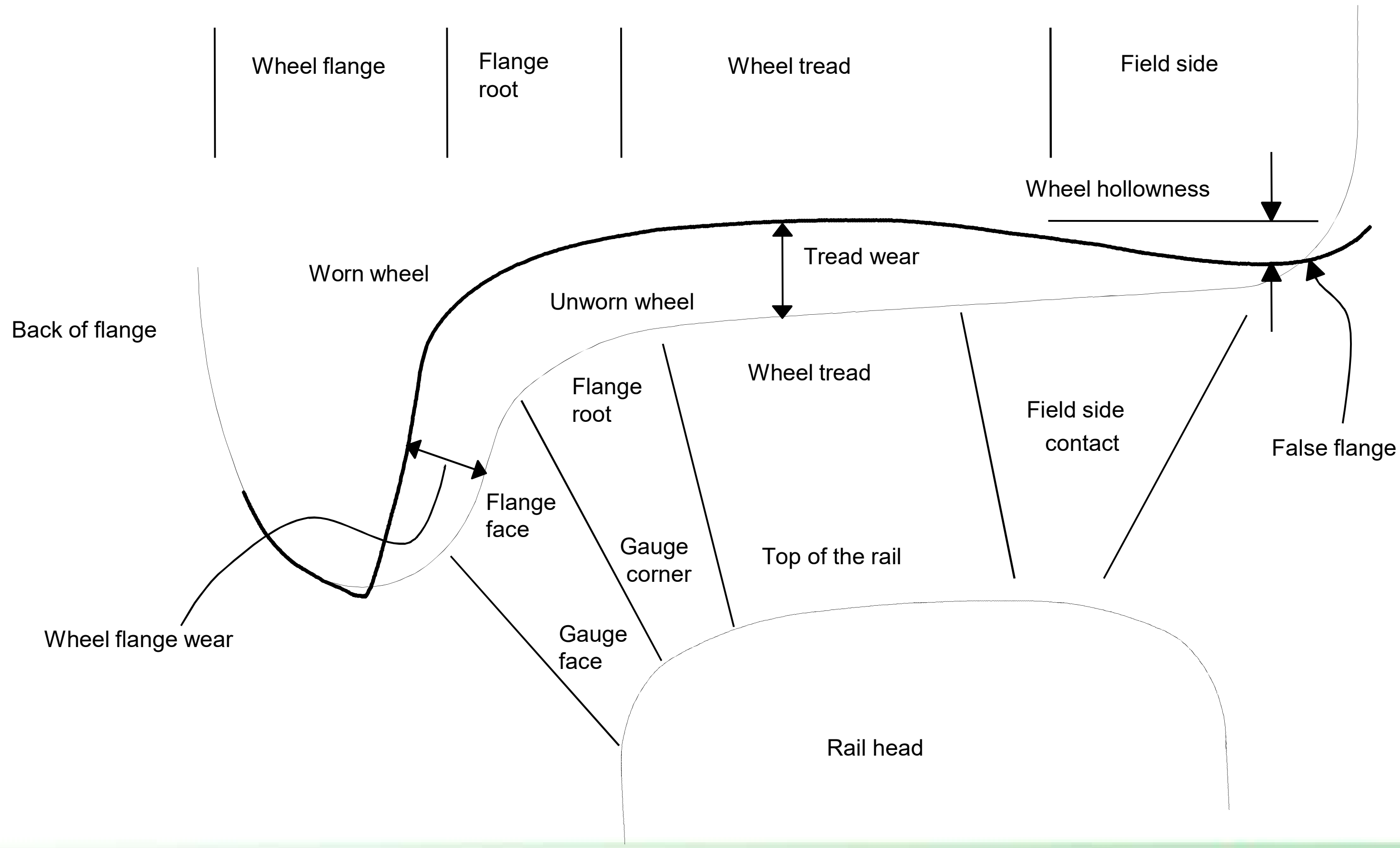
Vehicle-Track Interaction & Dynamics

Eric Magel, NRC Canada

Agenda

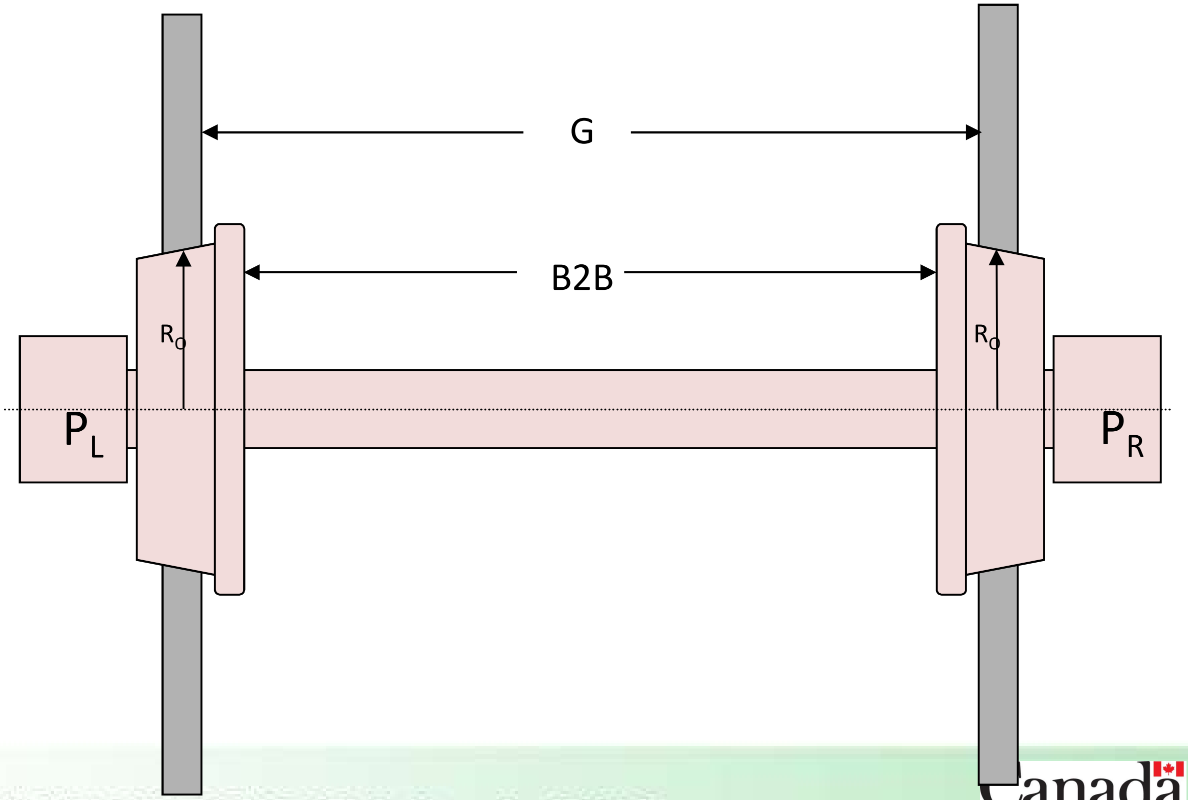
1. Vehicle steering, stability and curving forces
2. Wheel-Rail profile design and performance
3. VTI Derailment Mechanisms and Risk Assessment
4. Impact and Dynamic Loads

Terminology

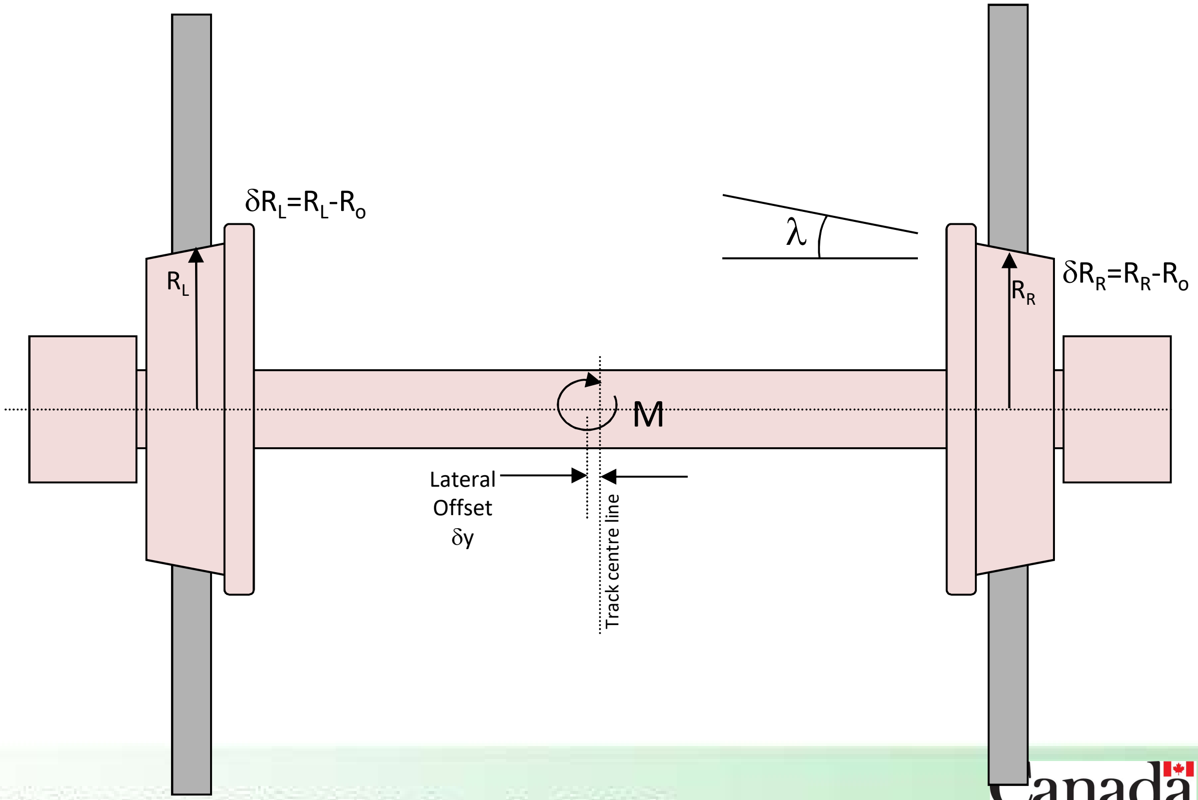


VEHICLE STEERING

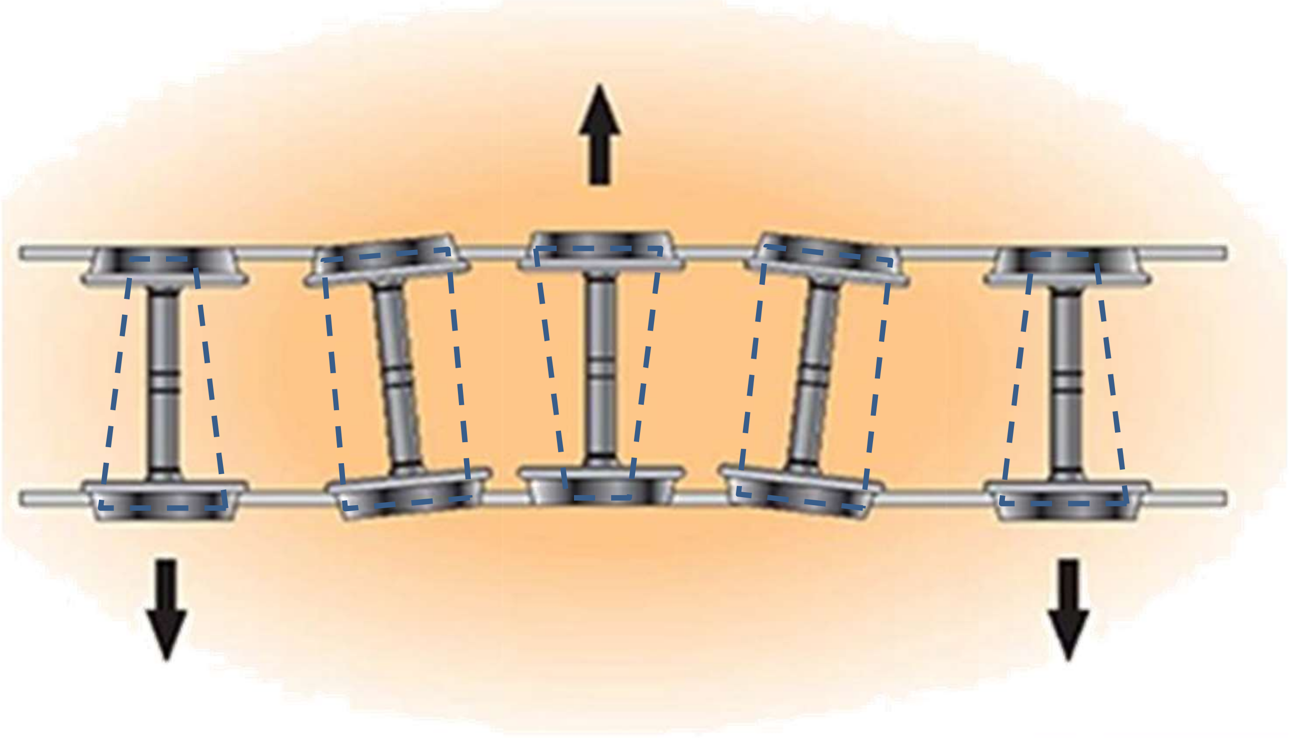
The free rolling wheelset

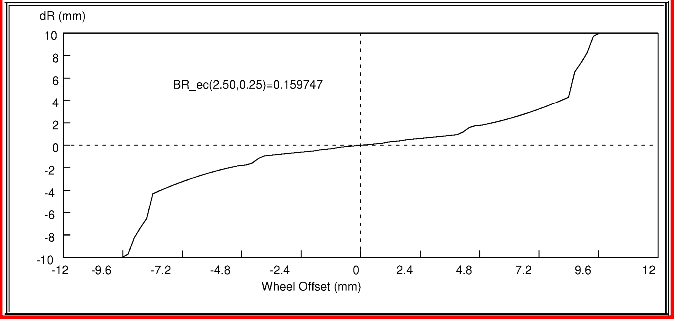


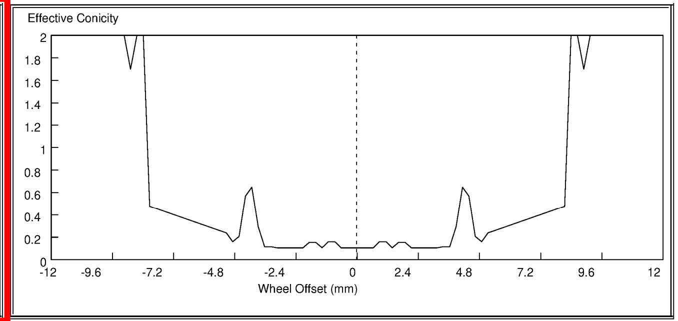
The free rolling wheelset



The Free Wheelset - Hunting

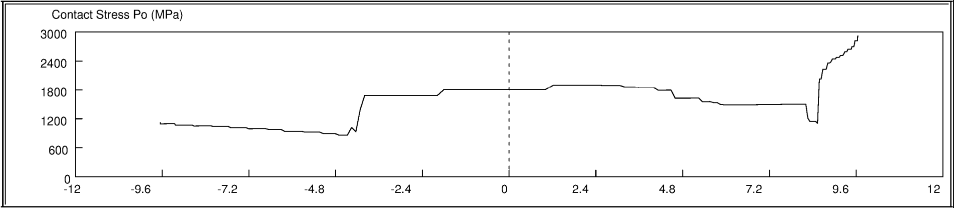


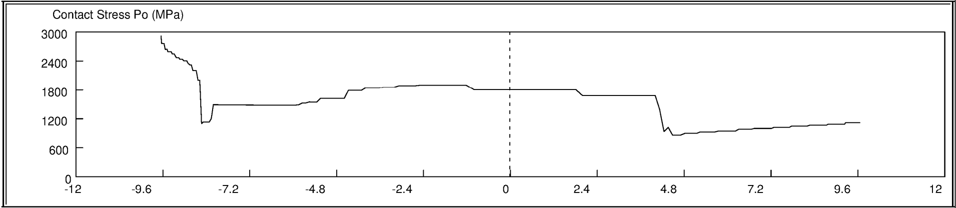


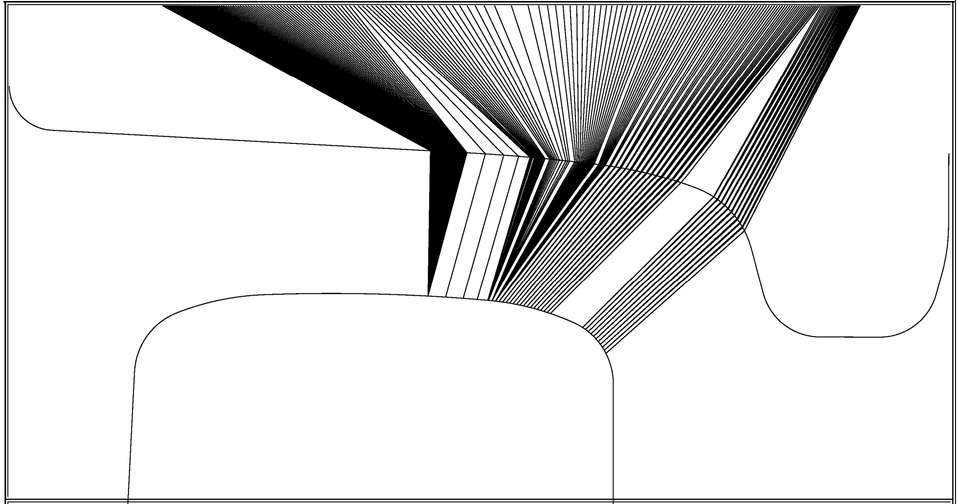


Input/Output Values

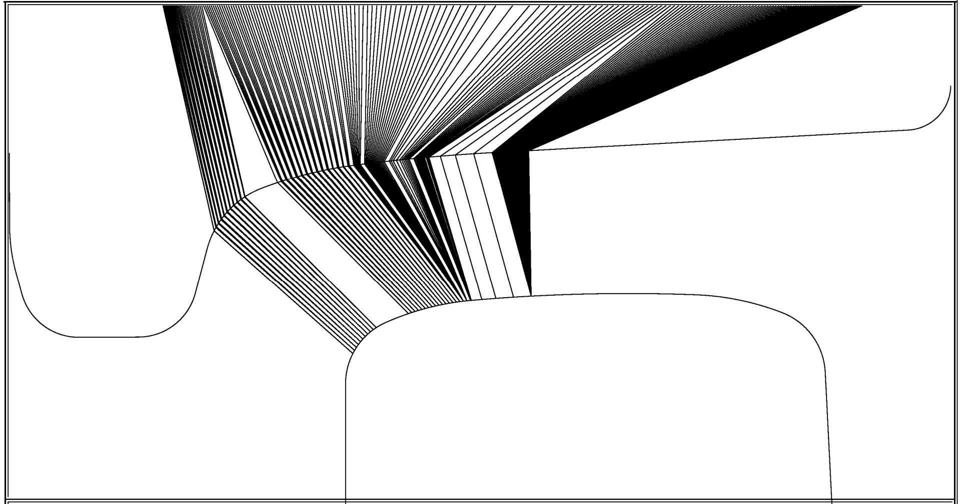
L_load : 5100
R_load : 5100
aoa : 0
Rleft : 0.33
Rright : 0.33
CurveRadius : 10000
dR : 0.0
Smoment : 0.0
Drag : 0.0







freshmeasured.trc re115_40



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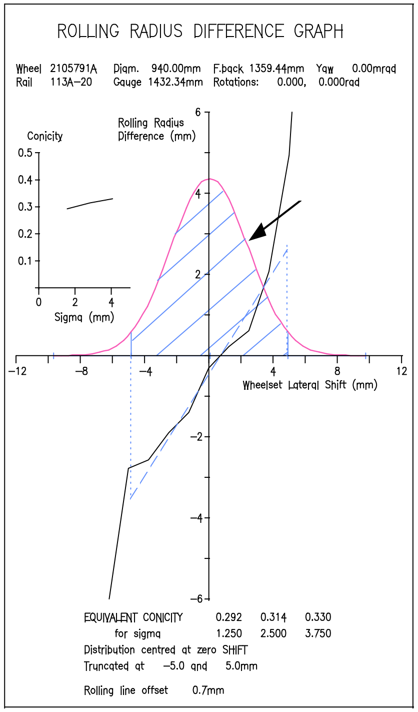
Equivalent Conicity from the ΔR plot

- British Rail derivation

$$\lambda_e = \frac{1}{2}\int \frac{N(y)\,(r_R - r_L)}{y}\,dy$$



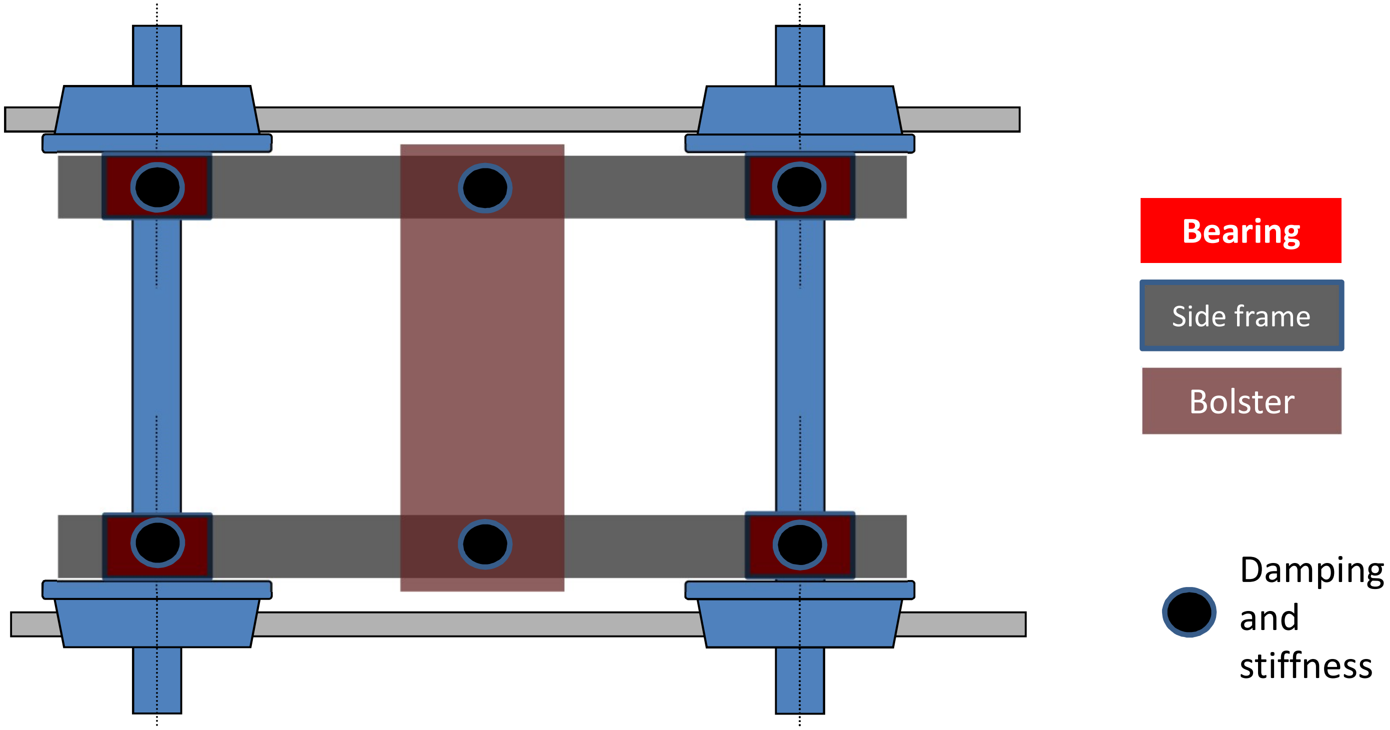
ROLLING RADIUS DIFFERENCE GRAPH

Wheel 2105791A Diam. 940.00mm F.back 1359.44mm Yaw 0.00mrad
Rail 113A–20 Gauge 1432.34mm Rotations: 0.000, 0.000rad

EQUIVALENT CONICITY 0.292 0.314 0.330
for sigma 1.250 2.500 3.750
Distribution centred at zero SHIFT
Truncated at –5.0 and 5.0mm

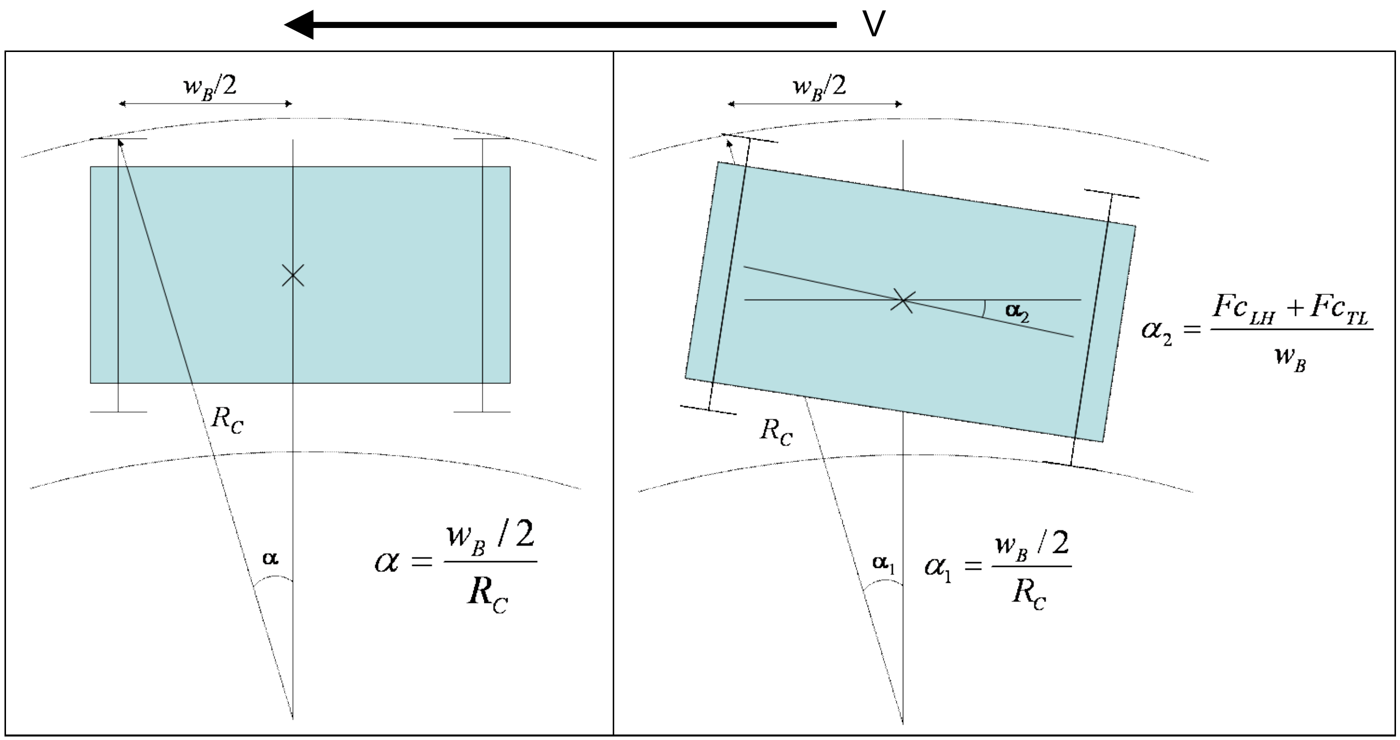
Rolling line offset 0.7mm

A Truck can provide Stability



Leading wheelset: yaw angle

- Rigid Truck
- Self-steering (flexible)
- Steered



$$\alpha = \frac{w_B/2}{R_C}$$

$$\alpha_1 = \frac{w_B/2}{R_C}$$

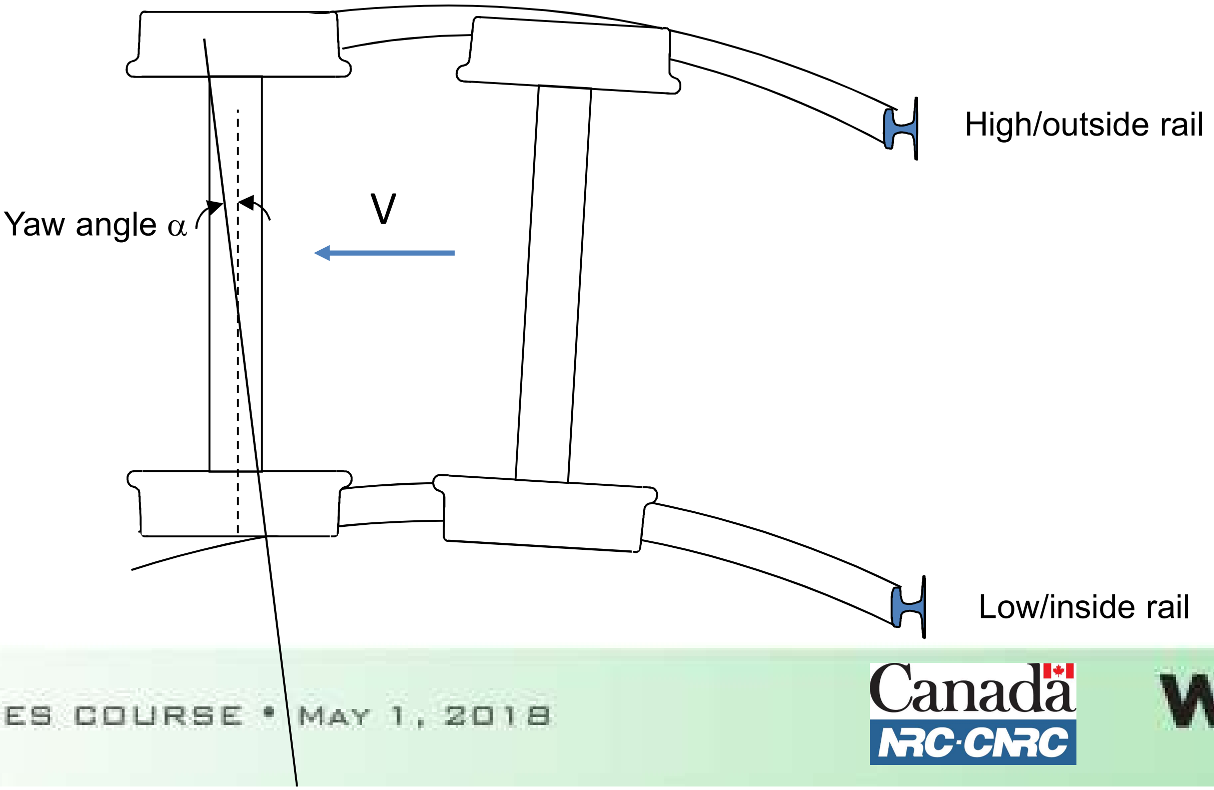
$$\alpha_2 = \frac{Fc_{LH} + Fc_{TL}}{w_B}$$

Also, yaw angle due to deflection of suspension (bending and shear)

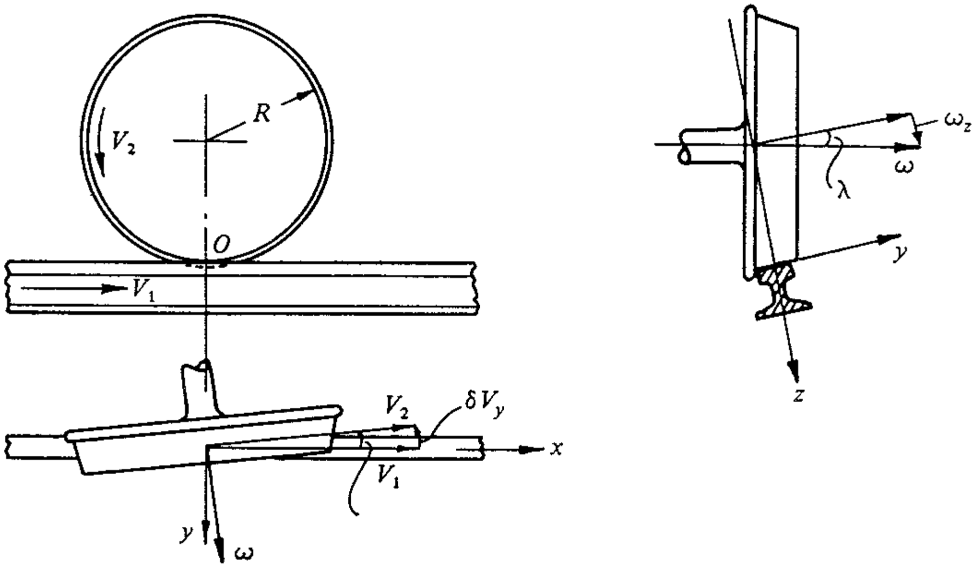
CURVING FORCES

The Wheelsets (in a curve)

(leading) wheelset shifts to outside of curve



Creepage in a Single Wheel/rail Contact



Longitudinal Creepage

$$\psi_X = \frac{V_2 - V_1}{V_1}$$

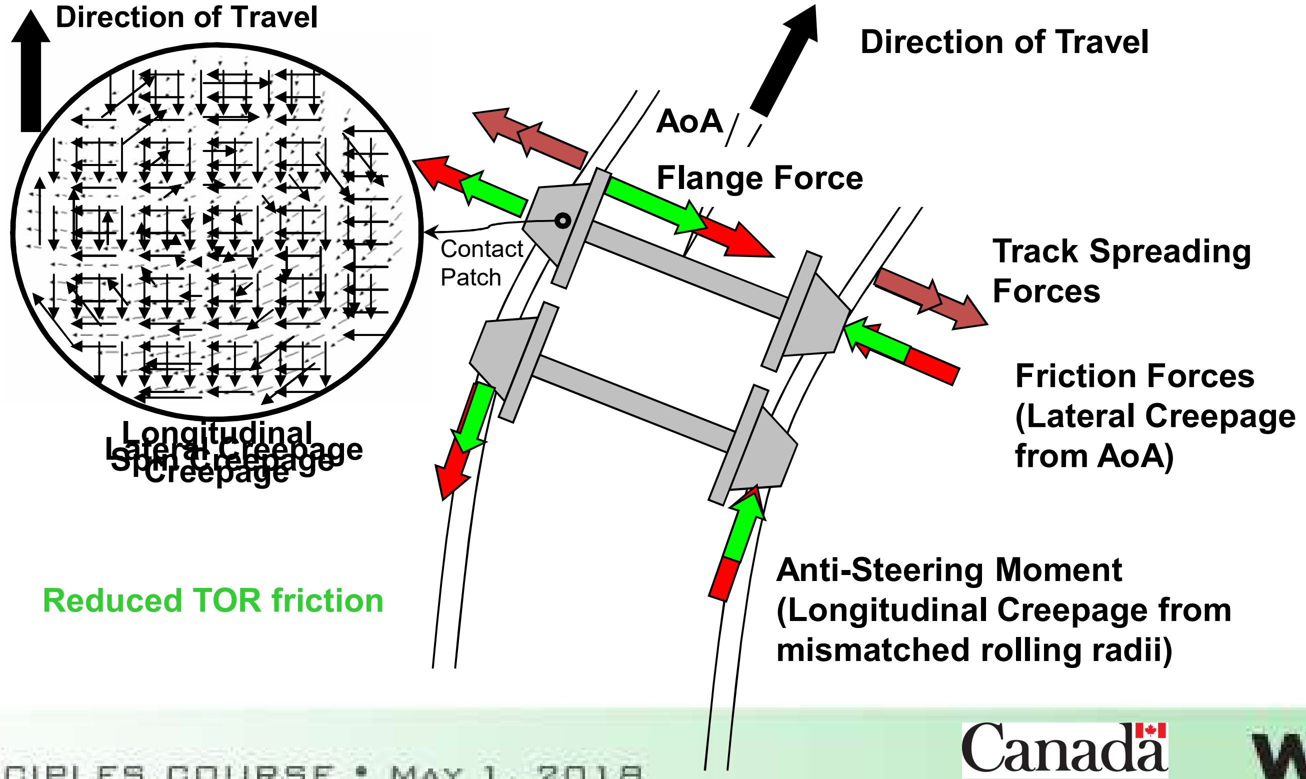
Lateral Creepage

$$\psi_Y = \frac{\delta V_Y}{V_1} = \tan\gamma$$

Spin Parameter

$$\Phi = \omega\frac{(ab)^{1/2}}{V_1 R} = \left(\frac{(ab)^{1/2}}{R}\right)\tan\lambda$$

Lateral Forces (Creep) in Curves



Direction of Travel

Longitudinal Creepage

Lateral Creepage

Spin Creepage

Contact Patch

Direction of Travel

AoA

Flange Force

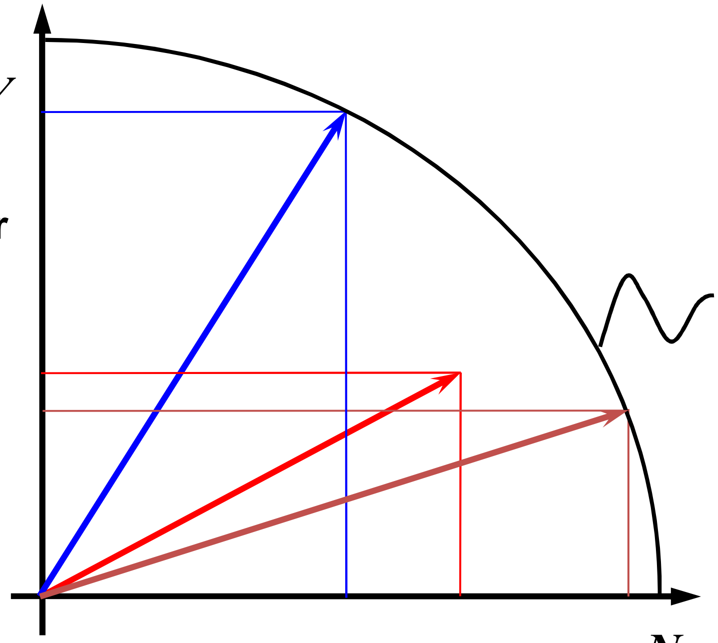
Track Spreading Forces

Friction Forces (Lateral Creepage from AoA)

Anti-Steering Moment (Longitudinal Creepage from mismatched rolling radii)

Reduced TOR friction

Friction Circle



$L=\mu V$

L: Lateral Creepage or Force

$T=\mu V$

Where $T=\sqrt{L^2+N^2}$

$N=\mu V$

N: Longitudinal Creepage or Force

WHEEL-RAIL PROFILE DESIGN AND PERFORMANCE

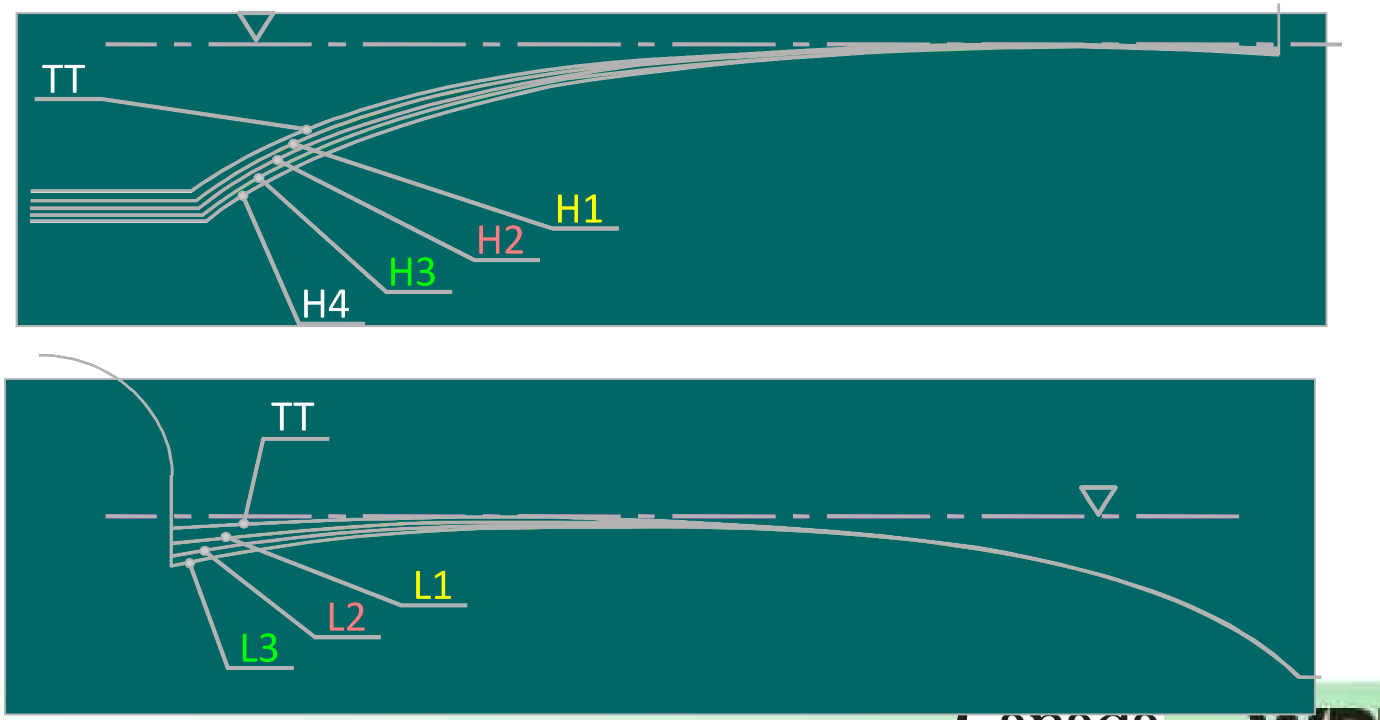
Design of Engineered Rail Profiles

Rail design considers:

- **Track curvature**
- **Worn wheel shapes**
- **Types of vehicle and speed (hunting)**
- **Dynamic rail rotation**
- **Rail hardness**
- **Grinding interval (profile deterioration between intervals)**
- **Static gage**

- control contact stress
- inhibit hunting
- minimize wear

The NRC family of heavy haul rail templates (1990's)



Rail Profile Design Criteria

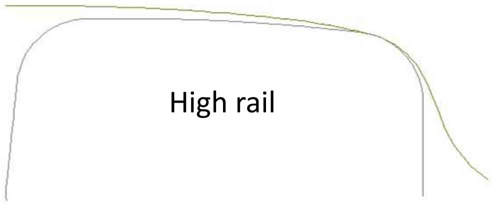
Goals are to reduce/control:

- Gauge face and TOR wear
- Rolling contact fatigue (RCF)
- Dynamic instability (hunting)
- Corrugation formation
- Wheel hollowing

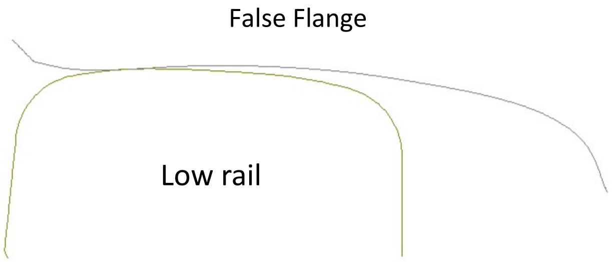
And are easily or practically implemented by grinding

Wheel/rail contact stresses

- Stress and damage depend on:
  - wheel radius
  - wheel load
  - friction coefficient
  - wheel/rail profiles (contact geometry)

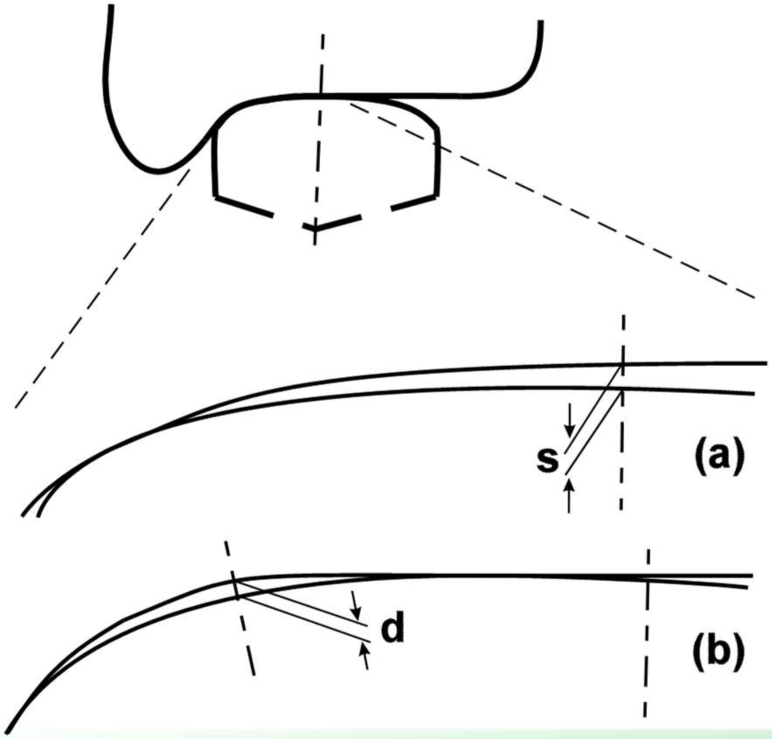


Severe gauge-corner contact



Hollow wheels

Wheel / Rail Conformality



- closely conformal (as per hertzian spring)

  0.1 mm (0.004”) or less
- conformal

  0.1 mm to 0.4mm

  (0.004” to 0.016”)
- non-conformal

  0.4 mm (0.016”) or larger

Some Typical Issues Associated with Wheel/Rail Conformality

Closely conformal profiles

- Dynamic instability (hunting)
- Corrugation formation by spin creepage

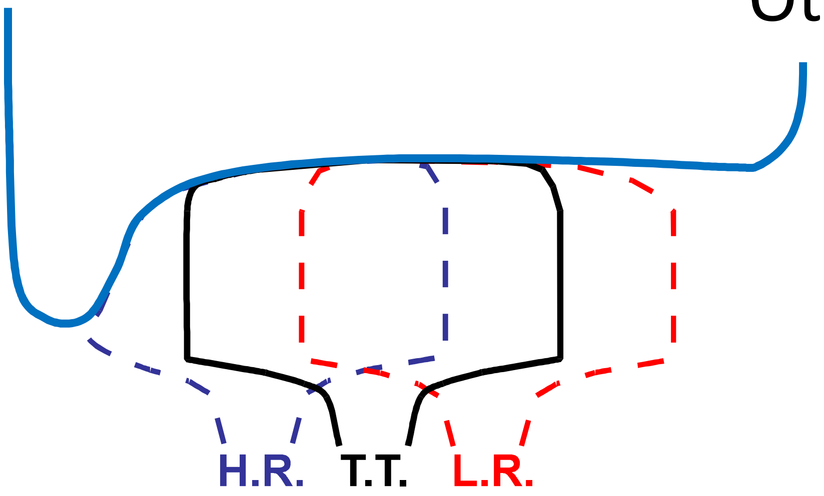
Conformal profiles

- Low stress state W/R interface
- Used for mass transit and high speed lines = 1PT conformal (good for steering)
- Heavy haul = 2PT conformal (balance contact stress steering and wear)

Non-conformal profiles

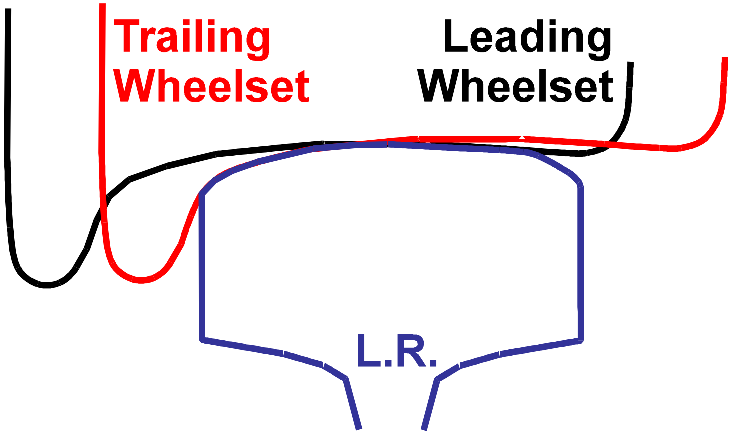
- High stress state W/R interface
- 1PT: cracks (RCF) at GC of HR and FS of LR
- 2PT: high gauge face wear in curves

Worn Wheel and Rail Profiles are Envelopes of Each Other



- Worn wheel is an envelope of all rail profiles it encounters on a particular route

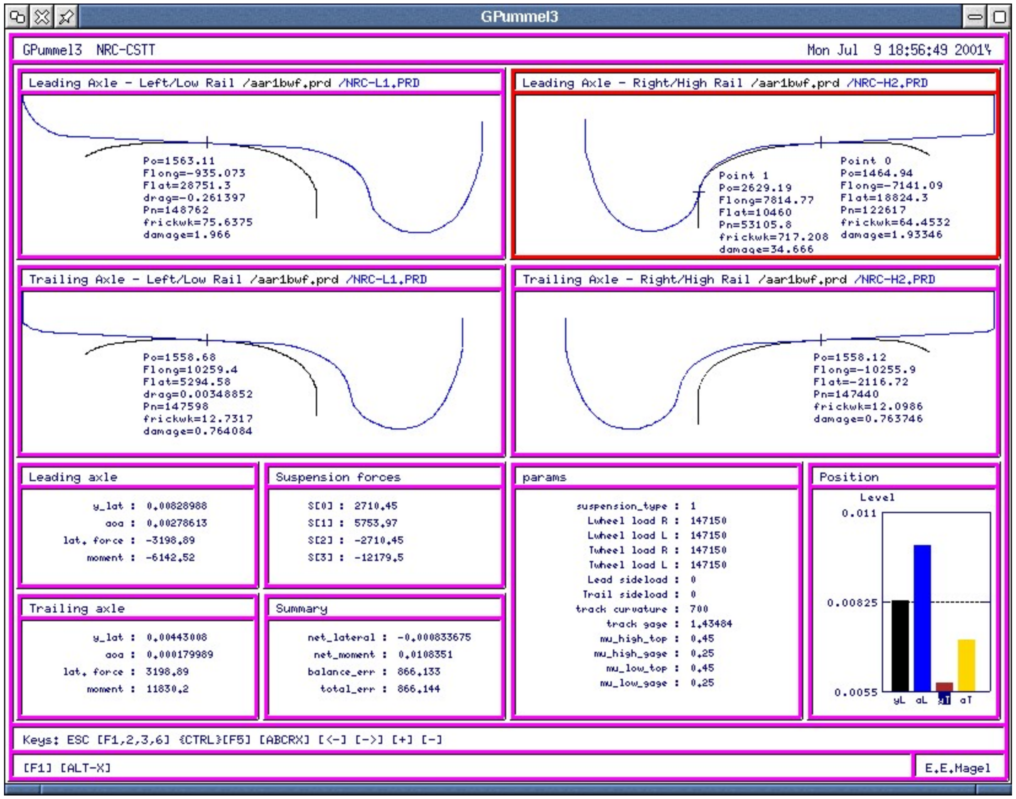
- Worn rail is an envelope of all wheel profiles that pass over it



Pummelling Analysis

- Simulation
  - Measured wheel profiles
  - vehicle characteristics (stiffness, wheelbase etc.)
  - rail hardness (for damage evaluation)
  - rail curvature, super-elevation, dynamic rail rotation etc.
- Evaluate distributions of
  - contact stress
  - steering moments
  - effective conicity

Pummelling: design/analysis tool



```
GPummel3  NRC-CSTT                                        Mon Jul  9 18:56:49 2001%

Leading Axle - Left/Low Rail /aar1bwf.prd /NRC-L1.PRD
Po=1563.11
Flong=-935.073
Flat=28751.3
drag=-0.261397
Pn=148762
frickwk=75.6375
damage=1.966

Leading Axle - Right/High Rail /aar1bwf.prd /NRC-H2.PRD
Point 1                 Point 0
Po=2629.19              Po=1464.94
Flong=7814.77           Flong=-7141.09
Flat=10460              Flat=18824.3
Pn=53105.8              Pn=122617
frickwk=717.208         frickwk=64.4532
damage=34.666           damage=1.93346

Trailing Axle - Left/Low Rail /aar1bwf.prd /NRC-L1.PRD
Po=1558.68
Flong=10259.4
Flat=5294.58
drag=0.00348852
Pn=147598
frickwk=12.7317
damage=0.764084

Trailing Axle - Right/High Rail /aar1bwf.prd /NRC-H2.PRD
Po=1558.12
Flong=-10255.9
Flat=-2116.72
Pn=147440
frickwk=12.0986
damage=0.763746

Leading axle
     y_lat :  0.00828988
       aoa :  0.00278613
lat. force :  -3198.89
    moment :  -6142.52

Suspension forces
  S[0] :  2710.45
  S[1] :  5753.97
  S[2] :  -2710.45
  S[3] :  -12179.5

Trailing axle
     y_lat :  0.00443008
       aoa :  0.000179989
lat. force :  3198.89
    moment :  11830.2

Summary
net_lateral :  -0.000833675
 net_moment :  0.0108351
balance_err :  866.133
  total_err :  866.144

params
suspension_type :  1
  Lwheel load R :  147150
  Lwheel load L :  147150
  Twheel load R :  147150
  Twheel load L :  147150
  Lead sideload :  0
 Trail sideload :  0
track curvature :  700
     track gage :  1.43484
    mu_high_top :  0.45
   mu_high_gage :  0.25
     mu_low_top :  0.45
    mu_low_gage :  0.25

Keys: ESC [F1,2,3,6] {CTRL}[F5] [ABCRX] [<-] [->] [+] [-]
[F1] [ALT-X]                                                              E.E.Magel
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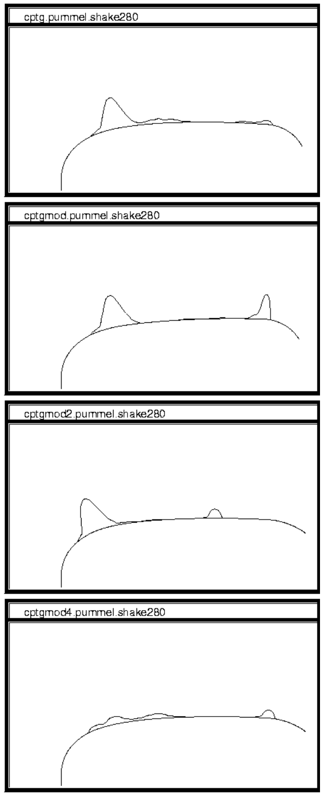
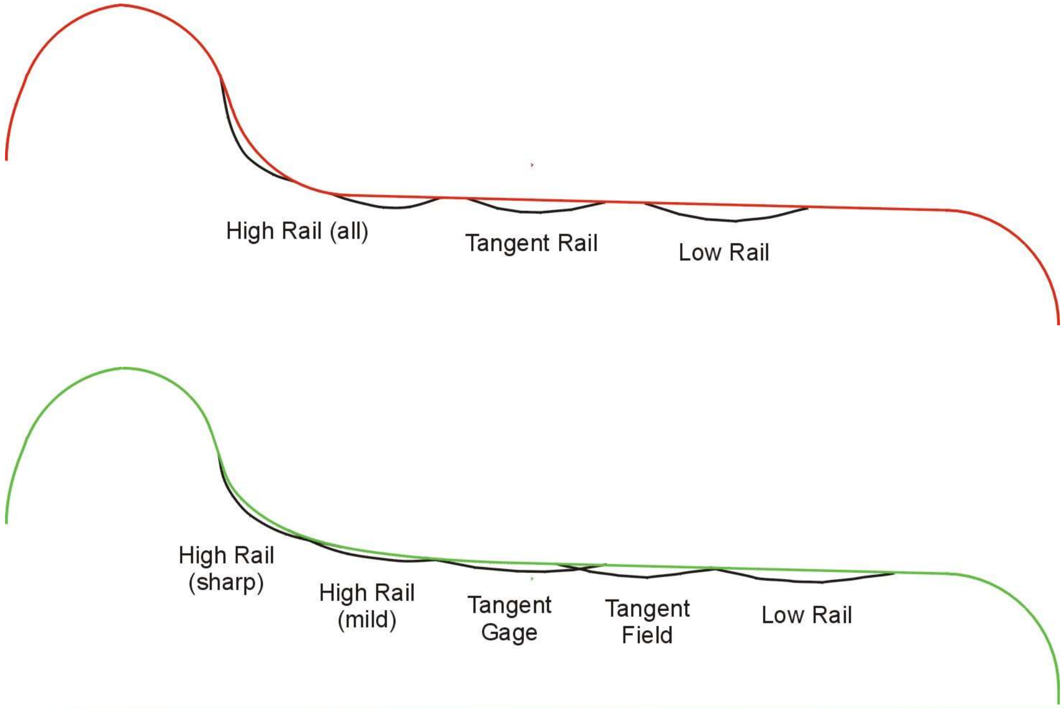
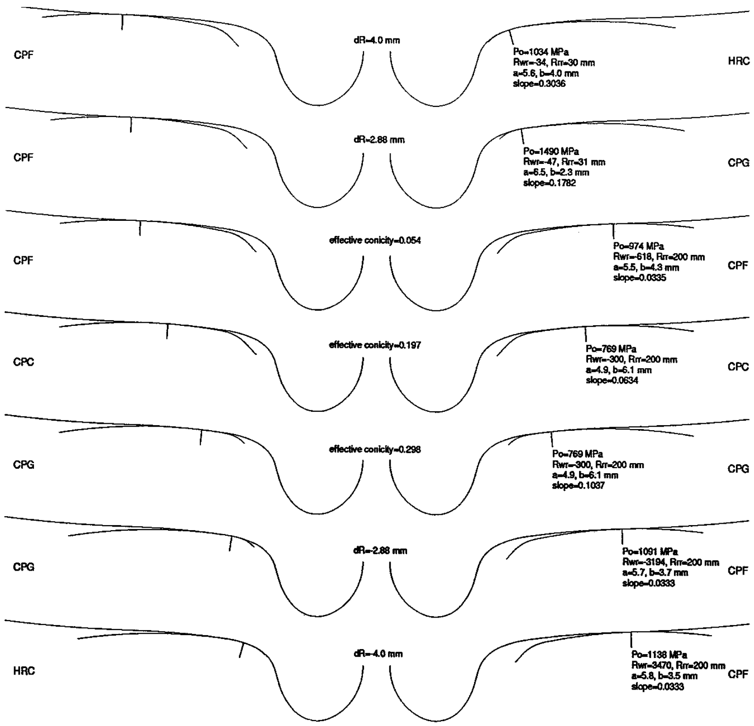


A Family of Rail Profiles

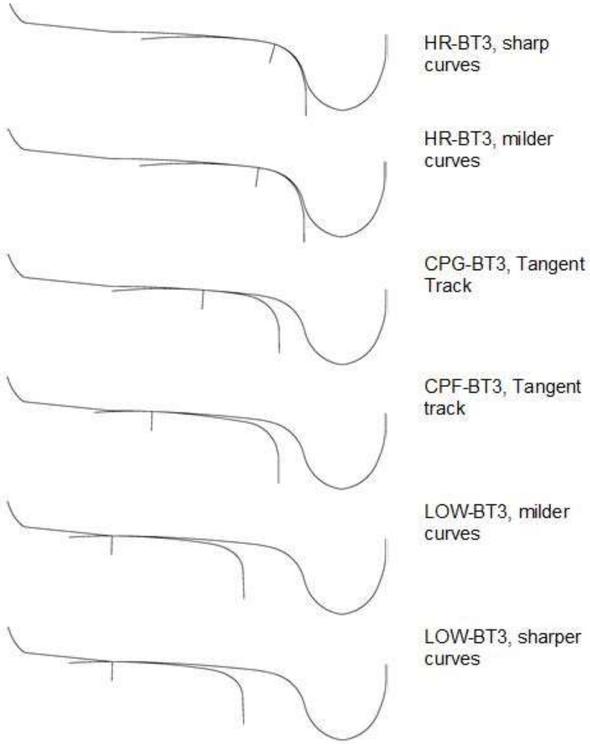


Families of Rail Profiles

Los Angeles



San Francisco



VTI DERAILMENT MECHANISMS AND RISK ASSESSMENT

Wheel climb

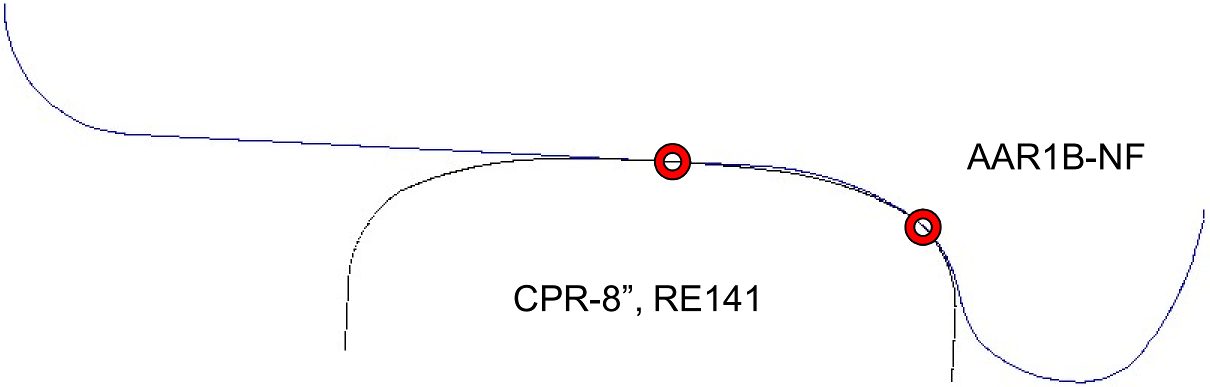
Hunting

Low rail rollover

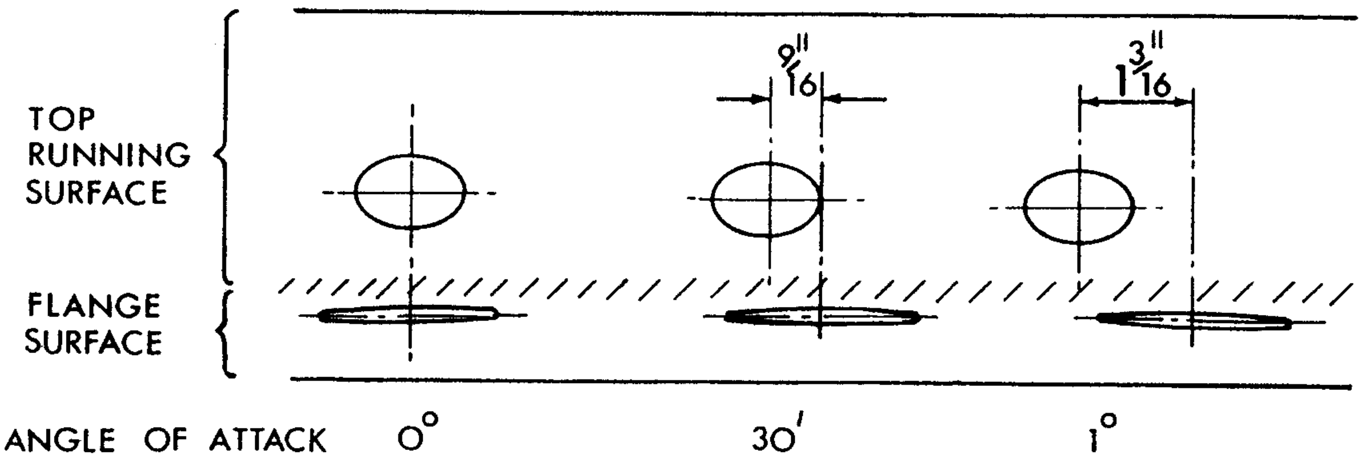
WHEEL CLIMB

Wheel/Rail Contact

- W/R contact often takes place at two points simultaneously (some new wheels especially)

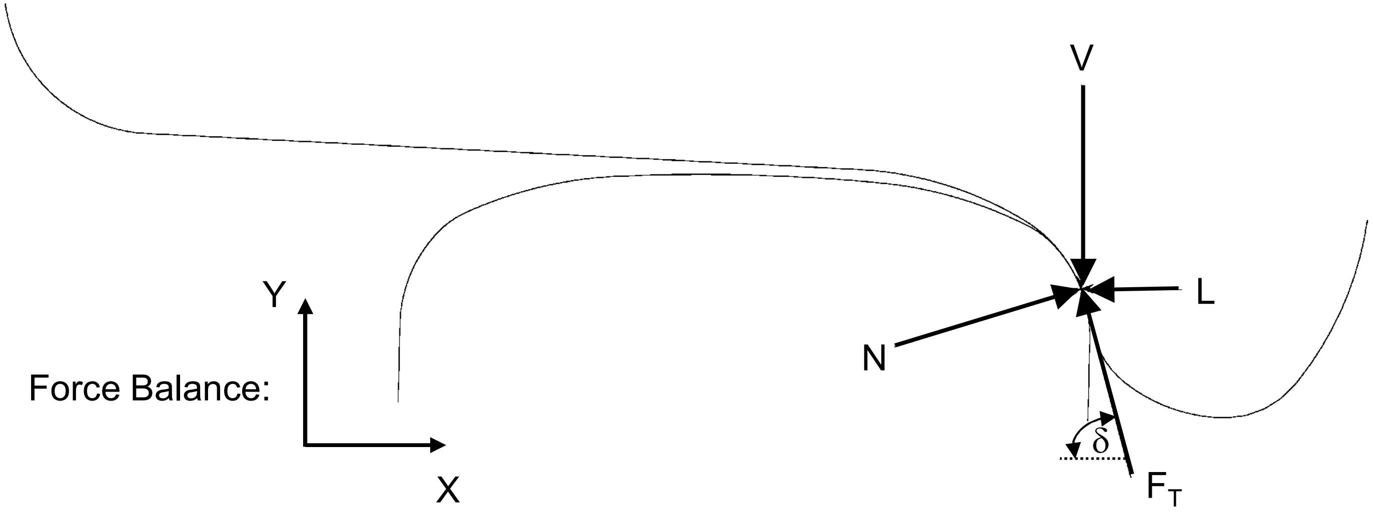


Wheel/Rail Contact (cont’d)



- Plan view of contact ellipses on high rail for different angles of attack

Deriving Nadal

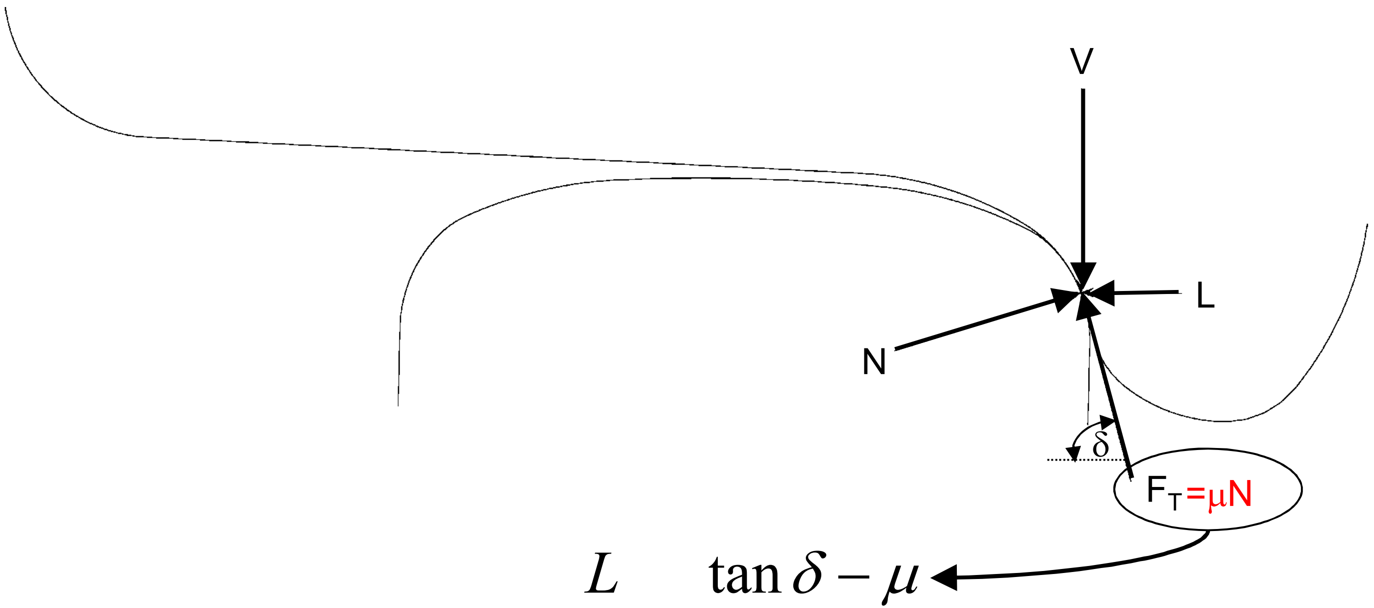


Force Balance:

$$\left.\begin{aligned}\sum Fx = 0 &\Rightarrow N\sin(\delta) - F_T\cos(\delta) - L = 0\\ \sum Fy = 0 &\Rightarrow F_T\sin(\delta) + N\cos(\delta) - V = 0\end{aligned}\right\}$$

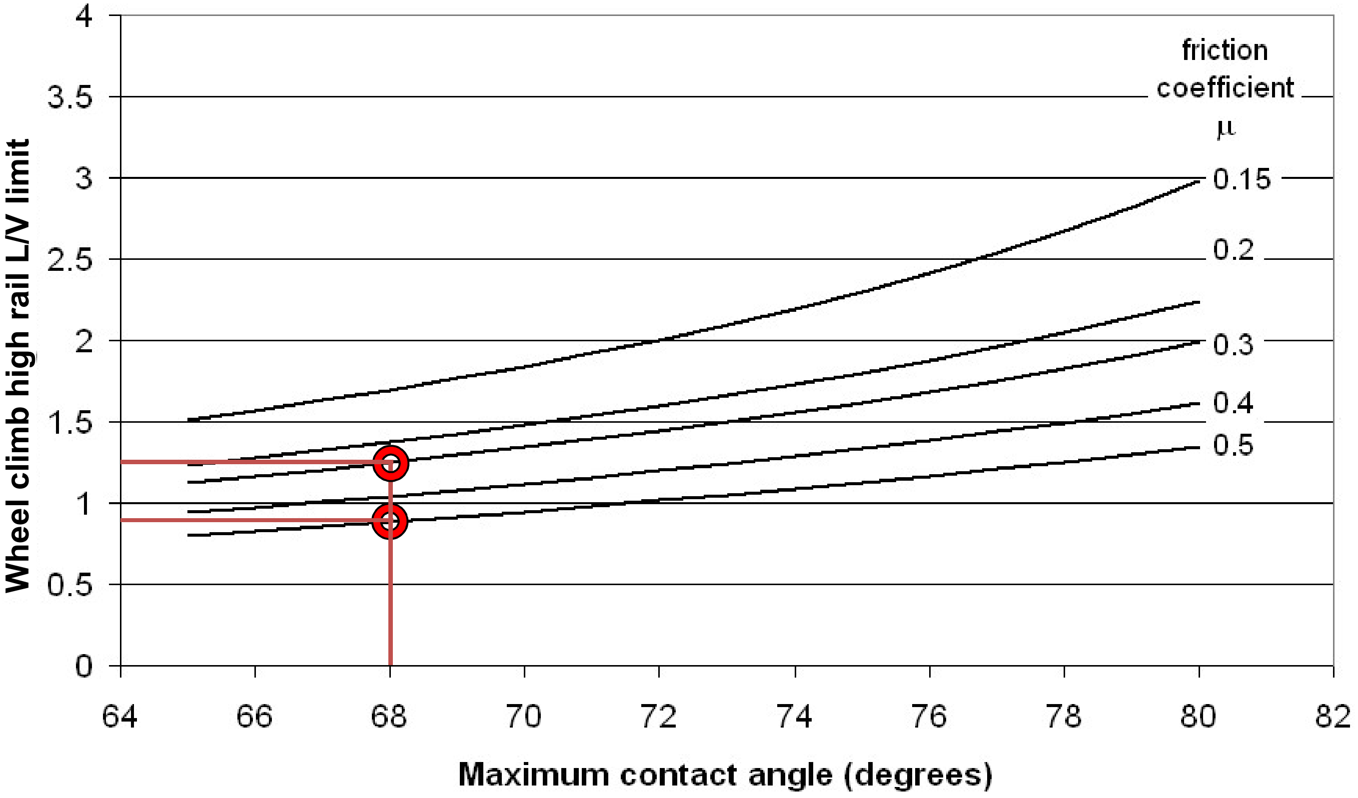
$$\frac{L}{V} = \frac{\tan\delta - F_T / N}{1 + F_T\tan\delta / N}$$

Nadal’s Relationship

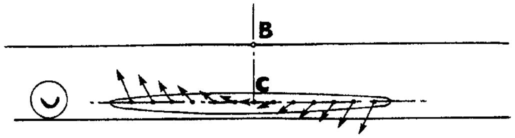


$$\frac{L}{V} = \frac{\tan\delta - \mu}{1 + \mu \tan\delta}$$

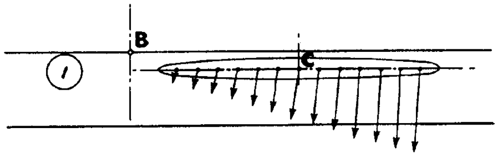
Nadal Index (1908)



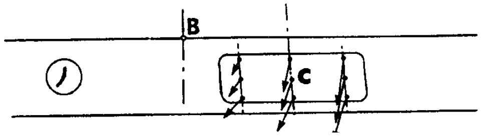
Slip vectors at the gage face contact



δ>β, α=0



δ<β, large α



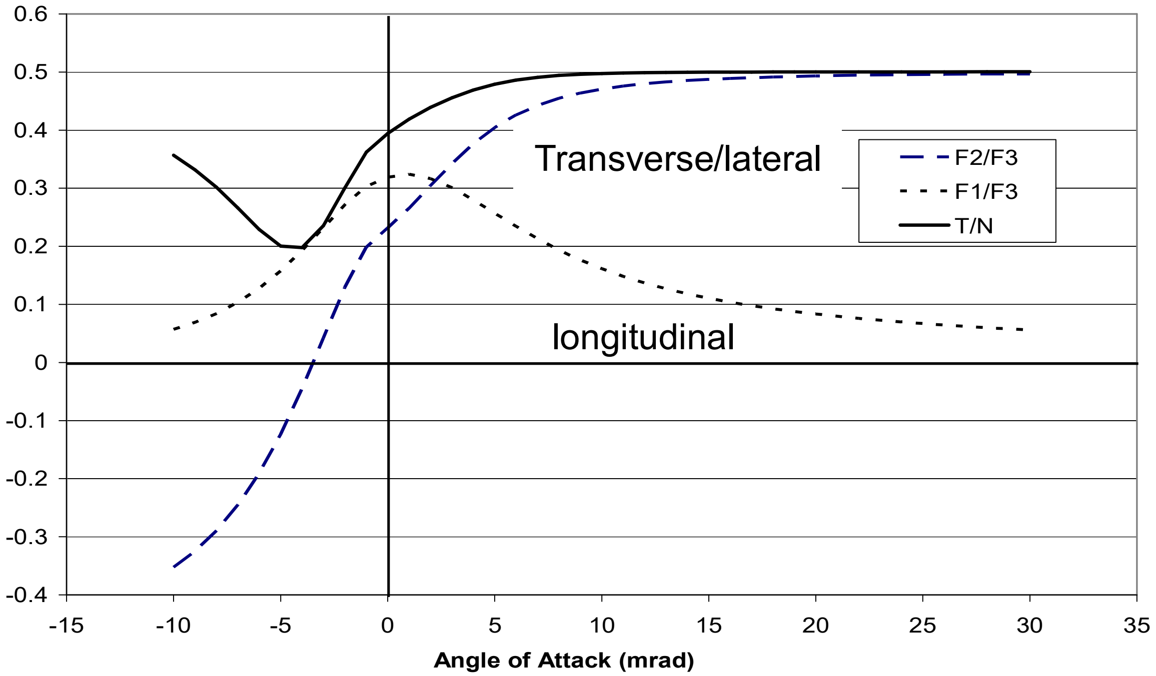
δ=β, moderate α

α = angle of attack
δ = wheel flange angle
β = gage face angle

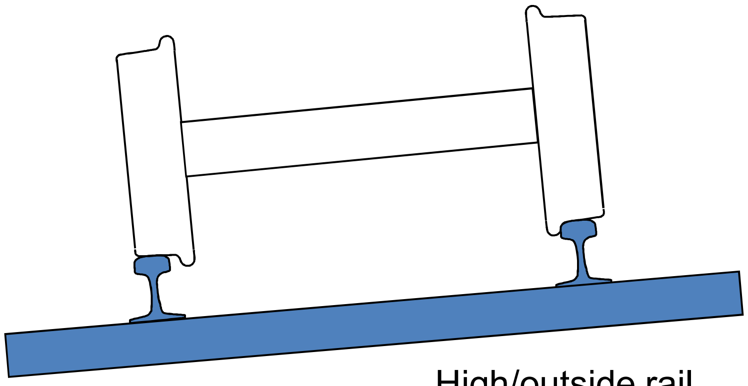
Gilchrist and Brickle

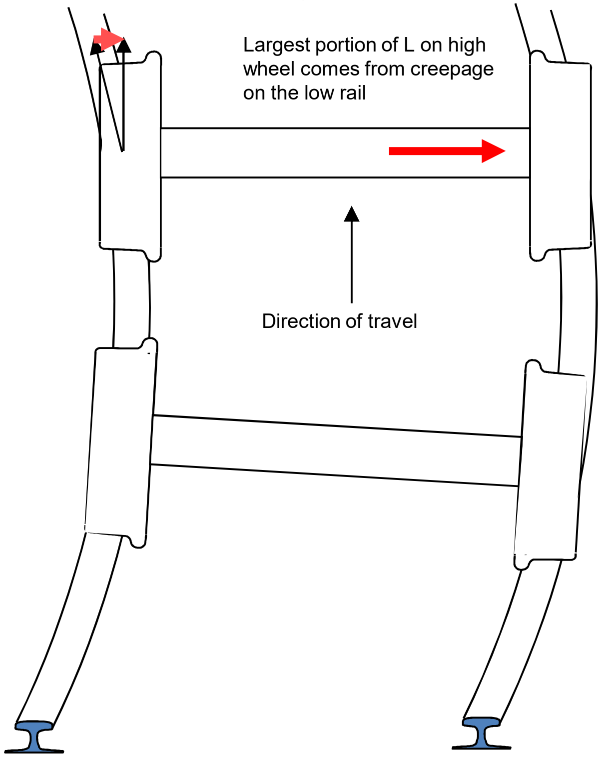
- Nadal is not generally correct, even in steady state conditions since longitudinal creepage is also present at the gage-face wheel/rail contact patch
- Nadal provides a conservative threshold limit (i.e. indicates that the risk is higher than it actually is)
  - Very conservative for small or negative angles of attack
  - Though adequate for more than about 5mrad yaw.
- Considered the out of plane geometry
- Note that the most dangerous condition occurs when there is no longitudinal slip across the contact patch (i.e. the creep force is completely transverse, lifting the wheel)
  - Braking the wheelset in a curve
  - Independently rotating wheelsets

Long/Lateral Slip (single wheelset, δ =63.5°, μ=0.5, a/b=7.5) from Weinstock

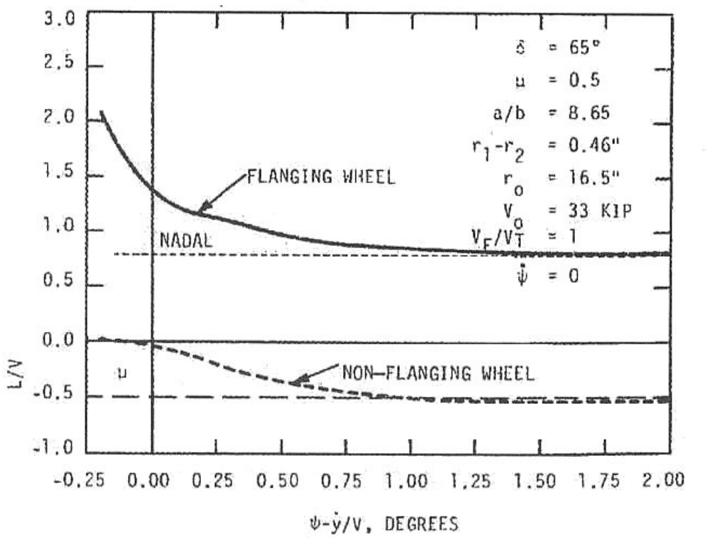


Lateral wheel/rail forces




Largest portion of L on high wheel comes from creepage on the low rail

Weinstock derailment criterion

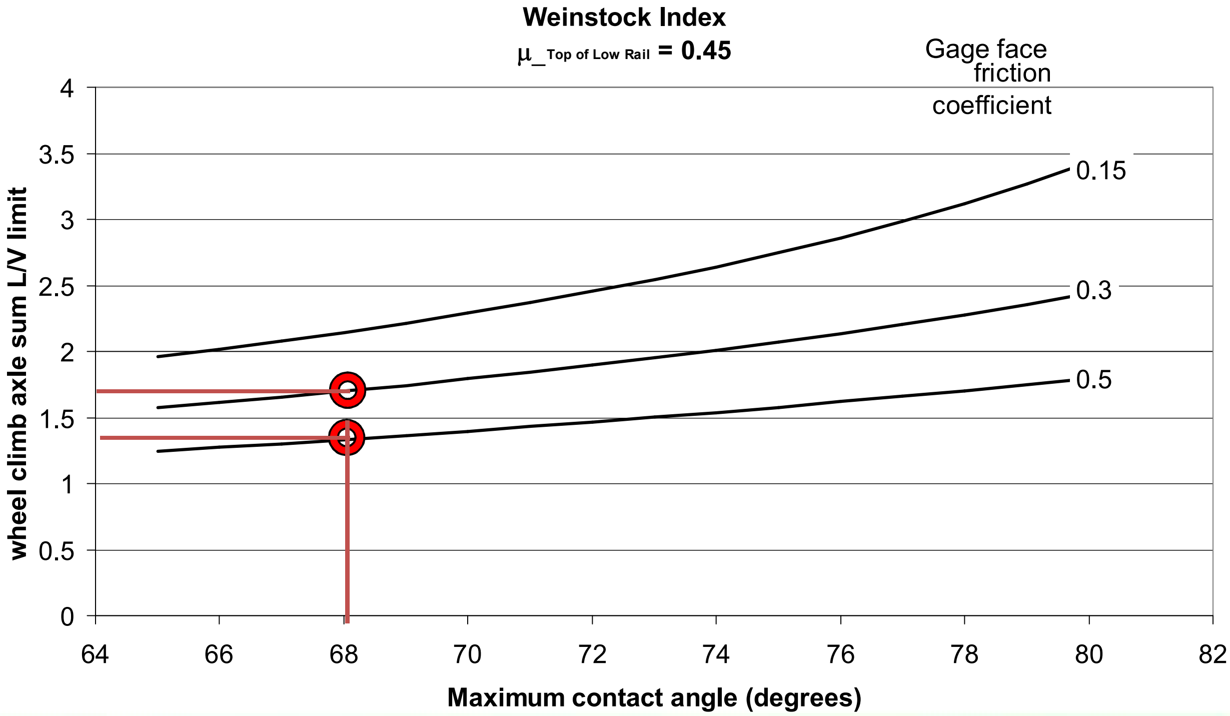


$$|L/V|_{flanging} + |L/V|_{non\_flanging} > (L/V_{NADAL} + \mu)$$

- Holds for all positive angles of attack,
- Less accurate for +ve cant deficiency

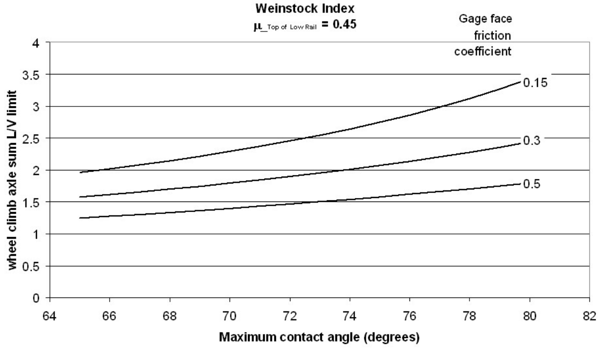
- At incipient wheel climb, the L/V values on the flanging and non-flanging wheels are, for positive angles of attack, separated by a roughly constant value equal to the Nadal limit plus the coefficient of friction on the top of the low rail

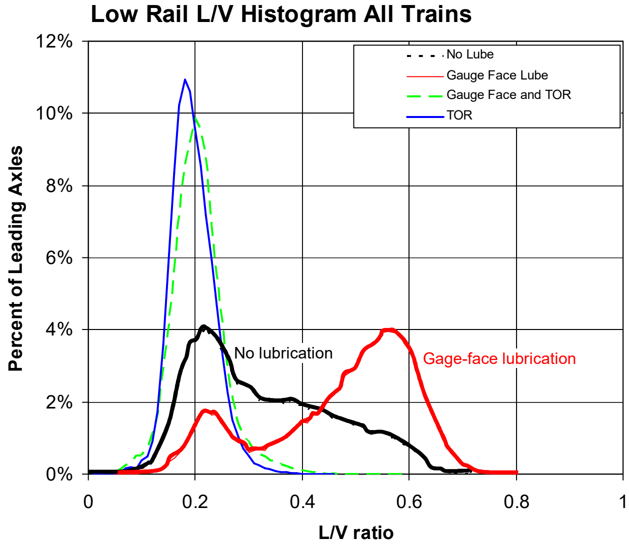
Weinstock Criterion



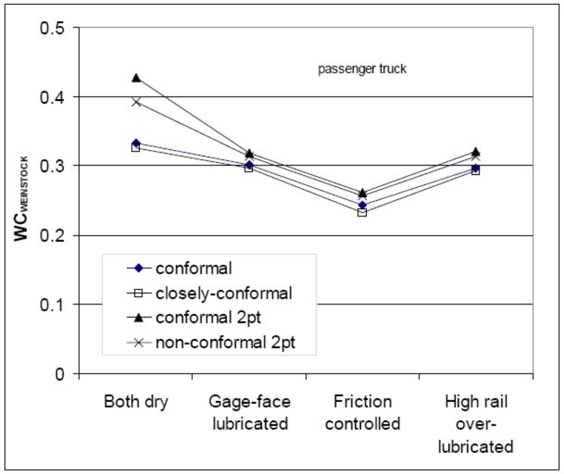
An Example

- Is lubrication a good thing?





L/V goes up, but Weinstock limit also.

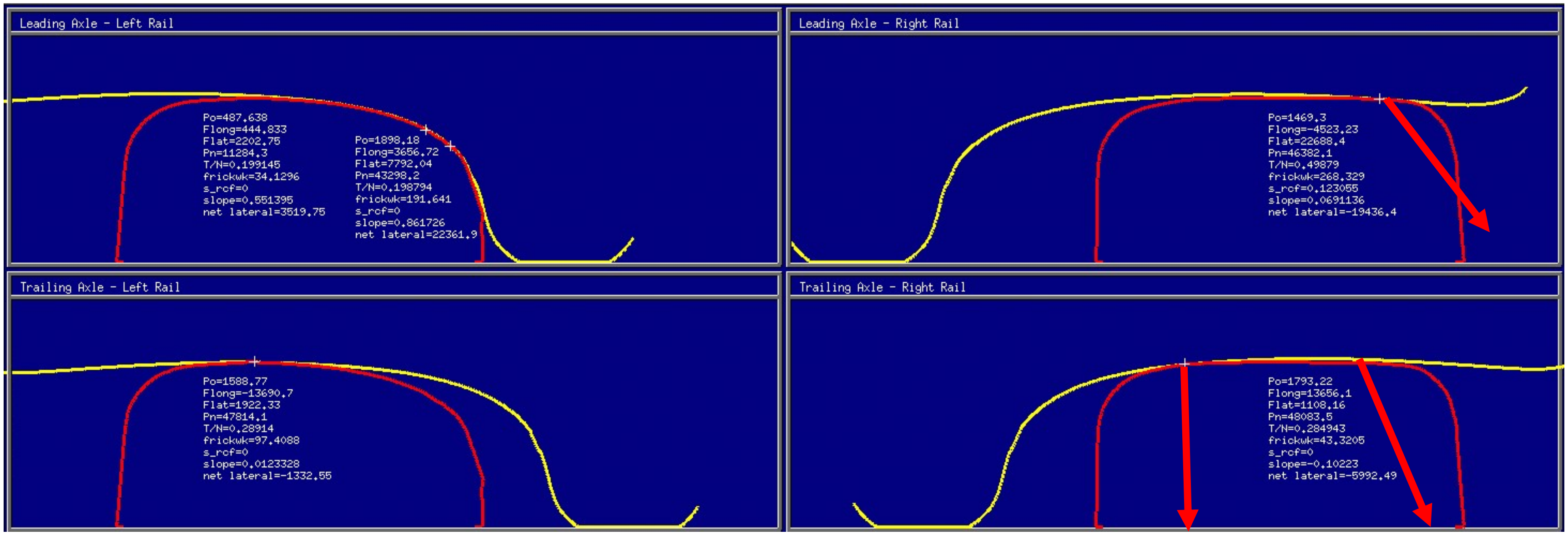


Wheel climb - conclusions

- Nadal – provides a relationship between contact angle and friction coefficient
- Is based upon simplified view of the slip conditions
- Wheel climb threshold matches Nadal at most practical angles of attack, but not for low aoa.
- Weinstock rectifies that (for positive angles of attack) and includes explicitly the effect of friction on top of low rail.
- A safe L/V is some fraction of the (Nadal or Weinstock) threshold value, say 60-80%.
- These are static and quasi-static derivations.

Low rail rollover

- Wide gauge, hollow wheels, poor restraint, underbalanced running, high friction



Leading Axle - Left Rail

```
Po=487.638
Flong=444.833
Flat=2202.75
Pn=11284.3
T/N=0.199145
frickwk=34.1296
s_rcf=0
slope=0.551395
net lateral=3519.75
```

```
Po=1898.18
Flong=3656.72
Flat=7792.04
Pn=43298.2
T/N=0.198794
frickwk=191.641
s_rcf=0
slope=0.861726
net lateral=22361.9
```

Leading Axle - Right Rail

```
Po=1469.3
Flong=-4523.23
Flat=22688.4
Pn=46382.1
T/N=0.49879
frickwk=268.329
s_rcf=0.123055
slope=0.0691136
net lateral=-19436.4
```

Trailing Axle - Left Rail

```
Po=1588.77
Flong=-13690.7
Flat=1922.33
Pn=47814.1
T/N=0.28914
frickwk=97.4088
s_rcf=0
slope=0.0123328
net lateral=-1332.55
```

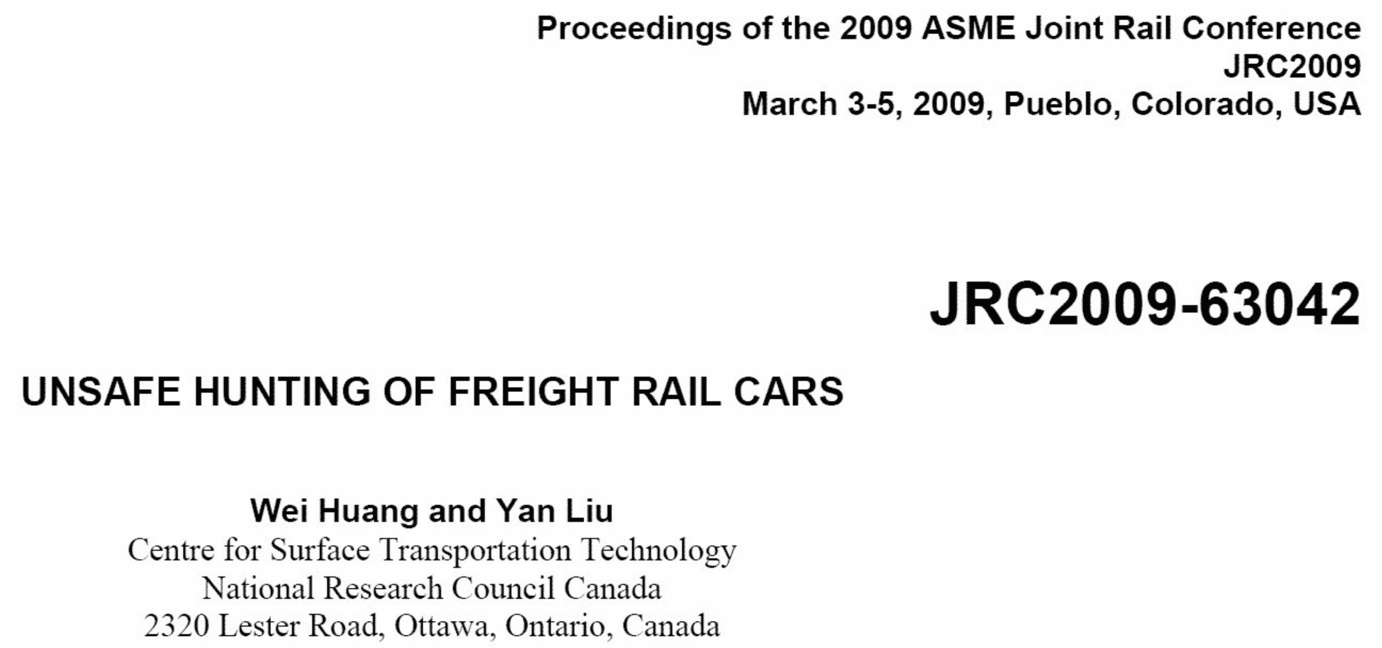
Trailing Axle - Right Rail

```
Po=1793.22
Flong=13656.1
Flat=1108.16
Pn=48083.5
T/N=0.284943
frickwk=43.3205
s_rcf=0
slope=-0.10223
net lateral=-5992.49
```

Hunting

- FRA statistics: E4TC (truck hunting) + E4TL (locomotive hunting)
  - 6 per year prior to 2007
  - Gondolas, tank and covered hopper
  - Class 4 and 5 tracks, 40-60 mph
  - 98% under dry conditions
  - Empty cars mostly (7/8)
- After 2007 – none: AAR Rule 46?
- Hunting truck detectors

Unsafe hunting



Proceedings of the 2009 ASME Joint Rail Conference
JRC2009
March 3-5, 2009, Pueblo, Colorado, USA

JRC2009-63042

UNSAFE HUNTING OF FREIGHT RAIL CARS

Wei Huang and Yan Liu
Centre for Surface Transportation Technology
National Research Council Canada
2320 Lester Road, Ottawa, Ontario, Canada

Unsafe Hunting

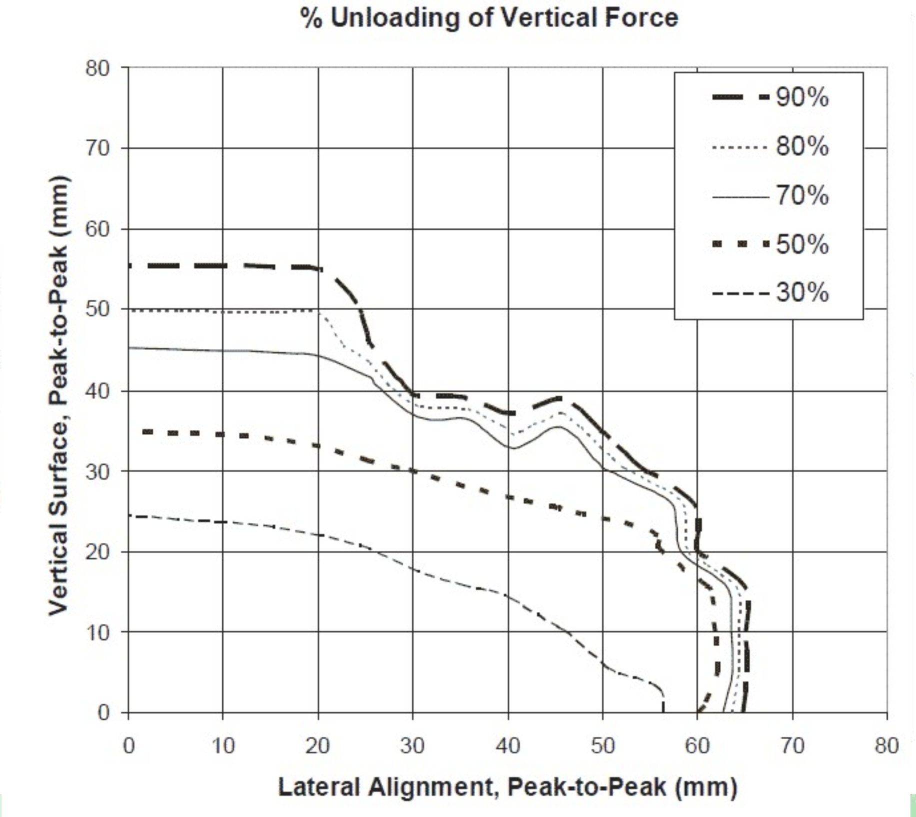
- Criteria:
  - RMS lateral acceleration of car body exceeding 0.13 g sustained
  - L/V greater than Nadal limit
  - Wheel unloading >0.9
- Contributing factors
  - Yaw damping at centre-plate/center-bowl (esp. for loaded car)
  - Truck warp stiffness
  - Track quality (smoother track is more prone to hunting, but derailment more likely on rougher track).

Dynamic and Impact Loads

- Even perfect vehicle/track will have dynamic forces (though typically within 10% of static load).
- Rail irregularities – welds, joints, switches, crushed heads, corrugation, spalling
- Wheel irregularities – out of roundness (ovality), polygonization (corrugation), wheel flats

Track geometry imperfections

- Force exceptions (AAR Ch. 11)
  - L/V>1.0, V/Vs<0.1
- vs Track Geometry
  - IWS: poor correlation e.g. 1 in 8

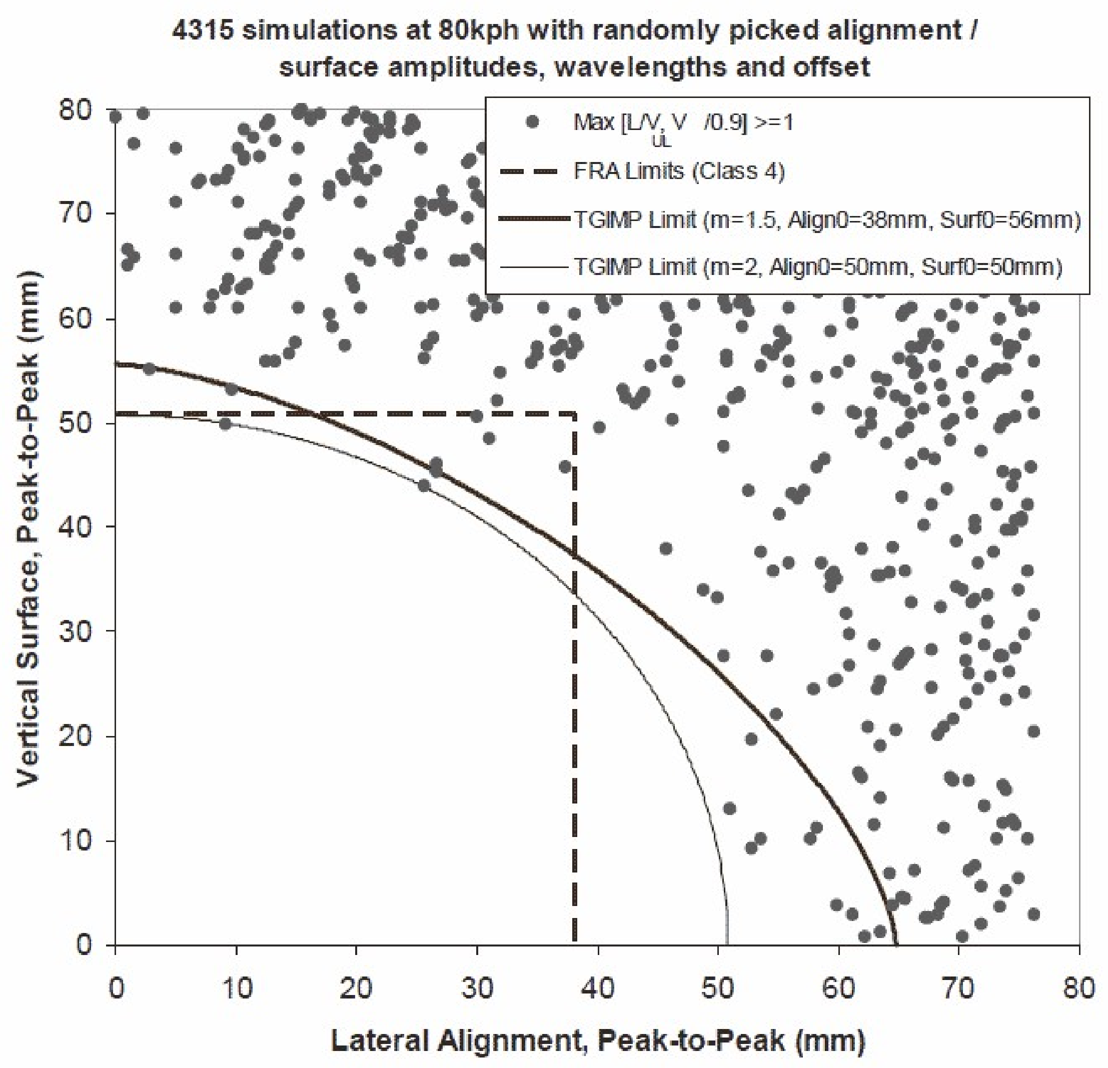


Simulation:

- Covered hopper
- Moderately worn truck
- 80 kph (50 mph)
- μ=0.5
- Unworn AAR1B wheel

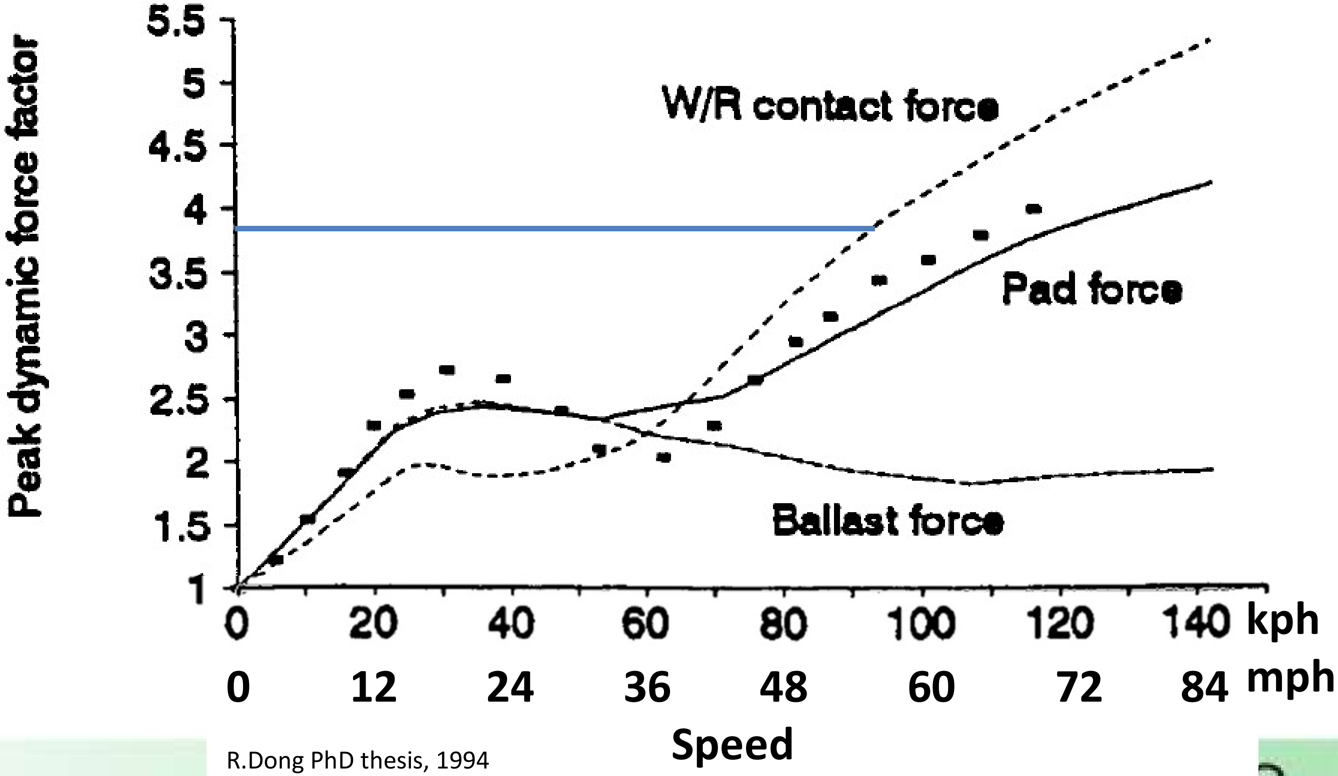
TGIMS

- Current FRA limits are conservative with respect to single geometry errors and do not capture all high risk combinations



Forces due to Wheel flats

Freight cars:
Nominal wheel load: 36kips
Extreme values 140 kips

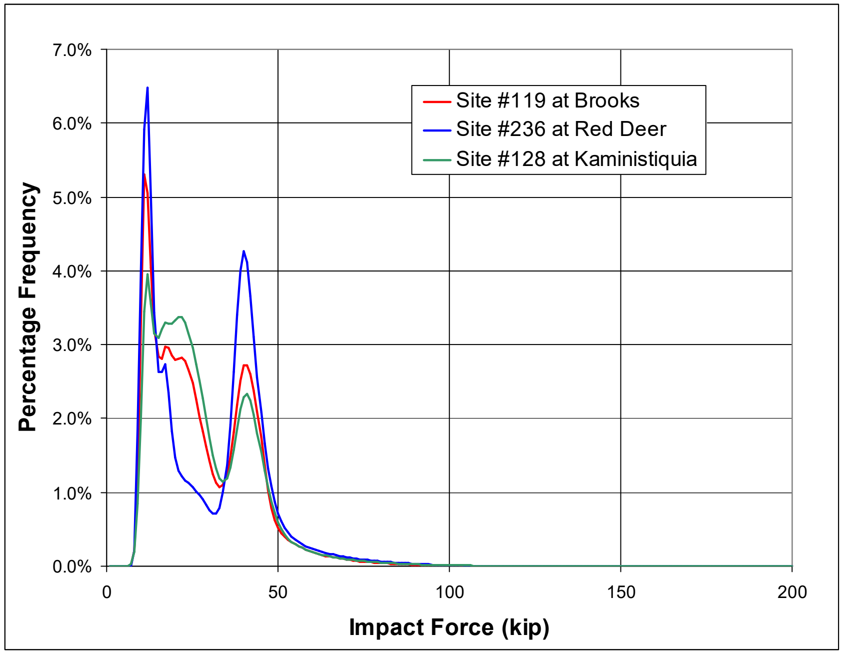


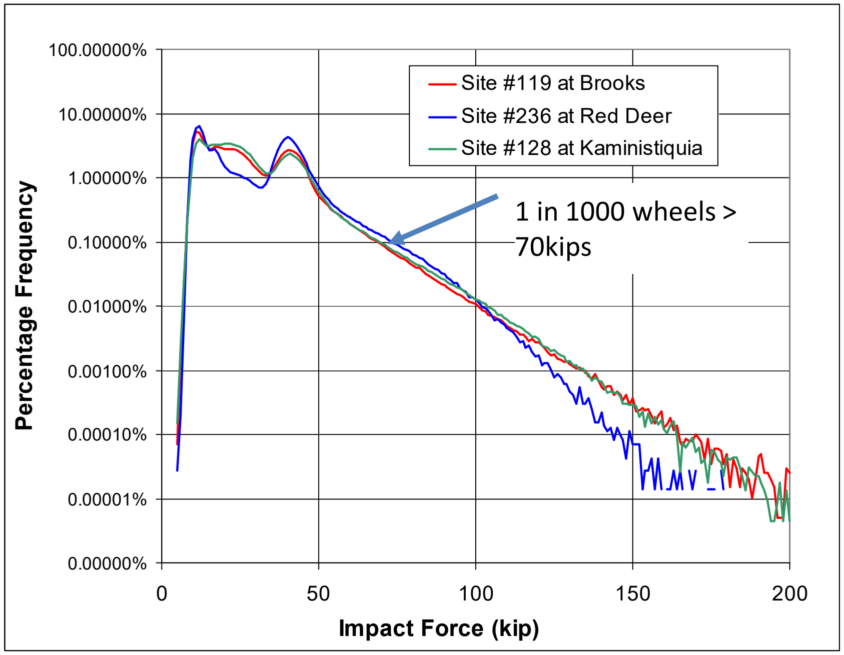
R.Dong PhD thesis, 1994

Heavy haul example: 30 trains/d x 150cars/t x 4 axles/car = 18000 axles/day

Wheel Impact Forces

Liu et al, IHHA 2009





1 in 1000 wheels > 70kips

Conclusions

- Matching of wheel/rail profiles
  - Rolling radius difference: stability and curving
  - Strong impact on stress, curving forces, stability, surface damage, safety/derailment (with friction conditions, truck suspensions, track geometry etc.)
  - Must consider both new and worn shapes (pummeling)
- Nadal formula is adequate for most wheel climb analyses
- Track Geometry and dynamic forces strongly affect safety