Dimensionieren und prinzipielles Verhalten von Luftfeder-/Dämpfermodulen

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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”

Spring

**mg = c_0 z_0**

Air spring

\[ mg = p_0 V_0 \]

Air spring damper

\[ mg = A_T (p_0 - p_u) \]

**no leveling**

**leveling**
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The Bridgman Postulate and the Buckingham $\Pi$ Theorem

- Predefined basic physical quantities (e.g. MLT, FLT, FL, ....) and physical units (e.g. 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

\[ 72 \text{km/h} = f(y_1, y_2, ..., y_r), \quad 20 \text{m/sec} = f(c_1 y_1, c_2 y_2, ..., c_r y_r) \]

\[ 36 \text{km/h} = f(x_1, x_2, ..., x_r), \quad 10 \text{m/sec} = f(c_1 x_1, c_2 x_2, ..., c_r x_r) \]

\[ \frac{x}{y} = \frac{x'}{y'} \quad \Rightarrow \quad x = x_1^{a_1} x_2^{a_2} ... x_r^{a_r} \]
Store kinetic energy „the inertia mass“

\[ K = \frac{1}{2} m \dot{z}^2 \omega^2 = 2\pi^2 m \left( \frac{\dot{z}}{T} \right)^2 \]

\[ E = \frac{1}{2} c_0 \dot{z}^2 \]

oscillation period \[ T = 2\pi \sqrt{\frac{m}{c_0}} \]
Store kinetic energy

\[ T = fn(m, g, c_0) \quad T = fn(m, g, L, p_0(mg)) \quad T = fn(m, g, L) \]
Store kinetic energy

\[ T = fn(m, g, c_0) \]

\[ \begin{array}{c|cccc}
T & m & g & c_0 \\
\hline
L & 1 & & \\
M & 1 & 1 & \\
T & 1 & -2 & -2
\end{array} \]

\[ T = fn(m, g, L, p_0(mg)) \]

\[ \begin{array}{c|cccc}
T & m & g & L \\
\hline
L & 1 & 1 & \\
M & 1 & \\
T & 1 & -2
\end{array} \]
## Store kinetic energy

<table>
<thead>
<tr>
<th>spring</th>
<th>air spring</th>
<th>air spring damper</th>
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</thead>
</table>

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

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

<table>
<thead>
<tr>
<th>(T)</th>
<th>(\frac{1}{\sqrt{m}})</th>
<th>(g)</th>
<th>(c_0)</th>
<th>(L)</th>
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Store kinetic energy

\[ T = \text{const} \sqrt{\frac{m}{c_0}} \]

\[ T = \text{const} \sqrt{\frac{L}{g}} \]
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.

\[
\frac{T}{\sqrt{L/g}} = \text{const}
\]
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 \( f >> f_\gamma \) \( \sim \left( \frac{S}{V_0} \right)^2 \frac{\lambda}{\rho_0 c_p} \sim 0.001 \ldots 0.01 \text{Hz} \)

\[ p = p_0 \left( \frac{V_0}{V} \right)^\gamma \]

for \( f << f_\gamma \)

\[ p = p_0 \left( \frac{V_0}{V} \right) \]

\[
c_0 = -\gamma p_0 A_T \frac{1}{V_0} \frac{dV}{dz} + \left( p_0 - p_u \right) \frac{dA_T}{dz}
\]

„volume part“ „piston part“
The typical length

For \( p_0 \gg p_u \) : \[ mg = (p_0 - p_u)A_T \approx p_0 A_T \]

it follows

\[ c_0 = \frac{mg}{L} \left( \gamma + \frac{1}{A_T} \frac{dA_T}{dz} \right) \]

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

\[ T = 2\pi \sqrt{\frac{L / g}{\sqrt{\gamma + \frac{1}{A_T} \frac{dA_T}{dz} / L}}} \]
The typical length \( L = \frac{V_0}{A} \) for \( p_0 >> p_u \) and

\[
\frac{1}{A_T} \frac{dA_T}{d(z/L)} \equiv 0
\]

\[ T = 2\pi \sqrt{\frac{L}{\gamma g}} \]
Advantage of an air suspension system: Separation of function

- Carry the load at a given pressure
  - diameter \( D^2 \sim mg \)

- Store the kinetic energy and leave the eigenfrequency unchanged
  - height \( L \sim \sqrt{T} \)

\[
\begin{align*}
D^2 & \sim mg \\
L & \sim \sqrt{T}
\end{align*}
\]

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

- 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
Damping

\[ \text{Re}(ze^{i\Omega t}) \]

\[ \text{Re}(z_0 e^{i\Omega t}) \]
Bode diagram for (linear) hydraulic suspension system

\[ c_0 = m \frac{g \gamma}{L} \quad d = c_0 f_0 \]

\[ f_0 = f_{n}(p_0) \]

\[ c(\Omega) = c_0 + id\Omega \]

\[ \overline{c}(\Omega) = 1 + i\Omega / f_0 \]

Bode-Diagram: stiffness

Bode-Diagram: phase angle

tuning frequency = cut off frequency \( f_0 \)
Bode diagram for (linear) air damping system

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

Bode-Diagram: stiffness

Bode-Diagram: phase angle
Air damping is easy to understand

- $f \ll f_0$
  - No piston
  - No dissipation
  - Soft air spring $V_1 + V_2$

- $f \sim f_0$
  - dissipation

- $f \gg f_0$
  - No valve
  - Stiff air spring $V_1, V_2$ in parallel
The stiffness ratio is a question of geometry

\[ V_0 = V_1 + V_2 \]

\[ \bar{c}_\infty := \frac{c_\infty}{c_0} = \frac{1 - \nu(1 - \alpha^2)}{\nu(1 - \nu)(1 - \alpha)^2} = fn(\kappa_i) = 4..20 \]

\[ \nu := \frac{V_2}{V_0}, \alpha := \frac{A_1}{A_2} \]
Why air damping?

- 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
Tranfer behavior of an air spring damper

\[ c_{\text{dyn}} \text{ in N/mm} \]

\[ c_{0} / \gamma \]

\[ c_{0} \]

\[ c_{\infty} \]

\[ \delta_{\text{max}} \]

\[ \delta_{\gamma} \]

\[ W \text{ in J} \]

\[ W_{\text{max}} \]

\[ f_{\gamma} \]

\[ f_{0} \]

\[ \text{Ventilfläche} = f(p_2 - p_1) \]

\[ \Delta p \text{ in bar} \]

\[ \text{Zug} \]
The max. dissipation follows out of a most general dimensional analysis

\[ W_{\text{max}} = fn(p_0, T_0, R, \gamma, L, A_V, \hat{z}, \kappa_i) \]
The dissipation per oscillation

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

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

From experiments it follows further

\[ \frac{W_{\text{max}}}{p_0 L^3} = \left(\frac{\hat{z}}{L}\right)^2 fn(\gamma, \kappa_i) \]
The dissipation per oscillation

\[ \frac{W_{\text{max}}}{p_0 L^3} = \left( \frac{\hat{Z}}{L} \right)^2 f_n(\gamma, \kappa_i) \]

- 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 selfadaptive with respect to stiffness and damping
The tuning frequency

\[ c_\text{dyn} \text{ in N/mm} \]

\[ c_\infty \]

\[ c_0 / \gamma \]

\[ f_\gamma \]

\[ f_0 \]

\[ W_{\text{max}} \]

\[ \delta_{\text{max}} \]

\[ \delta_\gamma \]

\[ \delta \text{ in }^\circ \]

\[ \text{Ventilfläche}=f(p_2-p_1) \]

\[ \text{Ventilfläche in mm}^2 \]

\[ \text{Druck} \rightarrow \Delta p \text{ in bar} \rightarrow \text{Zug} \]
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

\[ f_0 = f_n(p_0, T_0, R, \gamma, L, A_V, \hat{z}, \kappa_i) \]

\[ A_V = A_V(i) \]

\[ \Delta p \]
Valve B:
controlled valve or orifice with constant cross section

\[ f_0 \sim \hat{\omega}^{-1/2} \]
Valve B: controlled valve or orifice with constant cross section

\[ f_0 = f_n(p_0, T_0, R, \gamma, L, A_V, \hat{z}, \kappa_i) \]

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

for \( A_V / L^2 << 1 \)

Further from experiments

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

\[ f_0 \sim \hat{z}^{-1/2} \]
ADASS air damper air spring simulation

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

Experiment LFD
Simulation
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

- The dissipation \( \sim \) load self adjusting system

- The tuning frequency is independent of the load

- The tuning frequency shows a only weak dependence on temperature

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

\[
f_0 \sim \frac{\sqrt{\gamma RT_0}}{A_V^{1/2}} \left( \frac{\hat{z}}{L} \right)^{-1/2}
\]
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Design solutions which show small dry friction

UNICAR IAA 1981
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.