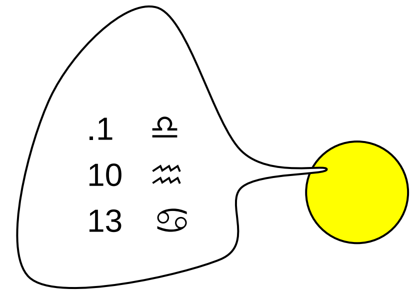
Tranfer behavior of an air spring damper

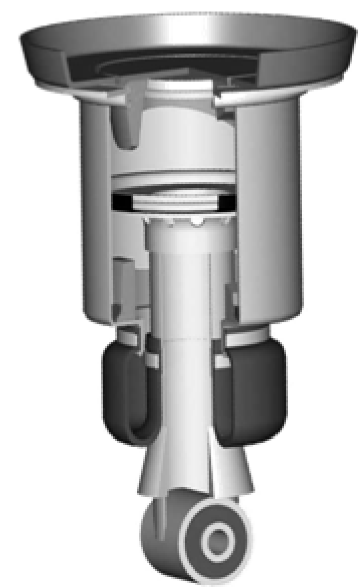




The max. dissipation follows out of a most general dimensional analysis





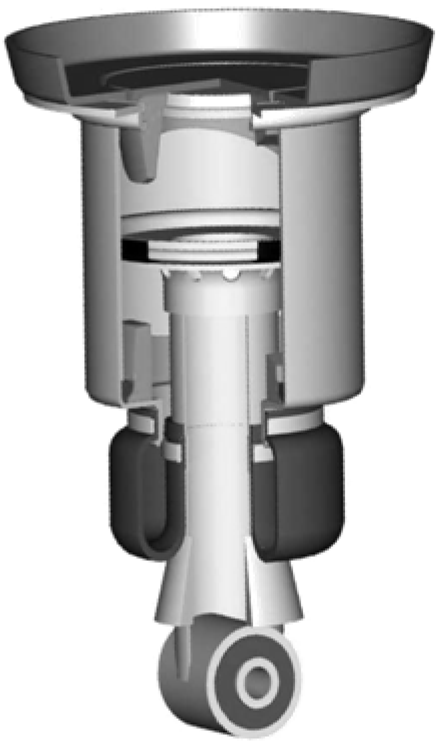


$$W_{\max} = fn(p_0, T_0, R, \gamma, L, A_V, \hat{z}, \kappa_i)$$

The dissipation per oszillation

$$W_{max} = fn(p_0, T_0, R, \gamma, L, A_V, \hat{z}, \kappa_i)$$

$$\frac{W_{max}}{p_0 L^3} = fn\left(\frac{\hat{z}}{L}, \gamma, \kappa_i\right) \qquad for \qquad A_V / L^2 << 1$$

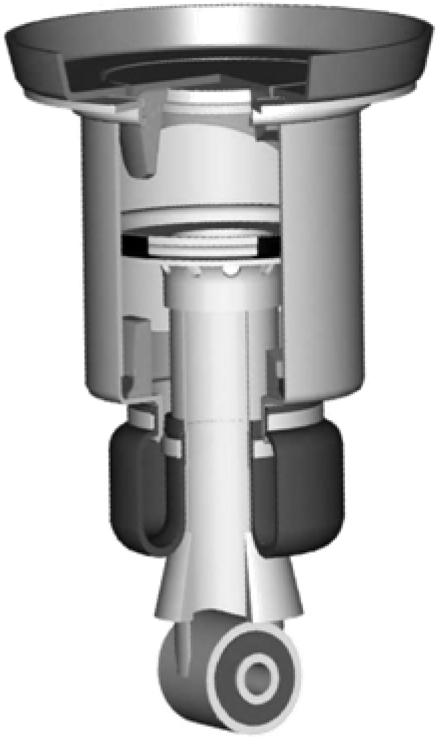


From experiments it follows further

$$\frac{W_{max}}{p_0 L^3} = \left(\frac{\hat{z}}{L}\right)^2 fn(\gamma, \kappa_i)$$

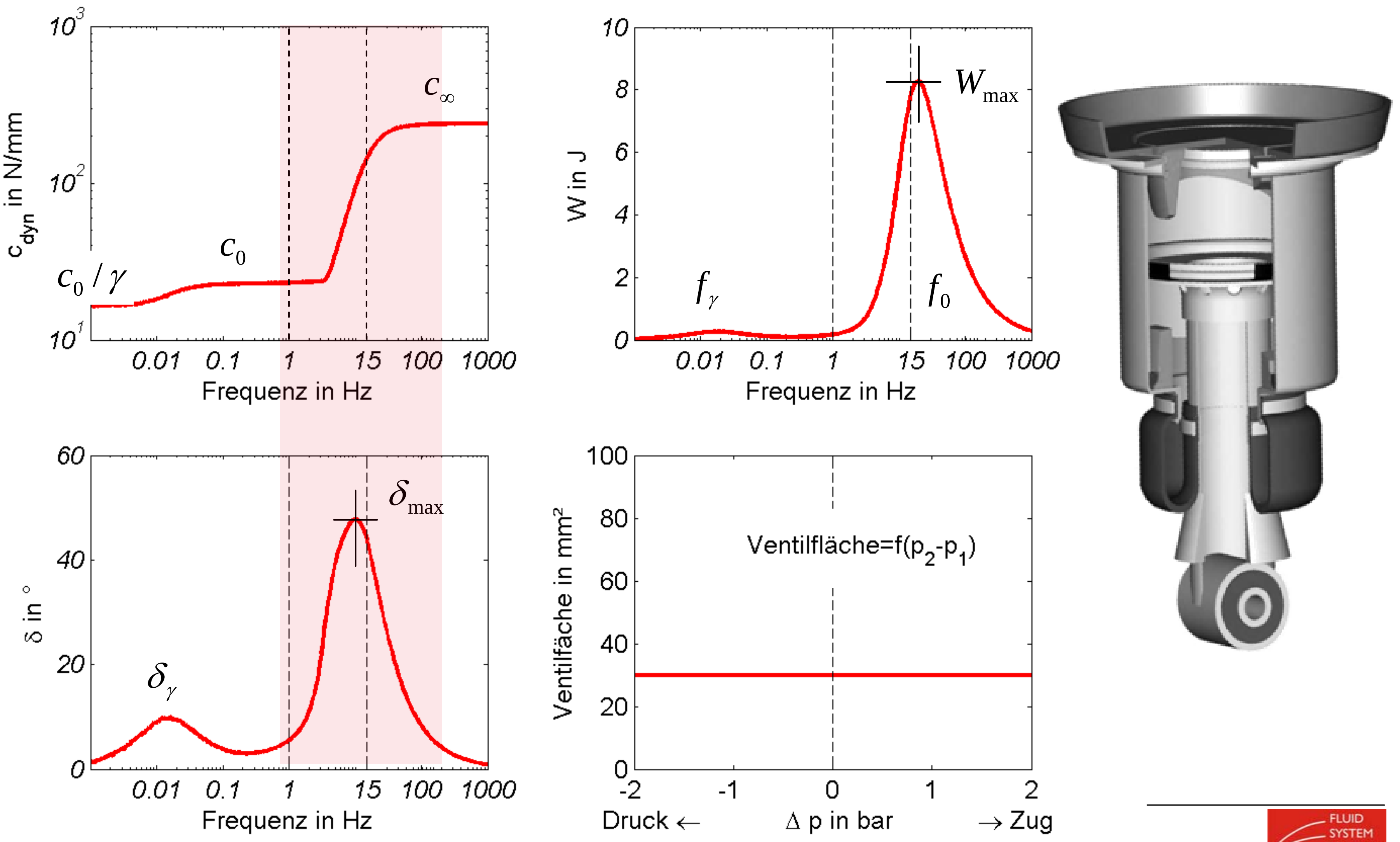
The dissipation per oszillation

$$\frac{W_{\max}}{p_0 L^3} = \left(\frac{\hat{z}}{L}\right)^2 fn\left(\gamma, \kappa_i\right)$$



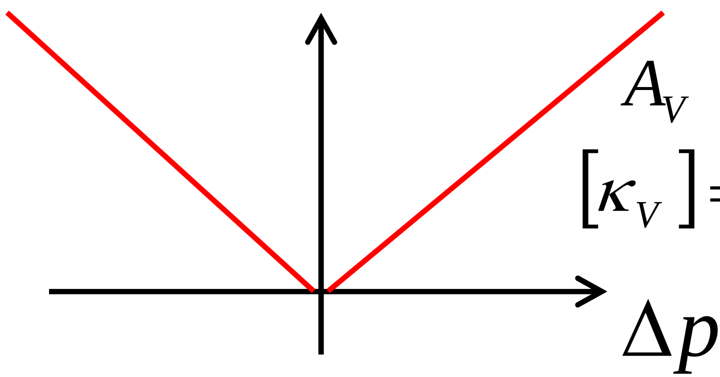
- The dissipation is independent of any temperature change!
- The max. dissipation is independent of the specific valve design
- The max. damping is proportional to the absolute pressure, hence
- The system is selfadaptiv with respect to stiffness and damping

The tuning frequency



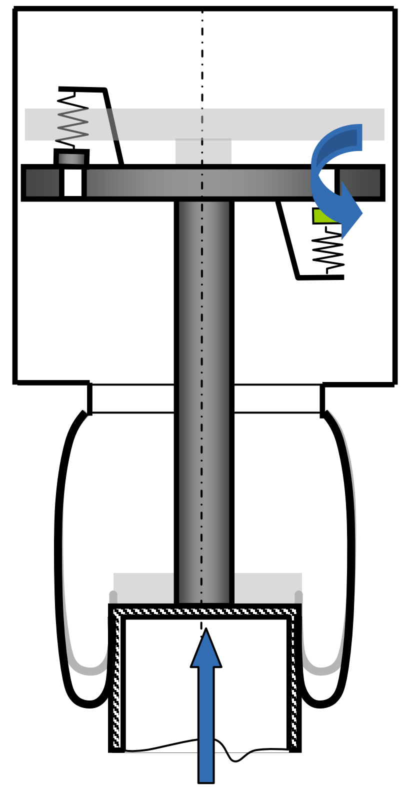
Valve A: check-valve

$$f_0 = fn(p_0, T_0, R, \gamma, L, \kappa_V, \hat{z}, \kappa_i)$$



$$A_V = \kappa_V \, |\Delta p|$$

$$[\kappa_V] = L^4 / F$$

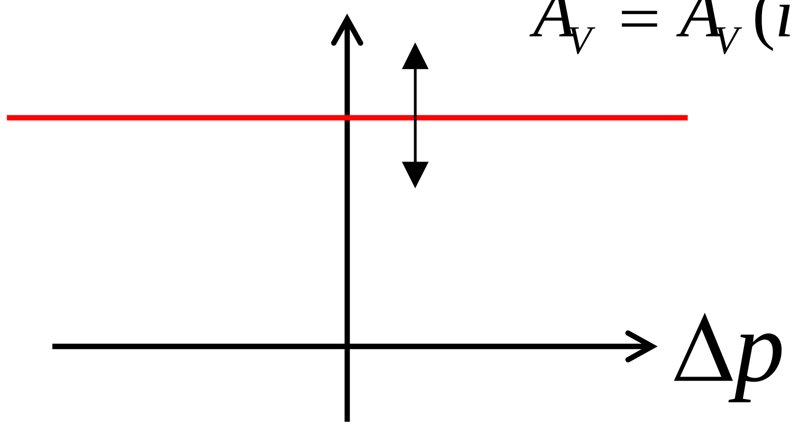


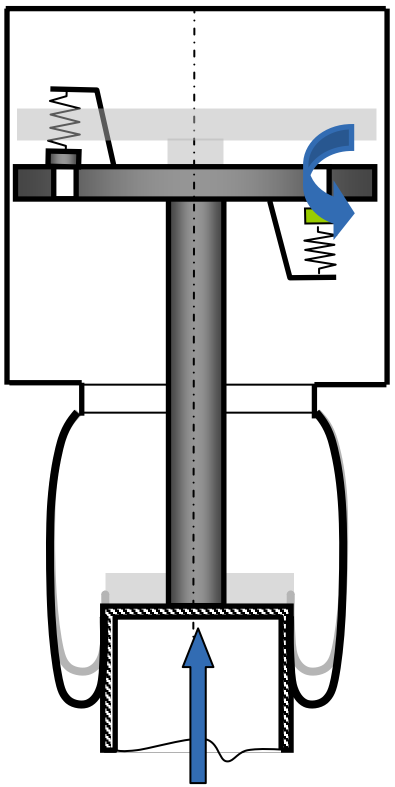
$$\frac{f_0}{\sqrt{\gamma R T_0}\,(p_0 \kappa_V)^{-1/2}} = \left(\frac{\hat{z}}{L}\right)^{1/2} fn(\gamma, \kappa_i)$$

Valve B: controlled valve or orifice with constant cross section

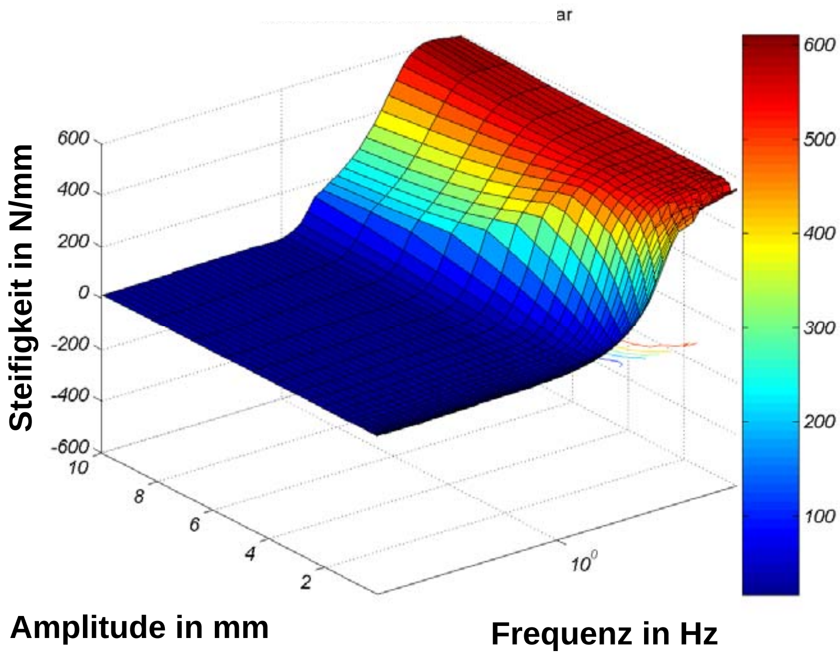
$$f_0 = fn(p_0, T_0, R, \gamma, L, A_V, \hat{z}, \kappa_i)$$

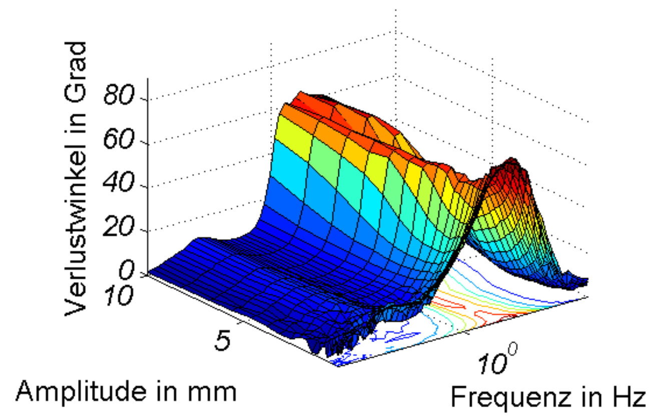
$$A_V = A_V(i)$$



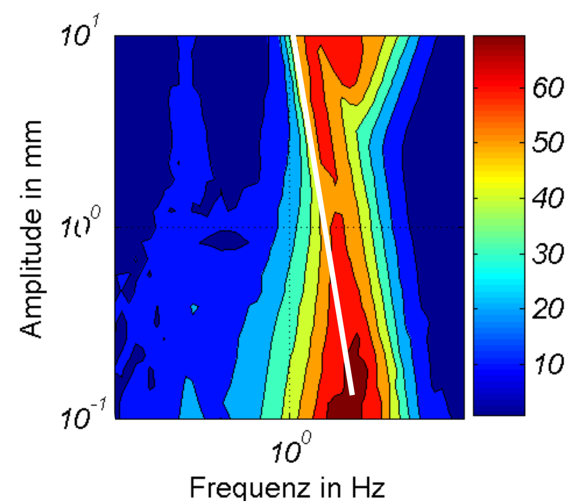


Valve B: controlled valve or orifice with constant cross section





$$f_0 \sim \hat{z}^{-1/2}$$



Valve B: controlled valve or orifice with constant cross section

$$f_0 = fn(p_0, T_0, R, \gamma, L, A_V, \hat{z}, \kappa_i)$$

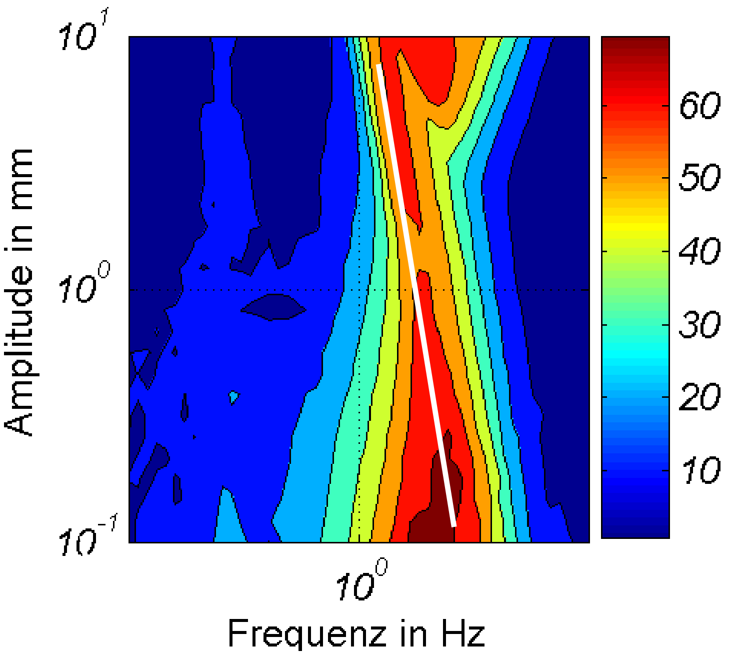
$$\frac{f_0}{\sqrt{RT_0}\, A_V^{-1/2}} = fn\left(\frac{\hat{z}}{L}, \gamma, \kappa_i\right)$$

$$for \qquad A_V / L^2 << 1$$

Further from experiments

$$\frac{f_0}{\sqrt{\gamma RT_0}\, A_V^{-1/2}} = \left(\frac{\hat{z}}{L}\right)^{-1/2} const(\gamma, \kappa_i)$$

Verlustwinkel in °



$$f_0 \sim \hat{z}^{-1/2}$$

ADASS air damper air spring simulation

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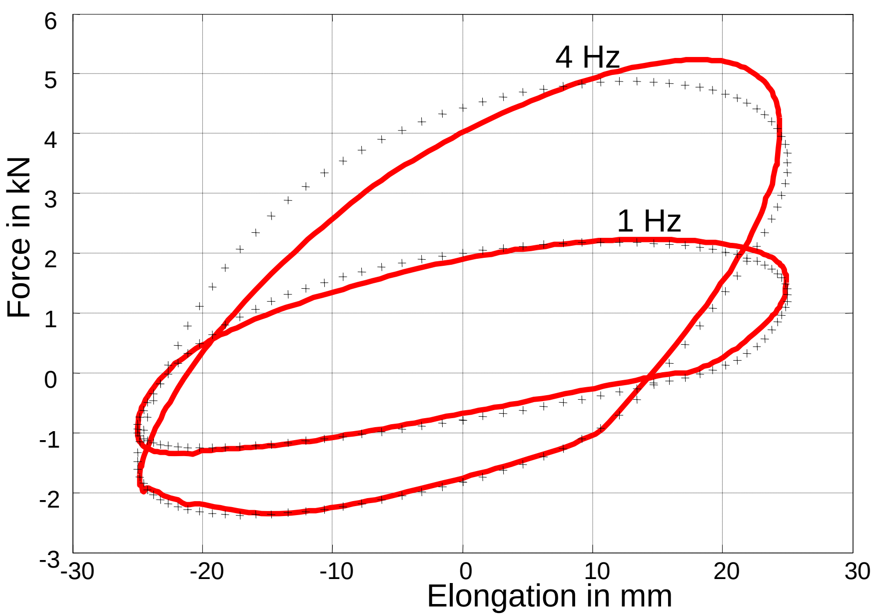
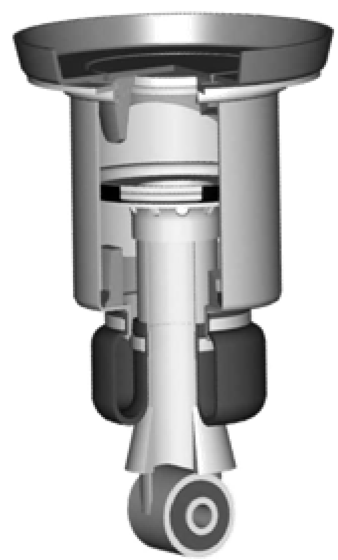


— Experiment LFD

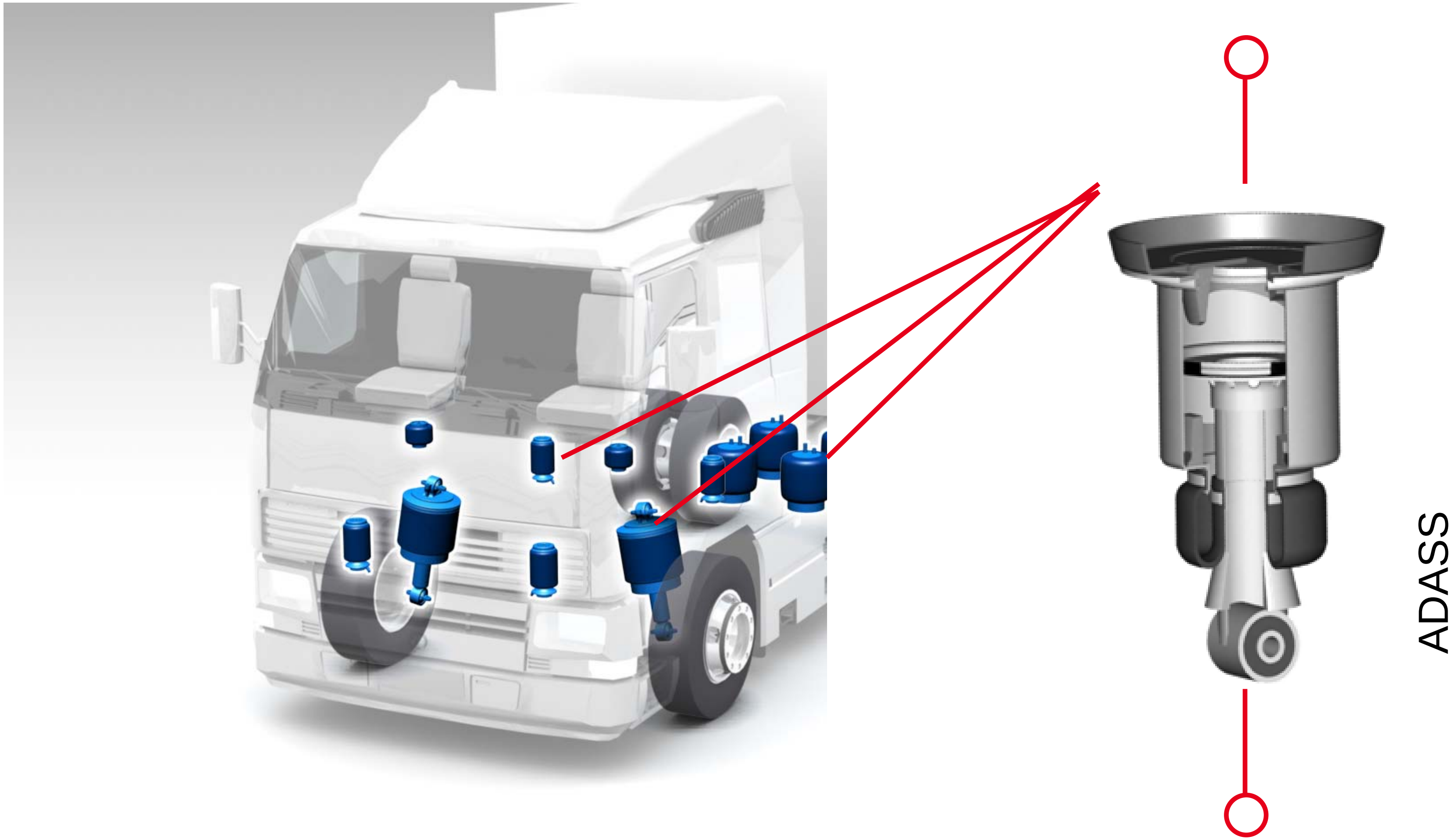
\+ Simulation

- ADASS is available by Vibracoustic for Matlab/Simulink, Simpack, ADAMS, Modelica/Dymola
- Truly physical description
- Simulation in the time domain



FLUID SYSTEM TECHNIK

Test the performance of a new suspension system by multi body simulation first with the help of ADASS



Advantages of using air as damping medium

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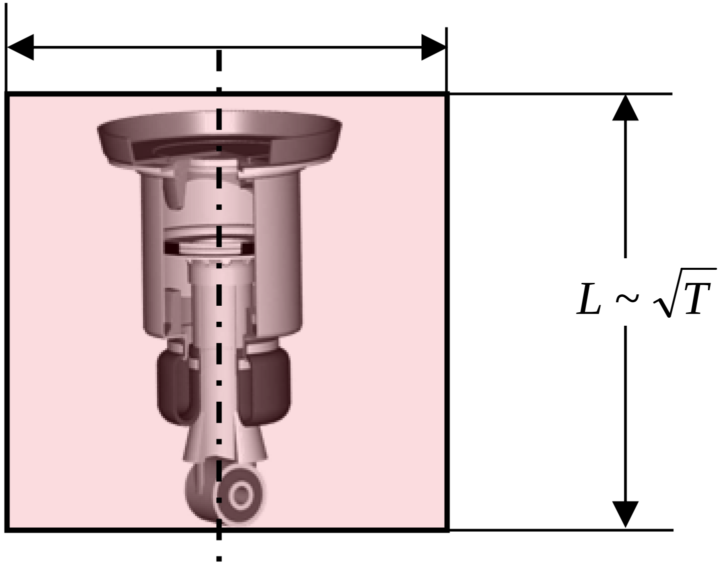
- The dissipation ~ load
  self adjusting system
- The tuning frequency is independent of the load
- The tuning frequency shows a only weak dependence on temperature



$$W_{max} \sim p_0 L^3 \left(\frac{\hat{z}}{L} \right)^2$$

$$f_0 \sim \frac{\sqrt{\gamma R T_0}}{A_V^{1/2}} \left(\frac{\hat{z}}{L} \right)^{-1/2}$$

FLUID SYSTEM TECHNIK

The tasks of any passive suspension system

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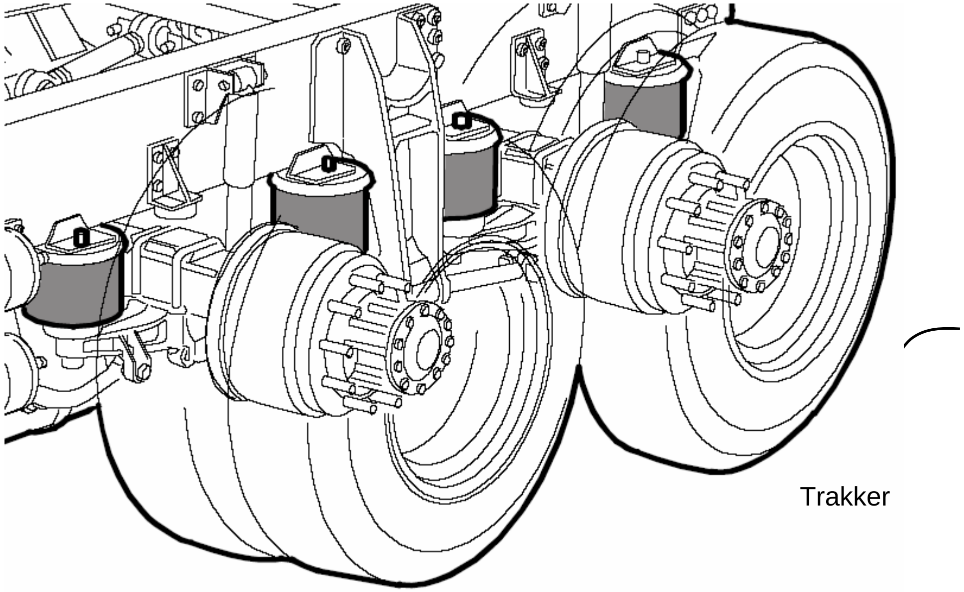
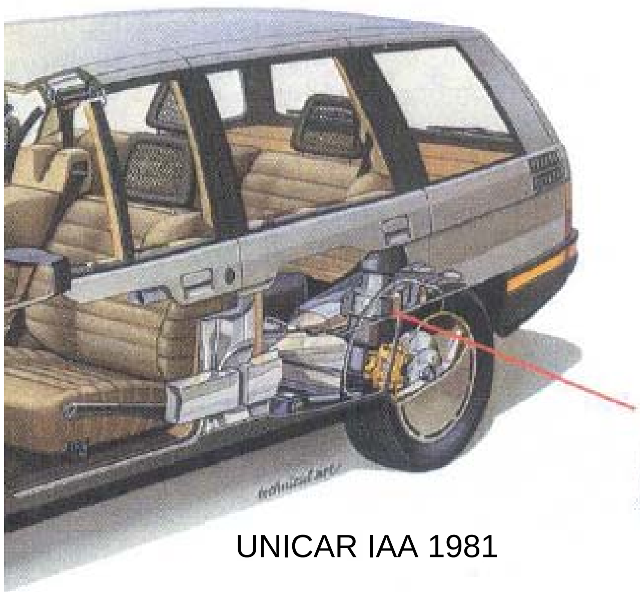
- Carry the load! ✓
- Store kinetic energy
  and leaving the eigenfrequency
  unchanged by loading! ✓
- Dissipate kinetic energy and keep the acceleration as low as possible no matter the load is!
- Have as less as possible dry friction within the suspension system to reduce harshness and not to restrict the tuning possibilities ✓
- Be robust, not to pricy,
  and in case easy to replace



FLUID SYSTEM TECHNIK

Design solutions which show small dry friction



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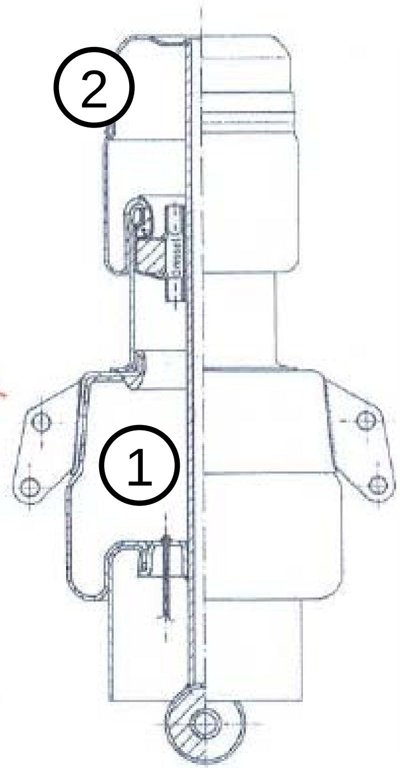

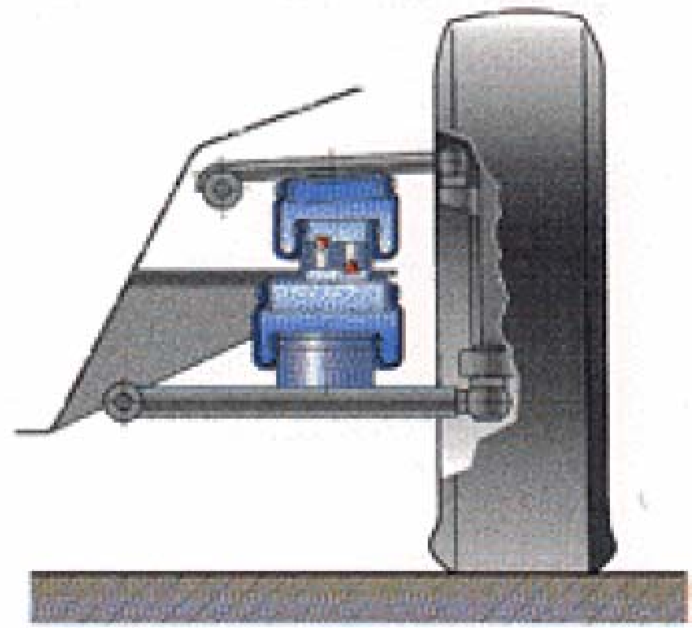

Summary

- Air suspension system in general allow a distinction between the functions “carry the load” and “store kinetic energy”.
- Air damping system in addition are self adaptive to a change in load/mass.
- The damping performance is independent of the temperature.
- The tuning frequency changes with the square root of the abs. temperature.
- The tuning frequency is independent of the pressure / load.
- Design solutions which show very low dry frictions are available.