NUMERO: **2519A** ANNO: 2021

A P P U N T I

STUDENTE: Sobrero Giovanni

MATERIA: Powertrain Components Design, vol. II,Transmission - Prof. Galvagno

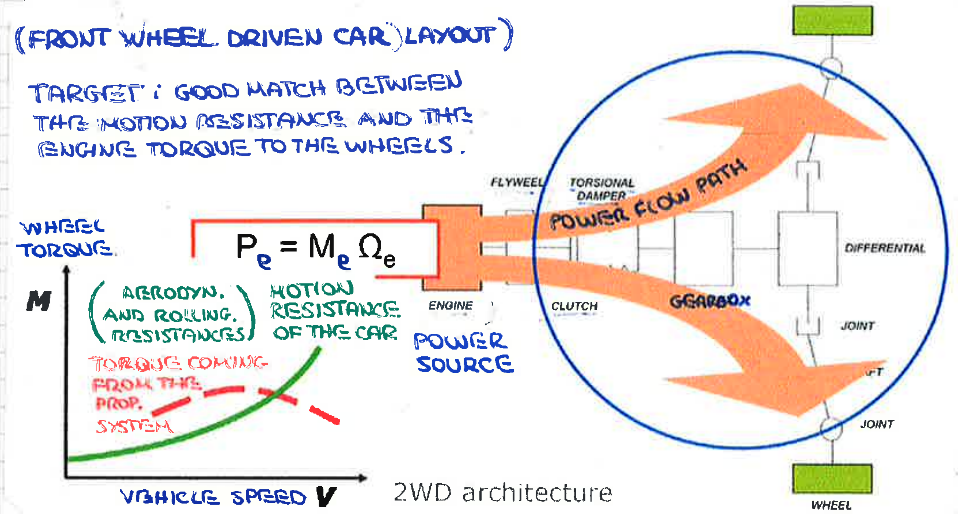
T1 TRANSMISSION FUNCTIONS AND ARCHITECTURES

- AUTOMOTIVE TRANSMISSION CHARACTERISTICS AND EVOLUTION
  - TRANSMISSION FUNCTION
  - GEARBOX FUNCTION
  - INFLUENCE ON VEHICLE PERFORMANCES AND FUNCTIONS
  - REQUIREMENTS AND CONSTRAINTS
  - SPECIFIC CHARACTERISTICS (VS INDUSTRIAL TRANSMISSIONS)
  - TECHNOLOGY
  - CURRENT TRENDS
- TRANSMISSION ARCHITECTURES
- TRANSMISSION COMPONENTS

• AUTOMOTIVE TRANSMISSION CHARACTERISTICS AND EVOLUTION

- TRANSMISSION FUNCTION

• THE MAIN FUNCTION OF A VEHICLE TRANSMISSION IS:
TO TRANSFER TO THE WHEELS THE POWER GENERATED BY THE ENGINE TO MOVE THE VEHICLE, BY MEANS OF PROPER ARCHITECTURES AND COMPONENTS.

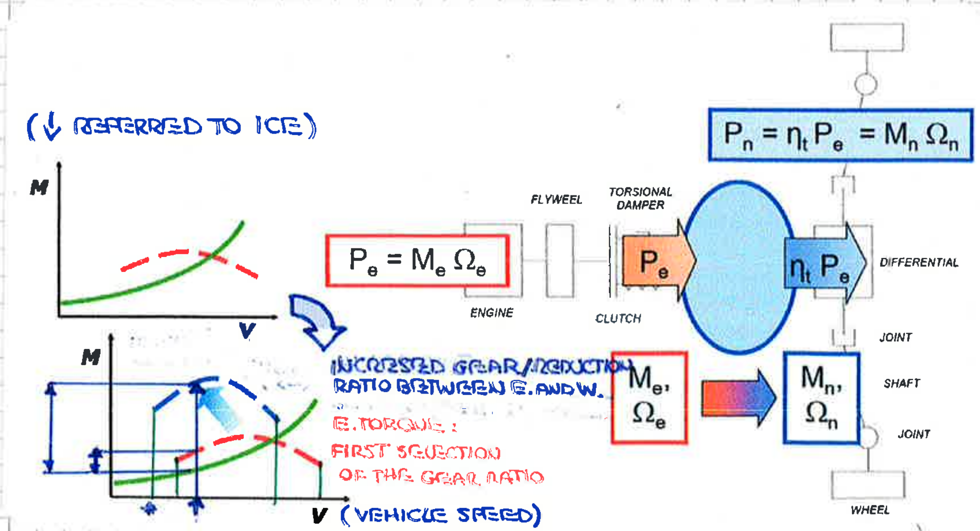


2WD architecture

- ENGINE
- FLYWHEEL → TO REDUCE TORQUE IRREGULARITIES.
1 - CLUTCH
2 - TORSIONAL DAMPER → PROPER ISOLATION FROM E. VIBRATION IN ORDER TO HAVE A LOW LEVEL OF TORSIONAL VIBRATION AT GEARBOX
3 - GEARBOX
4 - DIFFERENTIAL
5 - HALFSHAFTS
6 - JOINTS

- GEARBOX FUNCTION

• THE FUNCTION OF THE GEARBOX IS:
TO ADAPT THE TORQUE/SPEED VALUES TO THE VEHICLE NEEDS, DEFINED BY THE ROAD CHARACTERISTICS, THE DRIVER REQUESTS AND THE ENVIRONMENT LIMITATIONS.



RED SOLUTION IS BETTER IN TERMS OF FUEL CONSUMPTION. WE KEEP ENGINE WORKING AT HIGH LOAD AND (POSSIBLY) AT LOW SPEED TO INCREASE THE FUEL ECONOMY (η ↑)
BLUE SOLUTION HAS AN HIGHER CONSUMPTION

INCREASING THE DISTANCE BETWEEN THE TWO CURVES WE INCREASE THE CAPABILITY OF THE CAR TO OVERCOME THE ROAD SLOPE AND TO ACCELERATE.
HIGH LEVEL OF ACC. ⟷ LOW GEARS ⟷ HIGH REDUCTION RATIO → MORE LIMITED SPEED RANGE
*: ICE IS NOT ABLE TO PROVIDE TORQUE AT ZERO ROT. SPEED (AS INSTEAD E.V.: MAX T AT ZERO V. SPEED)

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 4 di 260

– INFLUENCE ON VEHICLE PERFORMANCES AND FUNCTIONS

• THE TRANSMISSION INFLUENCES THE MOST IMPORTANT VEHICLE PERFORMANCES AND FUNCTIONS: (REDUCTION OF NOISE/VIBRATIONS)

1. DYNAMIC PERFORMANCE (WE CAN IMPROVE V. ACCEL. CHANGING THE GEAR RATIO)
2. FUEL CONSUMPTION AND EMISSIONS (CHANGING THE E. OPERATING POINT)
3. "DRIVEABILITY" AND "COMFORT" ('HOW DRIVER FEELS THE CAR PERFORMANCE IN TERMS OF LONGITUDINAL MOTION OF VEHICLE; "ASSOCIATED WITH GEARSHIFT PHASE: LACK OF POWER ≠ POWERSHIFT TRAN.)
4. RELIABILITY (="AFFIDABILITÀ")

• THEREFORE, THE TRANSMISSION DESIGN MUST TAKE INTO ACCOUNT ALL OF THESE DIFFERENT ASPECTS, OFTEN CONFLICTING, IN ORDER TO FIND THE BEST SOLUTION FOR THE DRIVER AND THE PASSENGERS, FOR THE ENVIRONMENT AND FOR THE MANUFACTURER.

– REQUIREMENTS AND CONSTRAINTS

Integration engine-vehicle-environment (performance/fuel consumption):
- High gear ratios number (→ ∞)
- Reliability
- Minimum power loss (η → 1)

← **Powertrain, Vehicle, Environment**

Interface vehicle-driver (ergonomics, vibro-acoustics and ride comfort):
- Good driveability
- Good shiftability / dynamics
- Low vibration and low noise

← **Customer**

SHIFTABILITY: IS THE PERFORMANCE DURING GEAR SHIFT.

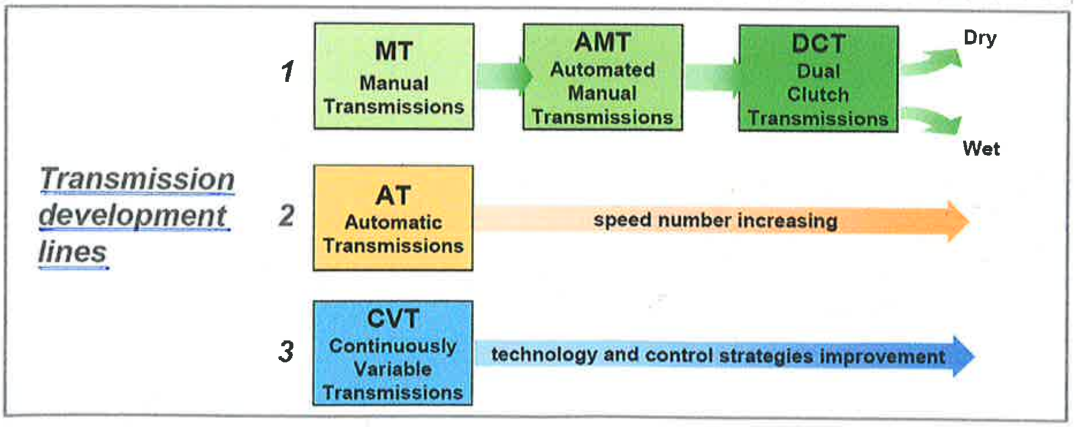
Characteristics requested by the manufacturer:
- Compactness (dimension and weight)
- Simplicity in manufacturing
- Low cost

← **Manufacturer**

$\tau \to \infty$: WE CAN CONTINUOUSLY CHANGE THE OPERATING POINT; MOREOVER, IN THIS WAY, WE CAN LIMIT THE DISCONTINUITY OF THE ACC. WHEN WE SHIFT GEARS.
IF WE INCREASE THE NUMBER OF GEAR RATIOS ⇒ WE INCREASE NUMBER OF COMP.S ⇒ WEIGHT, VOLUME INCREASING; TRANSM. LOSSES INCREASING
(TRADE OFF BETWEEN AN HIGH NUMBER OF GEAR RATIOS AND TRANSMISSION EFF.)

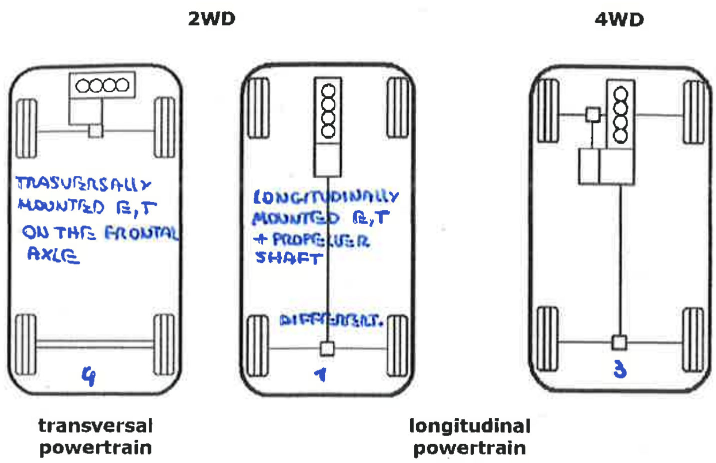
- CURRENT TRENDS

- INCREASING NUMBER OF NON-MT, IN PARTICULAR AMT (AUTOMATISED) AND DT (DUAL CLUTCH), IN EUROPE. [PRODUCT]
- INCREASING NUMBER OF GEAR RATIOS [LAYOUT]:
  - FROM 5 TO 6 (SOMETIMES 7) GEAR RATIOS FOR MT
  - FROM 5 TO 6-8 (SOMETIMES 10) GEAR RATIOS FOR NON-MT
- INCREASING PERFORMANCE AND COMFORT REQUIRENEMENTS FOR BOTH MT AND NON-MT [PERFORMANCES/FUNCTIONS]
  - INCREASING TRANSMITTED NOMINAL TORQUE AND REDUCTION OF VIBRATION/NOISE FOR ALL TYPES;
  - ERGONOMICS FOR MT;
  - DYNAMIC PERFORMANCE, DRIVEABILITY, ENERGY-SAVING SOLUTIONS AND STRATEGIES FOR NON-MT.
  - INCREASING RELIABILITY FOR ALL TYPES (COMPONENTS FOR LIFE)
- INCREASING USE OF ELECTRONIC AND ELECTRIC SYSTEMS FOR NON-MT [TECHNOLOGY]:
  - FROM SIMPLE TO SOPHISTICATED ELECTRONIC SYSTEMS AND COMPONENTS FOR GEARSHIFT CONTROL.
  - IN SOME APPLICATIONS, FROM HYDRAULIC TO ELECTRIC SYSTEMS FOR GEARSHIFT ACTUATION.

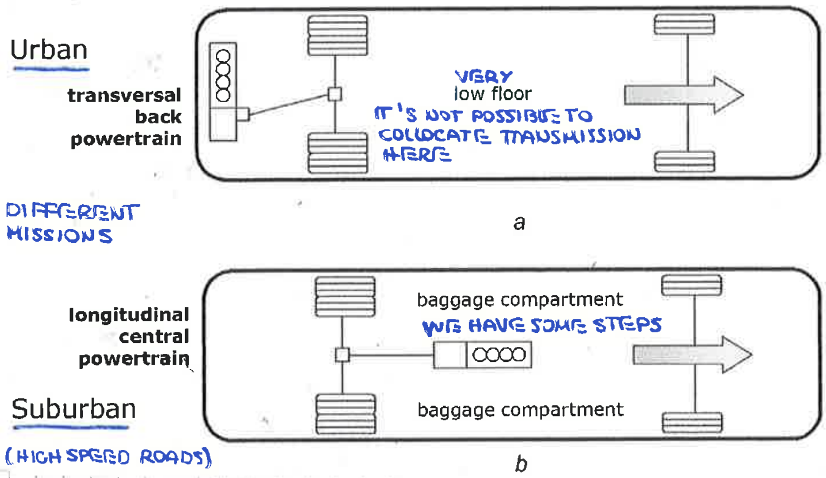


© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 7 di 260

Two axles industrial vehicles (GVW < 4t)



Buses transmission layouts



FOR THE URBAN BUSES WE USE A TRANSVERSAL BACK POWERTRAIN (BEHIND THE REAR AXLE). THIS CONF. ALLOWS EASIER PASSENGER ENTRY/EXIT.

FOR SUBURBAN BUSES WE USE A LONGITUDINAL CENTRAL POWERTRAIN => BETTER WEIGHT DISTRIBUTION

FOR BOTH CASES THE POWERTRAIN IS ON THE REAR AXLE.

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 9 di 260

T2 GEAR RATIOS DEFINITION

- RESISTANCE TO MOTION
  - RESISTANCES TO MOTION (R_R, R_a, R_w)
  - TOTAL RESISTANCE TO MOTION / ROAD LOAD (R)
  - POWER NEEDED FOR MOTION
  - GEAR RATIO DEFINITION
- TOP GEAR RATIO
  - TOP GEAR RATIO DEFINITION (MAX SPEED)
- BOTTOM GEAR RATIO
  - BOTTOM GEAR RATIO DEFINITION (MAX SLOPE)
  - BOTTOM GEAR RATIO COMPUTATION
  - BOTTOM GEAR RATIO CHOICE CRITERIA
- INTERMEDIATE GEAR RATIOS
  - INTERMEDIATE GEAR RATIOS DEFINITION
  - CHART FOR GEAR RATIOS DEFINITION
  - PRACTICAL GEAR RATIOS DEFINITION
  - EXAMPLES OF GEAR RATIOS DEFINITION
- VEHICLE ACCELERATION
  - EQUIVALENT MASS OF THE VEHICLE
  - AVAILABLE POWER FOR THE VEHICLE ACCELERATION
  - MAX ACCELERATION VS SPEED
  - OPTIMUM SPEEDS FOR GEAR SHIFTING
  - CRITERION FOR CHOOSING THE GEAR RATIOS
  - SPEED-TIME CURVE AT MAXIMUM POWER
- FUEL CONSUMPTION AT CONSTANT SPEED
  - EXAMPLE OF FUEL CONSUMPTION COMPUTATION
- FUEL CONSUMPTION IN ACTUAL DRIVING CONDITIONS
  - ASSUMPTIONS AND SIMPLIFICATIONS
  - NEW EUROPEAN DRIVING CYCLE (NEDC, UDC + EUDC)
  - WORLDWIDE HARMONIZED LIGHT VEHICLES TEST PROCEDURE (WLTP)
  - WLTP VS. NEDC
  - FUEL CONSUMPTION DURING THE CYCLE
  - MAP OF A DIRECT INJECTION DIESEL I.C. ENGINE

[TEXTBOOK: VOL II, PAR. 23.2, 23.3, 23.6, 23.7, 23.8, 23.10, 23.11; 22.2]

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 11 di 260

- TOTAL RESISTANCE TO MOTION (/ ROAD LOAD) (R)

$$R = R_R + R_a + R_W = \left[mg\cos(\alpha) - \frac{1}{2}\rho V^2 S C_z\right](f_0 + KV^2) + \frac{1}{2}\rho V^2 S C_x + mg\sin(\alpha)$$

WHERE, **ASSUMING** THAT THE **AIR IS STILL** ("FERMA"), THE VELOCITY WITH RESPECT TO THE AIR (**V_R**) BECOMES **CONFLATED** WITH VELOCITY (**V**).

TO HIGHLIGHT ITS (R) DEPENDENCE ON SPEED (V), THE **ROAD LOAD (R)** CAN BE WRITTEN AS:

$R = A + BV^2 + CV^4$ 4 DEGREE POLYNOMIAL (LAST TERM CAN BE NEGLECTED) NEGLIGIBLE AERODYNAMIC LIFT

WITH:

$A = mg[f_0\cos(\alpha) + \sin(\alpha)]$ $\rightarrow$ $A = mg[f_0 + i]$; **i : GRADE OF THE ROAD**

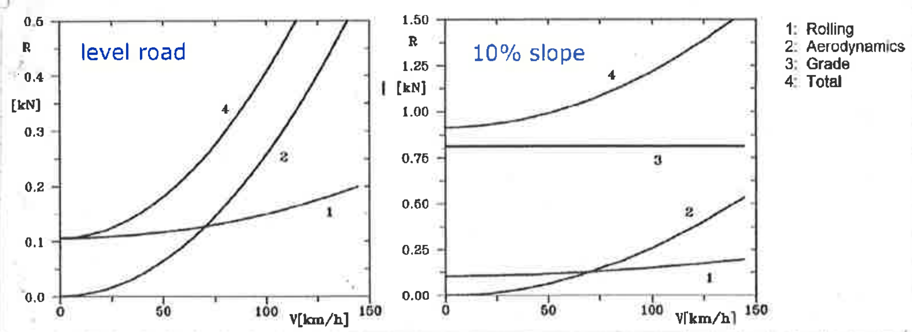
$B = mgK\cos(\alpha) + \frac{1}{2}\rho S[C_x - C_z f_0]$ $\rightarrow$ $B = mgK + \frac{1}{2}\rho S[C_x - C_z f_0]$

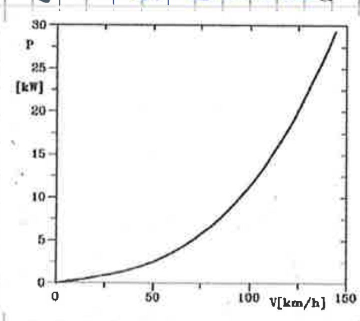
$C = -\frac{1}{2}\rho S K C_z$ (NOTE: **LEVEL ROAD $\Leftrightarrow$ $\alpha = 0$**)

NOTE: THE LAST **TERM (CV^4)** BECOMES **IMPORTANT ONLY AT VERY HIGH SPEED** IN THE CASE OF VEHICLES WITH **STRONG NEGATIVE LIFT**: IT IS USUALLY NEGLECTED EXCEPT IN RACING CARS.

NOTE: SINCE THE GRADE ANGLE OF ROADS OPEN TO VEHICULAR TRAFFIC IS USUALLY NOT VERY LARGE, IT IS **POSSIBLE TO ASSUME** THAT: **$\cos(\alpha) \approx 1$** ; $\sin(\alpha) \approx \tan(\alpha) \approx i$ WHERE i IS THE GRADE OF THE ROAD. IN THIS CASE COEFF. B IS INDEPENDENT OF THE GRADE OF THE ROAD AND: $A \approx mg(f_0 + i)$ DEPENDS LINEARLY ON IT. C NEVER DEPENDS ON GRADE.

[SMALL CAR EXAMPLE]





- POWER NEEDED FOR MOTION

THE **POWER NEEDED TO MOVE** THE VEHICLE **AT CONSTANT SPEED** (V=CONST) IS:

$P_m = VR = AV + BV^3 + CV^5$ (NEEDED / **REQUIRED BY** THE **WHEELS**)

(5 DEGREE POLYNOMIAL)

- GEAR RATIO DEFINITION

$\tau = \frac{\omega_{OUT}}{\omega_{IN}} = \frac{z_{p(in)}}{z_{g(out)}}$; **z : GEAR NUMBER OF TEETH**

SINGLE STAGE : 1 COUPLE OF GEARS $\begin{cases} \text{INPUT GEAR : PINION} \\ \text{OUTPUT GEAR : GEAR} \end{cases}$

$\tau < 1$ ALWAYS (OUR CONVENTION)

$i = \frac{1}{\tau} = \frac{\omega_{IN}}{\omega_{OUT}} = \frac{z_{p(OUT)}}{z_{p(IN)}}$ (WIDER RANGE OF VALUES)

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 13 di 260

- IF THE TRANSMISSION IS OF MECHANICAL TYPE, THE OVERALL GEAR RATIO (τ_t) IS THE PRODUCT OF THE GEAR RATIO AT THE GEARBOX (IN THE RELEVANT GEAR) (τ_g) AND THE GEAR RATIO AT THE FINAL DRIVE (τ_f)

$\tau_t = \tau_g \tau_f$; $V = R_e \Omega_e \tau_t$

MAXIMUM VEHICLE SPEED (V_{MAX}) IS NORMALLY OBTAINED USING THE HIGHEST GEAR RATIO (m), SOMETIMES USING ($m-1$) GEAR RATIO.

IF THE MAXIMUM VEHICLE SPEED (V_{MAX}) IS OBTAINED BY THE HIGHEST GEAR RATIO (τ_t'), THE INTERSECTION BETWEEN THE AVAILABLE POWER CURVE AT WHEELS (C_1) AND THE REQUIRED POWER CURVE (C_2), DEFINES THE PRODUCT OF THIS GEARBOX RATIO ($\tau_{g,m}$) BY THE FINAL DRIVE RATIO (τ_f)

$\tau_t' = \tau_{t,V_{MAX}} = \tau_{g,m} \cdot \tau_f$ (IN TOP GEAR USUALLY $\tau_{g,m} \cong 1$)

- THE GEARBOX RATIO ($\tau_{g,m}$) IS INFLUENCED BY THE TRANSMISSION LAYOUT:
  - IN THE COUNTERSHAFT TRANSMISSIONS NORMALLY: $\tau_{g,m} = 1$
  - IN THE SINGLE STAGE TRANSMISSIONS NORMALLY: $\tau_{g,m} > 1$

    (IT CAN REACH $\tau_{g,m} \approx 1.5$ IF THE GOAL IS TO REDUCE FUEL CONSUMPTION)
- THEN THE FINAL DRIVE RATIO (τ_f) CAN BE COMPUTED ($\tau_f \ll 1$) (CONST).

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 15 di 260

MAXIMUM GRADE / SLOPE i_{MAX} :

- IT CAN BE OBTAINED BY PLOTTING THE CURVES OF THE REQUIRED POWER (P_m) AT VARIOUS VALUES OF THE SLOPE (i) AND LOOKING FOR THE CURVE THAT IS TANGENT TO THE CURVE OF THE AVAILABLE POWER (P_a).
- THIS CONDITION IS HOWEVER **ONLY THEORETICAL**, SINCE IT CAN BE **MANAGED ONLY AT A SINGLE VALUE OF THE SPEED** :
  IF THE VEHICLE TRAVELS AT A **HIGHER SPEED**, IT SLOWS DOWN BECAUSE THE **POWER** (P_a) IS **NOT SUFFICIENT** ($P_a < P_m$), BUT **IF** ITS **SPEED** IS **REDUCED** THE **POWER** IS **INSUFFICIENT** AND THE **VEHICLE SLOWS DOWN FURTHER**.
  THE CONDITION IS THEREFORE **UNSTABLE** AND THE **VEHICLE STOPS**.
- IT IS THEN **NECESSARY** TO HAVE A **LARGE INTERSECTION** OF THE TWO CURVES, **NOT A TANGENCY** : THE CURVE OF THE **AVAILABLE POWER** (P_a) MUST BE **ABOVE** THAT OF THE **REQUIRED POWER** (P_m) **IN A WHOLE RANGE OF SPEEDS**, STARTING FROM A VALUE LOW ENOUGH TO ASSURE THAT STARTING ON THAT SLOPE IS POSSIBLE.

• INTERMEDIATE GEAR RATIOS

- INTERMEDIATE GEAR RATIOS DEFINITION

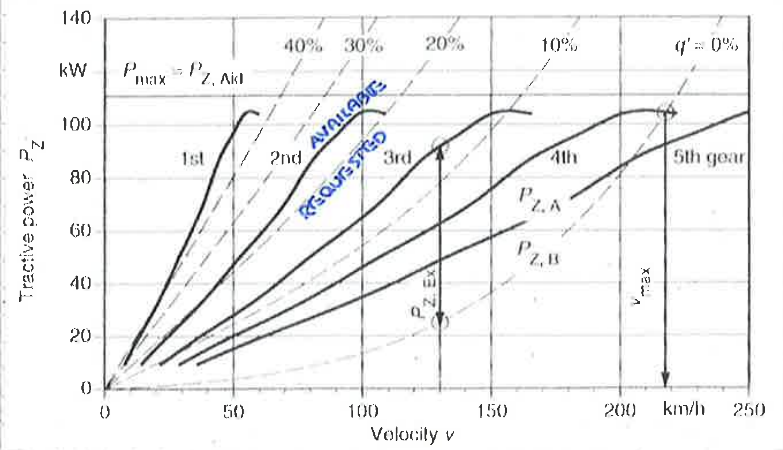
- THE **INTERMEDIATE GEAR RATIOS** BETWEEN THE TOP AND THE BOTTOM GEAR CAN BE STATED USING **DIFFERENT CRITERIA**:
- THEY CAN FOLLOW A "**GEOMETRIC PROGRESSION**"; (φ: RATIO OF GEOM. PROGR.) DEFINING: (FIRST CRITERION)

$\varphi = (\tau_{g,m} / \tau_{g,1})^{(1/m-1)}$: **RATIO BY TWO SUBSEQUENT GEAR RATIOS**

WHERE m: ORDER NUMBER OF THE MAXIMUM SPEED RATIO

- **THIS CONDITION** MAKES **BIGGEST** THE **AREA** BETWEEN THE **CURVE OF AVAILABLE POWER** (P_a) AND THE **CURVE OF REQUESTED POWER** (P_m), WITH THE **ASSUMPTION** THAT THE **TRANSMISSION EFFICIENCY** (η_t) IS **CONSTANT** FOR ALL THE **GEAR RATIOS**.
- OPERATING IN THIS WAY, THE AVAILABLE POWER (P_a) **CURVES** ON THE **P(V) LOGARITHMIC PLOT** ARE **ALL EQUISPACED**.

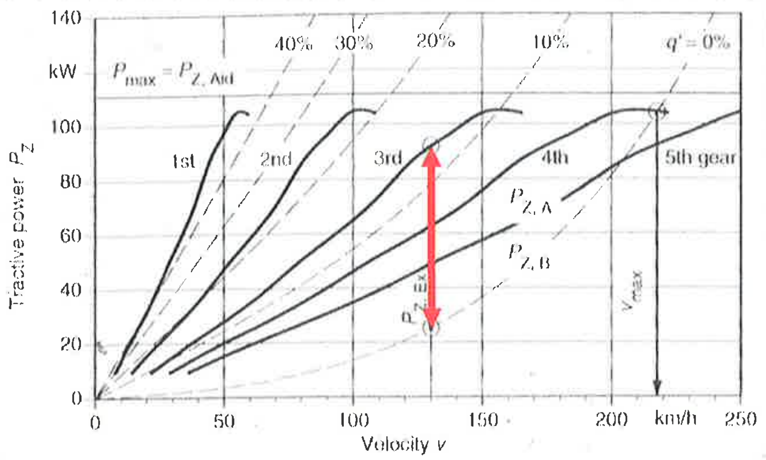
- CHART OF THE GEAR RATIOS DEFINITION



(HIGHER THE NUMBER OF GEAR RATIOS, BETTER THE OPTIMIZATION OF THE OPERATING POINT OF THE ENGINE)

VEHICLE ACCELERATION

- IF THE CURVE OF THE REQUIRED POWER (P_m) LIES, AT A CERTAIN SPEED, BELOW THAT OF THE POWER AVAILABLE AT THE WHEELS (P_a), THE DIFFERENCE ($P_a - P_m$) BETWEEN THE TWO CURVES IS THE POWER AVAILABLE TO ACCELERATE THE VEHICLE.



– EQUIVALENT MASS OF THE VEHICLE (m_e)

- THE KINETIC ENERGY OF THE VEHICLE (T) IS:

$$T = \frac{1}{2} m V^2 + \frac{1}{2} \sum_{\forall i} J_i \Omega_i^2 = \frac{1}{2} m_e V^2$$ (SUM OF TRANSLATIONAL PART AND ROT. PARTS CONTR.)

- THE EQUIVALENT / APPARENT MASS (m_e) OF THE VEHICLE IS THE MASS OF AN OBJECT THAT, WHEN MOVING AT THE SAME SPEED AS THE VEHICLE, HAS THE SAME TOTAL KINETIC ENERGY (T):

$$m_e = m + \frac{J_w}{R_e^2} + \frac{J_t}{R_e^2 \tau_f^2} + \frac{J_e}{R_e^2 \tau_f^2 \tau_g^2}$$

- WHERE, J_w IS THE TOTAL MOMENT OF INERTIA OF THE WHEELS, WHICH ARE ASSUMED TO HAVE THE SAME RADIUS AND HENCE TO ROTATE AT THE SAME SPEED, AND OF ALL ELEMENTS ROTATING AT THEIR SPEED.

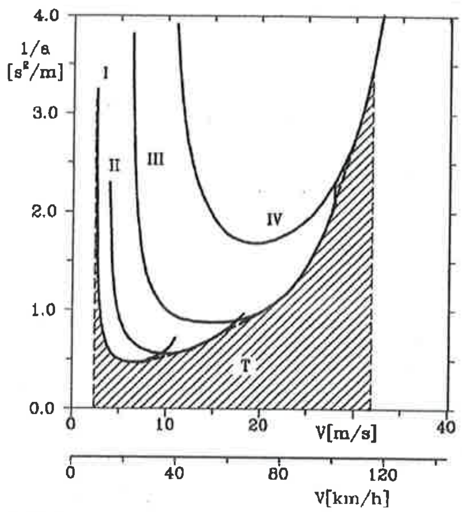
  J_t IS THE MOMENT OF INERTIA OF THE PROPELLER SHAFT AND OF ALL ELEMENTS OF THE TRANSMISSION.

  J_e IS THE MOMENT OF INERTIA OF THE ENGINE, THE CLUTCH AND ALL THE ELEMENTS ROTATING AT SPEED Ω_e.

- NOTE: TO ACCOUNT FOR THE FACT THAT THE ENGINE IS ACCELERATED DIRECTLY, AT LEAST IN AN APPROXIMATED WAY, THE LAST TERM IS SOMETIMES MULTIPLIED BY η_t.

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 21 di 260

– OPTIMUM SPEEDS FOR GEAR SHIFTING



- THE AREA UNDER THE CURVE

$$\frac{V m_e}{\eta_t P_e - P_m} = \frac{1}{a}$$

VERSUS V IS THE TIME REQUIRED FOR THE ACCELERATION.

- THE SPEEDS AT WHICH GEAR SHIFTING MUST OCCUR TO MINIMIZE ACCELERATION TIME ARE READILY IDENTIFIED ON THE PLOT $1/a(V)$:
- THE AREA MUST BE MINIMIZED AND GEARS MUST BE SHIFTED AT THE INTERSECTION OF THE VARIOUS CURVES.
- IF THEY DO NOT INTERSECT, THE SHORTER GEAR MUST BE USED UP TO THE MAXIMUM ENGINE SPEED.

– A CRITERION FOR CHOOSING THE GEAR RATIOS

- IF THE EQUIVALENT MASS CHANGES BETWEEN TWO RATIOS IN A NOT NEGLIGIBLE WAY, THE CONSTANT ACCELERATION CRITERION LEADS TO ANTICIPATE THE TIME FOR GEAR CHANGE.
- THE LOWER ENVELOPE OF THE CURVES (DASHED LINE) IS THE CURVE FOR AN OPTIMIZED CVT; THE AREA UNDER THE DASHED CURVE IS THEN THE MINIMUM TIME TO SPEED UNDER IDEAL CONDITIONS.
- THE AREAS BETWEEN THE DASHED AND THE CONTINUOUS LINES ACCOUNT FOR THE TIME WHICH MUST BE ADDED DUE TO THE PRESENCE OF A FINITE NUMBER OF SPEEDS: THE TRANSMISSION RATIOS CAN BE CHOOSEN IN SUCH A WAY AS TO MINIMIZE THIS AREA.
- BY INCREASING THE NUMBER OF SPEEDS THE ACCELERATION TIME IS REDUCED. HOWEVER, AT EACH GEAR SHIFTING THERE IS A TIME (0.2–0.5 [s]) IN WHICH THE VEHICLE, WITH A NON POWERSHIFT TRANSMISSION, DOES NOT ACCELERATE: INCREASING THE NUMBER OF SPEEDS LEADS TO AN INCREASE IN THE NUMBER OF GEAR SHIFTS AND THUS OF THE WASTED TIME. THIS IS ONE OF THE REASONS TO LIMIT THE NUMBER OF GEAR RATIOS.

Q. CVT IS A POWERSHIFT TRANSMISSION? IS IT REACHED THE IDEAL CONDITION REPRESENTED BY THE DASHED LINE? IF YES, HOW FAR IS GOD?

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 23 di 260

• FUEL CONSUMPTION AT CONSTANT SPEED

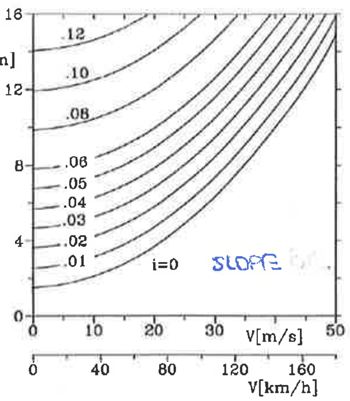
THEORETICALLY:

$e = P_m t = \frac{P_m d}{V}$: REQUIRED ENERGY (FOR CONSTANT SPEED) (AT WHEELS)

$Q = \frac{A + BV^2 + CV^4}{\eta_t \eta_e H \rho_s}$: FUEL CONSUMPTION (VOLUME) (PER UNIT DISTANCE)

STRONG hp: DENOMINATOR = CONST

STRONG hp: $\eta_e = \text{const.}$



$Q \left[\frac{l}{100km}\right] = \frac{R_R}{\eta \rho_s H_i}$

PRACTICAL PROCEDURE: (MORE REALISTIC)

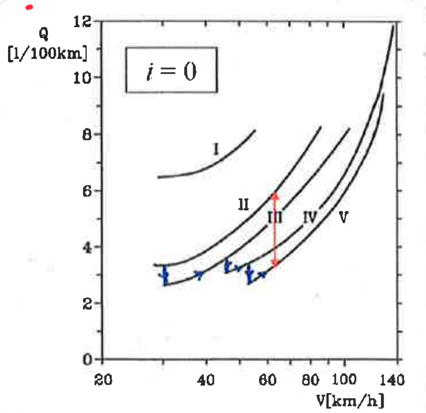
$P_E = \frac{P_m}{\eta_t}$

$Q = \frac{q P_m}{\eta_t V \rho_s}$

WITH:

$q(T_e, \omega_e) = \frac{1}{H \eta_e}$ SPECIFIC FUEL CONSUMPTION

H : LOWER HEATING VALUE OF THE FUEL



IF WE WANT TO MINIMIZE THE FUEL CONSUMPTION THE BEST POINT WHERE TO PERFORM THE GEAR SHIFT IS AT THE BEGINNING OF NEXT GEAR (AS SOON AS POSSIBLE)

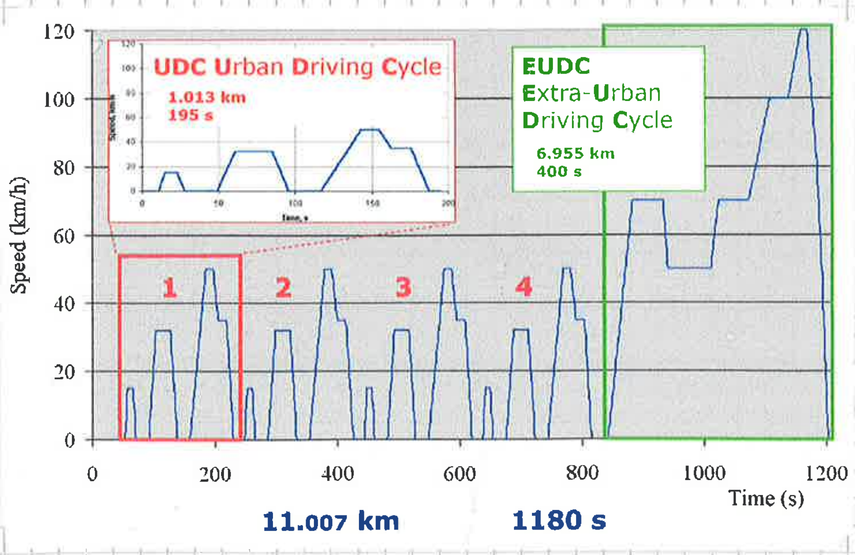
© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 25 di 260

• FUEL CONSUMPTION IN ACTUAL DRIVING CONDITIONS

– ASSUMPTIONS AND SIMPLIFICATIONS

- DESCRIPTION OF THE ACTUAL USE OF THE VEHICLE THROUGH THE STANDARDIZED EUROPEAN CYCLE **NEDC** OR THE NEW WORLD LIGHT TEST PROCEDURE **WLTP** (TIME HISTORY OF THE SPEED OF THE VEHICLE), NOT TAKING INTO ACCOUNT THE BEHAVIOUR OF AN ACTUAL DRIVER NEITHER THE ACTUAL ROAD AND TRAFFIC CONDITIONS.
- **FUEL CONSUMPTION** COMPUTATION AS A **SUM OF PARTIAL CONSUMPTIONS** DURING THE CYCLE (**1 [S]** IS **ACCEPTABLE FOR** THE **INTERVALS**), **ASSUMING QUASI-STEADY-STATE** OPERATION; THIS **ASSUMPTION** INTRODUCES SOME **ERRORS (2-5%) DUE** TO THE FACT THAT **IN NON-STATIONARY OPERATION**:
  - THE **THERMAL CONDITIONS** OF THE ENGINE ARE **VARIABLE**, SO THAT THE **THERMAL ENERGY LOSSES** ARE **DIFFERENT** FROM **STADY-STATE VALUES**;
  - **PART OF FUEL BURNS** WITH A **LOWER EFFICIENCY** DUE TO **CONDENSATION OF** THE **VAPOR** ON THE **INTAKE MANIFOLD** IN INDIRECT INJECTION ENGINES, OR TO A DIFFERENT EVAPORATION RATE OF THE FUEL DROPLETS IN DIRECT INJECTION ENGINES.

– NEW EUROPEAN DRIVING CYCLE (NEDC, UDC + EUDC)



© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 27 di 260

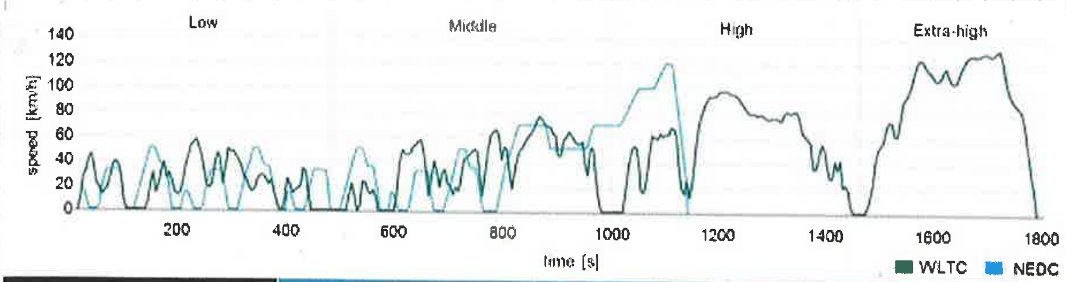
- THREE DIFFERENT TEST CYCLES ARE CONSIDERED, DEPENDING ON THE VEHICLE CLASS, WHICH IS BASED ON THE POWER/MASS RATIO PMR, ACCOUNTING FOR NOMINAL POWER TO DRY WEIGHT ([kW/t] OR [W/Kg]).

- Class 1 – PMR ≤ 22
  (mini-trucks etc...)
  LOW, MIDDLE
- Class 2 – 22 < PMR ≤ 34
  (bus/vans)
  LOW, MIDDLE, HIGH
- Class 3 – PMR > 34
  (most passenger cars)
  LOW, MIDDLE, HIGH, EXTRA-HIGH

$$PMR = \frac{POWER}{UNLOADEN\ MASS}$$

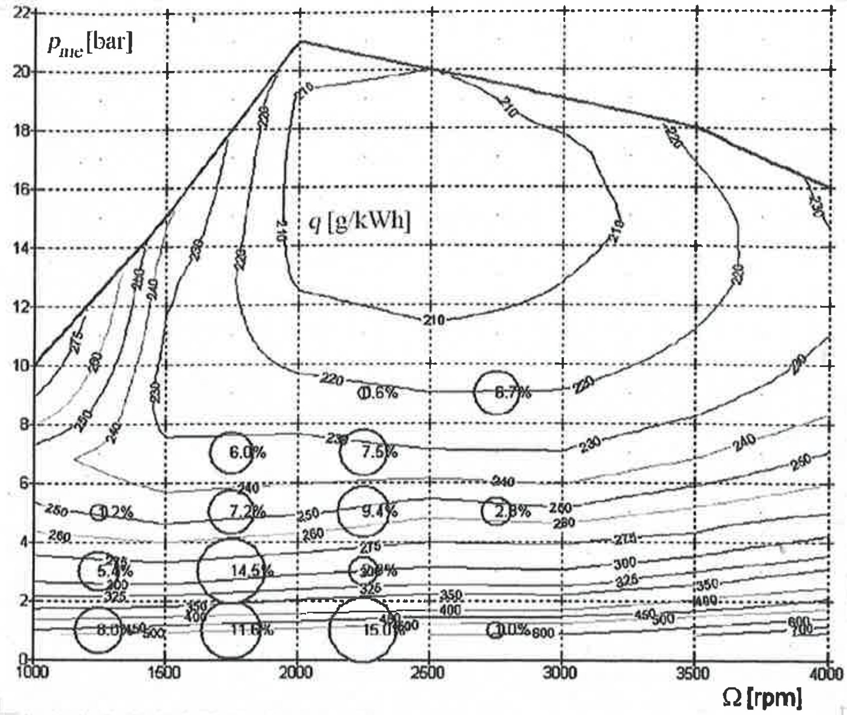
- EVERY CLASS IS CHARACTERIZED BY SPECIFIC DRIVING TESTS, DESIGNED TO REPRESENT THE WORKING OPERATION OF REAL VEHICLES ON URBAN/EXTRA-URBAN ROADS AND HIGHWAYS.
  EACH SEGMENT DURATION IS FIXED AMONG VARIOUS CLASSES; HOWEVER, THE ACCELERATION AND SPEED CURVES HAVE DIFFERENT SHAPES.

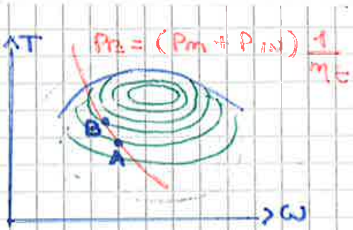
– WLTP VS. NEDC



	NEDC	WLTP
Test cycle	Single test cycle	Dynamic cycle more representative of real driving
Cycle time	20 minutes	30 minutes
Cycle distance	11 kilometres	23.25 kilometres
Driving phases	2 phases, 66% urban 34% non-urban driving	4 more dynamic phases 52% urban, 48% non-urban
Average speed	34 kilometres per hour	46.5 kilometres per hour
Maximum speed	120 kilometres per hour	131 kilometres per hour
Influence of optional equipment	Impact on CO2/fuel not considered	Additional features are taken into account
Gear shifts	Vehicles have fixed gear shift points	Different gear shift points for each vehicle
Test temperatures	Measurements at 20-30°C	Measurements at 23°C, CO2 values corrected to 14°C

- MAP OF A DIRECT INJECTION DIESEL I.C. ENGINE





POWER REQUESTED TO THE ENGINE TO MOVE THE VEHICLE

FOR A SPECIFIC POINT OF THE MISSION, TO REDUCE q, WE CAN CHANGE THE GEARSHIFT SCHEDULE (WORKING ON A ISO-POWER LINE) : A → B
IF WE HAD INFINITE AVAILABILITY OF GEAR RATIOS WE COULD MOVE SMOOTHLY (IN A GIVEN RANGE) IN ALL THE POINTS OF THE HYPERBOLE

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 31 di 260

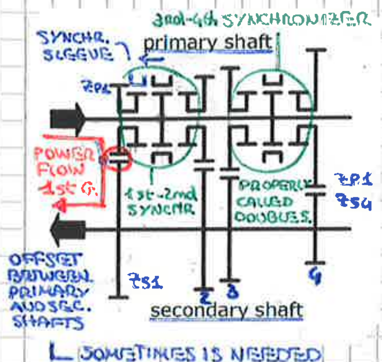
• FOUR SPEED GEARBOX

- FOUR SPEED GEARBOX CONFIGURATIONS

- SINGLE STAGE GEARBOXES
- DOUBLE STAGE (COUNTERSHAFT) GEARBOXES
- MULTISTAGE GEARBOXES

• SINGLE STAGE

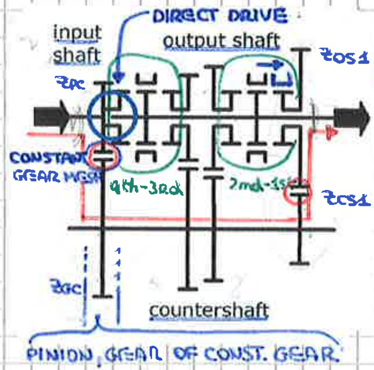
$\tau = \frac{\omega_{OUT}}{\omega_{IN}} = -\frac{z_{IN}}{z_{OUT}}$ ("-" BECAUSE SEC. S. ROTATES IN THE OPPOSIT DIR.)



- SINGLE CONTACT POINT BETWEEN THE GEARS INSIDE THE TRANSMISSION (ENABLE AT A CERTAIN POINT OF TIME) TO TRANSMIT TORQUE FROM INPUT (PRIMARY) TO OUTPUT (SECONDARY) SHAFT. $\tau = \left[-\frac{z_{P1}}{z_{S1}}, -\frac{z_{P2}}{z_{S2}}, -\frac{z_{P3}}{z_{S3}}, -\frac{z_{P4}}{z_{S4}}\right]$
- ONLY 1 COUPLE OF GEARS IS RESPONSABLE OF GEAR RATIO.
  1 SYNCHRONIZER BETWEEN TWO CONSECUTIVE GEARS
  └ SOMETIMES IS NEEDED

• DOUBLE STAGE (COUNTERSHAFT)

1st → 2nd → 3rd → 4th ⇒ $\begin{cases} T_e \nearrow \ \omega_e \searrow \ \text{(PRIME MOVER)} \\ T_{OUT} \searrow \ \omega_{OUT} \nearrow \end{cases}$

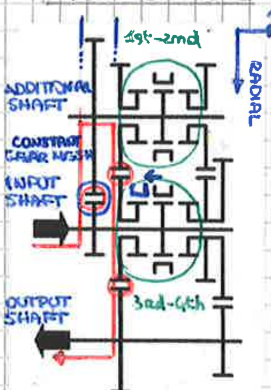


- TWO CONTACT POINTS BETWEEN THE G.
- THE OUTPUT IS COAXIAL WITH THE INPUT SHAFT; + COUNTERSHAFT
- THE DIRECT DRIVE IS A GEAR RATIO EQUAL TO 1 ($\tau = 1$); WE REALIZE A SORT OF SHORT CIRCUIT: WE CONNECT DIRECTLY THE INPUT TO THE OUTPUT SHAFT. ($\eta_t \nearrow$ BECAUSE NO POWER ON COUNTERSHAFT)
- TYPICALLY $\tau_{TOT} > 1 \Rightarrow z_{OUT} < z_{IN}$ — $\omega_{OUT} > \omega_{IN}$ — LOWER LOSSES IN COUNTS, NOT NULL, BECAUSE IT IS STILL ROTATING → VISC. LOSSES

$\tau = \left[\left(-\frac{z_{PC}}{z_{GC}}\right)\left(-\frac{z_{CS1}}{z_{OS1}}\right), \left(-\frac{z_{PC}}{z_{GC}}\right)\left(-\frac{z_{CS2}}{z_{OS2}}\right), \dots, 1\right]$

• MULTISTAGE

AXIAL (≡ TRANSVERSAL DIRECTION OF VEHICLE)



- IN SOME CASES WE HAVE MORE THAN TWO CONTACT POINTS TO REALIZE THE TOTAL TRANSMISSION RATIO OF A SPECIFIC GEAR.
- THE OUTPUT IS PARALLEL TO THE INPUT SHAFT; + ADDITIONAL UPPER PARALLELS.
- COMPACTNESS IN AXIAL DIRECTION; HIGHER RADIAL SPACE OCCUPANCY (WE ARE ABLE TO REDUCE THE TOTAL LENGTH OF THE TRANSM., TO THE DETRIMENT OF THE TRANSVERSAL LENGTH)
  CONVENIENT FOR: IN LINE CONF. WITH BIG ENGINE $\begin{cases} \text{LONGITUDINAL (TO RED. AXIAL)} \\ \to \text{TRANSVERSAL (TO RED. TR. L.)} \end{cases}$
  (ESPECIALLY FOR MORE THAN 4 CYL.)

- HIGHER THE NUMBER OF STAGES (CONTACT GEAR POINTS ↑), LOWER THE EFFICIENCY ($\eta_t \searrow$)
  $\eta_t = \prod \eta_{i,G}$ (EQUAL TO THE PRODUCT OF GEAR MESH EFFICIENCIES)
- IN MT ONLY ONE SYNCHRONIZER ACTION IS USED.

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 33 di 260

- FOUR SPEED GEARBOX APPLICATIONS

- **SINGLE STAGE GEARBOXES** ARE MAINLY APPLIED TO **FRONT WHEEL DRIVEN** VEHICLES, BECAUSE IN THIS CASE IT IS USEFUL THAT THE IMPUT AND THE OUTPUT SHAFT ARE OFFSET.
- ON THE CONTRARY, IN **CONVENTIONAL VEHICLES**, IT IS BETTER THAT THE INPUT AND OUTPUT SHAFTS ARE ALIGNED. THIS IS WHY **REAR WHEEL DRIVEN** VEHICLES ADOPT USUALLY A **DOUBLE STAGE GEARBOX**.
- THE **MULTISTAGE CONFIGURATION** IS SOMETIME ADOPTED ON **FRONT WHEEL DRIVEN** VEHICLES WITH **TRANSVERSAL ENGINE**, BECAUSE THE **TRANSVERSAL LENGTH OF THE GEARBOX** CAN BE **SHORTENED**; IT IS **USED** WHEN THE **NUMBER OF SPEEDS** OR THE WIDTH OF GEARS **DON'T ALLOW USING A SINGLE STAGE** TRANSMISSION.

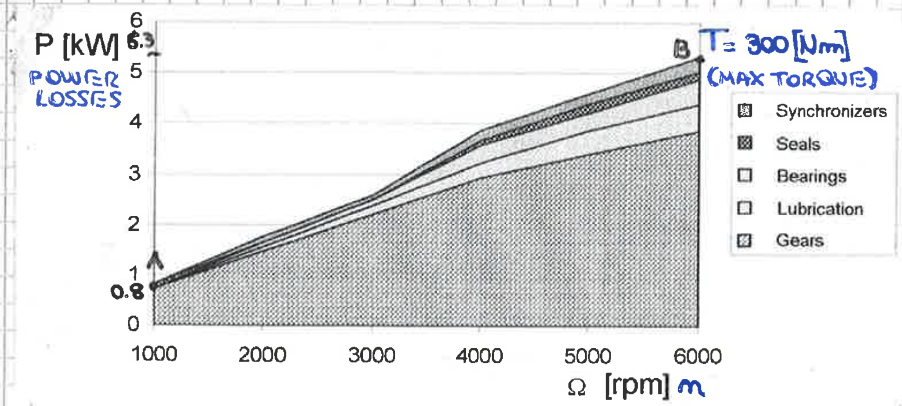
• TRANSMISSION POWER LOSSES AND EFFICIENCY

– CONTRIBUTIONS TO THE TOTAL TRANSMISSION LOSSES (POWER LOSSES)

• THE MAIN CONTRIBUTIONS TO THE TOTAL TRANSMISSION LOSSES ARE GIVEN BY: (IN MT)

1. GEAR LOSSES: GENERATED BY FRICTION BETWEEN ENGAGING TEETH FLANKS (TORQUE AND SPEED DEPENDENT) AND BY FRICTION OF WHEELS ROTATING IN THE AIR AND IN THE OIL (ONLY SPEED DEPENDENT);
2. SYNCHRONIZER LOSSES: GENERATED BY THE OIL FRICTION IN THE SYNCHRONIZERS (OR IN THE CLUTCHES) OF NOT ENGAGED GEARS (ONLY SPEED DEPENDENT); (RELAT. SPEED BETWEEN SYNCH. BODY AND ONE ADIACENT NOT-ENG. GEAR)
3. BEARING LOSSES: GENERATED BY THE EXTENSION OF THE CONTACT AREA OF ROLLING BODIES AND BY THEIR DEFORMATION (PARTLY DEPENDENT AND PARTLY INDEPENDENT ON POWER) AND BY THEIR ROTATION IN THE AIR AND IN THE OIL (ONLY SPEED DEPENDENT);
4. SEALING LOSSES: GENERATED BY FRICTION BETWEEN SEALS AND ROTATING SHAFTS (ONLY SPEED DEPENDENT); (ELASTOMERIC COMP. – EL. CLAMP – FRICTION)
5. LUBRIFICATION LOSSES: GENERATED BY THE LUBRICANT PUMP, IF ANY, (POWER INDEPENDENT). (+ VISCOUS EFFECT OF OIL IN WET CLUTCH)

• SINGLE STAGE GEARBOX FULL TORQUE CONDITION – POWER LOSSES:



LUBRIFICATION PUMP IS NOT ALWAYS PRESENT; AN ALTERNATIVE IS THE SPLASH LUBRICATION (TYPICAL IN MT)

$[W]\ P_{IN} = \omega\,[Rad/s] \cdot T\,[Nm] = \frac{2\pi n\,[Rpm]}{60} \cdot T\,[Nm]$

SYMBOL (T OR M)

$\eta(\omega, T) = \frac{P_{OUT}}{P_{IN}} = \frac{P_{IN} - P_{LOSS}}{P_{IN}}$

$\eta_A(1000\,[Rpm], 300\,[Nm])$ $\eta_B(6000\,[Rpm],$

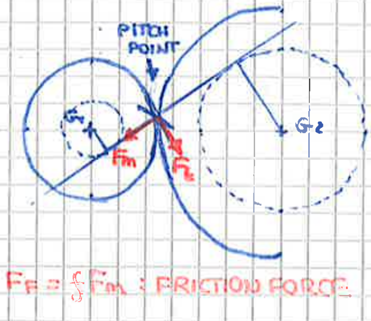
$P_{IN\,A} = 31.4\,[kW]$; $\eta_A = 0.974$

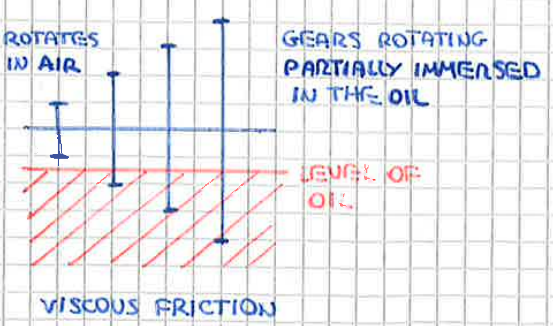
$P_{IN\,A} = 188.4\,[kW]$; $\eta_B = 0.972$

• GEARBOX EFFICIENCY MODELS HELP TO EVALUATE THE POWER LOSSES CONTRIBUTIONS AT DIFFERENT WORKING CONDITIONS.

$\eta \searrow \begin{cases} \omega \nearrow \\ T \searrow \end{cases}$

1.

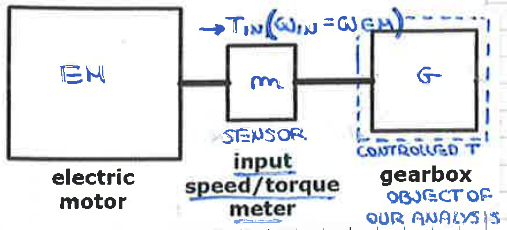




36

- TEST BENCH LAYOUTS FOR POWER LOSSES MEASUREMENT

• POWER LOSSES MEASUREMENT AT IDLE:



MEASUREMENTS PERFORMED AT:

- DIFFERENT TEMPERATURES
- DIFFERENT LOADS
- DIFFERENT ROTATIONAL SPEEDS

$\eta_g = \frac{P_G}{P_{BRAKE}} = \frac{\omega_E T_E}{\omega_{BRAKE} T_{BRAKE}}$

$T_{BRAKE} = \frac{\omega_E}{\omega_{BRAKE}} \frac{T_E}{\eta_g} = \frac{1}{\tau} \frac{T_E}{\eta_g}$

$T_E = T_{BRAKE} \cdot \tau \cdot \eta_g$

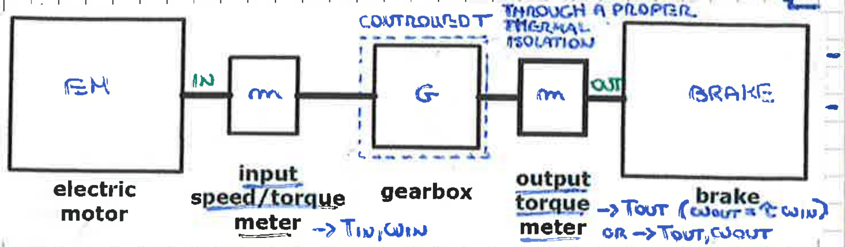
IN CASE OF REGENERATIVE BR.

$T_{OUT} = \frac{\omega_{IN} T_{IN}}{\omega_{OUT}} \cdot 1/\eta_g \; (T_{OUT} = T_{BRAKE})$

BACKWARD MODEL ≠ FORWARD MODEL

$\hookrightarrow T_{OUT} = \frac{\omega_{IN} T_{IN}}{\omega_{OUT}} \eta_g = \frac{T_{IN}}{\tau} \eta_g$, $\eta_g = \eta_g(\omega_{IN}, T_{IN}, T, GEAR)$

• POWER LOSSES MEASUREMENT UNDER LOAD: $\hookrightarrow \eta_g$



- TWO TORQUE ACTUATORS
- VERY HIGH ACCURACY INSTR. NEEDED

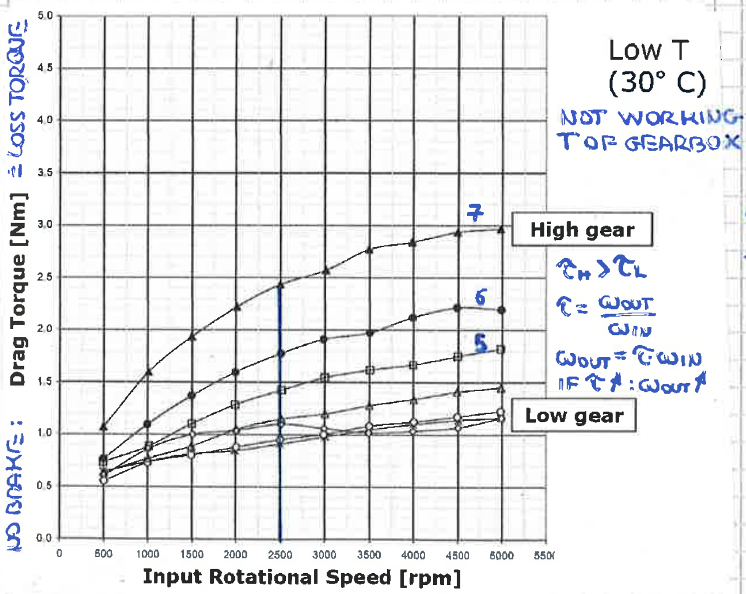
⇒ SET OF DIFFERENT SENSORS

(HIGHER LEVEL OF ACCURACY ESPECIALLY FOR OUTPUT SHAFT SENSORS)

0.1% OF ACCURACY W.R.T. FULL SCALE

$\eta_{UNDER\,LOAD} = \frac{P_{OUT}}{P_{IN}} = \frac{\omega_{OUT} T_{OUT}}{\omega_{IN} T_{IN}}$; $\eta_{AT\,IDLE} = 0$ BY DEFINITION: $T_{OUT} \sim 0$: $P_{OUT} \sim 0$

- GEARBOX DRAG TORQUE AT IDLE CONDITION (BENCH TESTS)



TO UNDERST. DRAG [↓ AT IDLE↓]

$\omega \uparrow$: DRAG TORQUE ↑
(OUT)

LOW → HIGH GEAR : DRAG TORQUE ↑

($\omega_{7th} > \omega_{6th}$ SPEED DEP. LOSSES ↑)

$\omega_{EM} (= CONST) T_{EM} \uparrow = \omega_{OUT} \uparrow T_{OUT} (= T_G = CONST \sim 0)$ (6→7)

$T_{DRAG} = T_{EM} \uparrow = \frac{\omega_{OUT} \uparrow \cdot T_{OUT} (= CONST)}{\omega_E (= CONST)}$

CONSIDERING A SINGLE STAGE GEARBOX, THE ROTATIONAL SPEED OF SECONDARY SHAFT (SO OF ALL CONNECTED GEARS) IS IMPOSED BY THE ENGAGED GEAR: IF 4th IS ENGAG. $\omega_{SS} \uparrow$ ⇒ LOSSES CONTR. (OF ALL CONNECT. G. !!) ↑

WE ARE IMPOSING, THROUGH THE E.M., THE $\omega_i^G [RPM] = \omega_i^{EM} [RPM]$, AND WE MEASURE, THROUGH THE TORQUE METER, THE **DRAG TORQUE** [Nm], THAT IS THE TORQUE **ABSORBED** BY THE TRANSMISSION **WITHOUT** PROVIDING ANY **POWER** TO THE **OUTPUT SHAFT**.

THE NUMBER OF CONTACT POINTS (CONSIDERING THE POWER PATH) IS NOT SO IMPORTANT BECAUSE WE ARE NOT PROVIDING TORQUE TO THE OUTPUT SHAFT (DURING THE DRAG TORQUE (<-> AT IDLE) MEASUREMENT TEST.

DRAG TORQUE DEPENDS ON **OUTPUT ROTATIONAL SPEED**

$T_{DRAG} \uparrow \propto \omega_{OUT} \uparrow$ BECAUSE OF LOSSES DUE TO THE ROTATION OF GEARS INSIDE THE OIL INCREASE

($\omega_{OUT} \uparrow$: **CHURNING LOSSES** ↑ : $T_{DRAG} \uparrow$)

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 39 di 260

TG MANUAL TRANSMISSIONS : PASSENGER CARS

- PASSENGER CAR MANUAL TRANSMISSIONS
- SINGLE STAGE
  - SINGLE STAGE SCHEME - 5 SPEED TRANSMISSION
  - SINGLE STAGE LAYOUT - 5 SPEED TRANSMISSION
  - SINGLE STAGE LAYOUT - 6 SPEED TRANSMISSION
- DOUBLE STAGE
  - DOUBLE STAGE SCHEME - 5 SPEED TRANSMISSION
  - DOUBLE STAGE LAYOUT - 6 SPEED TRANSMISSION
- TRIPLE STAGE
  - TRIPLE STAGE SCHEME - 5 SPEED TRANSMISSION
  - TRIPLE STAGE POWER FLOW - 5 SPEED TRANSMISSION

[TEXTBOOK : VOL I, PAR. 9.3]

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 41 di 260

• SINGLE STAGE

– SINGLE STAGE SCHEME – 5 SPEED TRANSMISSION

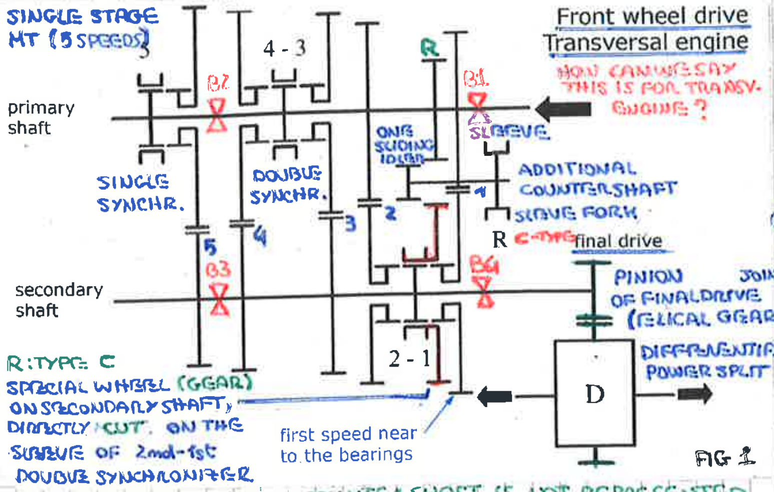
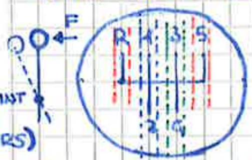


FIG 1

HERE, THE ADDITIONAL COUNTERSHAFT IS NOT REPRESENTED (IT ENGAGES R (ON PRIMARY) AND THE SPECIAL WHEEL)

ON THE UPPER SURFACE OF THE GEARSHIFT LEVER KNOB: ("MANOPOLA DELLA LEVA DEL CAMBIO") WE FIND THE SELECTION SCHEME



WE CAN SEE 4 SELECTION PLANES

THE SELECTION PLANES ARE THE PLANE ON WHICH WE HAVE TO MOVE GEARSHIFT LEVER IN ORDER TO ENGAGE ONE OF THE TWO GEARS.

– SINGLE STAGE LAYOUT – 5 SPEED TRANSMISSION

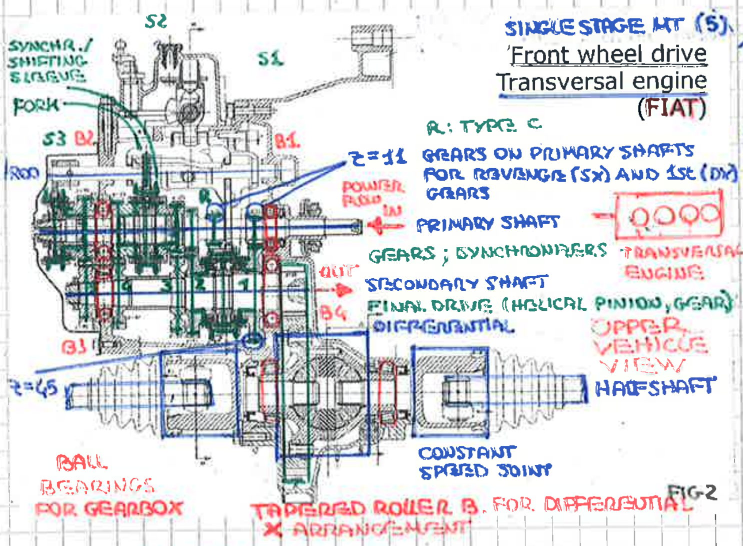


FIG 2

$$\tau_G = \left[\frac{11}{45}, \frac{19}{41}, \frac{25}{37}, \frac{33}{37}, \frac{39}{35}\right] = [0.24, 0.46, 0.68, 0.89, 1.11]$$

REDUCTION RATIOS $\left(\tau = \frac{\omega_{OUT}}{\omega_{IN}} = \frac{z_{IN}}{z_{OUT}} < 1 \Rightarrow \omega_{OUT} < \omega_{IN}\right)$

1st → 4th : $\tau \uparrow \Rightarrow \omega_{OUT} \uparrow$ ($\omega_{OUT} \leq \omega_{IN} = \omega_E$)

4th → 5th : $\tau \uparrow \Rightarrow \omega_{OUT} \uparrow$ ($\omega_{OUT} > \omega_{IN}$)

$$\tau_{FD} = \left[\frac{16}{55}\right] = [0.29]$$

$$\tau_t = \tau_G \cdot \tau_{FD} = [0.07, 0.13, 0.20, 0.26, 0.32]$$

BEARINGS MUST BALANCE NOT ONLY RADIAL BUT ALSO AXIAL FORCES BECAUSE WE ARE DEALING WITH HELICAL GEARS.

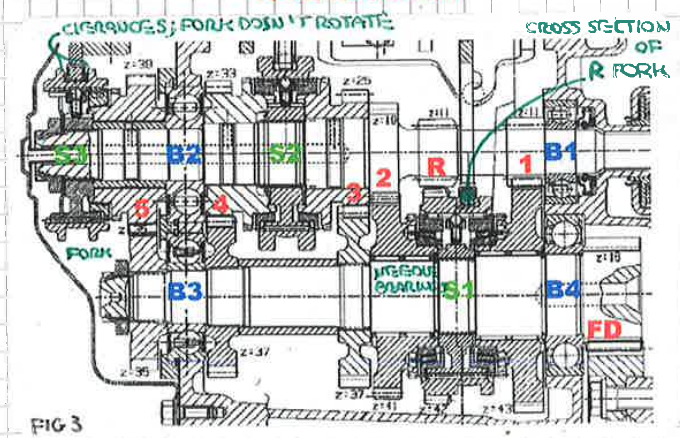


FIG 3

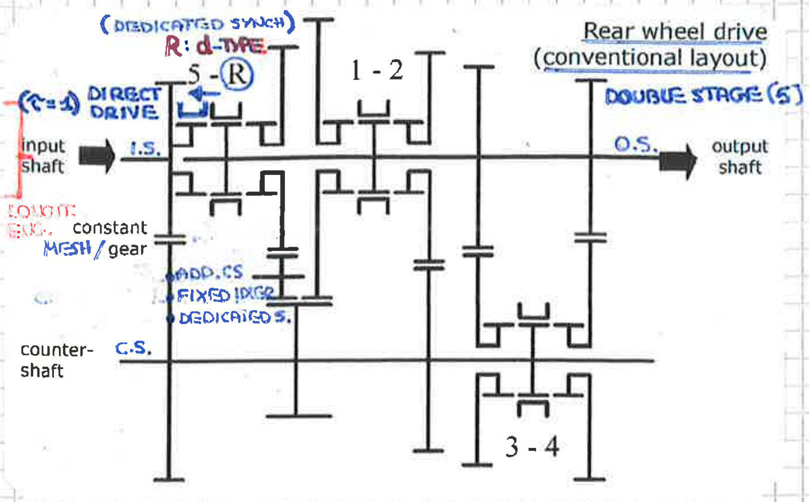
FIRST GEARS AND BEARINGS ARE POSITIONED SO CLOSE IN ORDER TO REDUCE THE AMOUNT OF DEFLECTION OF THE SHAFT (TO ASSURE A PERFECT COUPLING BETWEEN ALL THE GEARS) (WE COULD ALSO INCREASE THE SECTION OF SHAFTS BUT THIS WOULD PENALIZE IN TERM OF WEIGHT)

DUE TO THE FACT THAT THE PRIMARY 1st GEAR HAS A VERY LIMITED PITCH DIAMETER, WE ARE NOT ABLE TO COUPLE THE SYNCHRONIZER ON THE PRIMARY SHAFT; THEREFORE, THE 2nd-1st DOUBLE SYNCRONIZER IS COUPLED ON THE SECONDARY SHAFT. [RECOGNIZABLE BY NEEDLE B. PRESENCE]

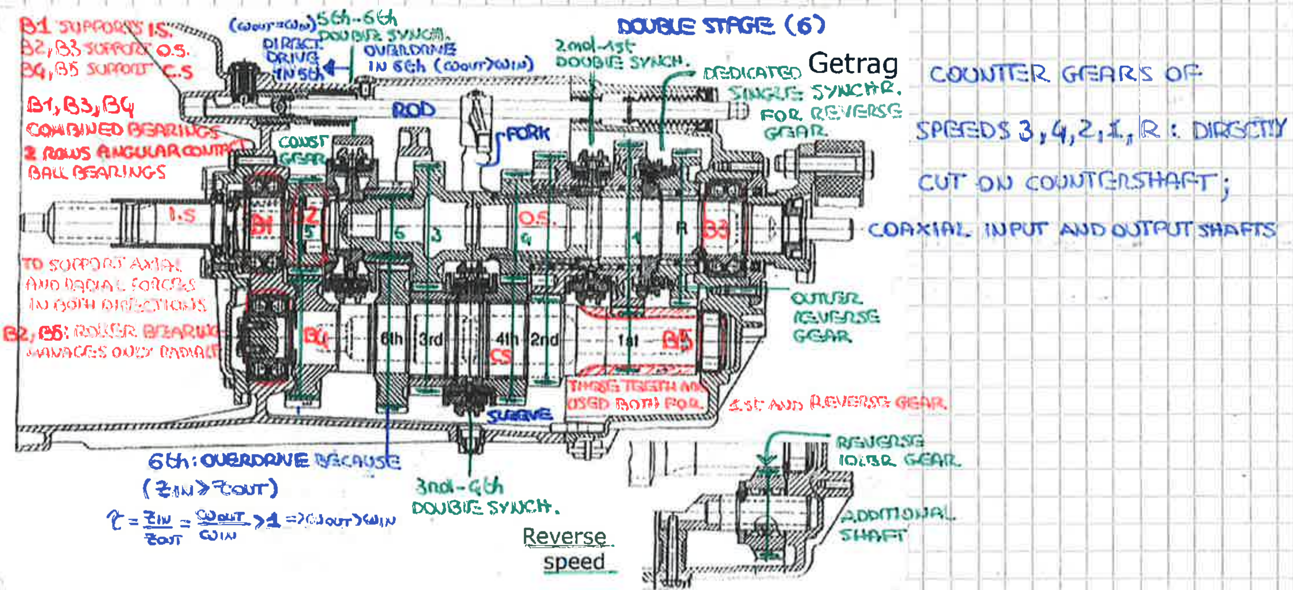
SECONDARY 1st AND 2nd GEARS, PRIMARY 3rd AND 4th GEARS ARE IDLER GEARS: THEY ARE MOUNTED THROUGH NEEDLE BEARINGS W.R.T. SECONDARY (1st, 2nd) AND PRIM.

• DOUBLE STAGE (COUNTERSHAFT)

- DOUBLE STAGE SCHEME - 5 SPEED TRANSMISSION



- DOUBLE STAGE LAYOUT - 6 SPEED TRANSMISSION

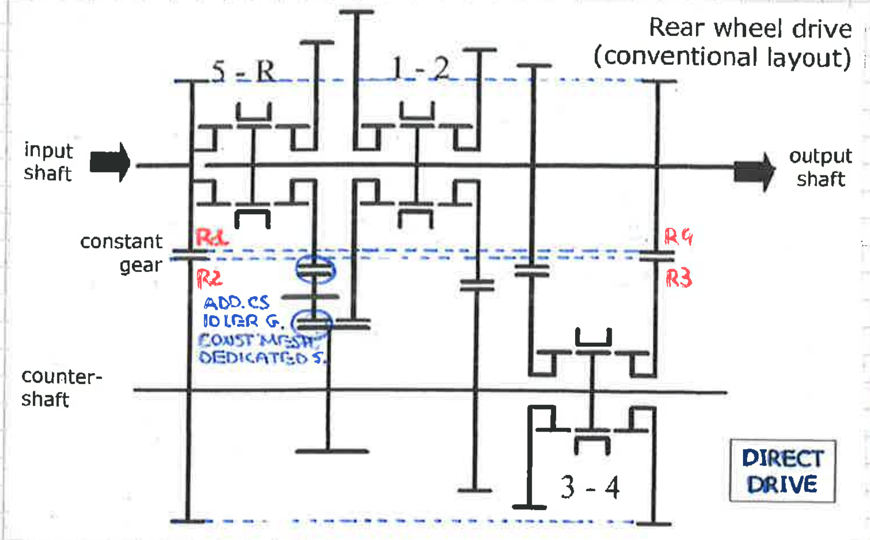


T5 MANUAL TRANSMISSION : INDUSTRIAL VEHICLES

- **LAYOUT SCHEMES AND CHARACTERISTICS**
  - GENERALITIES
  - OVERDRIVE AND DIRECT DRIVE SCHEMES
  - DOUBLE STAGE - 5 SPEED TRANSMISSION - DIRECT DRIVE
  - GENERALITIES
- **MULTIPLE RANGE TRANSMISSIONS**
  - GENERALITIES
  - SCHEME OF A 16 SPEED TRANSMISSION FOR I.V.
  - TRANSMISSION RATIOS OF A 16 SPEED TRANSMISSION
  - POWER FLOW SCHEME FOR A 16 SPEED TRANSMISSION
  - ALTERNATIVE ARCHITECTURES
  - TRUCK GEARBOX - 8 SPEEDS
  - TRUCK GEARBOX - 16 SPEEDS
  - 16 SPEED SEMIAUTOMATIC FULLER GEARBOX
  - SCHEME OF A 12 SPEED FULLER GEARBOX
  - POWER FLOW IN A 12 SPEED FULLER GEARBOX

[TEXTBOOK : VoL. I , PAR 9.4]

- DOUBLE STAGE 5 SPEED TRANSMISSION : DIRECT DRIVE



$\hat{\tau}_5 = 1$

$\hat{\tau}_4 = \frac{R1}{R2} \cdot \frac{R3}{R4} = \underbrace{\frac{R1}{R4}}_{=1} \cdot \underbrace{\frac{R3}{R2}}_{=1} = 1$

WHY?

Q. IS THERE AN ERROR IN THE DRAWING?

- GENERALITIES

- THE **CHOICE** BETWEEN THE TWO ALTERNATIVES (OVERDRIVE OR DIRECT DRIVE) CAN BE JUSTIFIED BY THE **DIFFERENT VEHICLE MISSION** ; ALMOST THE SAME GEARBOX CAN BE USED ON DIFFERENT VEHICLES WITH DIFFERENT SPEEDS OF MORE FREQUENT USE (FOR INSTANCE A TRUCK AND A BUS).
- THE **CONSTANT GEAR COUPLE** IS USED FOR ALL SPEEDS BUT NOT FOR THE HIGHEST (OR FOR THAT BEFORE THE HIGHEST ONE).
- **SOMETIMES** THE **CONSTANT GEAR** IS **SET AFTER** THE **DIFFERENT SPEED GEARS**; THIS CONFIGURATION OFFERS THE FOLLOWING **ADVANTAGES**:
  - **REDUCTION** OF THE **SYNCHRONIZATION WORK**, BECAUSE OF THE **SMALLER GEAR DIMENSION**, AT THE **SAME TORQUE** AND **TOTAL TRANSMISSION RATIO**;
  - **LESS STRESS** ON THE **INPUT SHAFT** AND ON THE **COUNTERSHAFT**.
- ON THE CONTRARY, THEY SHOW THE FOLLOWING **DISADVANTAGES**:
  - **BEARINGS ROTATE FASTER**;
  - **CONSTANT GEAR WHEELS** ARE MUCH **MORE STRESSED**.

- MULTIPLE RANGE TRANSMISSIONS

- GENERALITIES

- MULTIPLE RANGE TRANSMISSIONS FEATURE, IN ADDITION TO THE MAIN GEARBOX, OTHER GEARBOXES THAT MULTIPLY THE NUMBER OF SPEEDS OF THE MAIN GEARBOX BY THEIR SPEED NUMBER.
- THIS ARRANGEMENT IS USED WHEN MORE THAN 6 SPEEDS ARE NECESSARY. A MULTIPLE RANGE TRANSMISSION IS THEREFORE MADE OUT OF A COMBINATION OF DIFFERENT COUNTERSHAFT GEARBOXES, SINGLE RANGE GEARBOXES OR EPICYCLOIDAL GEARBOXES.
- THE ADDED ELEMENT IS CALLED RANGE CHANGER IF IT IS CONCEIVED IN ORDER TO USE THE MAIN GEARBOX SPEEDS IN SEQUENCE, IN TWO COMPLETELY NON OVERLAPPING SERIES OF VEHICLE SPEEDS.
- THE ADDED ELEMENT IS CALLED SPLITTER IF IT IS CONCEIVED IN ORDER TO CREATE SPEEDS THAT ARE SET BETWEEN THE MAIN GEARBOX SPEEDS.
- IN OTHER WORDS, THE SPLITTER IS A GEARBOX THAT COMPRESSES THE GEAR SEQUENCE, BECAUSE IT REDUCES THE GAP BETWEEN SPEEDS; THE RANGE CHANGER IS A GEARBOX THAT EXSPANDS THE GEAR SEQUENCE, BECAUSE IT INCREASES THE TOTAL RANGE OF THE TRANSMISSION.
- RANGE CHANGER USUALLY ADD A LOW RANGE OF SPEEDS → ADDITIONAL (LOWER) TRANSMISSION RATIO τ (+L) (SPLITTER +H INTERMEDIATE SPEEDS).
- RANGE CHANGER CAN BE REALIZED WITH ORDINARY GEARS OR EPICYCLOIDAL GEAR SET. (/EPICYCLIC GEAR SET) (EPICYCLIC IS MORE CORRECT)

(AN ADVANTAGE OF USING EPICYCLIC GEAR SET CAN BE, FOR EXAMPLE, THE POSSIBILITY OF AN EASIER AUTOMATIC ACTUATION).

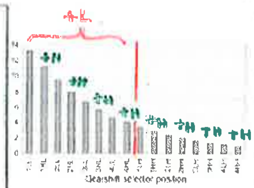
- ALTERNATIVE ARCHITECTURES

3-range gearbox

Main gearbox with front-mounted splitter unit and rear-mounted range-change unit

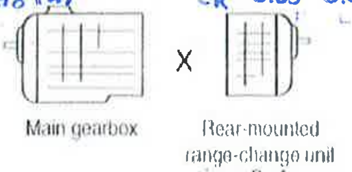
$i_{split} = 1.1–1.3$ (TYPICAL) $i_R = 3–4$

$\tau_S = 0.91 - 0.78$ (H) $\tau_R = 0.33 - 0.25$ (L)

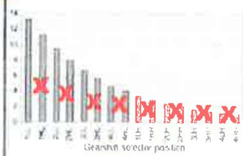


2-range gearbox

Main gearbox with rear-mounted range-change unit

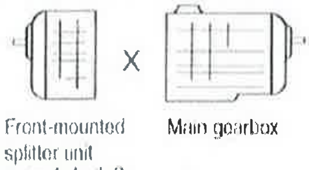


$i_R = 3–4$



2-range gearbox

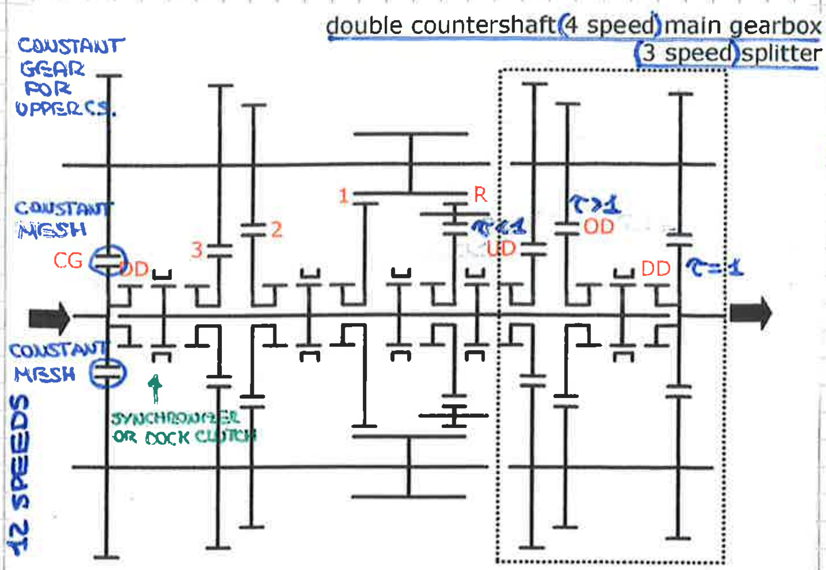
Main gearbox with front-mounted splitter unit



$i_{split} = 1.1–1.3$

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 53 di 260

- SCHEME OF A 12 SPEED FULLER GEARBOX

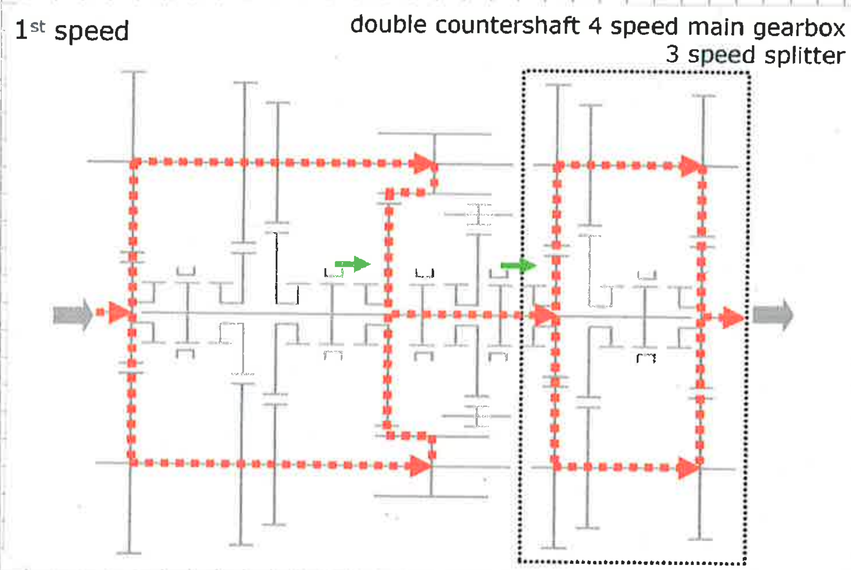


MAIN GEARBOX (4) + SPLITTER (3)
WITH TWO COUNTERSHAFTS — WITH TWO COUNTERSHAFTS

WHY, IN THIS CONFIGURATION, WE USE TWO COUNTERSHAFTS?
(X THE ADD. DOESN'T INCREASE m OF τ)
- WE REDUCE THE AXIAL LENGTH BECAUSE, SPLITING THE POWER AMONG THE TWO C.S. (PART OF POWER GOES ON THE UPPER C.S. AND PART GOES ON THE LOWER C.S.), WE CAN REDUCE THE WIDTH OF THE GEARS (BUT WE INCREASE THE TOTAL RADIAL WIDTH).

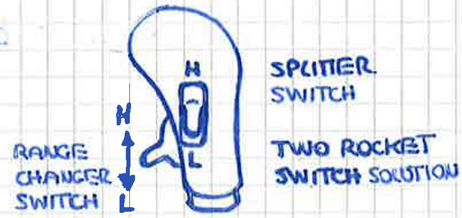
N.W. IT'S NOT POSSIBLE TO HAVE A PERFECTLY SYMMETRIC SYSTEM. WE COULD IMAGINE TO HAVE:
40% P → UPPER C.S.
60% P → LOWER C.S.

- POWER FLOW IN A 12 SPEED FULLER GEARBOX

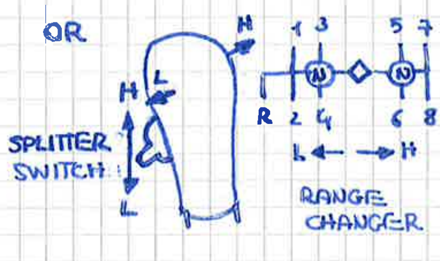


GEAR 1 + UNDER DRIVE ⇒ LOWEST TRANSMISSION RATIO ($\tau \ll 1$) ($\Rightarrow \omega_{out} \ll \omega_{in}$)
(LOWEST τ FOR MAIN G.) (LOWEST τ FOR SPLITTER)

- THE PNEUMATIC ACTUATOR MOVES 1 CYLINDER FOR THE ACTUATION OF THE SPLITTER, 2 CYLINDERS FOR THE MAIN GEARBOX AND 1 CYLINDER FOR THE RANGE CHANGER.
- ON THE LEVER KNOB WE FIND TWO SWITCHES:
  ONE TO MANAGE THE RANGE CHANGER (H, L)
  ONE TO MANAGE THE SPLITTER (H, L)
- WHEN WE WANT TO OPERATE ON THE SPLITTER SWITCH WE HAVE TO PRESS THE CLUTCH PEDAL; THERE IS A TRIGGERING VALVE CONNECTED TO THE CLUTCH PEDAL; PRESSING THE CLUTCH PEDAL WE TRIGGER THE SWITCH EVENT (I THINK ALSO FOR THE RANGE CHANGER SWITCH).



OR



© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 57 di 260

- MISSION OF VEHICLE AND TRANSMISSION

- CONTENT:

- **MISSION**: DESCRIPTION OF THE **OPERATING CONDITIONS** OF THE VEHICLE (OR TRANSMISSION) DURING THE VEHICLE LIFE, TO UNDERSTAND:
  - WHICH LOADS ARE APPLIED;
  - HOW LONG THEY MUST BE WITHSTOOD WITHOUT DAMAGE.
- **TRANSMISSION LIFE**, AS OTHER VEHICLE SYSTEMS LIFE, **CANNOT BE DESCRIBED BY DETERMINISTIC TERMS**, BUT **ONLY BY STATISTIC TERMS**, BECAUSE THE LOADS COMING FROM THE ROAD AND THE DRIVING STYLE HAVE **STATISTIC NATURE**.
- THE ENDURANCE SPECIFICATIONS ALSO, NORMALLY ADOPTED BY MANUFACTURERS, ARE ASSIGNED STATISTICALLY THROUGH THE **MAGNITUDE B_{10}** THAT CORRESPONDS TO THE **ENDURANCE** WHICH **ISN'T REACHED** BY THE **10% OF PRODUCED TRANSMISSIONS**.
- THE **ENDURANCE** IS THE **MAXIMUM DISTANCE** TRAVELED ACCORDING TO THE FORESEEN MISSION, WITHOUT ANY MAJOR DAMAGE.

- ENDURANCE SPECIFICATIONS AND CRITERIA

- **ENDURANCE** REFERENCE VALUES **FOR B_{10}** ARE:

- **CARS**	> 150-200'000 [km]
- **CONSTRUCTION INDUSTRIAL VEHICLES**	> 300'000 [km]
- **URBAN BUSES**	> 600'000 [km]
- **LONG HAULAGE TRUCKS** ("CAMION DA TRASPORTO")	> 800'000 [km]

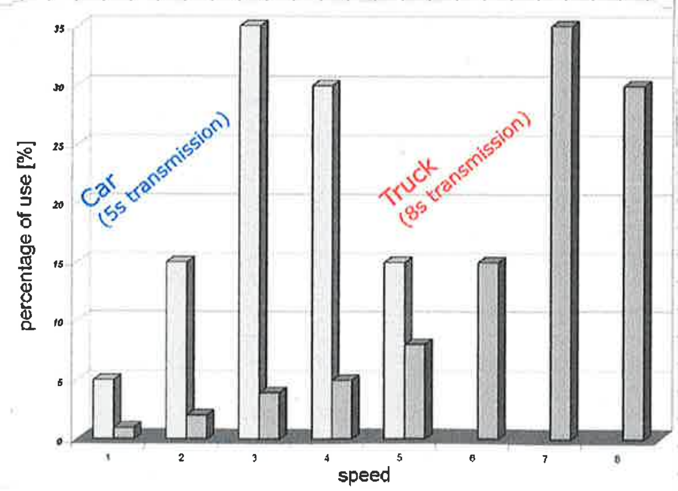
- IN THE TRANSMISSIONS, NOT ONLY THE COMPONENTS STRUCTURAL INTEGRITY HAS TO BE PRESERVED, BUT ALSO THEIR FUNCTIONAL CHARACTERISTICS: THE EASE OF USE, THE MECHANICAL EFFICIENCY, THE GENERATED VIBRATION AND NOISE. THEREFORE, THE MOST IMPORTANT ASPECTS AFFECTING THE ENDURANCE ARE:
  - STRUCTURAL INTEGRITY
  - SHIFT STICK MANOEUVRABILITY
  - VIBRATION AND NOISE
  - OIL LEAKAGE
  - SHIFTS QUALITY (AUTOMATIC GEARBOXES)

• FROM MISSION PROFILE TO COMPONENTS LOAD PROFILE

- CONTENT

- TO TRANSFORM A MISSION PROFILE INTO A LOAD PROFILE, EXPERIMENTAL TESTS ON THE VEHICLE (ROAD OR BENCH) ARE PERFORMED.
- A TYPICAL DATA ACQUISITION INCLUDES TIME HISTORIES OF CAR SPEED, GEAR SHIFT POSITION, CLUTCH POSITION, ACCELERATOR POSITION AND ROAD SLOPE; THEY ARE STATISTICALLY SYNTHESIZED TO OBTAIN INFORMATION EASIER TO BE USED.
- FROM THESE, MANY OTHER DATA ARE DERIVED, SUCH AS THOSE RELATED TO THE TRANSMISSION; TRANSMISSION OPERATION IS IN PARTICULAR IDENTIFIED BY INPUT TORQUE, ENGAGED SPEED AND CLUTCH PEDAL POSITION.
- THE TIME HISTORY OF THESE PARAMETERS CAN BE USED AS REFERENCE MISSION FOR DESIGN CALCULATIONS AND EXPERIMENTAL ENDURANCE VALIDATIONS.
- THE INPUT TORQUE IS STRONGLY CHANGING VERSUS TIME AND CAN ALSO BE NEGATIVE; ALL STRUCTURAL TRANSMISSION COMPONENTS SERVICE LIFE IS THEREFORE LIMITED BY FATIGUE PHENOMENA.
- GEARS AND OTHER ROTATING ELEMENTS INTRODUCE ADDITIONAL LOAD VARIATIONS VERSUS TIME.

- DIAGRAM OF USE PERCENTAGE (%) OF DIFFERENT SPEEDS (NEDC)



The use percentages are in distance (km).

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 61 di 260

• ASSUMPTIONS AND STEPS FOR A PRELIMINARY DESIGN

- CONTENT

• FIRST ASSUMPTION : NO MUTUAL INFLUENCE BETWEEN:
- LOAD CONDITIONS : USED FOR DESIGNED GEARS, SHAFTS, BEARINGS, HOUSING AND SEALS.
- NUMBER OF UP AND DOWNSHIFTS : USED TO DESIGN SYNCHRONIZERS, SHIFTING MECHANISMS AND CLUTCH.

• THE STEPS TO BE FOLLOWED TO DEFINE THE REQUIRED ENDURANCE AND THE CORRESPONDING LOAD CONDITIONS TO BE APPLIED TO EACH COMPONENT ARE:
- DEFINITION OF VEHICLE LIFE (PRODUCT TARGET)
- ASSUMPTION OF CYCLE MIX (VEHICLE MISSION)
- DEFINITION OF SINGLE CYCLES (URBAN, HIGHWAY, MOUNTAIN, ...)
- COMPUTATION OF SPEEDS % USE IN SINGLE CYCLES AND LIFE.
- EVALUATION OF LOAD DISTRIBUTION (INCL. MISUSE) → LOAD SPECTRA
- DEFINITION OF TEST REQUESTED LIFE AND LOAD CONDITIONS.

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 63 di 260

- PRODUCT DEVELOPMENT STEPS

– CONTENT

- PRODUCT DEVELOPMENT STEPS CAN BE SYNTHETIZED AS FOLLOWS:
  - PRODUCT STRATEGY (INCLUDING VOLUMES AND COSTS)
  - PRODUCT CONCEPT FEASIBILITY
  - PRODUCT AND PROCESS SPECIFICATIONS (INCLUDING PRODUCT MISSION)
  - PRODUCT DESIGN, CALCULATIONS AND DRAWING
  - PROCESS DEFINITION (INCLUDING TECHNOLOGY SELECTION)
  - PROTOTYPING (α AND β PROTOS)
  - FEW ITEMS BENCH TESTING (DESIGN RELEASE)
  - PRE-PRODUCTION (γ PROTOS)
  - MANY ITEMS BENCH TESTING (PRODUCTION RELEASE)
  - PROTOTYPE CAR BENCH TESTING (RELIABILITY AND QUALITY DEMONSTRATION)

• GEAR TYPES

- CONTENT

• TYPES OF GEARS IN AUTOMOTIVE TRANSMISSIONS:

Type	Applications		Typical failures				Efficiency
	Cars	I.V.	Bending	Pitting	Scuffing	Wear	
Cylindrical:	SMALL CAR (FOR REVERSE)						
Spur	(X)	X	XX	XX	X		XXX ($\eta_S > \eta_H$)
Helical	XXX	XXX	XX	XX		X	XXX
Bevel:							
Spiral	NOT USED	XX	XX	XX		X	XX
Hypoid	OFFSET X	XX	XX	XX		X	X

— MOST USED IN AUTOMOTIVE SECTOR

HELYCAL GEARS OPERATE MORE SMOOTHLY

— POSSIBILITY TO HAVE AN OFFSET BETWEEN CENTER LINES OF INPUT AND OUTPUT (EX. OF APP.: REAR W. DRIVEN CAR FINAL DRIVE)

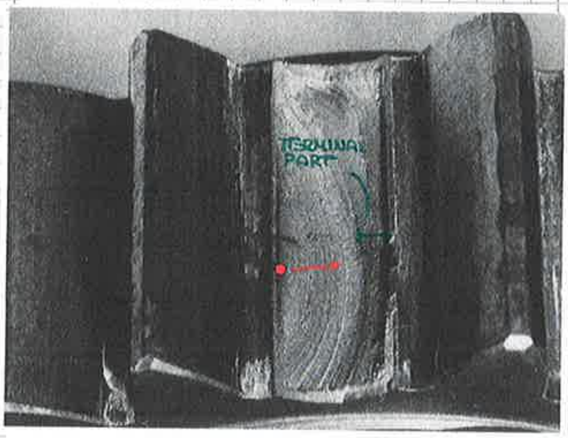
+ STRAIGTH TEETH BEVEL GEAR (FOR OPEN DIFF.)

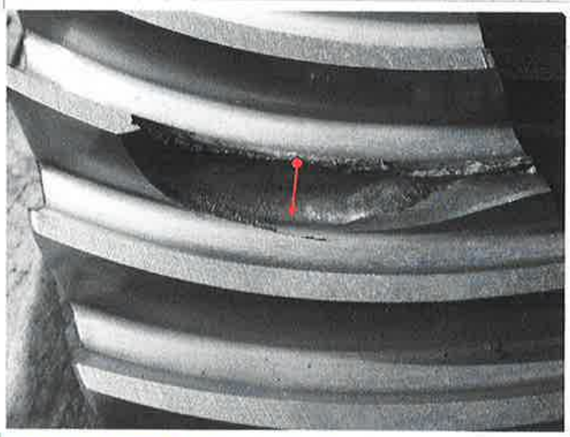
© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 67 di 260

- GEAR FAILURE (a)

- TOOTH BENDING FAILURES OCCUR WHEN A SUBSTANTIAL PART OF THE TOOTH IS REMOVED AS A RESULT OF THE FAILURE.
- IT IS USEFUL TO SEPARATE FAILURES DUE TO OVERLOAD FROM THESE DUE TO FATIGUE:
  - THESE LAST SHOW VERY CLEARLY A POINT WHERE THE FAILURE STARTED; A SET OF LINES WHICH ARE ALMOST CONCENTRIC TO THE STARTING POINT, SHOW THE PROPAGATION DIRECTION OF THE FAILURE; FAILURE SURFACES ARE POLISHED BY THE CONTINUOUS SHOCKS ON THE TWO PARTS, DUE TO LOAD PULSATION; THE TERMINAL PART OF THE FAILURE IS CHARACTERIZED BY A VERY IRREGULAR SURFACE WITH CRYSTAL ASPECT, DUE TO A SUDDEN RUPTURE WHEN THE RESISTANT SECTION HAS DECREASED TOO MUCH;
  - OVERLOAD FAILURES HAVE THE ASPECT OF THE TERMINAL PART OF FATIGUE FAILURES.

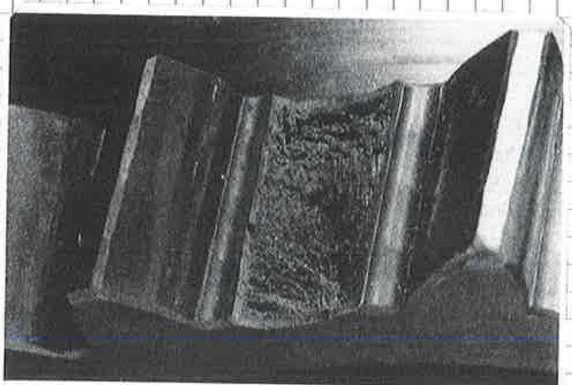
- EXAMPLES: BENDING FATIGUE FAILURE





WELL POLISHED BY CONTINUOUS SHOCKS DUE TO LOAD PULSATION

- EXAMPLE: BENDING STRESS FAILURE DUE TO OVERLOAD

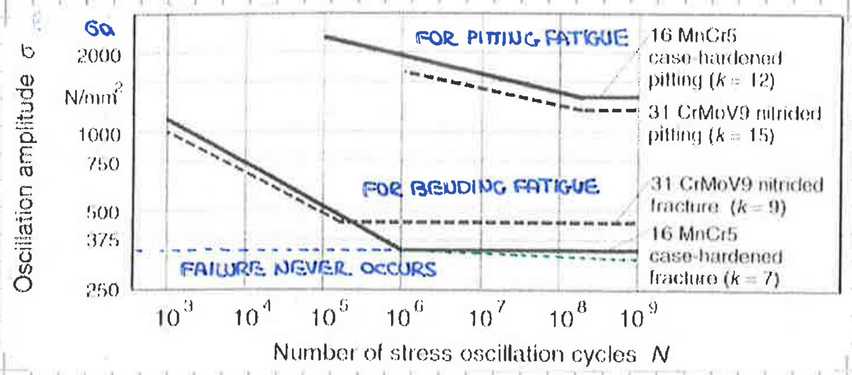


THE POINT WHERE BENDING FATIGUE STARTS CAN BE DUE TO THE PRESENCE OF A DEFECT OR A NOTCH OR A NON-FAILURE OVERLOAD.

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 69 di 260

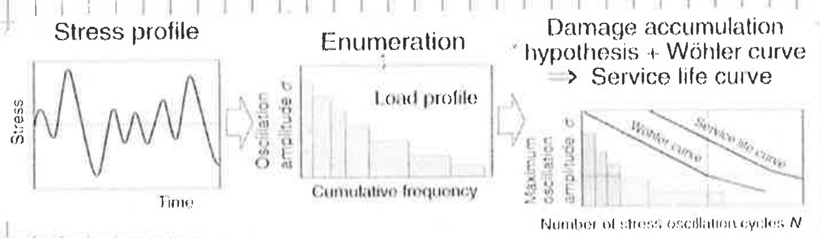
- ALLOWABLE STRESS : WÖHLER CURVES

- THE ALLOWABLE STRESS FOR BOTH BENDING AND PITTING PHENOMENA, ARE GIVEN BY THE FATIGUE CURVES (WÖHLER CURVES) OF THE MATERIAL; THEY TAKE INTO ACCOUNT ALSO HEAT TREATMENT CHARACTERISTICS. (W. CURVES ARE PLOTTED ON DOUBLE-LOG)
- IT IS A GOOD PRAXIS, TO CONSIDER NOT ONLY AVERAGE FAILURE CURVES, BUT ALSO B10, B50, B90 CURVES, IF RELIABILITY HAS TO BE PREDICTED.

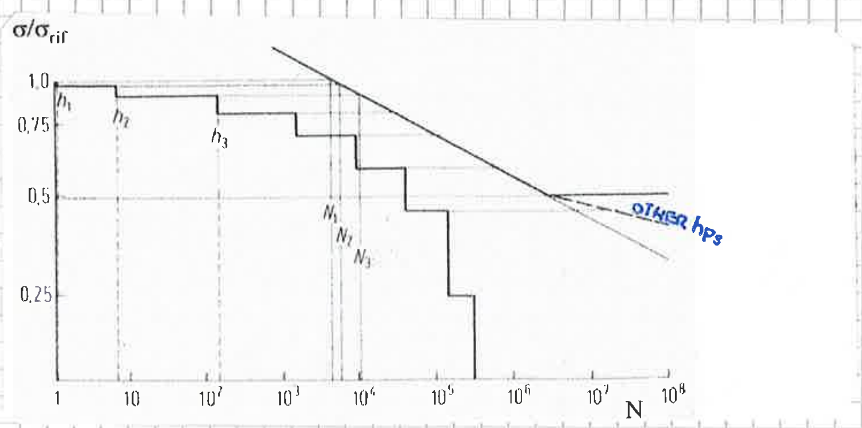


- DAMAGE ACCUMULATION PROCEDURE : MINER'S RULE

- IN ORDER TO COMPARE WORKING STRESS WITH WÖHLER LIMIT, IT IS USEFUL TO TRANSFORM, AS A FIRST STEP, TORQUE TIME HISTORY INTO STRESS TIME HISTORY AND THEN INTO A HISTOGRAM, CALLED "LOAD PROFILE" AND IN WHICH EACH STRESS AMPLITUDE CLASS IS REPRESENTED WITH THE CORRESPONDING CYCLE NUMBER.



- WÖHLER CURVE AND ASSIGNED LOAD PROFILE (STAIRCASE LINE)



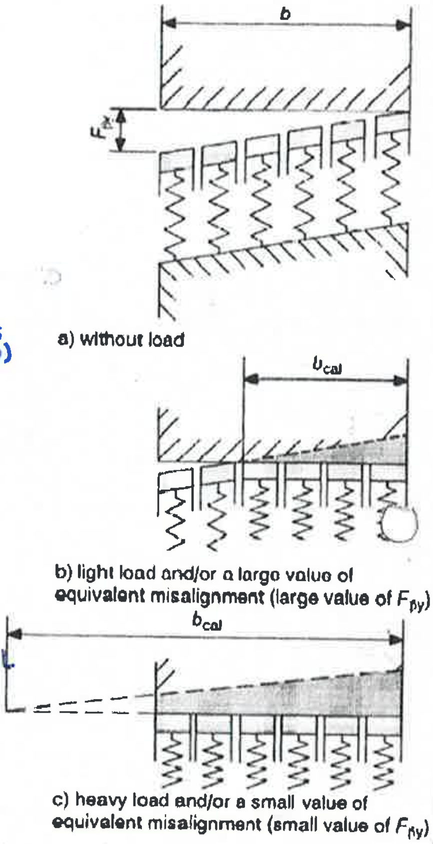
MINER ACCUMULATION RULE:

EACH GROUP OF LOAD PRODUCES A PARTIAL DAMAGE EQUAL TO: $\frac{(h_i - h_{i-1})}{N_i}$

NO DAMAGE CONDITION : $\sum \frac{(h_i - h_{i-1})}{N_i} < 1$

- MICRO - GEOMETRY

- ISO STANDARD 6336 TAKES INTO ACCOUNT, FOR BOTH BENDING AND PITTING CALCULATIONS, TWO IMPORTANT EFFECTS ON GEAR TEETH ACTUAL STRESS (INCREASE OF THE STRESS VALUE):
  - LOCAL EFFECTS (→)
  - DYNAMIC EFFECTS
- THESE EFFECTS ARE GENERATED BY ACTUAL MESHING CONDITIONS, DUE TO:
  - MANUFACTURING ERRORS (TOLERANCES);
  - TEETH DEFLECTION UNDER THE APPLIED LOAD; (CONTACT COND. IS RE-EVALUATED)
  - GEAR DISPLACEMENT DUE TO GEARBOX ERRORS AND DISPLACEMENT. (MISALIGNMENT ERR. → SURFACE PARTIALIZATION)
- LOCAL AND DYNAMIC EFFECTS INFLUENCE, TOGETHER WITH MACRO-GEOMETRY, ALSO GEAR TRANSMISSION ERROR → GEAR MESHING NOISE.
- MICRO-GEOMETRY OPTIMISATION IS A VARIATION OF THEORETICAL GEOMETRY IN ORDER TO REDUCE THESE EFFECTS ON STRESS AND TRANSMISSION ERRORS.



a) without load

b) light load and/or a large value of equivalent misalignment (large value of $F_{\beta y}$)

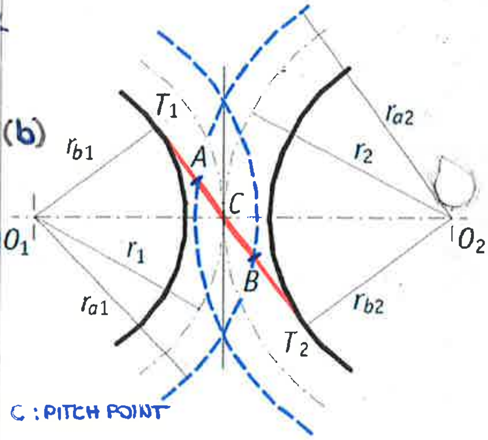
c) heavy load and/or a small value of equivalent misalignment (small value of $F_{\beta y}$)

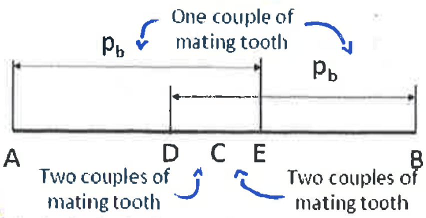
- MACRO - GEOMETRY : CONT. LINE, CONT. POINTS, CONT. RATIO

- FOR A SPUR GEAR PAIR, THE BASIC DESIGN PARAMETER TO DEFINE THE MACRO-GEOMETRY ARE: MODULE (m), PRESSURE ANGLE (α), NUMB. OF TEETH (z) AND FACEWIDTH (b)
- THE TEETH CONTACT POINT IS MOVING ALONG THE CONTACT LINE $\overline{T_1T_2}$; THE SEGMENT OF ALL THE ACTUAL CONTACT POINTS ($\overline{AB}$) IS DEFINED BY THE INTERSECTIONS OF THIS LINE WITH THE TWO ADDENDUM CIRCLES (SEGMENT $\overline{AB}$)
- THE CONTACT RATIO (ε) IS: $\left(P_b = \frac{2\pi R_b}{z}\right)$

$\varepsilon = \frac{\overline{AB}}{P_b}$, P_b = BASE PITCH $\overline{AB}$: SEGMENT OF CONTACT

- FOR AN HELICAL GEAR PAIR, THE ADDITIONAL DESIGN PARAMETER IS THE HELIX ANGLE.





ALTERNANCE OF 1 MATING TOOTH COUPLE - 2 M.T. COUPLE - 1 M.T. COUPLE

THIS ALTERNATION OF NUMBER OF TEETH ENGAGED AT THE SAME TIME PRODUCES A SORT OF PARAMETRIC EXCITATION TO THE SYSTEM → CHANGE IN THE MESHING STIFFNESS

• CALCULATION OF TOOTH BENDING STRENGTH

- CONTENT

• CALCULATION ACCORDING TO ISO STANDARD 6336-1 AND 6336-3 - METHOD B

1) NOMINAL TOOTH ROOT STRESS (σ_{F0}) (LOOK THE FIGURES IN SLIDES 22, 23, 24) (I'M NOT SURE THEY ARE RELEVANT...)

$$\sigma_{F0} = \frac{F_t}{b\,m_m} \cdot Y_F \cdot Y_S \cdot Y_\beta \cdot Y_B \cdot Y_{DT}$$

WITH:

F_t : NOMINAL TANGENTIAL LOAD : TANGENTIAL FORCE APPLIED TO THE TOOTH.

b : FACEWIDTH

m_m : NORMAL MODULE

Y_F : FORM FACTOR

Y_S : STRESS CORRECTION FACTOR

Y_β : HELIX ANGLE FACTOR

Y_B : RIM THICKNESS FACTOR

Y_{DT} : DEEP TOOTH FACTOR

CORRECTION FACTORS ARE GIVEN IN FORM OF EQUATIONS OR VALUES FROM TABLES.

2) TOOTH ROOT STRESS (σ_F) (ACTUAL)

$$\sigma_F = \sigma_{F0} \cdot K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha}$$

OTHER CORRECTION FACTORS

WITH:

σ_{F0} : NOMINAL TOOTH ROOT STRESS

K_A : APPLICATION FACTOR

K_V : DYNAMIC FACTOR (TO CONSIDER RESONANCE PHEN.)

$K_{F\beta}$: FACE LOAD FACTOR

$K_{F\alpha}$: TRANSVERSE LOAD FACTOR

(TO CONSIDER MISALIGNMENT EFFECTS)

3) PERMISSIBLE BENDING STRESS (σ_{FP})

$$\sigma_{FP} = \frac{\sigma_{Flim} Y_{ST} \cdot Y_{NT}}{S_{Fmin}} \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X$$

WITH:

σ_{Flim} : BENDING STRESS LIMIT VALUE

Y_{ST} : STRESS CORRECTION FACTOR

Y_{NT} : LIFE FACTOR FOR TOOTH ROOT STRESS (TAKES INTO ACC. THE LIFE EXPECTATIONS)

S_{Fmin} : MINIMUM REQUIRED SAFETY FACTOR

$Y_{\delta relT}$: RELATIVE NOTCH SENSIVITY FACTOR

Y_{RrelT} : RELATIVE SURFACE FACTOR

Y_X : SIZE FACTOR

4) STRESS COMPARISON

$$\sigma_F \leq \sigma_{FP}$$

- CALCULATION OF SURFACE DURABILITY (PITTING)

- CONTENT

- CALCULATION ACCORDING ISO STANDARD 6336-1 AND 6336-2 - METHOD B

1) NOMINAL CONTACT STRESS (σ_{H0}) (LOOK THE FIGURES 24, 25, 26)

$$\sigma_{H0} = Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot \sqrt{\frac{F_t}{d_1 \cdot b} \frac{u+1}{u}}$$

WITH:

F_t : NOMINAL TANGENTIAL LOAD

b : CONTACT FACEWIDTH

d_1 : REFERENCE DIAMETER OF PINION

u : GEAR RATIO (z_2/z_1)

Z_H : ZONE FACTOR

Z_E : ELASTICITY FACTOR

Z_ε : CONTACT RATIO FACTOR

Z_β : HELIX ANGLE FACTOR

2) CONTACT STRESS (σ_H)

$$\sigma_H = Z_B \cdot \sigma_{H0} \cdot \sqrt{K_A \cdot K_V \cdot K_{H\beta} \cdot K_{H\alpha}}$$

WITH:

Z_B : SINGLE PAIR TOOTH CONTACT FACTOR

σ_{H0} : NOMINAL CONTACT STRESS

K_A : APPLICATION FACTOR

K_V : DYNAMIC FACTOR

$K_{H\beta}$: FACE LOAD FACTOR

$K_{H\alpha}$: TRANSVERSE LOAD FACTOR

3) PERMISSIBLE CONTACT STRESS (σ_{HP})

$$\sigma_{HP} = \frac{\sigma_{Hlim} \cdot Z_{NT}}{S_{Hmin}} \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_W \cdot Z_X$$

σ_{Hlim} : ALLOWABLE STRESS NUMBER (CONTACT)

Z_{NT} : LIFE FACTOR FOR CONTACT STRESS

S_{Hmin} : MINIMUM REQUIRED SAFETY FACTOR

Z_L : LUBRICANT FACTOR   LUCIA VA(A)ROMA x (IL)WEEKEND

Z_V : VELOCITY FACTOR

Z_R : ROUGHNESS FACTOR

Z_W : WORK HARDENING FACTOR

Z_X : SIZE FACTOR

4) STRESS COMPARISON

$\sigma_H \leq \sigma_{HP}$

- GEAR AND TRANSMISSION NOISE

- CONTENT

- TYPES OF TRANSMISSION NOISE:
  1. GEAR WHINE (WHISTLE) - LOADED GEARS - PURE TONE NOISE (SPECIFIC HIGH f)
  2. GEAR RATTLE - UNLOADED GEARS - BROADBAND NOISE (NOT SPECIFIC CHARACTERIZING f) (f-RANGE: 500-2000 [Hz])
  3. BEARING WHINE (CHARACTERISTIC OF INITIAL DAMAGE OF B.)
  4. SPEED SHIFT GRATING (DUE TO PROBLEMS IN THE SYNCHRONIZATION SYSTEMS)
  - ...

(CHANGE IN N. OF T. IN CONT.)

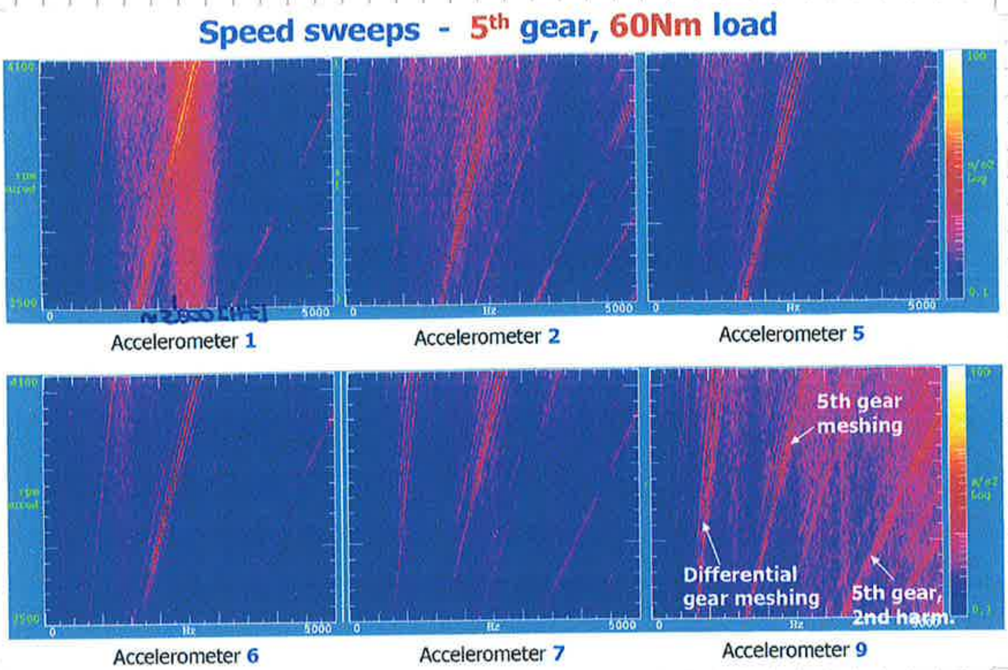
- THE GEAR WHINE IS DUE TO THE TORSIONAL VIBRATION OF THE LOADED GEARS GENERATED BY THE MESHING STIFFNESS VARIATION DURING THE TEETH ENGAGEMENT (1→2 OR 2→3 COUPLES OF TEETH IN CONTACT); IT IS A PURE TONE NOISE, AT THE MESHING (MEDIUM-HIGH) FREQUENCY:

$$f_m = \Omega z$$ (ONOMATOPEIC WORD)

- THE GEAR RATTLE IS DUE TO THE SHOCKS BETWEEN THE TEETH OF THE UNLOADED GEARS GENERATED BY THE ROTATIONAL SPEED VIBRATION (AT LOW-MEDIUM FREQ.s) CAUSED BY THE RESONANCE OF THE DRIVELINE EXCITED BY THE ENGINE IRREGULARITIES (IN PARTICULAR, 2nd ORDER); IT IS A BROADBAND NOISE, NORMALLY IN THE RANGE 500-2000 [Hz]. (ESPECIALLY FOR GT 4 CYL. ENGINE)

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 79 di 260

- EXAMPLE : GEAR WHINE

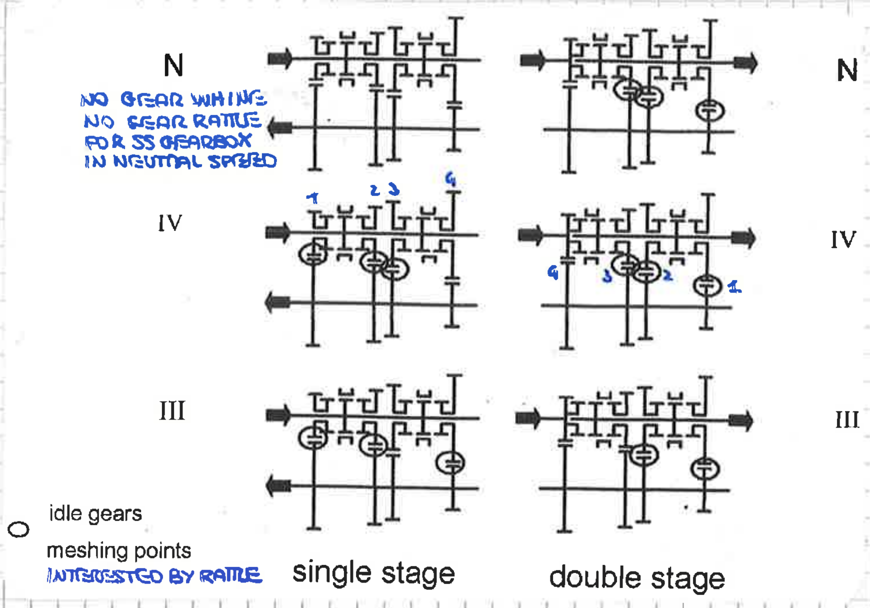


THE **INCLINED LINES** REPRESENTS THE **EXCITATIONS** DUE TO THE **TRANSMISSION ERRORS** (→ GEAR WHINE)

THE **VERTICAL LINES** REPRESENTS THE **STRUCTURAL RESONANCES** OF THE TRANSMISSION.

WHEN THE **EXCITATIONS CROSS** THE **STRUCTURAL RESONANCES** WE CAN OBSERVE A **MAGNIFICATION** OF THE **AMPLITUDE OF VIBRATIONS** (EXCITATION).

- MESHING POINTS OF ROTATING IDLE GEARS (GEAR RATTLE)

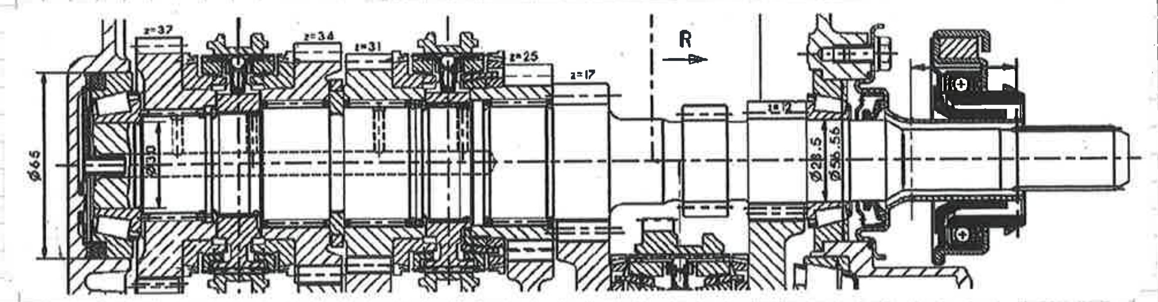


• SHAFTS

- CONTENT (FUNCTIONS)

• THE GEARBOXES SHAFTS HAVE TWO MAIN FUNCTIONS:

- TO TRANSMIT TORQUE BETWEEN THE DIFFERENT ENGAGED GEAR PAIRS (TORSIONAL FUNCTION);
- TO SUPPORT RIGIDLY THE MESHING GEARS. (RADIAL AND AXIAL FUNCTIONS).



• THE MAXIMUM STIFFNESS IS REQUIRED TO REDUCE SHAFT DISPLACEMENTS UNDER LOAD, IN ORDER TO AVOID TOO LARGE MISALIGNMENT OF BEARINGS AND GEARS:

- BEARINGS HAVE PRECISE LIMITATIONS IN ALIGNMENT ANGLE (WITH LARGE ANGLES THEIR LIFE DECREASES);
- GEARS TOOTH FLANKS CAN ACCEPT LIMITED MISALIGNMENT: HIGHER VALUES GENERATE HIGH LOCAL STRESS (REDUCING LIFE) AND NOISE.

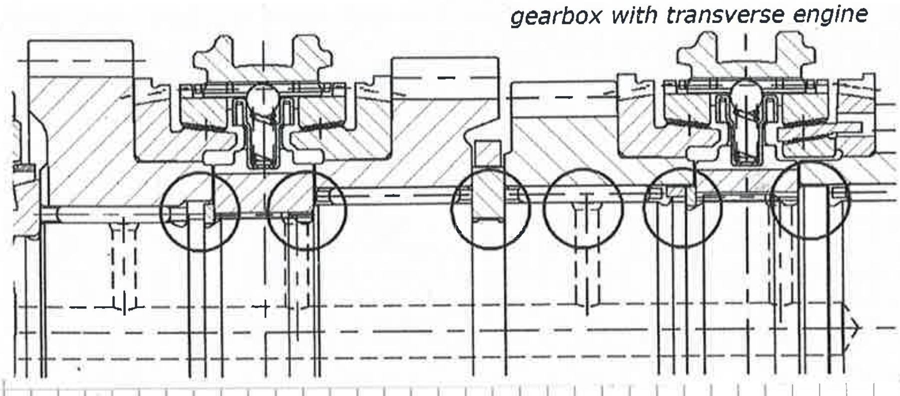
- SHAFT STRESS

• NOMINAL STRESS IN GEARBOX SHAFTS IS NORMALLY A NON-CRITICAL ASPECT.

• POTENTIALLY DANGEROUS STRESS LEVELS ARE GENERATED BY FREQUENT DIAMETER VARIATIONS.

• THEREFORE, CHAMFERS ("SMUSSI") AND EDGE ROUNDING ARE REQUESTED TO REDUCE TORSION AND BENDING STRESS CONCENTRATION.

Details of a shaft for a front wheel drive gearbox with transverse engine



• BEARINGS

- CONTENT (FUNCTIONS AND TYPES)

- GEARBOX BEARINGS ARE NORMALLY ROLLER BEARINGS; BUSH BEARINGS ("CUSCINETTI A BOCCOLA") ARE LIMITED TO LOWLY STRESSED IDLE GEAR WHEELS AND TO SLIDING BEARINGS FOR INTERNAL SHIFT MECHANISMS.
- IF POSSIBLE, BEARINGS ARE LIMITED TO TWO PER SHAFT, AVOIDING HYPERSTATIC MOUNTING, TOO SENSITIVE TO MACHINING TOLERANCES.
- THE ROLLING BEARINGS APPLIED TO SHAFTS ENDS ARE MAINLY OF FOUR DIFFERENT TYPES:
  - DEEP GROOVE BALL BEARINGS
  - FOUR CONTACTS BALL BEARINGS
  - CYLINDRICAL ROLLER BEARINGS
  - TAPERED ROLLER BEARINGS
- DEEP GROOVE BALL BEARINGS: WIDELY APPLIED BECAUSE THEY CAN WITHSTAND BOTH RADIAL AND AXIAL LOADS, ARE EASILY ASSEMBLED ON THE SHAFT AND ARE REASONABLY CHEAP; DISADVANTAGES: LARGE DIMENSION AND OIL POLLUTION SENSIVITY. SOMETIMES SELF LUBRIFICATED SEALED BALL BEARINGS ARE PREFERRED.
- FOUR CONTACT BALL BEARINGS: ALMOST EQUIVALENT TO THE PREVIOUS ONES, WITH THE ADVANTAGE OF SMALLER DIMENSIONS AT A SLIGHTLY INCREASED COST.
- CYLINDRICAL ROLLER BEARINGS: HIGH RADIAL LOAD CAPABILITY, BUT CAN'T WITHSTAND AXIAL LOADS; THEY ARE USUALLY COUPLED TO BALL BEARINGS AT THE OTHER END OF THE SHAFT. DISADVANTAGES: HIGHER COST AND BIGGER SENSIVITY TO ANGULAR DISPLACEMENTS.
- TAPERED ROLLER BEARINGS: MORE AND MORE EXTENSIVELY APPLIED BECAUSE OF THEIR OPTIMUM RADIAL AND AXIAL LOAD CAPACITY, WITH LIMITED DIMENSIONS. DISADVANTAGES: THEY REQUIRE, FOR A CORRECT OPERATION, AN AXIAL APPROPRIATE PRELOAD, WHICH MUST BE MAINTAINED AT EVERY WORK CONDITION; MOREOVER, THEIR ASSEMBLY ON THE GEARBOX IS DIFFICULT, BECAUSE INNER RACE WITH ROLLER CAGE AND OUTER RACE ARE SEPARABLE COMPONENTS.
- ON IDLE GEAR WHEELS NEEDLE BEARINGS ARE OFTEN USED.

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 87 di 260

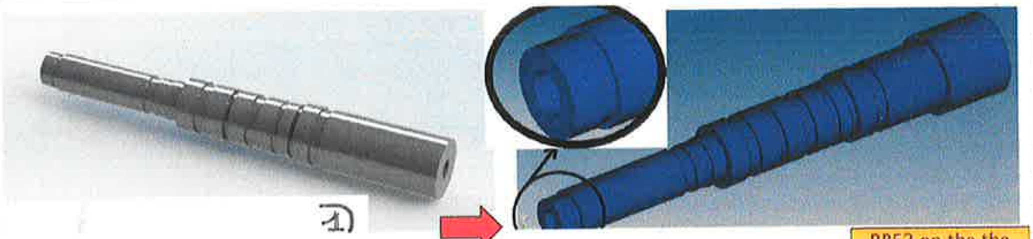
- SHAFTS DEFLECTIONS: CALCULATION AND DESIGN RULES

- IN A GEARBOX DESIGN, IT IS VERY IMPORTANT TO CALCULATE SHAFT DEFLECTIONS, TAKING INTO ACCOUNT HOUSING AND BEARING STIFFNESS AND SHAFT ACTUAL GEOMETRY; NORMALLY A FEM COMPUTATION IS THEN PERFORMED.
  (HOUSING AND BEARING STIFFNESS CAN BE REPRESENTED BY STIFFNESS MATRIXES)

- FOLLOWING DESIGN RULES HELP TO REDUCE SHAFT DISPLACEMENTS (AND STRESS):
  - REDUCING THE SHAFT SPAN BETWEEN BEARINGS, BY LIMITING AS MUCH AS POSSIBLE GEAR WHEELS AND SYNCHRONIZERS WIDTH;
  - INSTALLING THE WHEELS SUBJECTED TO HIGHEST LOADS (LOW GEARS) AS CLOSE TO THE BEARINGS AS POSSIBLE;
  - AVOIDING TOO STEEP DIAMETER TRANSITION, ORGANIZING COMPONENTS BY INCREASING OR DECREASING DIAMETERS;
  - AVOIDING FEATHER KEYS ("LINGUETTE"), PREFERRING SPLINE CONNECTIONS;
  - SMOOTHENING DIAMETER TRANSITIONS AND DRILLING, AS SHOWN PREVIOUSLY;
  - USING CIRCLIPS ("ANELLI DI SICUREZZA") AT THE ENDS OF THE SHAFT ONLY.

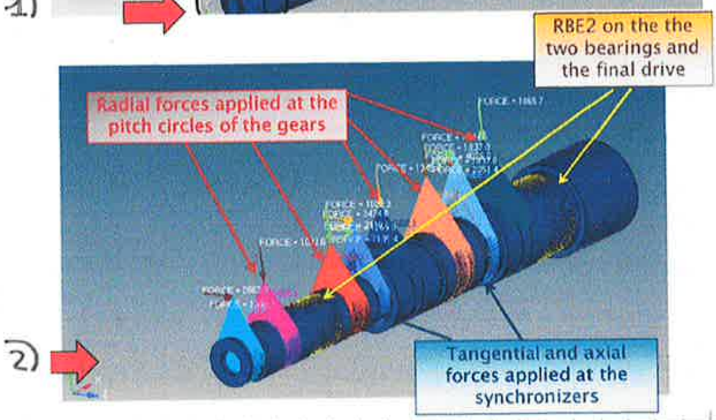
- SHAFT DEFLECTIONS: FE MODEL, LOADS AND CONSTRAINTS

1) THE FEM MESH MUST BE STRONGLY REFINED FOR STRESS CONCENTRATION (IN PARTICULAR NEAR NOTCHES).
2) LOADS AND CONSTRAINTS HAVE TO BE APPLIED IN EXACT POINTS OF FORCE / REACTION APPLICATION.



FEM
- MESH
- LOADS AND CONSTR.

FEA



• TRANSMISSION LUBRIFICATION

- CONTENT (FUNCTIONS)

• THE MAIN LUBRICANT FUNCTIONS ARE:
- REDUCING FRICTION AND WEAR OF METALLIC AND NON-METALLIC (ROTARY AND SLIDING SEALS) PARTS;
- BUILDING UP LUBRICATED HYDRODYNAMIC FILMS (EHD LUBRICATION);
- DISTRIBUTING TO COLDER AREAS THE HEAT GENERATED IN HOT ONES;
- PROTECTING COMPONENTS AGAINST CORROSION;
- DETERGING RESIDUAL PARTICLES PRODUCED BY WEAR;

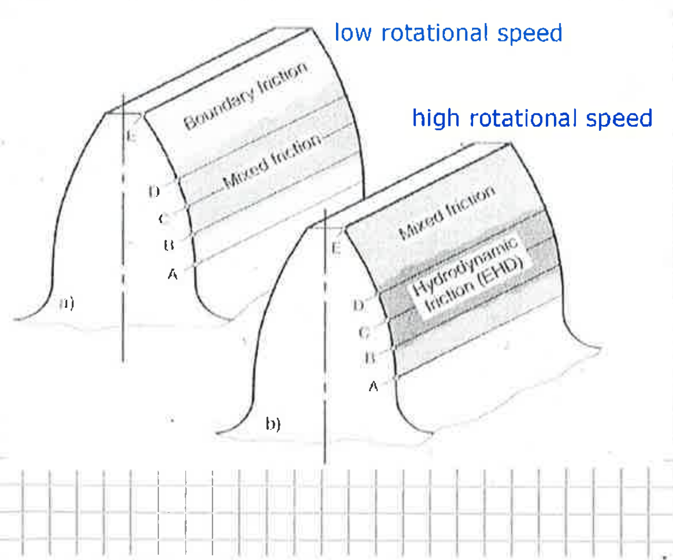
• PERFORMING ALL OF THE ABOVE FUNCTIONS FOR LONG TIME AND AT ANY POSSIBLE WORKING TEMPERATURE.

• GEARBOX COMPONENTS LUBRICATION IS USUALLY MADE BY SPLASHING AND SPRAYING OIL BY ROTATING GEARS (CHANNELS IN THE HOUSINGS, TRANSVERSAL DRILLING ON SHAFTS). ON HEAVY DUTY MANUAL GEARBOXES AND ON AUTOMATIC ONES PRESSURE LUBRICATION IS USED (GEAR PUMP).

• THE LUBRICANT DISTRIBUTION STUDY BY MATHEMATICAL MODELS WOULD BE RATHER DIFFICULT: EXPERIMENTAL ANALYSIS ARE SET UP, USING MODIFIED GEARBOXES WITH TRANSPARENT WINDOWS.

- TYPES OF LUBRIFICATION

• ON MESHING TOOTH FLANKS DIFFERENT PHENOMENA CAN BE IDENTIFIED:
- ELASTO-HYDRODYNAMIC LUBRICATION (EHD):
- MIXED LUBRICATION
- BOUNDARY LUBRIFICATION: SURFACES ARE IN CONTACT WITHOUT INTERPOSITION OF LUBRICANT OIL.



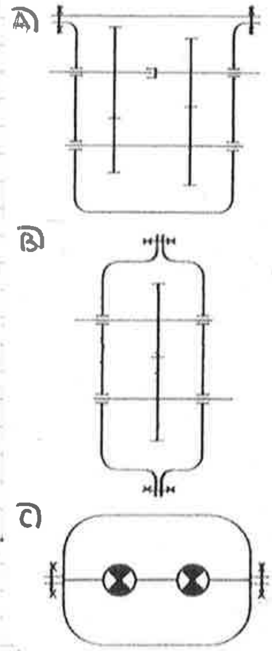
- HOUSING LAY-OUT

• HOUSING LAY-OUT CAN BE CLASSIFIED ACCORDING TO THREE ALTERNATIVE ARCHITECTURES:

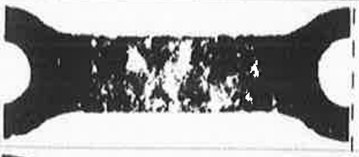
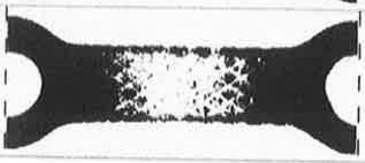
A) THROUGH HOUSING, WHEN BEARINGS SEATS ARE CUT ON THE SAME HOUSING ELEMENT WHICH RESULTS PARTICULARLY STIFF AND SIMPLE TO BE MACHINED; THERE ARE OPENINGS, CLOSED BY REMOVABLE COVERS WHICH ALLOW ASSEMBLING AND DISASSEMBLING INTERIOR PARTS.

B) END LOADED HOUSINGS: THE HOUSING IS CUT TRANSVERSELY TO SHAFTS IN TWO HALVES, THEREFORE THE BEARING SEATS OF THE SAME SHAFT REST ON DIFFERENT HOUSING PARTS.

C) TOP LOADED HOUSINGS: THEY ARE CUT ALONG SHAFTS IN TWO HALVES, THEREFORE EACH BEARING RESTS ON TWO DIFFERENT HALF SEATS.



- SEALS ANALYSIS: PHOTOGRAPHIC PRESSURE SENSITIVE FILM

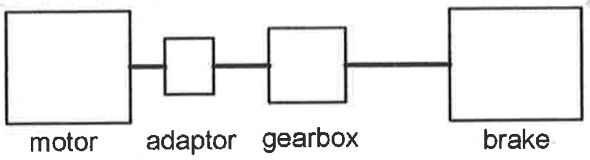
Contact imprint	Characteristic feature
	Very **uniform imprint** with high values in the middle between the bolts. "Very good".
	Marked **inhomogeneities** caused by the structure of the seal material.
	High contact pressure in the **peripheral zones of the seal**, caused by deformations during production of the seal (purching, cutting).
	**Uneven distribution**, no contact pressure in centre between the bolts.
	Effect of the **sheet steel carrier**.

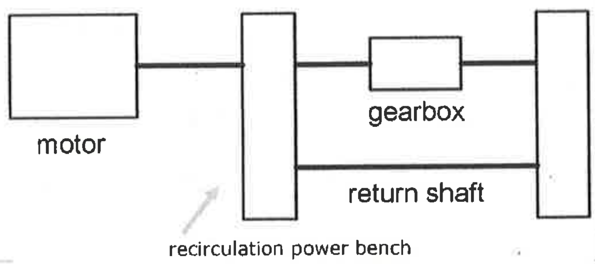
© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 95 di 260

- ENDURANCE TESTS

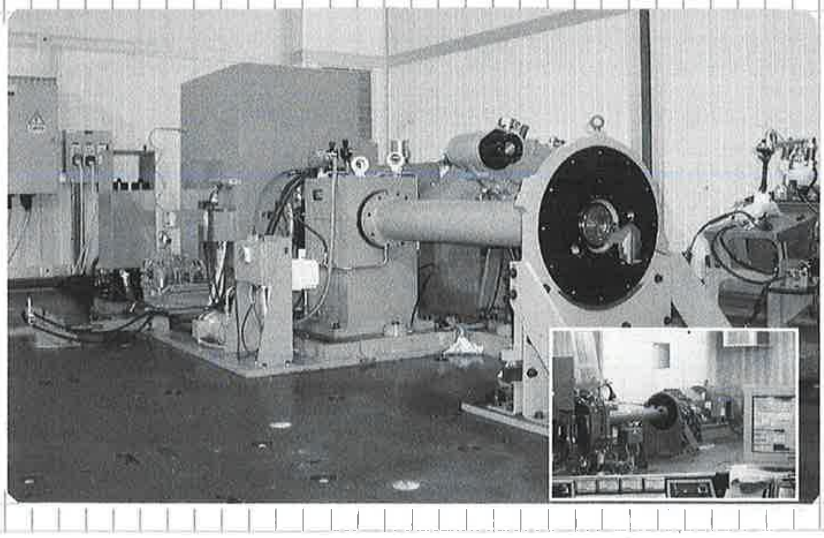
- ENDURANCE TESTS: THE COMPONENT IS WORKING FOR THE EXPECTED LIFE ACCORDING TO DIFFERENT POSSIBLE MISSION PROFILES.
- EXPECTED RESULTS CONSIST OF FAILURES THAT MUST OCCUR AFTER USEFUL LIFE ONLY; IF THEY OCCUR PREMATURELY, THEY MUST BE ANALYSED TO DESIGN CORRECTIVE COUNTERMEASURES.
- REPETITION OF ALL SCHEDULED TESTS IS IN ANY CASE REQUESTED, UNTIL SUCCESS IS REACHED. RELIABILITY CAN BE DEMONSTRATED BY REPEATING ENDURANCE TESTS ON A STATISTICALLY SIGNIFICANT PROTOTYPES NUMBER.
- ALMOST ALL FUNCTIONAL AND ENDURANCE TESTS CAN BE PERFORMED ON A BENCH OR ON A VEHICLE; IT IS PREFERABLE TO MAKE TESTS ON A VEHICLE ONLY FOR RESULT CONFIRMATION.
- BASIC TRANSMISSION TEST BENCHES ARE SIMPLE AND INCLUDE A FOUNDATION BLOCK ON WHICH A COMPLETE TRANSMISSION CAN BE INSTALLED; IT CAN BE PUT IN ROTATION BY AN ELECTRIC MOTOR OR BY AN ACTUAL ENGINE.

- SCHEMES FOR TRANSMISSION TEST BENCHES





- EXAMPLE: TRANSMISSION TEST BENCH



© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 97 di 260

- FUNCTION PERFORMED BY THE SYNCHRONIZER

- CONTENT (FUNCTIONS)

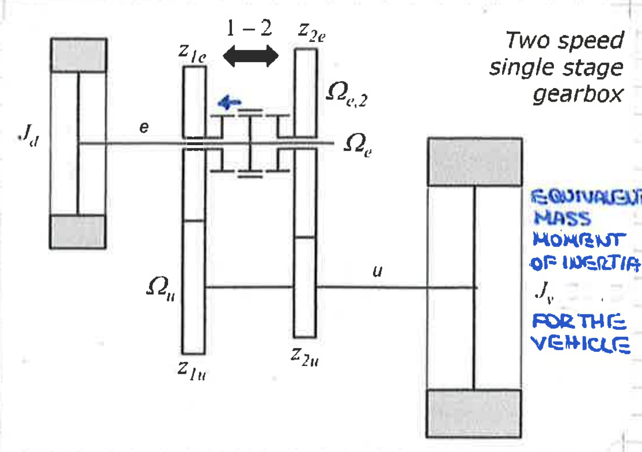
- SYNCHRONIZERS MAIN FUNCTION IS TO ENABLE CHANGING OF ACTIVE GEARS, ON A MOVING VEHICLE, WITHOUT NEGATIVE CONSEQUENCES ON GEARS MECHANICAL INTEGRITY AND ON DRIVER AND PASSENGER COMFORT.
- DURING SYNCHRONIZATION THE FRICTION CLUTCH MUST BE DISENGAGED.
- IN PRACTICE, THE MAIN FUNCTIONS ARE THEN:
  - ADAPTING SPEED OF PARTS TO BE SYNCHRONIZED BY AN ENERGY TRANSFER FROM ONE TO THE OTHER. THE MEASURE OF THIS FUNCTION IS THE SYNCHRONIZATION TIME; IT IS A SIGNIFICANT PART OF SHIFTING TIME DURING WHICH THE VEHICLE IS WITHOUT TRACTION. (AFTER THE SYNCHRONIZATION PHASE IN TERMS OF SPEED, S. HAS TO ENSURE:)
  - POSITIVELY JOINING OF SYNCHRONIZED PARTS TO TRANSMIT THE NECESSARY TORQUE; JOINT MUST BE STABLE ON TIME, WITH NO DANGER OF GEAR SELF DISENGAGEMENT
- IN ADDITION, ADDITIONAL FUNCTIONS MUST BE ASSURED:
  - MEASURING ROTATING PARTS SPEED DIFFERENCE, TO IDENTIFY THE SUITABLE MOMENT TO ENGAGE THE SPEED. (THERE IS A MECHANICAL SENSING ELEMENT) ($\Delta\omega$)
  - ENABLING POSITIVE ENGAGEMENT ONLY WHEN SPEEDS ARE EQUAL. ($\Delta\omega = 0$)

– SIMPLIFIED SCHEME OF MASSES TO BE SYNCHRONIZED

- TRANSMISSION RATIOS:

$\tau_1 = z_{1e} / z_{1u}$

$\tau_2 = z_{2e} / z_{2u}$



hp: SINCE THE VEHICLE INERTIA IS HUGE COMPARED TO THE INERTIA OF THE COMP. THAT MUST BE SYNCHRONIZED, WE CAN ASSUME THAT, DURING SYNCHRONIZATION PROCESS, $\Omega_{OUT} = CONST$

WITH 1st SPEED ENGAGED GEAR 1E ROTATES AT SAME SPEED OF THE ENGINE WHILE 2E NOT.

- WITH 1st SPEED ENGAGED:

$$\tau_1 = \frac{\Omega_u}{\Omega_e} = \frac{z_{1e}}{z_{1u}} \Rightarrow \Omega_u = \Omega_e \frac{z_{1e}}{z_{1u}} \quad \left(\tau = \frac{\Omega_{OUT}}{\Omega_{IN}} = \frac{z_{IN}}{z_{OUT}} = \frac{R_{IN}}{R_{OUT}} \Rightarrow \Omega_{OUT} = \Omega_{IN} \frac{z_{IN}}{z_{OUT}} \right)$$

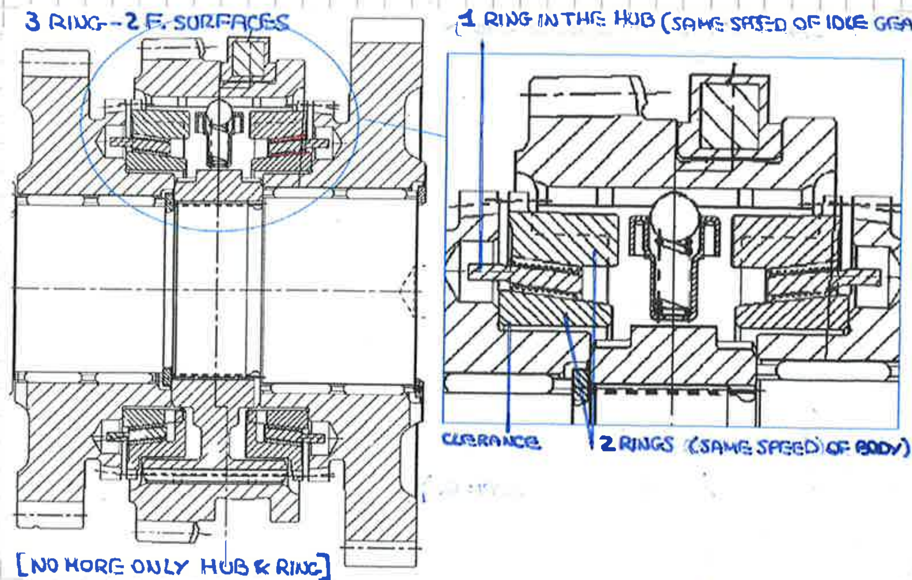
$$\Omega_{e,2} = \Omega_e \frac{z_{1e}}{z_{1u}} \frac{z_{2u}}{z_{2e}} = \Omega_e \underbrace{\frac{z_{1e}}{z_{2e}}}_{<1} \underbrace{\frac{z_{2u}}{z_{1u}}}_{<1} = \Omega_e \underbrace{\frac{\tau_1}{\tau_2}}_{<1} < \Omega_e$$

- TO ENGAGE 2nd SPEED WE MUST REDUCE Ω_e TO $\Omega_{e,2}$ VALUE, THEN

$\Delta\Omega_e = \Omega_e - \Omega_{e,2}$ (SPEED DIFF. THAT MUST BE SYNCHRONIZED)

- THE ROTATING MASSES HAVE INERTIA J_d

- DOUBLE CONE SYNCHRONIZER

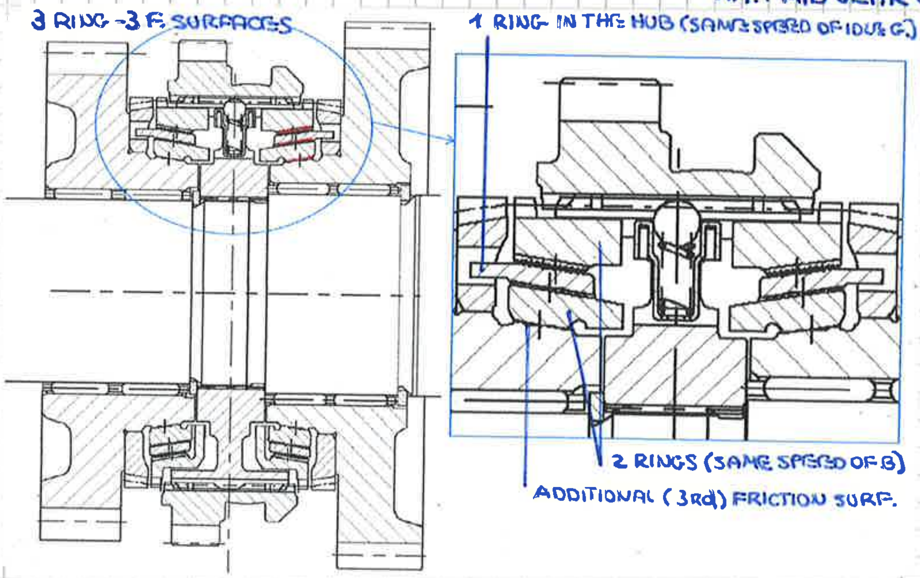


- IN ORDER TO REDUCE THE SYNCHRONIZATION TIME (t_s) (WITHOUT INCREASING LONG. DIM) WE INCREASE THE m. OF FRICTION CONE SURFACES.
- DOUBLE CONE = 2 F.C.S. (AT SAME APPLIED LEVER F.) WE DOUBLE THE AMOUNT OF T_s THAT CAN BE GENERATED FOR THE SAME AMOUNT OF DRIVER FORCE APPL. TO SLEEVE.

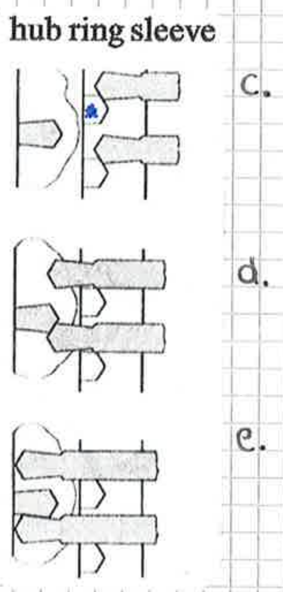
=> 2 RINGS + 1 RING INSERTED IN A HUB GROOVE

APP.: MID GEAR OF LARGE G. OR LOWEST FOR SMALL

- TRIPLE CONE SYNCHRONIZER



- TRIPLE CONE: 3 F.C.S. THE MEAN DIAMETER OF THE 3 FRICTION SURFACES IS LIGHTLY DIFFERENT BUT WE CAN SAY WE ARE INCREAS. (BY A FACTOR OF 3) THE FRICTION T_s WITHOUT INCREASING (JUST A LITTLE) THE SPACE OCCUPANCY.
- FINE GROOVES ON F.S. TO AVOID LUBRICATION.



c. WHEN RING AND HUB ARE SYNCHRONOUS, THE FRICTION TORQUE IS ZERO AND DOG TEETH TAPERED ENDS* ROTATE THE HUB OF THE ANGULAR THICKNESS OF HALF TOOTH, UNDER THE PRESSURE EXERTED BY THE SHIFT STICK.

d. THE SLEEVE IS NOW FREE TO MOVE ACROSS THE RING AND TO MATCH THE HUB; AGAIN TEETH TAPERED ENDS WILL ROTATE THE HUB, IF NECESSARY; IN THIS PHASE THE PUSH ELEMENTS PLUNGERS ("PISTONI") ARE COMPLETELY RETRACTED AND DO NOT WITHSTAND SLEEVE MOTION ANYMORE.

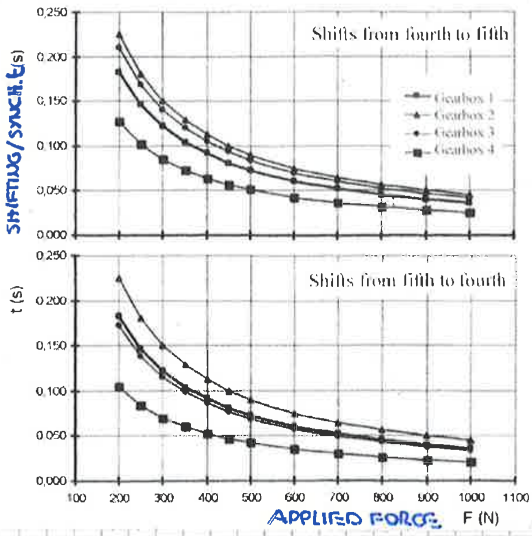
e. NOW A POSITIVE ENGAGEMENT IS MADE; AT THE END OF THIS PHASE THE DRIVER WILL STOP PUSHING SHIFT STICK; THE COUNTER TAPERED SHAPE OF DOG TEETH WILL RETAIN THE SLEEVE IN THE ENGAGED POSITION UNDER THE ACTION OF THE ENGINE TORQUE.

- SYNCHRONIZATION DIAGRAMS : TIME VS SHIFTING FORCE

4 different gearboxes

(UPSHIFT)
SHIFTS IV → V GEAR

(DOWNSHIFT)
SHIFTS V → IV GEAR



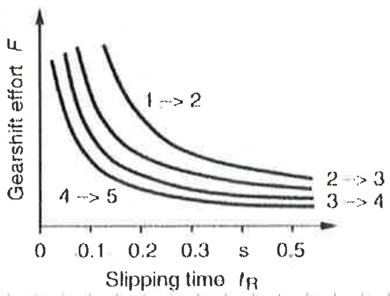
DIFFERENT PERFORMANCES DEPENDING ON TYPE OF GEARB. THE TRENDS ARE SIMILAR.

CHANGING TYPE OF G. WE CHANGE THE PERFORMANCE.

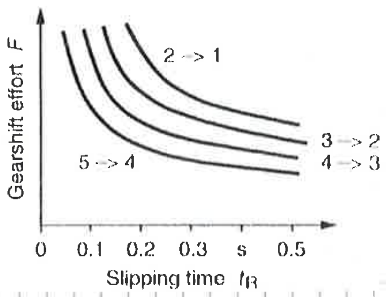
- GEARSHIFT EFFORT VS SLIPPING TIME

for different gearshifts

upshifts



downshifts



IF WE COMPARE DIFFERENT GEARSHIFTS:

- THE MOST CRITICAL PERFORM. IS THE ONE ASSOCIATED TO THE 1st GEAR (1st->2nd | 2nd->1st) (IN PARTICULAR THE DOWNSHIFT)
- PASSING FROM LOW TO HIGH GEAR THE SYNCH. EFFORT REDUCES (F, t COMBINATION IS LESS CRITICAL)

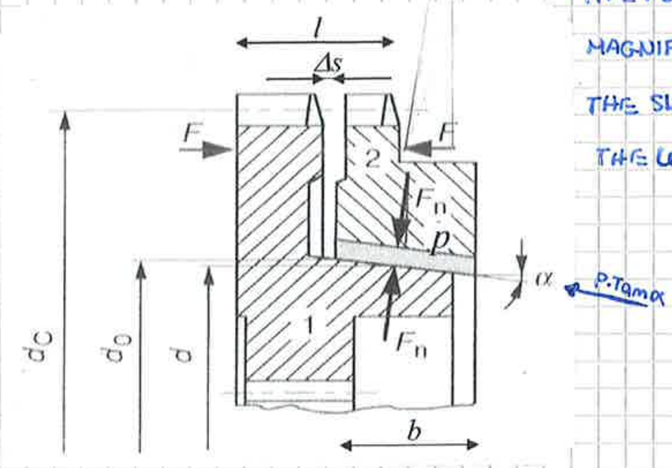
• GEOMETRIC CRITERIA

- SCHEME FORCES AND PRESSURE FOR THE SYNCHRONIZATION PHASE

$F_n = \frac{F}{\sin\alpha}$ F: FORCE APPLIED ON THE SLEEVE.

THE FORCE ACTING ON FRICTION SURFACES IS MAGNIFIED WITH RESPECT TO THE FORCE ACTING ON THE SLEEVE.

THE LOWER THE ANGLE (α) THE HIGHER THE FORCE (F_n)



- SYNCHRONIZATION TORQUE (M_s) AND CONE ANGLE (α)

- A FORCE (F) APPLIED TO THE SLEEVE ORIGINATES A NORMAL PRESSURE (P) ON CONTACTING SURFACES:

$F = F_n \sin\alpha = P \cdot b\pi d \sin\alpha \Rightarrow P = \frac{F}{b\pi d \sin\alpha}$ (STARTING FROM WE EVALUE P)

WITH:

b: CONICAL SURFACE WIDTH

d: MEAN DIAMETER

α: CONE ANGLE

- THE SYNCHRONIZATION TORQUE IS:

$$M_s = \frac{P \cdot b\pi d^2 f}{2} = \frac{F d f}{2\sin\alpha} = f \cdot F_n \cdot \frac{d}{2}$$

WITH:

f: FRICTION COEFF.

- M_s WILL INCREASE ($M_s\uparrow$) IF $\begin{cases} f\uparrow \\ \alpha\downarrow \end{cases}$

- FOR UNLOCKING RINGS, THE LIMIT CONDITION ON FORCES IS:

$P\tan\alpha = Pf$ $\begin{cases} P\tan\alpha: \text{PROJECTION OF THE AXIAL F ALONG THE FRICTION S.} \\ Pf: \text{FRICTION FORCE PER UNIT SURFACE} \end{cases}$

- AND THEN:

$\tan\alpha > f$

$\tan\alpha$ MUST BE LOW TO MAGNIFY THE SYNCH. TORQUE $\left(\tan\alpha\downarrow \Rightarrow f\frac{F}{\sin\alpha\downarrow}(d/2) = M_s\uparrow\right)$ BUT NOT TOO MUCH TO AVOID THE BLOCKING OF THE AXIAL MOTION OF THE RING ($\tan\alpha > f$).

• FUNCTIONAL CRITERIA

– THERMAL POWER (P_{MAX}) AND ENERGY (E)

• THE TORQUE BALANCE DURING THE SYNCHRONIZATION PHASE IS:

$$M_S + M_a = -J_{R,EQ}\frac{d\Omega}{dt}$$

: FOR UPSHIFTS ($\Omega_2 < \Omega_1$) $M_S = -J_{R,EQ}\frac{d\Omega}{dt} - M_{LOSS}$

← WE ASSUME TO BE CONST.

$P_{MAX} = M_S(\Omega_2 - \Omega_1)$

$E = \frac{1}{2}J_{R,EQ}(\Omega_2^2 - \Omega_1^2) + \frac{1}{2}M_a(\Omega_2 - \Omega_1)\Delta t$

WITH:

M_S: SYNCHRONIZATION TORQUE

M_a: FRICTION TORQUE OF MECHANICAL LOSSES (DRAG TORQUE) (SUM OF ALL CONTR. FOR EX: FRICT. BETWEEN TEETH OF GEARS INVOLVED IN THE DEC/ACC)

Ω: ROTATIONAL SPEED UPSTREAM ELEMENTS (PRIMARY SHAFT)

$J_{R,EQ}$: ROTATING MASS TO BE SYNCHRONIZED

• THERMAL POWER (P_{MAX})

$$P_{MAX} = M_S(\Omega_2 - \Omega_1)$$

• THERMAL ENERGY (E)

$$E = \frac{1}{2}J_{R,EQ}(\Omega_2^2 - \Omega_1^2) + \frac{1}{2}M_a(\Omega_2 - \Omega_1)\Delta t$$

WITH:

1: INITIAL CONDITION

2: FINAL CONDITION

$$M_S = -J_{R,EQ}\frac{(\Omega_2 - \Omega_1)}{\Delta t} - M_{LOSS}$$

THE ACTION OF THE FRICTION (M_a) IN CASE OF UPSHIFT HELPS THE SYNCH. PROCESS (BECAUSE FRICTION FORCE TENDS ALWAYS TO DECELERATE COMPONENTS); IN THE DOWSHIFT CASE IT ACTS AGAINST THE ACCELERATION OF THE PRIMARY SHAFT (WE NEED AN **EXTRA TORQUE** TO BALANCE THE ACTION OF THE FRICTION)

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 111 di 260

- MATERIAL REFERENCE DESIGN VALUES

Material	Friction coefficient f	Reference speed V m/s	Specific Energy E_s J/mm²	* Specific Power P_s W/mm²	Contact pressure p N/mm²
Steel / Bronze	0,08 – 0,12	5	0,09	0,45	3
Steel / Steel Mo	0,08 – 0,12	7	0,53	0,84	6

* WE HAVE TO TAKE INTO CONS. THE EVENTUAL PRESENCE OF THE FINE GROOVES. (→ NOMINAL SURFACE IS REDUCED)

← TO REDUCE SYNCH. DIMENSION WE CAN USE STEEL MO INSTEAD OF BRONZE FOR THE RING TO INCREASE P (WE DOUBLE P_s, P AND INCREASE E_s!!)

• SYNCHRONIZER REFERENCE SPEED (V):

$$V=\frac{d}{2}(\Omega_2-\Omega_1)$$

- MATERIALS MATCHING FOR SYNCHRONIZER RING AND HUB

• THE ABOVE MATERIALS HAVE TO BE DEFINED TAKING INTO ACCOUNT THE FOLLOWING NEEDS:

1 - WEAR MUST BE SMALLER ALSO WITHOUT THE HELP OF HYDRO-DYNAMIC LUBRICATION;

2 - MATERIALS MUST BE EASILY MACHINED;

3 - FRICTION COEFF. (f) MUST BE AS CONSTANT AS POSSIBLE IN PRODUCTION;

4 - FRICTION COEFF. (f) MUST BE INSENSITIVE TO WEAR AND TEMPERATURE;

5 - MATERIALS MUST WITHSTAND POSSIBLE OVERLOADS.

• THEN, THE AVAILABLE OPTIONS ARE:

- HUBS: Cr Mn STEEL OR Cr Mo STEEL;
- RINGS: SPECIAL SHOT PEENED BRASSES OR BRONZES, OR STEELS, COATED WITH A Mo LAYER OR PAPER LAYER, WITH ADVANTAGE ON THE FRICTION COEFF. (f); RECENTLY, OTHER MATERIALS HAVE BEEN USED, LIKE SINTERED STEELS, Cu-Zn OR Al ALLOYS, CARBON.

[LOOK AT SLIDE 23/24, T9: LINK OF AN EXPLANATION]

T10 START-UP DEVICES FOR MT

- **CLUTCH MAIN AND ADDITIONAL FUNCTIONS**
  - CONTENT
- **DRYCLUTCH**
  - CONTENT
  - DRYCLUTCH COMPONENTS
  - DRYCLUTCH ENGAGED / DISENGAGED SCHEME
  - CLUTCH INTERNAL DISENGAGEMENT MECHANISM
  - DIAPHRAGM SPRING
  - ADVANTAGES OF DIAPHRAGM SPRING
  - PRESSURE PLATE FORCE
  - PRESSURE PLATE TRAVEL VARIATION WITH WEAR
  - INTERNAL DISENGAGEMENT MECHANISM PULLED/PUSHED
- **DRIVEN PLATE**
  - CONTENT (SCHEME)
  - COMPONENTS
  - COMPONENTS AND CHARACTERISTICS
  - DRIVEN PLATE HUB WITH SECONDARY TORSIONAL DAMPER
  - CLUTCH WITH DOUBLE MASS FLYWHEEL (DMF)
  - SINGLE MASS FLYWHEEL (SMF) VS DUAL MASS FLYWHEEL (DMF)
  - SMF VS DMF: VIBRATION ISOLATION
  - SMF VS DMF: MODE SHAPES
  - SMF VS DMF: FRF IN 1st SPEED
  - SMF VS DMF: FRF IN 5th SPEED
  - DRIVEN PLATE FRICTION MATERIAL PROPERTIES
  - EXTERNAL DISENGAGEMENT MECHANISMS - THRUST BEARING
  - SELF ADJUSTING TECHNOLOGY (S.A.T)
  - WEAR COMPENSATION MECHANISM
- **DESIGN CRITERIA**
  - CONTENT
  - MECHANICAL DESIGN SPECIFICATIONS
  - THERMAL DESIGN SPECIFICATIONS

- MAIN FUNCTIONS
- DRY CLUTCH
- DMF CLUTCH
- DIAPHRAGM SPRING
- PRESS/PULL SPRING
- DRIVEN PLATE (SMF)
- DMF CLUTCH
- LUMPED P. MODELS
- DMF VS SMF
- MATERIALS
- EXT. DIS. MECH.
- S.A.T.
- DESIGN CRITERIA
- TH. DESIGN PARAMET.

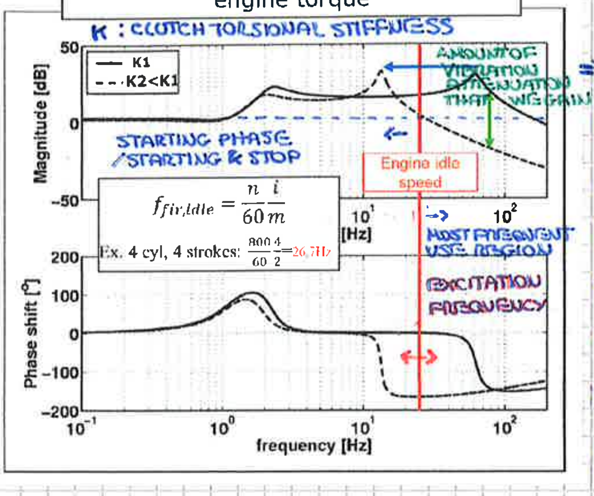
[TEXTBOOK: VOL. I, PAR. 11.1]

+2 - TO CONTROL THE DRIVELINE TORSIONAL DYNAMICS, IN SUCH A WAY: THROUGH THE COMPONENTS INTEGRATING INTO THE C.

+2.2 → TO ATTENUATE A↓ VIBRATION AND NOISE WHEN HARMONICS OF THE ENGINE TORQUE MATCH RESONANCE FREQUENCIES OF TRANSMISSION MODES (DAMPING / HYSTERESIS); (+ INCREASING DAMPING OR WORKING ON H. OF COMPONENTS)

+2.1 → TO REDUCE VIBRATION TRANSMISSIBILITY FROM ENGINE TO TRANSMISSION INPUT SHAFT (STIFFNESS AND INERTIA). (+ REDUCING STIFFNESS OR INCREASING INERTIA OF INPUT SHAFT)

Torsional vibration isolation: primary shaft acceleration / engine torque



2.1 + $\dot{\omega}/T_E$ = 0 [dB] ⇒ NO AMPLIFICATION, NO ATTENUATION OF VIBRATIONS GENERATED BY ICE;

⇒ WITH $K_{CLUTCH} = K2$ (<K1) ⇒ ATTENUATION OF VIBRATIONS BECAUSE, REDUCING THE TORSIONAL STIFFNESS, WE ANTICIPATE THE DAMPING MODE <-> WE HAVE NO RESONANCE PEAKS IN THE MOST USED WORKING RANGE (WE DON'T CROSS RESONANCE P.) → 1 AND WE ALSO IMPROVE THE ISOLATION OF VIBRATION / VIBRATION ATTENUATION → 2; 2 EFFECTS

IN THIS WAY, HOWEVER, WE EXPECT VIBRATIONS DURING START & STOPS.

THE EXCITATION OF THE SYSTEM IS DIRECTLY ASSOCIATED TO THE FREQUENCY APPLIED THROUGH THE TORQUE IRREGULARITY OF THE ICE.

⇒ REDUCING THE TORSIONAL STIFFNESS OF THE CLUTCH WE REDUCE THE NATURAL FREQ. TO VIBRATE OF THE (TORSIONAL DAMPER OF THE) CLUTCH

THE RED LINE REPRESENTS THE EXCITATION FREQUENCY OF THE ENGINE WHEN ROTATES AT IDLE ($f_{FIR,IDLE} = \frac{n}{60}\frac{i}{m} = \frac{800}{60}\frac{4}{2} = 27.7$ [Hz] : EXC. FREQ. AT IDLE)

WE CAN CALCULATE THE FIR FREQUENCIES (FIR: FINITE IMPULSE RESPONSE), THAT ARE THE MAIN EXCITATION FREQUENCIES DUE TO THE IGNITION OF THE CYL.

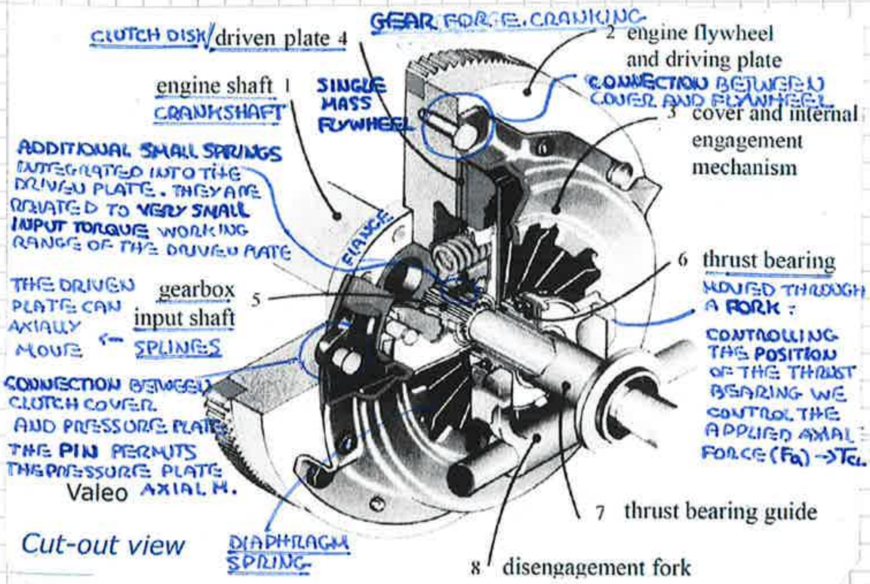
$f_{FIR} = \frac{n}{60}\frac{i}{m}$ (COMPARED TO THE f_m (MODE 1, 2, ...) PROPER OF THE CLUTCH)

n: [RPM] ENGINE

i: NUMBER OF CYLINDERS

m { 2 : FOR 4T ENGINE (4 STROKES ENG.) ; 1 : FOR 2T EN. }

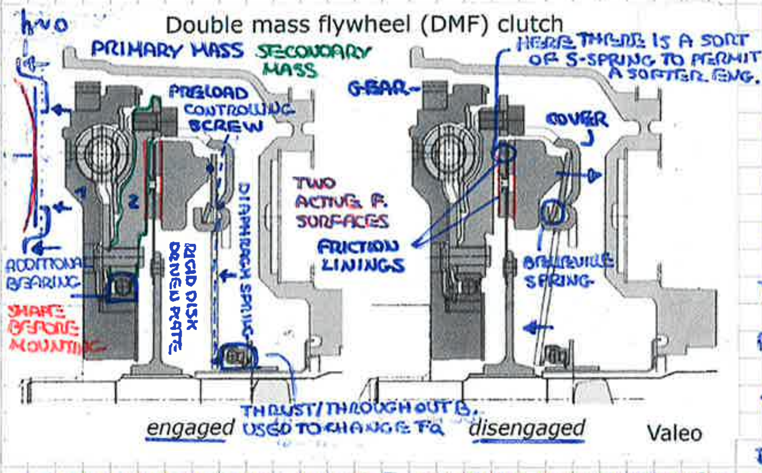
2.2 + TO ATTENUATE VIBRATION AND NOISE, SO TO REDUCE THE PEAKS MAGNITUDE, WE ADD SOME DAMPING INTO THE SYSTEM (CLUTCH) = ADD SOME FRICTION (FRICTION PLATES). (DAMPING THROUGH FRICTION) (MORE NEAR TO AN HYSTERESIS DAMPING THAN A VISCOUS ONE).



ON THE FLYWHEEL THERE IS A DIRECTLY CUT GEAR:
→ FOR ENGINE CRANKING (START) TRANSMISSION CONNECTION WITH THE ENGINE STARTER, THROUGH A PINION ⇒ HIGH REDUCTION RATIO $\left(i = \frac{\omega_{in}}{\omega_{out}}\right)$

- DRY CLUTCH ENGAGED/DISENGAGED SCHEME

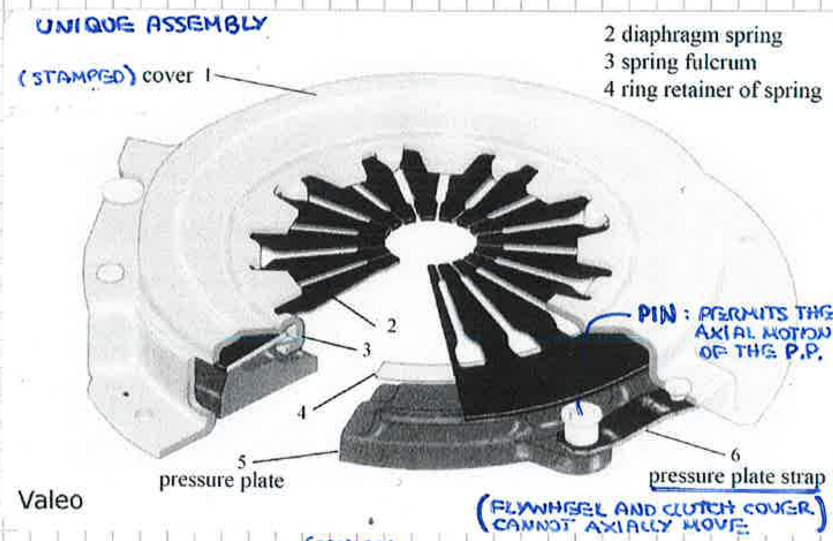
* (BENDED COIL SPRINGS) ⇒ PRIMARY-SEC. MASSES: LOW STIFFNESS CONN.



DOUBLE/DUAL MASS FLYWHEEL:
+ VERY BIG COIL SPRINGS (NO MORE STRAIGHT AXES): BENDED AXES = ARC AXES; THEY ARE INSERTED IN A DEDICATED GROOVE IN WHICH THEY CAN SLIDE, AND ARE USED TO TRANSFER THE TORQUE FROM THE PRIMARY MASS TO THE SECONDARY MASS.*
+ WE HAVE TO ADD AN ADDITIONAL BEARING TO ALLOW THE RELATIVE MOTION OF P. & S. MASSES
- NO MORE SPRINGS INTEGRATED INTO DRIVEN (THE TORSIONAL ELASTICITY OF THE SYSTEM IS MOVED UPWARD TO THE LEFT, CLOSER TO THE ENGINE)

• 3 CIRCUMFERENCES: WHERE F_a IS APPLIED TO P.P. THROUGH D.S. ...

- CLUTCH INTERNAL DISENGAGEMENT MECHANISM



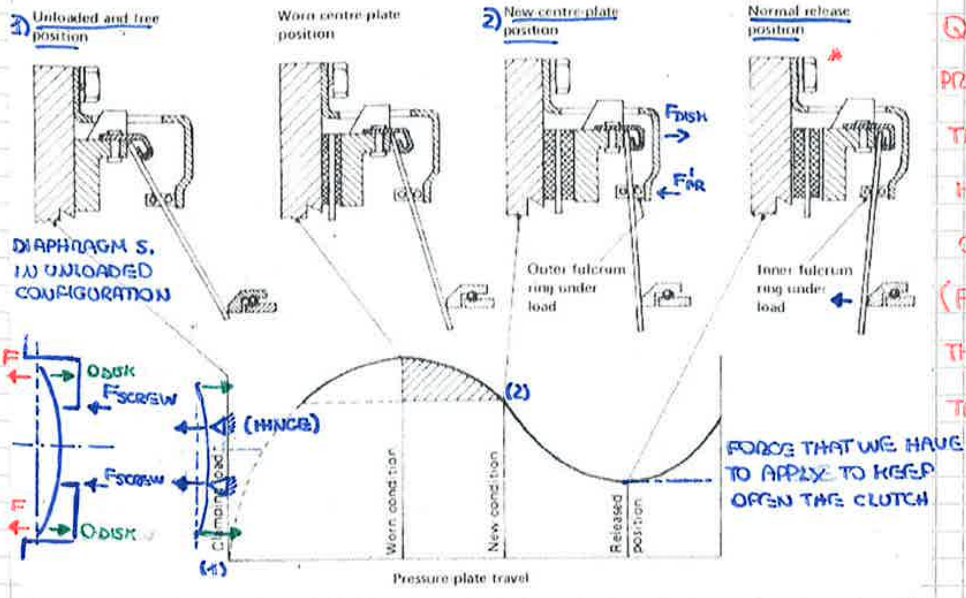
• THE DIAPHRAGM SPRING IS INSTALLED THROUGH A FULCRUM REALIZED THROUGH A BELLEVILLE SPRING. THE FULCRUM IMPOSES A SORT OF HINGE

→ CONSTRAINTS THE P.P. TO ROTATE WITH THE COVER (TORSIONAL RIGID CONN.) BUT PERMITS AXIAL MOV. (FREE AX. CON.)

DIAPHRAGM SPRING: TAPERED/CONICAL STEEL DISK: IT SHOWS SOME RADIAL CUTS TO INCREASE FLEXIBILITY

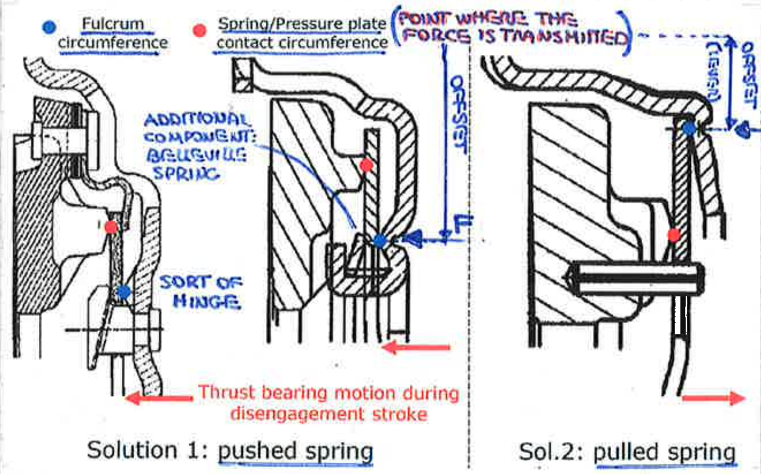
FINGERS: WE APPLIE THE FORCE, THROUGH THE THRUST B., ON THE FINGER ENDS; THE FORCE IS TRANSFERED TO THE EXTERNAL CIRCUMFERENCE AND SO TO THE PRESSURE PLATE.

- PRESSURE PLATE TRAVEL VARIATION WITH WEAR



Q. TO REACH THE DESIRED PREL., DO WE ONLY INTERPOSE THE NEW CLUTCH DISK, OR HAVE WE ALSO TO ACT ON OTHER COMPONENTS (FOR EX. THE SCREW* OR THE POSITION OF THE THRUST BEARING)?

- INTERNAL DISENGAGEMENT MECHANISM PULLED/PUSHED



SOLUTION 1: TO DISENGAGE THE CLUTCH WE HAVE TO MOVE THE THRUST BEARING : **dx → sx (PUSH)**

SOLUTION 2: **sx → dx (PULL)**

(DIFFERENT CONTACT CIRCUMF. POINTS)

SOL. 1 { EXTERNAL **DS/P.P.** C. POINT; INTERNAL **FULCRUM** C. POINT

WE HAVE TO **PUSH** THE **THRUST B.** IN ORDER TO ACHIEVE THE **ROTATION OF D.S.** AROUND THE FULCRUM.

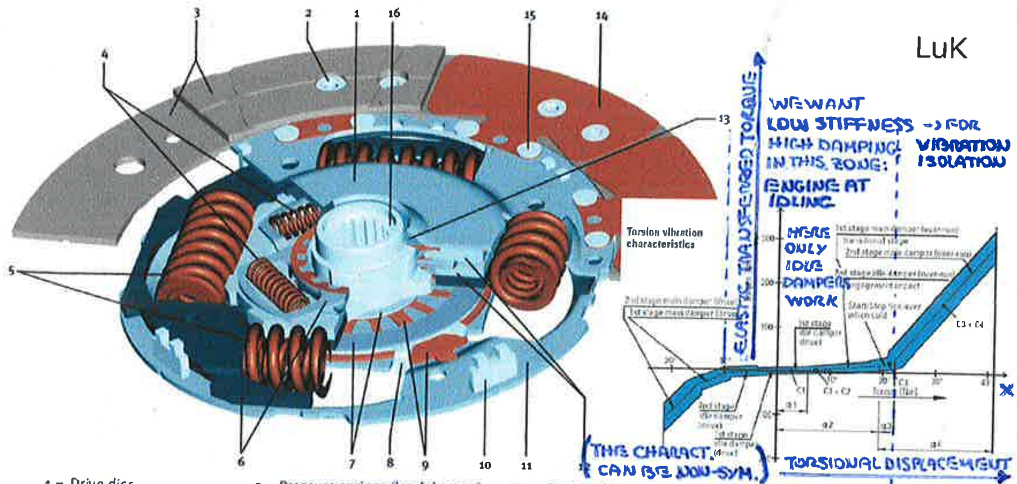
THE **BELLEVILLE SPRING** IS USED TO CREATE A **SPECIFIC CIRCUMFERENCE** WITH RESPECT TO WHICH THE DESIRED ROTATION, SO THE DEF., OF THE B.S. CAN BE OBTAINED.

SOL. 2 { EXTERNAL **FULCRUM** C. POINT; INTERNAL **D.S/P.P.** C. POINT

WE HAVE TO **PULL** THE **THRUST** B. IN ORDER TO DISENGAGE THE CLUTCH.

- USUALLY USED FOR **HIGH TORQUE ENGINES** FOR WHICH WE HAVE TO APPLY AN HIGHER F, WHICH MEANS HIGHER STRESS IN THE CLUTCH COVER.
- **BETTER USE OF THE SPACE**;
- THE **CLUTCH COVER** IS **LESS STRESSED** BY BENDING WITH RESPECT TO SOLUTION 1 WHICH IS STRESSED BY AN HIGHER BENDING MOMENT DUE TO THE LONGER **LEVER/OFFSET** ⇒ **THINNER COVER** (SAME F COMPARISON) OR **HIGHER FORCE**, SO HIGHER $T_{cl,t}$ (SAME THICKN. COMPARISON)
- **MORE EXPENSIVE**

- COMPONENTS AND CHARACTERISTICS



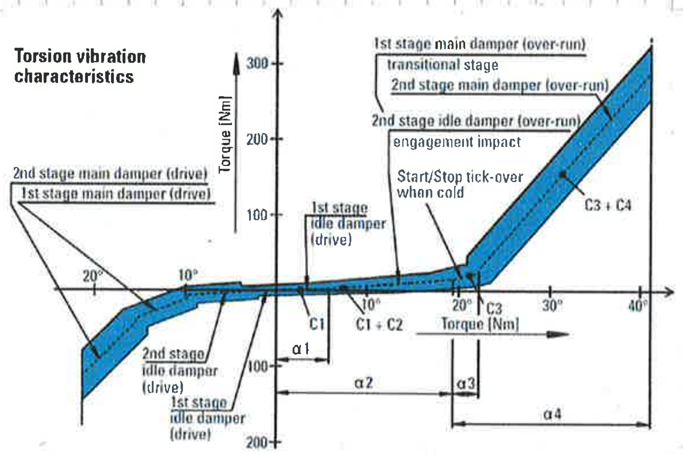
TWO DAMPING SYST.

- MAIN DAMPERS (BIGGEST SPRINGS) ARE CALLED: PRESSURE SPRINGS OR LOAD DAMPERS
- SECONDARY DAMPERS MORE CLOSE TO THE AXES ARE CALLED: PRESSURE SPRINGS OR LOW-LOAD DAMPERS OR IDLE DAMPERS

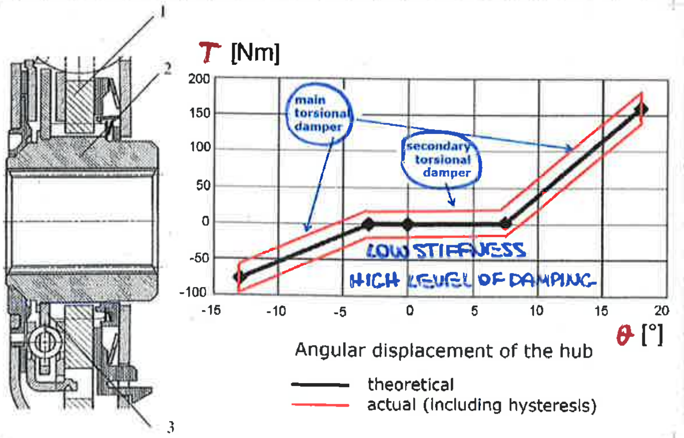
1 = Drive disc
2 = Friction lining rivet
3 = Friction lining
4 = Pressure springs (idle/low-load damper) RESPONSABLE OF THE HYSTERESIS OF S.
5 = Pressure springs (load damper)
6 = Hub flanges
7 = Friction rings/DISKS/WASHERS
8 = Support disc
9 = Diaphragm springs
10 = Spacer plate
11 = Mating disc
12 = Damper cage assemblies (idle/low-load damper)
13 = Centering taper
14 = Axial spring cushion segment (PRELOADED)
15 = Spring cushion rivet
16 = Hub

FOR HIGHER TORQUE WE NEED HIGHER STIFFNESS FOR THE DAMPING SYSTEM (TO AVOID THE PACK OF THE COIL SPRING)

- ZOOM:



- DRIVEN PLATE HUB WITH SECONDARY TORSIONAL DAMPER



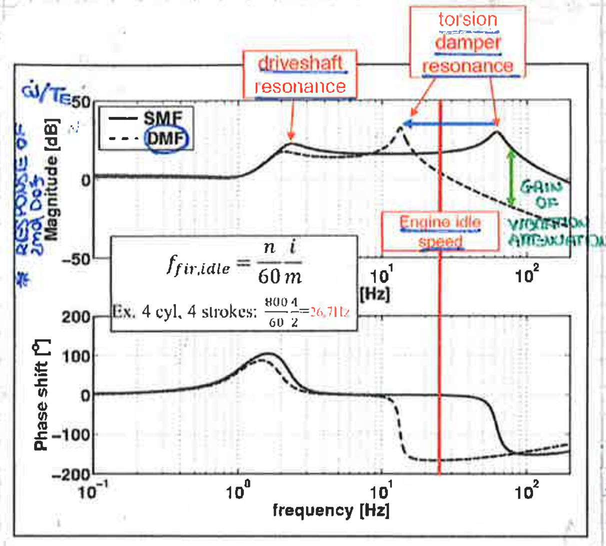
1 : HUB FLANGE (DISK 1)

2 : CLUTCH HUB

3 : ADDITIONAL DISK TORSIONALLY FIXED TO C. HUB; IT IS CONNECTED TO THE HUB FLANGE THROUGH THE SECONDARY DAMPER SYSTEM (MADE OF IDLE DAMPERS + FRICTION WASHERS + BELLS) IN THIS CONFIGURATION WE HAVE RELATIVE DAMPED MOTION ALSO BETWEEN THE HUB AND HUB FLANGE (DISK 1), IN ORDER TO ABSORB VIBRATIONS WHEN THE ENGINE IS AT IDLE.

- DMF VS SMF : VIBRATION ISOLATION

- BACKWARDS SHIFT OF 2nd NATURAL FREQUENCY (TORSIONAL DAMPER RESONANCE) OUT OF NORMAL OPERATING RANGE.
- + THEREFORE, REDUCING THE STEADY-STATE TORSIONAL VIBRATIONS OF THE TRANSMISSION.
- + SOLVES THE TD RESONANCE ISSUE FOR ENGINE SPEEDS ABOVE IDLING SPEED. * THE VIBRATION ISOLATION INCREASES *(HERE ONLY 2nd FREQ. RESPONSE FUNCT., SO THE RESPONSE OF 2nd DoF, IS REPORTED SO THE ω OF THE TRASM. WRT T. EXCIT. OF ICE)
- − DETERIORATES ENGINE CRANKING
- − DETERIORATES VEHICLE ACCELERATION PERFORMANCE (PARTICULARLY AT HIGH SP.)



3rd MODE: $f = 78.8$ [Hz]: TORSIONAL DAMPER MODE
MASSES 1,3 STAY STATIONARY <-> NO OSCILLATION OF ENGINE AND VEHICLE; HIGH OSCILLATION OF MASS/INERTIA 2, SO OF THE TRANSMISSION.

PASSING TO ANALYSE THE DMF SOLUTION:

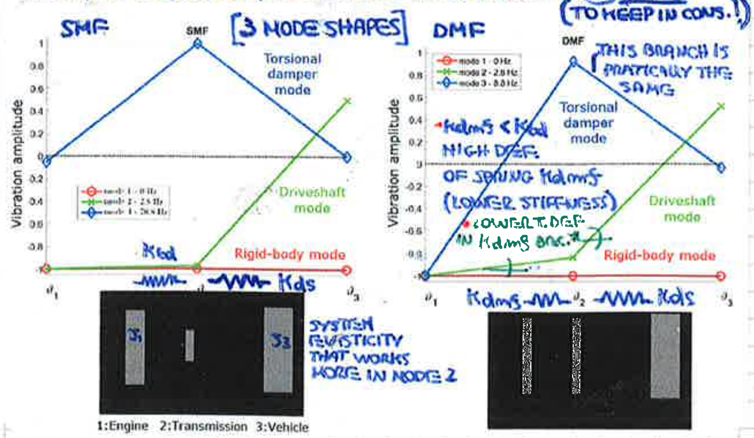
1st MODE: $f = 0$ [Hz]: SAME RIGID-BODY MODE

2nd MODE: $f = 2.8$ [Hz]: QUITE SIMILAR DRIVE-SHAFT MODE
WE JUST HAVE LESS OSCILLATION OF MASS 2 ($K_{dmf} < K_{td}$)
HIGH TORSIONAL DEFORM. IN K_{ds} <-> BIG OSC. A. DIFF J_2, J_3
* LOWER T. DEF. IN K_{dmf} BECAUSE 1,2 ARE CLOSE TO EACH O.

(*' ω_m ↓ BECAUSE K ↓ AND WE MOVE PART OF IN. TO J_2) 3rd MODE: *' $f = 8.8$ [Hz]: TORSIONAL DAMPER MODE: ENGINE IS NO MORE STATIONARY: HIGH OSC. OF J_1; J_1, J_2 IN PH. OPP.

- SMF VS DMF : MODE SHAPES

Example: 6-speed Dry Dual Clutch Transmission in 1st speed. (TO KEEP IN CONS.!)



-> 3 MASSES -> 3 DoF -> 3 ω_m -> 3 V. MODES

1st MODE: $f \sim 0$ [Hz]: RIGID BODY MOTION (THE SYSTEM IS FREE TO ROTATE RIGIDLY)

2nd & 3rd MODES: FLEXIBLE MODES: WE CAN ASSOCIATE THE MODE TO A SPECIFIC COMPONENT IN THE POWERTRAIN.

$\theta_1, \theta_2, \theta_3$: AMPLITUDES OF OSCILLATIONS OF EACH MASSES (J_1, J_2, J_3)

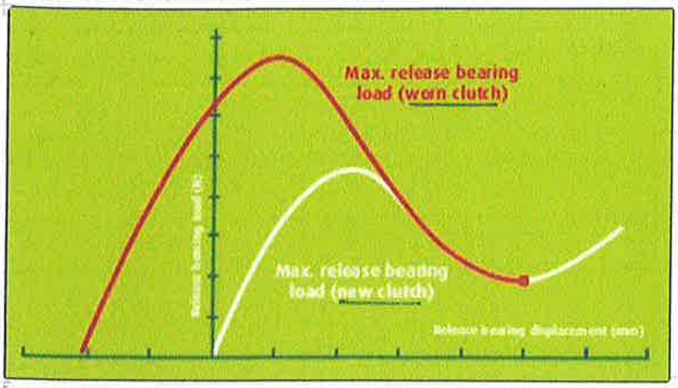
2nd MODE: $f = 2.8$ [Hz]: DRIVESHAFT MODE: ENGINE (θ_1) AND TRANSMISSION (θ_2) MASSES OSCILLATES WITH THE SAME AMPLITUDE: NO DEFORMATION OF SPRING K_{td} (J_1 AND J_2 RIGIDLY CONNECTED THROUGH AN INFINITELY RIGID SHAFT; THE FLEXIBILITY OF THE TORSIONAL DAMPER DOES NOT ENTER TO PLAY)
HIGH AMOUNT OF TORSIONAL DEFORMATION IN SPRING K_{ds} <-> VERY BIG DIFF. IN AMPL. OF OSC. OF J_2, J_3
IF SIGN OF A. IS THE SAME THE A. ARE IN PHASE; IF THERE IS A CHANGE OF SIGN: A. OUT OF PHASE
WHEN θ_2 IS IN THE INCREASING PART OF THE WAVE, θ_3 IS IN THE DECREASING PART (180° OF PHASE SHIFT)

- DRIVEN PLATE FRICTION MATERIAL PROPERTIES

- FRICTION MATERIALS USED IN CLUTCH LININGS HAVE SEVERE PERFORMANCE REQUIREMENTS. THE ENERGY CONVERSION (MECHANICAL TO THERMAL) MUST BE ACCOMPLISHED WITH A MINIMUM OF WEAR ON THE CONTACTING PARTS.
- FRICTION COMPOSITES ARE COMPOSED OF A BALANCE MIXTURE OF RESIN AND ADDITIVES TO ACHIEVE DESIRED CHARACTERISTICS.
- FRICTION MATERIALS COMMONLY CONSIST OF A SINTERED METAL MATRIX (E.G. SINTERED LEAD BRONZE AND IRON POWDERS) INTO WHICH MINERAL, METALLIC, NON-METALLIC OR CERAMIC AGENTS, AS WELL AS ABRASIVES AND DRY/SOLID LUBRICANTS ARE EMBEDDED.
- GRAPHITE AND MOLYBDENUM DISULPHIDE ARE SUITABLE AS DRY LUBRICANTS, HOWEVER CERAMIC ADDITIVES AND MINERALS, SUCH AS QUARTZ AND CORUNDUM, MAY BE USED TO INCREASE THE FRICTION COEFF. (FRICTION REINFORCES).
- SEMI-METALLICS ARE GOOD SUBSTITUTIONS FOR ORGANIC AND ASBESTOS ("AMIANTO") MATERIALS; SOME ORGANIC COMPONENTS ARE HOWEVER USED TO REACH DESIRABLE PROPERTIES, ADVANTAGES OF SEMI-METALLICS INCLUDE:
  - IMPROVED FRICTION STABILITY; (WE HAVE TO MANTAIN CONST. FRICTION COEFF.)
  - HIGH TEMPERATURE WEAR RESISTANCE;
  - HIGH PERFORMANCE WITH MINIMAL NOISE.

- SELF ADJUSTAMENT TECHNOLOGY (S.A.T)

RELEASE BEARING LOAD [N]



THIS VALUE IS **8 TO 12 [Kg]** (1 [Kg] = 9.8 [N]) WHEN THE CLUTCH IS **NEW** AND INCREASES TO **15-20 [Kg]** WHEN THE CLUTCH IS **WORN.**

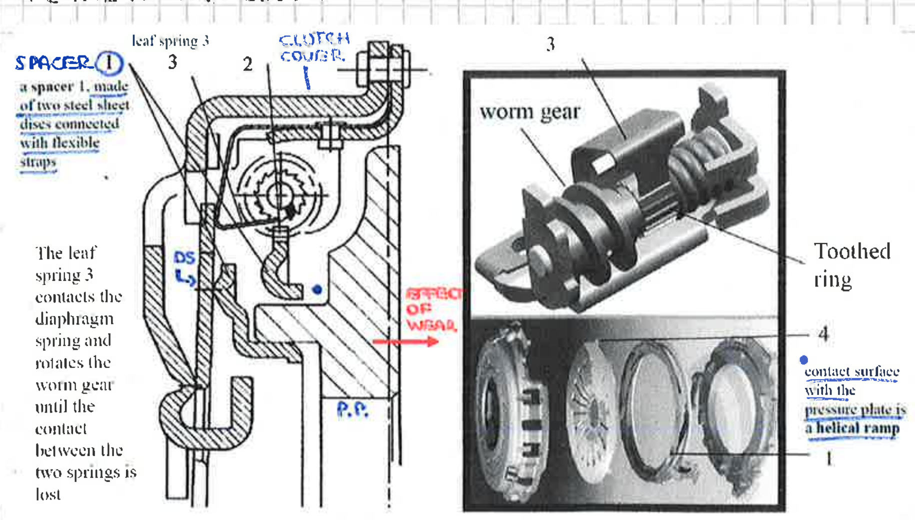
← WE ARE CONSIDERING THE **EFFECT OF WEAR**

RELEASE BEARING DISPLACEMENT [mm]

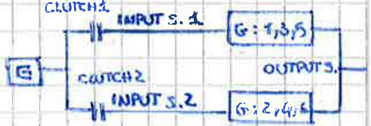
- **DUE TO THE LOSS OF THICKNESS** OF THE FACINGS, THE **PRESSURE PLATE** MOVES **TOWARDS THE FLYWHEEL**, AND THE **DIAPHRAGM IN CONTACT** WITH THE PRESSURE PLATE **CHANGES ANGLE** ACCORDINGLY.
- THE **ACTUATION FORCE** THAT NEEDS TO BE APPLIED TO THE SPRING VIA THE CLUTCH PEDAL **INCREASES**. AS A RESULT THE **PEDAL EFFORT** FOR THE DRIVER BECOMES **GREATER**.
- THE **SAME PEDAL EFFORT** CAN BE **MAINTAINED** THROUGHOUT THE **ENTIRE CLUTCH LIFE** THANKS TO THE **SELF ADJUSTING TECHNOLOGY** (S.A.T.). THE CONSTANT PEDAL EFFORT IS OBTAINED BY **INCREASING** THE **EFFECTIVE THICKNESS** OF THE **PRESSURE PLATE.**

- WEAR COMPENSATION MECHANISM

- **SELF ADJUSTING MECHANISMS FOR WEAR COMPENSATION** ON THE DRIVEN PLATE THICKNESS:



IN CASE OF **2 INPUT SHAFTS:** (1 OUTER, 1 INNER) → **DCU** (DUAL CLUTCH UNIT): 2 D.S., 2 THRUST BEARINGS, 2 P.P., 2 CLUTCH DISKS, ... (WE DOUBLE ALL COMPONENTS)



(2 MT IN PARALLEL) [25/11 1:25:00] ←!

WE **COMPENSATE** FOR THE **P.P. MOVEMENT** WITH A **RELATIVE ROTATION** BETWEEN THE **TWO STEEL DISKS** COMPOSING THE SPACER. THE **LEAF SPRING** IS A SORT OF **WEAR SENSOR**: IF WEAR IS PRESENT, **P.P. MOVES** TO **RIGHT**, ESTABLISHING A **CONTACT** BETWEEN THE **D.S.** AND THE **LEAF SPRING**. THIS CONTACT PRODUCE A **DISCRETE ROTATION** OF THE **WARM GEAR** ⇒ **RELATIVE ROTO-TRASL.** OF THE **TWO DISKS.**

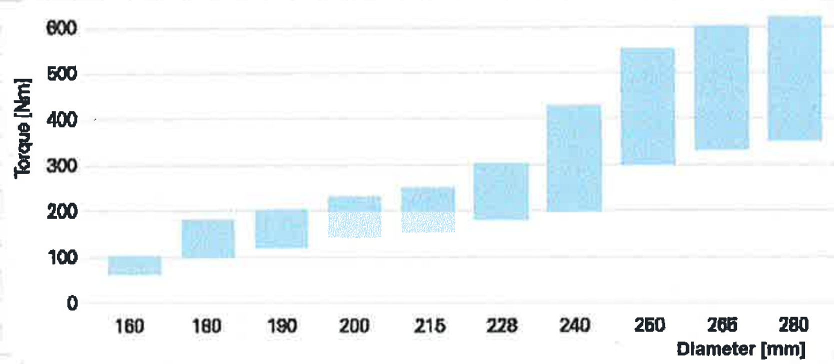
© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 129 di 260

- THERMAL DESIGN SPECIFICATIONS

- AN ESSENTIAL CLUTCH DESIGN **PARAMETER** IS THE **ENERGY WASTED** IN STANDARD OPERATIONS, IN PARTICULAR **AT START-UP AT MAXIMUM SLOPE** WITH THE VEHICLE **AT MAXIMUM WEIGHT**; IT CAN BE COMPUTED AND **REFERRED TO** THE CLUTCH **USEFUL FRICTION SURFACE** AND THE OBTAINED VALUE CAN BE COMPARED WITH THE VALUES OBTAINED IN PREVIOUS APPLICATIONS. **E/A_F : SPECIFIC ENERGY WASTE**

MOST CRITICAL OPERATION → MAXIMUM VALUE OF E.WASTE

- THE SAME CAN BE MADE FOR THE **TEMPERATURE** PROFILE OBTAINED **AT THE END OF A START-UP MANOEUVRE AT MAXIMUM SLOPE** OR DURING A SERIES OF REPEATED START-UP.
- **REFERENCE PARAMETER** FOR THE CLUTCH CLASSIFICATION IS THE **LININGS DIAMETER.** (MEAN DIAMETER) ($d \uparrow$ $T_{cl,c} \uparrow$) (WE CAN CHOOSE BETWEEN A FINITE n. OF d)
- AS A FINAL REMARK, THE **CLUTCH MUST OPERATE** PROPERLY **WITH A REASONABLE WEAR** FOR **AT LEAST 100 000 [Km]**, BUT CURRENT TRENDS ARE TO CONSIDER THE CLUTCH A **COMPONENT "PER LIFE."**



- **DCT LINEAR TORSIONAL DYNAMIC MODEL**
  - 3 DOFS
  - 4 DOFS
  - DCT MODELS IMPLEMENTATION IN MATLAB/SIMULINK
  - DCT CROSS-SHIFT PHASE SIMULATION
  - DCT CLUTCH ENGAGEMENT (INERTIA PHASE) SIMULATION
  - DCT COMPLETE UPSHIFT SIMULATION

[TEXTBOOK : VOL. I, PAR. 15.2]

- **INTRODUCTION : POWERSHIFT CONCEPT**

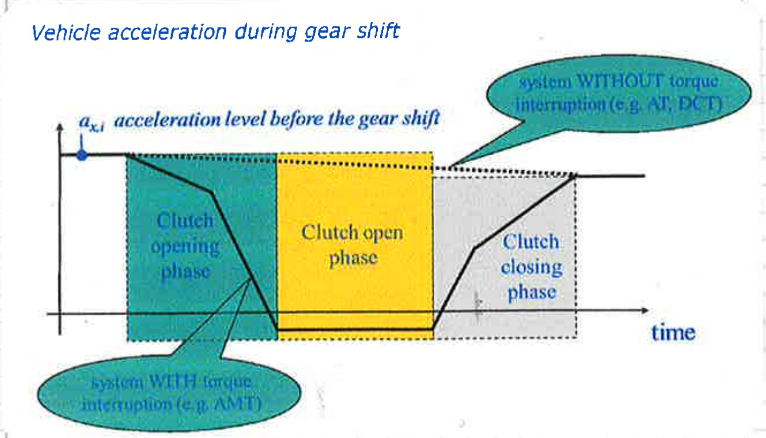
- CONTENT

1 • A DUAL CLUTCH TRANSMISSION (DCT) IS A PARTICULAR TYPE OF AUTOMATIC TRANSMISSION, BASED ON CONVENTIONAL MT COMPONENTS BUT ABLE TO AVOID THE INTERRUPTION OF THE TORQUE DURING THE GEARSHIFT OPERATIONS (POWERSHIFT CONCEPT) AND TO ASSURE THE ADVANTAGES OF AN ELECTRONICALLY CONTROLLED TRANSMISSION.

2 • A POWERSHIFT TRANSMISSION GIVES BETTER PERFORMANCE (ACCELERATION) TO THE VEHICLE AND BETTER COMFORT (LESS VARIATION OF ACCELERATION) DURING THE GEARSHIFT TO THE DRIVER AND THE PASSENGERS, WITH RESPECT TO A MANUAL OR AUTOMATIZED TRANSMISSION. (BUT LESS EFF. (F.C.↑) AND MORE COMPLEX)

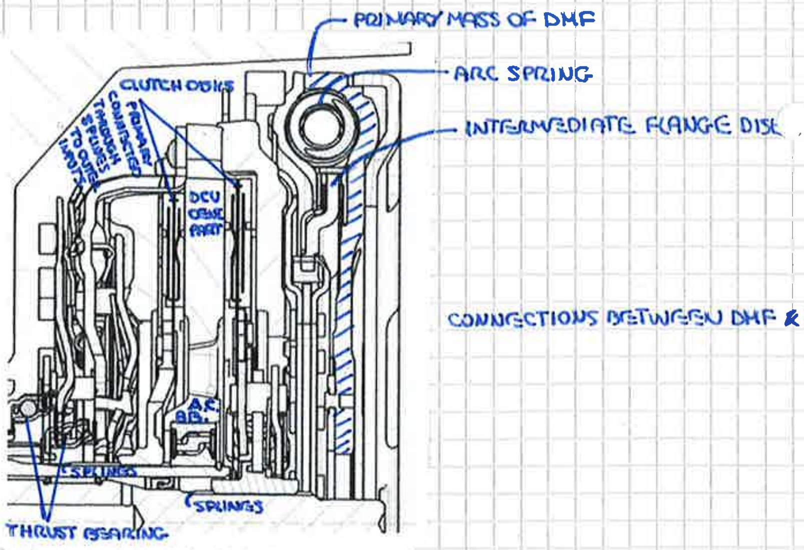
3 • TO REALISE THIS TYPE OF TRANSMISSION THE MOST IMPORTANT NEW COMPONENT TO BE DESIGNED IS A DOUBLE CLUTCH UNIT. (DCU)

4 • MOREOVER, A COMPLETE CONTROL SYSTEM, FOR BOTH CLUTCH AND GEARBOX, HAS TO BE DEVELOPED.



- DRY DCU

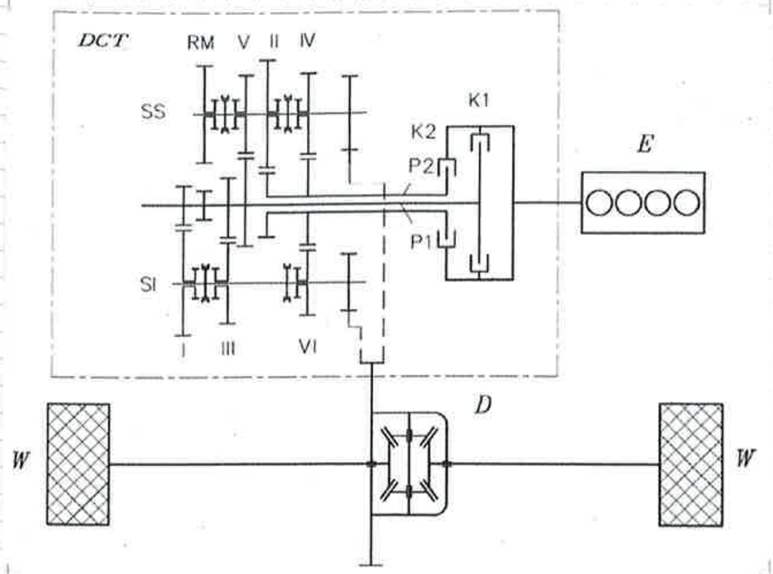
Volkswagen



CONNECTIONS BETWEEN DMF & DCU

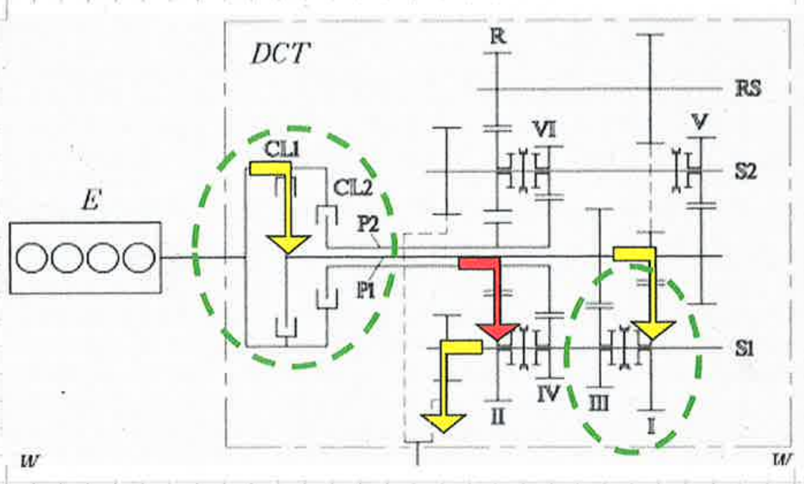
- ADVANTAGES / DISADVANTAGES:
  - LIMITED THERMAL PROPERTIES → MISUSE HAS TO BE AVOIDED
  - LOW DAMPING → LOWER COMFORT, BUT HIGHER EFFICIENCY
  - LOWER COST
  - MORE COMPLEX CONTROL STRATEGIES

- EXAMPLE: 6-SPEED DCT LAYOUT



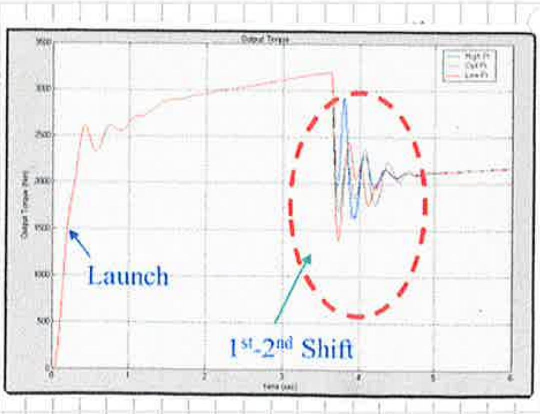
- DCT WORKING PRINCIPLE

- DOUBLE CLUTCH (WET OR DRY):
  - 2 COAXIAL PRIMARY SHAFTS (P_1 ODD GEAR, P_2 EVEN GEAR)
  - SYNCHRONIZERS NOT BETWEEN CONSECUTIVE GEARS.
  - PRESELECTION OF THE NEXT GEAR IN ADVANCE.
  - SMOOTH TRANSITION WITH TORQUE TRANSFER TO THE DRIVE WHEELS (CLUTCH TO CLUTCH)



- GEAR RATIO τ_{P1} ENGAGED AND τ_{P2} PRESELECTED
- TORQUE MODULATION THROUGH THE 2 CLUTCHES

$$\begin{cases} T_{ENG} - T_{CL1} - T_{CL2} = J_{ENG}\dot{\omega}_{ENG} \\ (T_{CL1}\tau_{P1} + T_{CL2}\tau_{P2})\tau_{S1} - T_? = J_{EQ?}\dot{\omega}_? \end{cases}$$



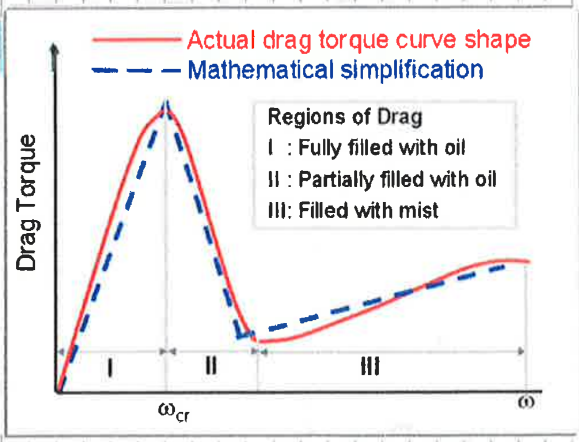
- **WET OR DRY?**

- CONTENT (LOOK AT SLIDE 34 : VIDEOS' LINKS)

- WET DUAL CLUTCH TRANSMISSION (WET-DCT):
  - IT USES TWO WET MULTI-PLATES CLUTCHES WORKING IN A COOLING LUBRICATING FLUID, WHICH KEEPS THE SURFACES CLEAN, GIVES SMOOTHER PERFORM. (HIGHER COMF.) AND LONGER LIFE, ENSURE GOOD TH. PROP. (HIGHER LOAD CAPACITY, LOWER SENSITIVITY TO MISUSE)
  - WET CLUTCHES HAVE A BEHAVIOUR SIMILAR TO A TORQUE CONVERTER AND DUE TO OIL FRICTIONING WASTE ENERGY TO THE OIL, THEY INCREASE THE VEHICLE FUEL CONSUMPTION (↑)(LOWER EFFICIENCY). IT HAS AN HIGHER COST.

- DRY DUAL CLUTCH TRANSMISSION (DRY-DCT):
  - IT USES TWO CONVENTIONAL CLUTCHES LIKE THE ONES USED IN THE MANUAL TRANSMISSION (MT). (HIGHER EFFICIENCY, LOWER COST) (LIMITED TH. PROPERTIES, MISUSE TO AV.)
  - THEY REQUIRE MORE COMPLEX CONTROL STRATEGIES, BUT AVOID TO INCREASE SO MUCH VEHICLE FUEL CONSUMPTION IN COMPARISON WITH MANUAL TRANSM. (MT).

- DRAG-TORQUE OF A WET DCT

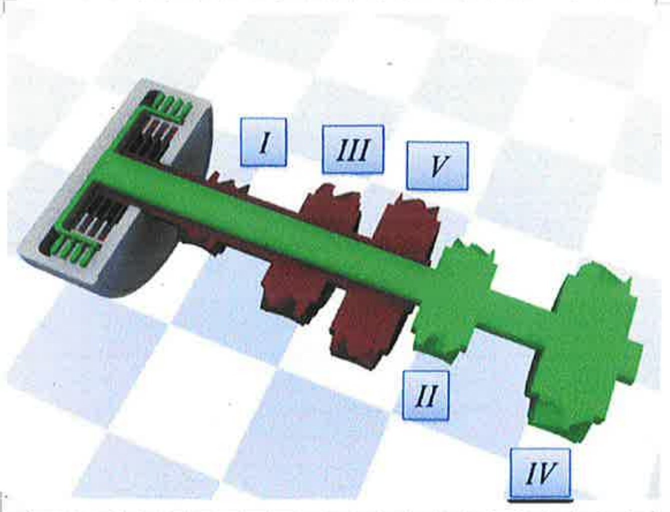


I : FULL FILM OF OIL EXISTS BETWEEN THE DISCS, SURFACE TENSION FORCES PREDOMINATES

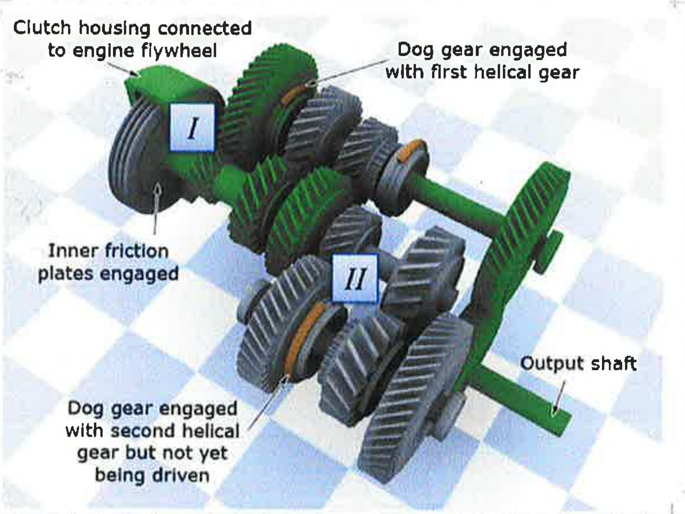
II : CENTRIFUGAL FORCES DOMINATES, OIL FILM DIMINISHING.

III : FLUID FORMS A MIST BETWEEN THE CLUTCH PLATES.

- 5-SPEED DCT-DSG : WET CLUTCHES AND COAXIAL PRIMARY SHAFTS

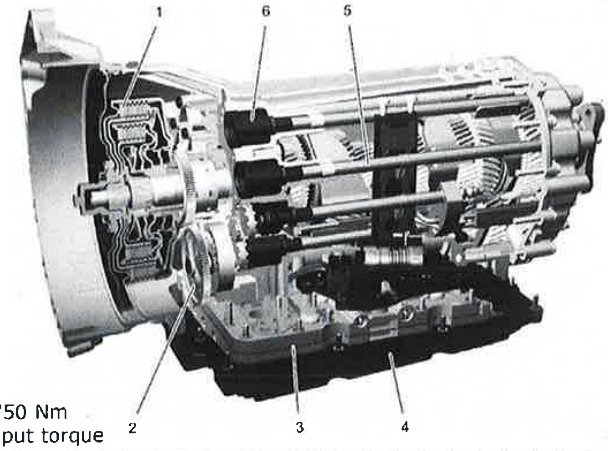


- DCT-DSG : POWER FLOW IN 1st GEAR WITH 2nd GEAR PRESELECTED



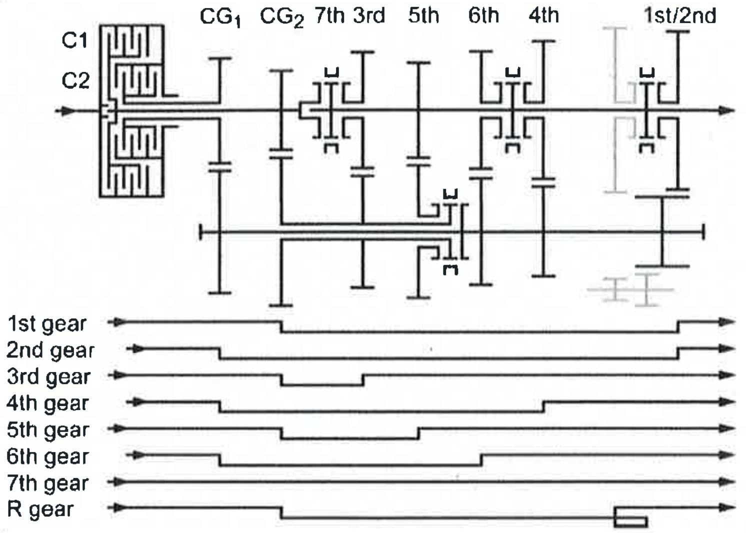
- DCT-DSG : POWER FLOW IN 2nd GEAR WITH 3rd GEAR PRESELECTED

- ZF 7-SPEED WET DCT

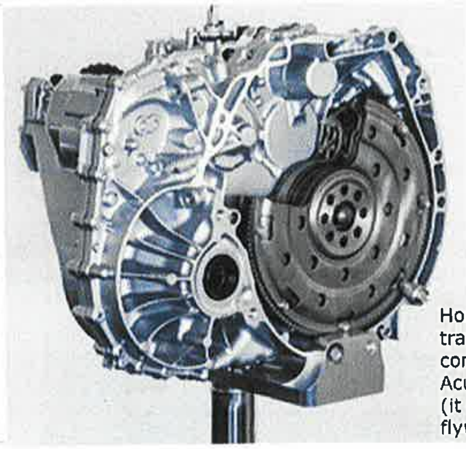


390 Nm to 750 Nm
maximum input torque

- ZF 7-SPEED WET DCT : POWER PATH ALONG THE TRANSMISSION



- HONDA DCT WITH TC

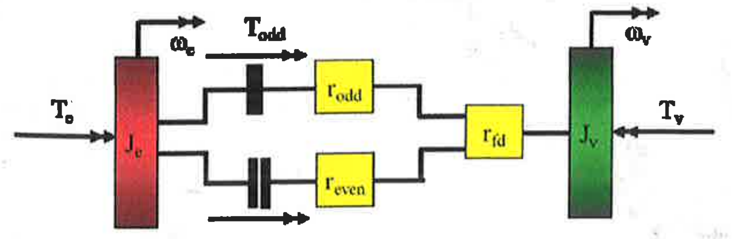


Honda's 8-speed dual-clutch transmission with torque converter as fitted to the 2015 Acura TLX
(it doesn't need a dual-mass flywheel)

• DCT GEARSHIFT CONTROL

- UPSHIFT FROM ODD TO EVEN GEAR

Upshifting from odd to even ($r_{even} > r_{odd}$), e.g. 1st → 2nd gear



• FIRST TORQUE PHASE THEN INERTIA PHASE:

1) TORQUE PHASE : EVEN CLUTCH TORQUE INCREASES, WHILE ODD CLUTCH REMAINS IN STICK UNTIL THE EVEN CLUTCH CAN SUPPORT THE TORQUE TRANSFER TO VEHICLE SUFFICIENTLY:

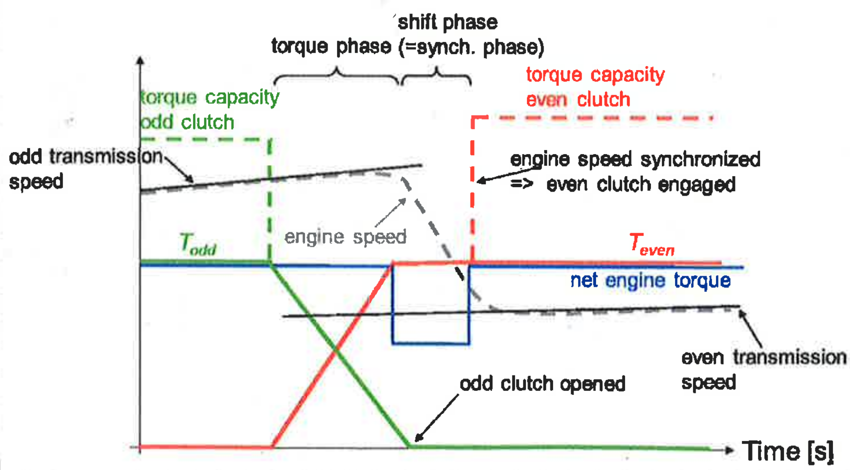
$$T_{EVEN} = P_{EVEN} A_{EVEN} R_{EVEN} \mu_{EVEN} SIGN(\Delta\omega_{EVEN})$$

$$T_{ODD} = \frac{(T_E - T_{EVEN}) J_V R_{ODD}^2 R_{fd}^2 + [T_V R_{ODD} R_{fd} - T_{EVEN}(R_{ODD}/R_{EVEN}) J_E]}{J_V R_{ODD}^2 R_{fd}^2 + J_E}$$

2) INERTIA PHASE: THEN ODD CLUTCH IS OPENED AND EVEN CLUTCH SYNCHRONIZES THE ENGINE TO THE EVEN TRANSMISSION SPEED.

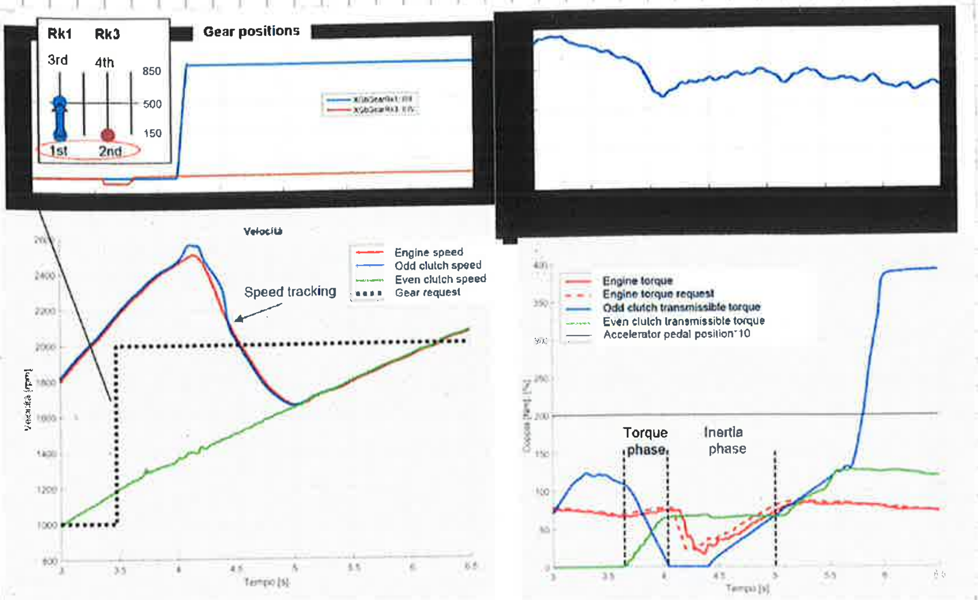
$$T_{fd} = \frac{T_{ODD}}{R_{ODD} R_{fd}} + \frac{T_{EVEN}}{R_{EVEN} R_{fd}}$$

Total upshift graphically

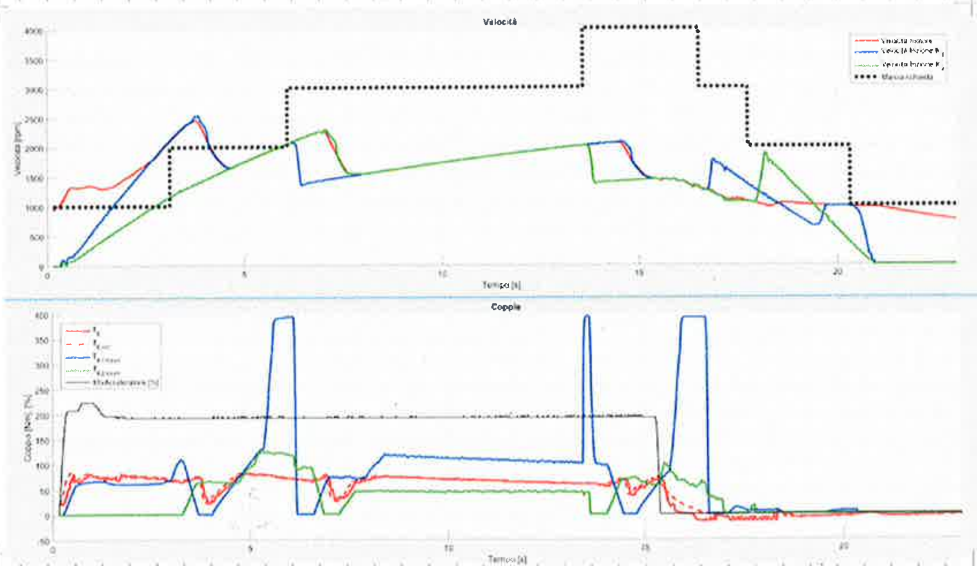


- ESPERIMENTAL DATA

- DCT CONTROL 1st → 2nd UPSHIFT, APP = 20%



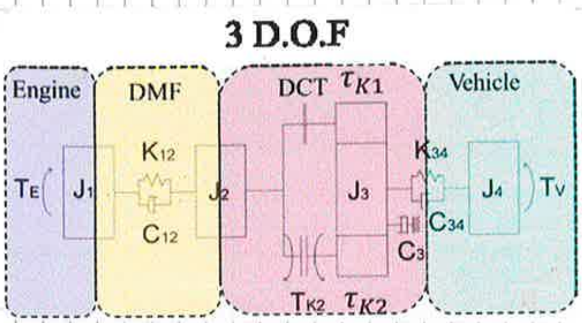
- DCT GEARSHIFT CONTROL : EXPERIMENTAL DATA



- **DCT LINEAR TORSIONAL DYNAMIC MODEL**

- 3 DOFs

- DEPENDING ON THE SPECIFIC WORKING CONDITION, DUE TO CLUTCHES AND SYNCHRONIZERS STATES, THE TRANSMISSION CAN BE MODELED USING A DIFFERENT NUMBER OF DEGREES OF FREEDOM.
- TWO MODELS ARE HERE SHOWN: WITH 3 D.O.F. WHEN ONE CLUTCH IS ENGAGED AND THE OTHER IS SLIPPING; WITH 4 D.O.F. IF BOTH THE CLUTCHES ARE SLIPPING.



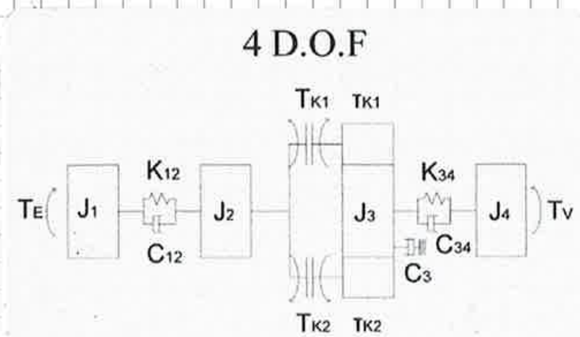
E.G. CROSS SHIFT, TIP-IN/TIP-OUT
- K_i = ENGAGED
- K_j = SLIPPING/OPEN/ENGAGED FOR SPEED TRACKING

- MODEL PARAMETERS:

- K_{12} AND C_{12}: REPRESENTS THE ELASTIC AND DISSIPATIVE BEHAVIOUR OF THE DUAL MASS FLYWHEEL.
- K_{34} REPRESENTS THE STIFFNESS OF THE DCT AND OF THE HALFSHAFTS.
- C_3 REPRESENTS THE CHURNING LOSSES OF THE GEARS INSIDE THE TRANSMISSION.
- C_{34} REPRESENTS THE HALFSHAFTS, JOINTS AND THE TIRE DAMPING.

- 4 DOFs

- THE MODEL INPUTS ARE THE TORQUE DELIVERED BY THE ENGINE T_E AND THE TORQUE TRANSFERRED BY THE TWO CLUTCHES T_{K1} AND T_{K2}. A CLUTCH (FRICTION) MODEL NEEDS TO BE COUPLED TO THE DCT TORSIONAL MODEL IN ORDER TO CALCULATE FROM THE TRANSMISSIBLE TORQUE THE EFFECTIVE TORQUE TRANSFERRED BY EACH CLUTCH.



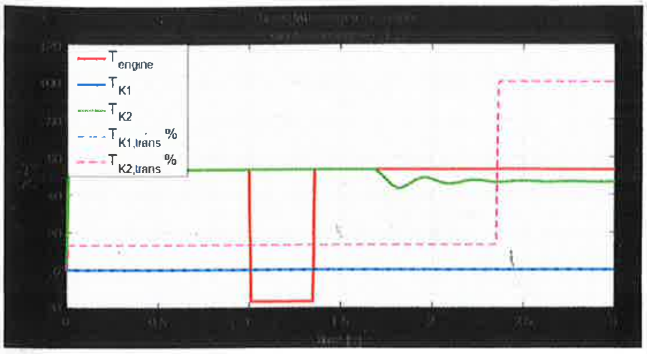
E.G. CLUTCH ENGAGEMENT
- K_i = SLIPPING/OPEN
- K_j = SLIPPING/OPEN/ENGAGED FOR SPEED TRAK.

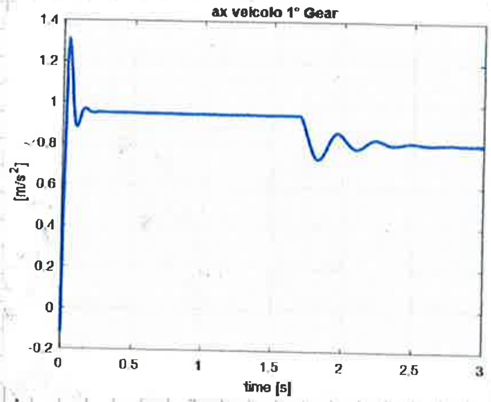
- THE 4 DOFS ARE THE ANGULAR POSITION OF:

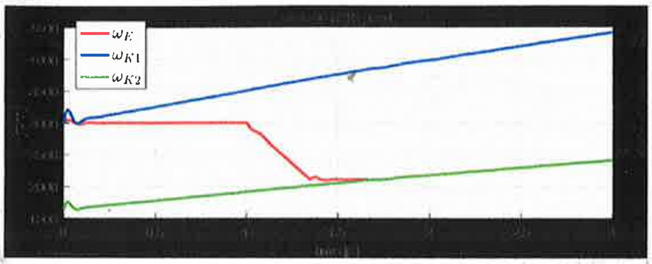
1. ENGINE CRANKSHAFT OR DMF PRIMARY MASS,
2. DMF SECONDARY MASS,
3. DIFFERENTIAL,
4. WHEELS

- DCT CLUTCH ENGAGEMENT (INERTIA PHASE) SIMULATION

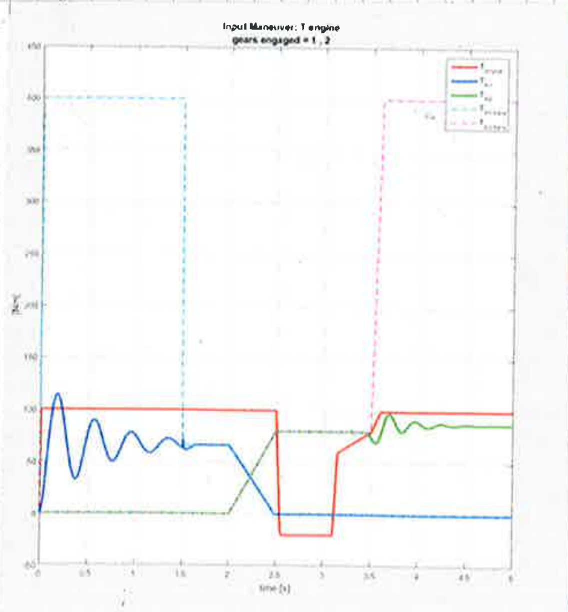
- EXAMPLE OF CLUTCH ENGAGEMENT (INERTIA PHASE) DURING A 1st TO 2nd GEAR UPSHIFT MANOEUVER USING THE 4 DOF MODEL.
- INERTIA CONDITIONS ARE THE EXIT CONDITIONS OF THE TORQUE PHASE: INCOMING CLUTCH TRANSFER THE ENGINE TORQUE $T_{K1} = T_{ENG}$, OFF-GOING CLUTCH IS DISENGAGED $T_{K1} = 0$.

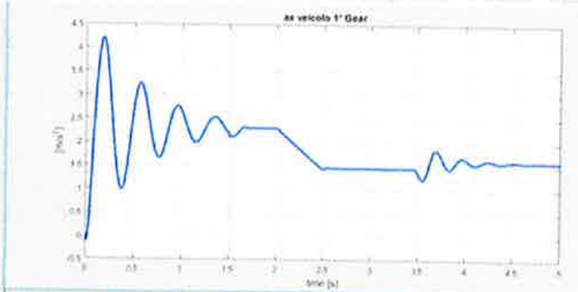


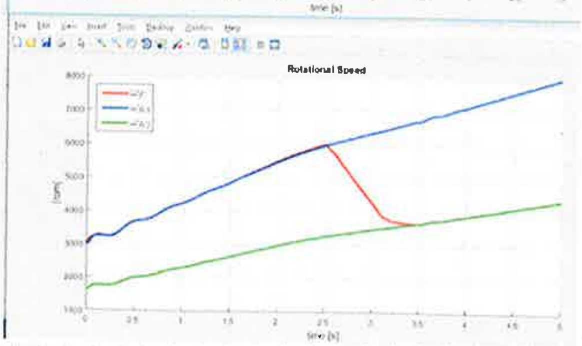




- DCT COMPLETE UPSHIFT SIMULATION







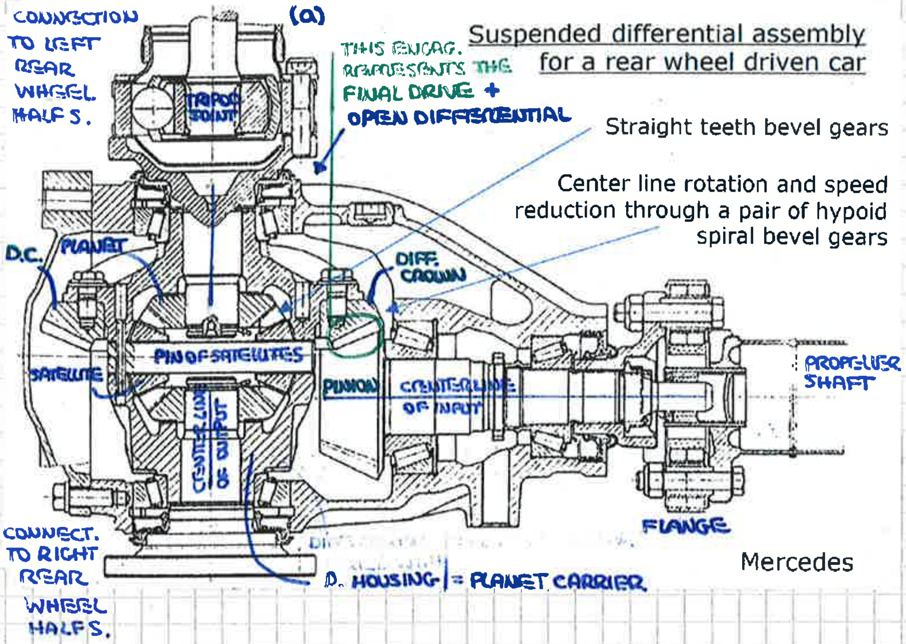
© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 151 di 260

DIFFERENTIAL, FINAL DRIVES AND TRANSFER BOXES

- CONTENT

- THE DIFFERENTIAL IS A MECHANISM THAT ALLOWS DIVIDING THE TORQUE COMING FROM AN INPUT SHAFT IN TWO PREDETERMINED PARTS, FLOWING THROUGH TWO OUTPUT SHAFTS : TORQUE RATIOS ARE INDEPENDENT OF SPEED RATIOS OF THE SAME SHAFTS.
- THIS MECHANISM CAN BE EITHER USED TO DIVIDE THE TORQUE COMING OUT OF THE FINAL DRIVE INTO ALMOST EQUAL PARTS ACTING ON THE TRACTION WHEELS OF THE SAME AXLE, OR THE TORQUE COMING OUT OF THE GEARBOX IN TWO PREDETERMINED PARTS ACTING ON DIFFERENT AXLES OF THE SAME VEHICLE; THIS SECOND APPLICATION IS NAMED SOMETIMES TRANSFER BOX DIFFERENTIAL OR CENTRAL DIFFERENTIAL.
- THE FINAL DRIVE IS A GEAR TRAIN THAT FURTHER REDUCES THE SPEED OF THE GEARBOX OUTPUT SHAFT TO ADAPT IT TO THE TRACTION WHEELS; THIS GEARTRAIN IS USUALLY INTEGRATED WITH A DIFFERENTIAL MECHANISM.
- THE TRANSFER BOX IS A MECHANISM THAT PROVIDES TO MOVE TWO OR MORE DRIVELINES WITH THE ONLY OUTPUT SHAFT OF THE GEARBOX; THIS IS USED ON VEHICLES WITH MULTIPLE TRACTION AXLES.

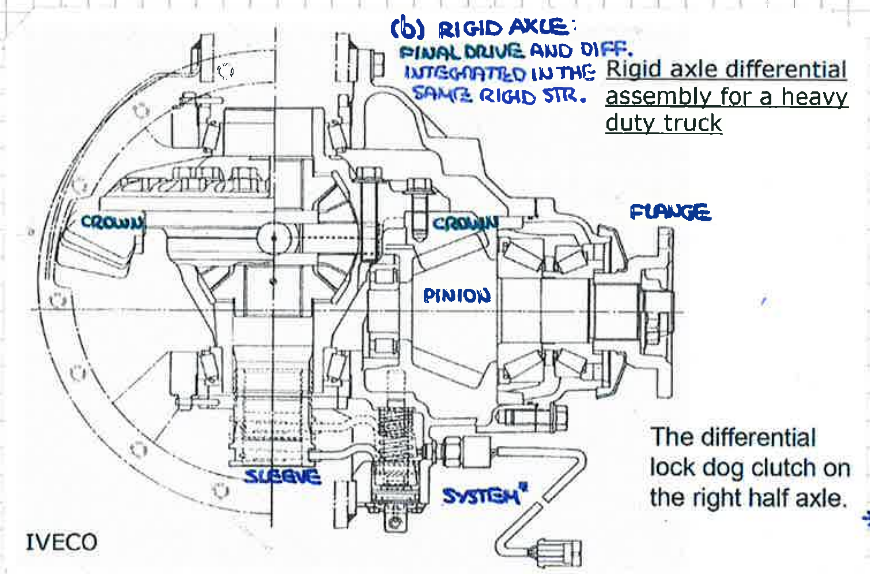
- DIFFERENTIAL AND FINAL DRIVE FOR A REAR WHEEL DRIVEN CAR



THE DIFFERENTIAL IS COMPOSED BY A PLANETARY GEAR SET WHICH SPLITS THE TORQUE:

- STRAIGHT TEETH BEVEL GEARS
- THE CROWN RECEIVES MOTION FROM THE PINION THAT IS RIGIDLY CONNECTED TO THE PROPELLER S. THROUGH A FLANGE.
- PINION AND CROWN ARE HYPOID SPIRAL BEVEL GEARS. (TO PERMIT TO HAVE AN OFFSET BETWEEN THE CENTER LINES)

- RIGID AXLE DIFFERENTIAL ASSEMBLY FOR A HEAVY DUTY TRUCK



a) IT IS A SUSPENDED DIFF. ASSEMBLY A DEDICATED SUSPENSION SYSTEM FOR THE ATTACHMENT OF THE DIFF. WITH RESPECT TO THE V. CHASSIS IS NEEDED (TO COUNTERACT THE T AMPLIFICATION OF THE FINAL DRIVE)

a) b) DIFF. MUST BE PROPERLY SUPP. WE HAVE TO COUNTER ACT THE F_a GENERATED BY THE FINAL DRIVE.

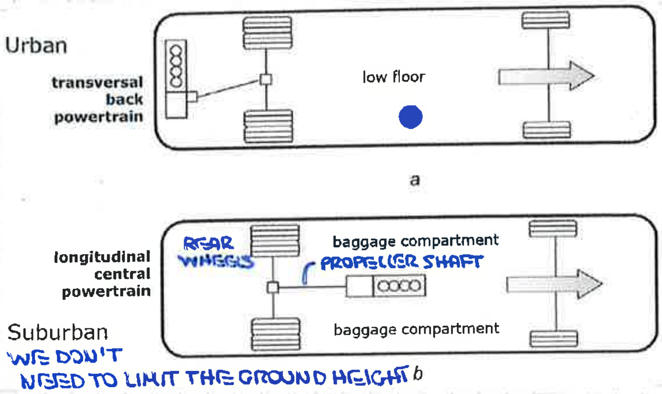
→ TAPERED ROLLER BEARINGS (TO SUPP. DIFF. HOUSING W.R.T. THE CASE)

a) THE PINION PUTS IN ROTATION THE DIFF. CROWN; THE DIFF. CROWN PUTS IN ROTATION THE PIN OF SATELLITES WHICH PUT IN ROTATION THE PLANETS CONNECTED TO THE DIFF. OUTPUT SHAFTS THROUGH SPLINES. THE DIFF. OUTPUT SHAFTS ARE CONNECTED TO THE HALF SHAFTS THROUGH PARTICULAR JOINTS: CONSTANT VELOCITY JOINTS (TRIPOD JOINTS); THEY ALLOWS NOT ONLY THE INCLINATION OF THE HALF S. WRT THE DIFF. OUTPUT S. BUT ALSO AN AXIAL MOTION OF THE CENTER OF THE JOINT WRT JOINT HOUS.

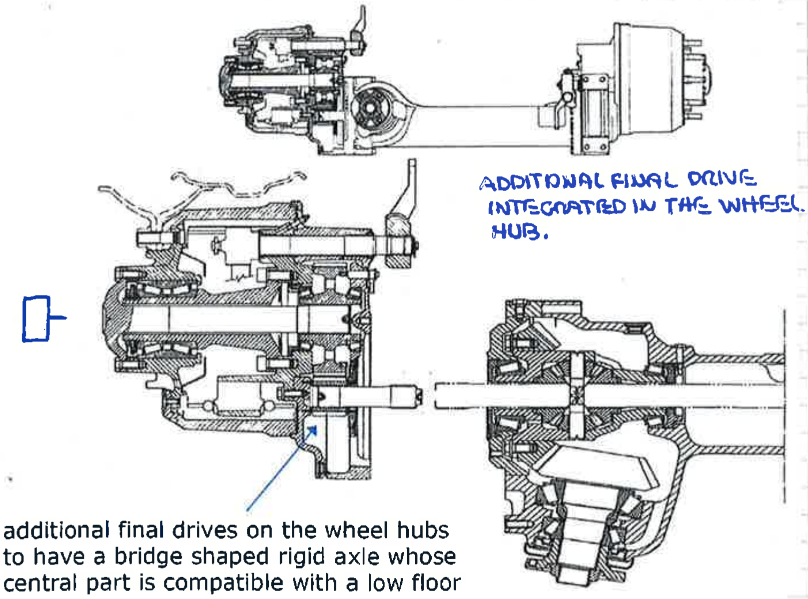
b) IN THE RIGID AXLE CONF. WE HAVE ALSO THE POSSIBILITY TO LOCK THE DIFFERENTIAL THROUGH A DOG CLUTCH ON THE RIGHT HALF SHAFT; THROUGH SYSTEM* IS POSSIBLE TO MOVE A SLEEVE USED TO PUT IN CONNECTION THE RIGHT PLANET WITH THE TRANSM. HOUSING: IN THIS WAY, IF WE LOCK TWO PARTS TO ROTATE AT THE SAME SPEED ALSO THE THIRD PART (LEFT SIDE) WILL ROTATE AT THE SAME SPEED.
(RIGID AXLE DIFFERENTIAL IS USED, FOR EX., ON OFF-ROAD VEHICLES)

- DIFFERENTIAL FOR BUSES

- ON URBAN BUSES IT IS IMPORTANT TO LIMIT THE GROUND HEIGHT OF THE AISLE ("CORRIDOIO").
- FLOOR HEIGHT AND TIRE ("PNEUMATICO") ROLLING RADIUS DEFINE THE MAXIMUM DIAMETER OF THE BEVEL GEAR WHEEL; THEREFORE, A SPUR GEAR FINAL DRIVE IS NECESSARY ON THE OUTPUT SHAFT OF THE DIFFERENTIAL. IT'S AN ADDITIONAL FINAL DRIVE TO OBTAIN THE DESIRED FINAL τ. IT'S INTEGRATED IN THE WHEEL HUB.



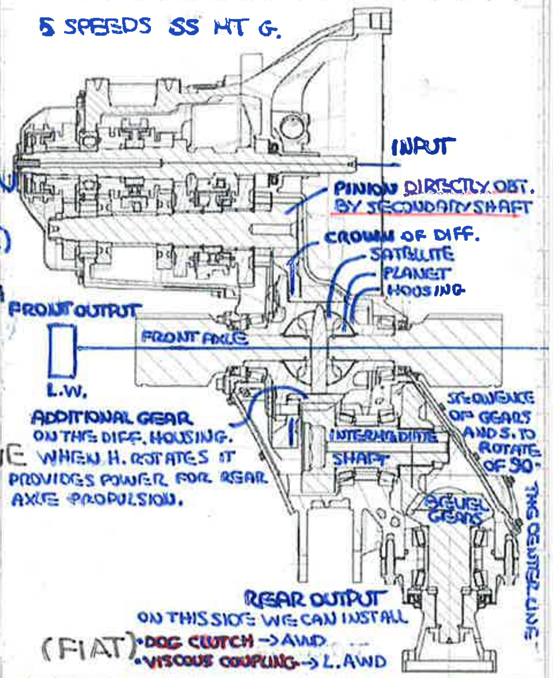
- RIGID REAR AXLE FOR URBAN BUSES



additional final drives on the wheel hubs to have a bridge shaped rigid axle whose central part is compatible with a low floor

– MODIFIED FRONT WHEEL DRIVES (TRANSVERSAL ENGINE)

- GEARBOX, DIFFERENTIAL AND TRANSFER BOX OF AN AWD CAR WITH TRANSVERSAL FRONT E.:
  - FRONT AXIAL DIFFERENTIAL IS COUPLED TO A SIMPLE BEVEL GEAR DRIVE WITH A TRANSMISSION RATIO CLOSE TO 1. (TO DO NOT HAVE S. DIFF. BET. F&R)
- IN CASE OF NON-PERMANENT AWD A DOG CLUTCH CAN BE PRESENT TO MOVE THE REAR AXLE;
- IN CASE OF PERMANENT TRACTION, A VISCOUS COUPLING IS USUALLY FIT ON THE REAR AXLE DRIVE LINE: THIS JOINT PROVIDES SUBTRACTING PART OF THE TORQUE AVAILABLE TO THE FRONT AXLE, ONLY WHEN THIS AXLE IS SHOWING AN AVERAGE SPEED GREATER THAN THAT OF THE REAR AXLE; A SELF LOCKING DIFFERENTIAL CAN BE INSTALLED IN THE TRANSFER CASE, AVOIDING THE VISCOUS COUPLING.

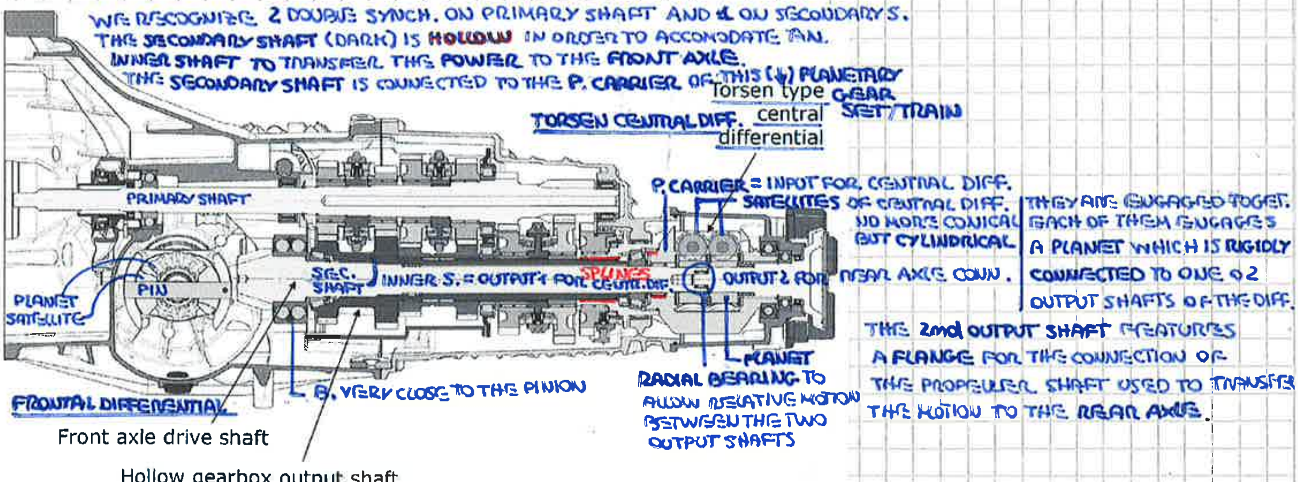


INSTEAD OF A JOINT → VISCOUS COUPL.: PERMITS A CERTAIN AMOUNT OF RELAT. VELOCITY BETWEEN THE 2 AXLES.

WE USE AXLES SPEED DIFF. AS A SENSING SIGNAL TO AUTOMATICALLY OPERATE THE VISCOUS COUPLING TO INCREASE THE T WHERE THE ω IS LOWER.

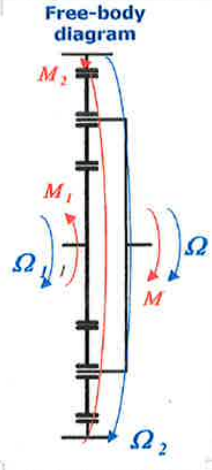
– MODIFIED FRONT WHEEL DRIVES (LONGITUDINAL ENGINE)

- GEARBOX, DIFFERENTIAL AND TRANSFER BOX OF A PERMANENT AWD WITH LONGITUDINAL ENGINE (AUDI)



© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 159 di 260

- FRICTION FREE DIFFERENTIAL: SCHEME AND EQUATIONS



- **STARTING FROM** THE **ORDINARY TRANSMISSION RATIO**:

$$R = (\Omega_2-\Omega)/(\Omega_1-\Omega) = -\frac{z_1}{z_2} = -\frac{z_S}{z_R}$$ (KNOWING $\Omega_1, \Omega_2 \rightarrow \Omega$) (1)

IT IS POSSIBLE TO WRITE THE SPEED EQUATIONS:

$$\Omega_1 = \Omega + (\Omega_2-\Omega)/R$$
$$\Omega_2 = \Omega + R(\Omega_1-\Omega)$$

OR $\Omega_2-\Omega_1 = \Omega-\Omega_1 + R(\Omega_1-\Omega) = (1-R)(\Omega-\Omega_1)$

- THE **SPEED DIFFERENCE** $(\Omega_2-\Omega_1)$ CAN **ASSUME ANY VALUE**.
- **FOR** THE **SYSTEM EQUILIBRIUM** IT MUST BE:

$M_1+M_2=M$ THEN $M_2 = M-M_1$

- AND IN CASE OF NO FRICTION, THE **POWER BALANCE** IS:

$P_{OUT} = P_{IN}$

$M_1\Omega_1 + M_2\Omega_2 = M\Omega$

- FRICTION FREE DIFFERENTIAL: TORQUE SPLIT

- **BY SUBSTITUTING** THE **SPEED Ω_2** AND **MOMENT M_2** EQUATIONS **IN** THE **LAST ONE**:

$$M_1\Omega_1 + (M-M_1)[\Omega + R(\Omega_1-\Omega)] = M\Omega$$
$$M_1\Omega_1 + \cancel{M\Omega} + MR(\Omega_1-\Omega) - M_1\Omega - M_1R(\Omega_1-\Omega) = \cancel{M\Omega}$$
$$M_1(\cancel{\Omega_1-\Omega}) + MR(\cancel{\Omega_1-\Omega}) - M_1R(\cancel{\Omega_1-\Omega}) = 0$$
$$M_1 + MR - M_1R = 0$$
$$M_1(1-R) = -MR$$

AND FINALLY:

$$M_1 = -MR/(1-R)$$
$$M_2 = M - M_1 = M + MR/(1-R) = M[1+R/(1-R)] = M/(1-R)$$

THEREFORE:

$$\frac{M_1}{M_2} = \frac{-\cancel{M}R}{\cancel{(1-R)}}\frac{\cancel{(1-R)}}{\cancel{M}} = -R \quad (2)$$

- THE **TORQUE** IS **SPLIT** IN **CONSTANT PARTS**, **DEPENDING ON ORDINARY TRANSMISSION RATIO** OF THE GEAR TRAIN, **NOT DEPENDING ON SPEED**.

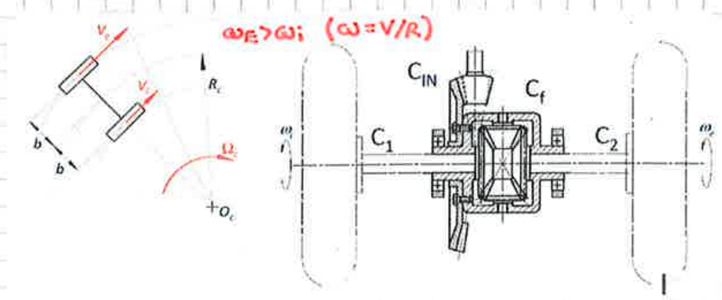
$$R = \frac{\Omega_2-\Omega}{\Omega_1-\Omega} = -\frac{z_1}{z_2} = -\frac{M_1}{M_2}$$

$M = M_1+M_2$

$M\Omega = M_1\Omega_1 + M_2\Omega_2$

– REAL OPEN DIFFERENTIAL : SIMPLE MODEL WITH FRICTION

$\omega_E > \omega_i$; $(\omega = V/R)$



$\omega_{IN} = \frac{\omega_1 + \omega_2}{2}$ $(\tau_0 = -1)$

$C_1 = \frac{C_{IN}}{2} \pm C_f$

$C_2 = \frac{C_{IN}}{2} \mp C_f$

CASE $\omega_1 > \omega_2$:

$C_1 = \frac{C_{IN}}{2} - C_f$

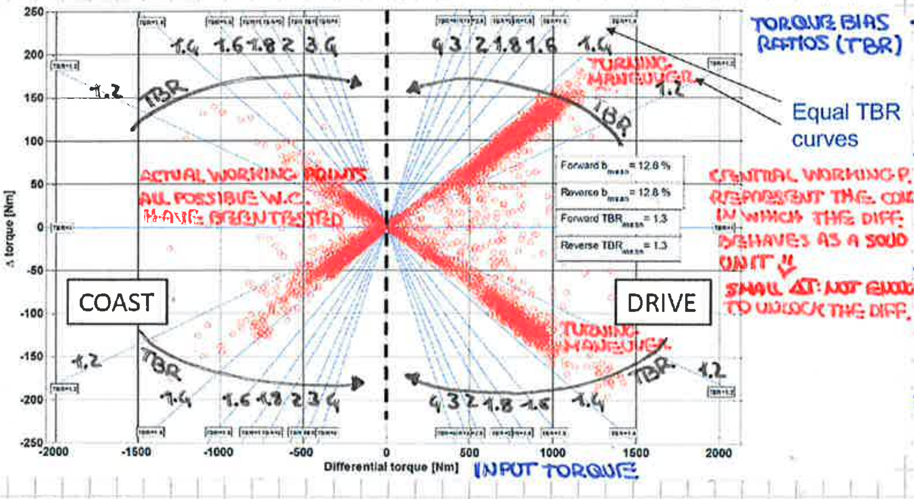
$C_2 = \frac{C_{IN}}{2} + C_f$

CASE $\omega_1 < \omega_2$:

$C_1 = \frac{C_{IN}}{2} + C_f$

$C_2 = \frac{C_{IN}}{2} - C_f$

– REAL OPEN DIFFERENTIAL : CHARACTERISTIC DIAGRAMS (EXPERIMENTAL DATA)



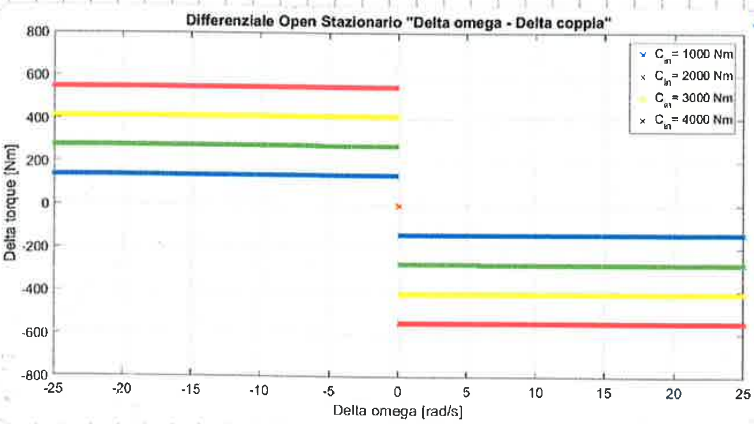
- THE CHARACT. DIAGRAM PRESENTS THE DIFF. INPUT T. ON THE HOR. AXIS AND THE ΔT OF THE TWO OUTPUT ON VERT. AXIS.
- THIS PLOT IS ACHIEVED EXP. BY MEASURING HALFSHAFTS T
  - ⇒ THE SUM = INPUT TORQUE
  - ⇒ THE DIFF = ΔT
- WE NOTICE THAT THE WORKING P. ARE CROWDED ALONG SPECIFIC LINES DEFINING SORTS OF BOUNDARY

⇒ ΔT IS PROPORTIONAL TO THE INPUT TORQUE OF THE DIFF.

$\Delta C = 2 C_f$

NOT CONSTANT BUT ALMOST PROPORTIONAL TO INPUT TORQUE

– REAL OPEN DIFFERENTIAL : CHARACTERISTIC DIAGRAMS (MODEL)



- WE CAN MODEL THE REAL OPEN DIFF IN ORDER TO HAVE A $\Delta T \propto$ INPUT T.
- IT IS QUITE INSENSITIVE TO $\Delta\omega$
  [WE WANT THIS PROPERTY: T SPLIT INDIPENDENT OF $\Delta\omega$]
- IN REALITY ALSO $\Delta\omega$ PLAYS A ROLE BUT NOT SO RELEVANT.
- IF WE INTRODUCE THIS CONCEPT $\Delta T \propto T_{IN}$ STARTING FROM THE DEF. OF THE TORQUE BIAS RATIO (TBR), THAT IS THE RATIO BET. MAXIMUM AND MINIMUM TORQUES, WE CAN ACHIEVE THAT THIS RATIO IS CONSTANT.
  TBR = CONST

– REAL OPEN DIFFERENTIAL : TORQUE BIAS RATIO (TBR)

$$TBR = \frac{\frac{C_{IN}}{2} + C_f}{\frac{C_{IN}}{2} - C_f} = \frac{\frac{C_{IN}}{2} + k C_{IN}}{\frac{C_{IN}}{2} - k C_{IN}} = \frac{k_1 C_{IN}}{k_2 C_{IN}} = CONST$$

$C_f = k C_{IN}$: FRICTION TORQUE IS $\propto T_{IN}$

- THE CASE OF THE **AXLE DIFFERENTIAL** CAN BE AGAIN OBTAINED BY SETTING $R = -1$.
- IN THIS CASE **ON A STRAIGHT ROAD** THE LIMITS BECOME:

$$\boxed{M_{min} = \frac{M\eta}{(1+\eta)}}$$

$$\boxed{M_{MAX} = \frac{M}{(1+\eta)}}$$

$$\boxed{\Delta M = \frac{M(1-\eta)}{(1+\eta)}} \cdot \begin{cases} \text{IF NOT EXCEEDED} \Rightarrow \text{LOCKED MODE} \\ \text{IF EXCEEDED} \Rightarrow \text{OPEN MODE : T SPLIT ACCORDING TO THE TBR} \end{cases}$$

E.G.: $\eta = 0.5$, $M_{min} = 0.33M$, $M_{MAX} = 0.66M$

- ON THE CONTRARY OF THE IDEAL NO FRICTION CASE, **IF ONE WHEEL** CAN BEAR A **LOWER TRACTION FORCE** BECAUSE OF A **LOWER TRACTION COEFFICIENT**, THERE WILL BE **NO RELATIVE MOTION** BETWEEN WHEELS **IF THE DIFFERENCE IN TORQUE IS INCLUDED IN THE ABOVE LIMITS**. (DIFF. WORKS IN LOCKED MODE)
- THE SAME APPLIES IN A CURVE, WHERE THE DIFFERENCE BETWEEN THE TORQUE VALUES ON THE TWO WHEEL IS:

$$\Delta M = \frac{M(1-\eta)}{(1+\eta)}$$

- DIFFERENTIAL WITH INTERNAL FRICTION: **INTERNAL POWER BALANCE** CONSID. ALSO η

$M\Omega = M_1\Omega_1 + M_2\Omega_2 + W_d$ ← POWER DISSIPATION

AXLE DIFFERENTIAL WITH FRICTION:

$(M_1 + M_2)\Omega = M_1\Omega_1 + M_2\Omega_2 + W_d$

$M_1(\Omega - \Omega_1) = M_2(\Omega_2 - \Omega) + W_d$

$(M_1 - M_2)(\Omega - \Omega_1) = W_d$

$$\boxed{(M_1 - M_2)\left(\frac{\Omega_2 - \Omega_1}{2}\right) = W_d}$$: INTERNAL POWER BALANCE

- W_d IS THE **POWER DISSIPATED** INSIDE THE **DIFFERENTIAL HOUSING**.
- **IT MUST BE POSITIVE**, THEREFORE **INPUT** AND **OUTPUT** CAN BE **IDENTIFIED**.

$W_d = W_i - W_o = M_i\Delta\Omega - M_o\Delta\Omega = (M_1 - M_2)\Delta\Omega$ → INPUT = 1 ; OUTPUT = 2

WITH:

W_i AND W_o INPUT AND OUTPUT POWER FOR THE INTERNAL MECHANISM OF THE HOUSING IF $\Delta\Omega \neq 0$.

- THE **DISSIPATED POWER** (W_d) $\propto \begin{cases} \text{DIFF. IN TORQUE } (M_1 - M_2) \\ \text{DIFF. IN SPEED } (\Omega_2 - \Omega_1) \end{cases}$, THEREFORE:

EVEN IF WE HAVE AN HIGH DIFF. IN TORQUE WE CAN LIMIT W_d IF WE LIMIT THE DIFF. IN SPEED.

- BEVEL GEAR DIFFERENTIAL: SCHEME AND EQUATIONS

- IN THE CASE OF A BEVEL GEAR DIFFERENTIAL, ON THE SATELLITE WILL ACT TANGENTIAL FORCES T_2 AND T_1, INCLINED ACCORDING TO THE PRESSURE ANGLE (θ).
- BUT, BECAUSE OF THE FRICTION, THESE FORCES WILL CAUSE ALSO NORMAL COMPONENTS N_1 AND N_2, ORIENTED TO OPPOSE THE RELATIVE MOTION.
- ON THIS SCHEME WE ASSUME THAT:

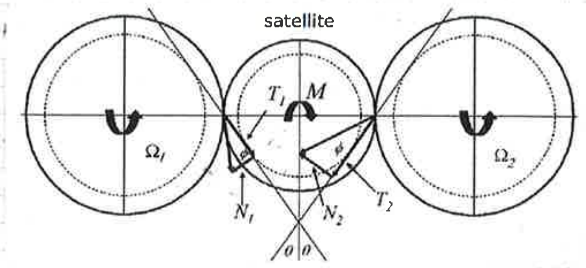
$\Omega_1 > \Omega > \Omega_2$

- IF f IS THE FRICTION COEFFICIENT AND ϕ THE FRICTION ANGLE, IT WILL BE:

$f = \tan\phi$

$N_1 = T_1 \tan\phi$

$N_2 = T_2 \tan\phi$



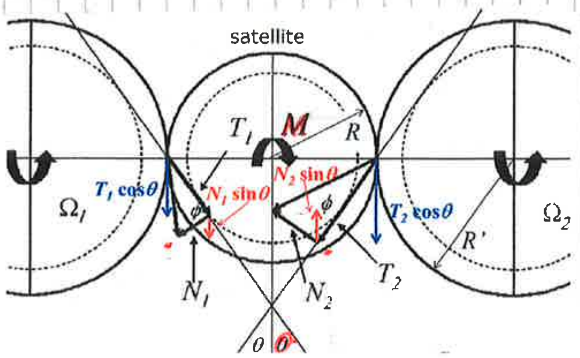
- BEVEL GEAR DIFFERENTIAL: EQUILIBRIUM EQUATION

- FOR THE ROTATION EQUILIBRIUM OF THE SATELLITE:

$(T_2 - T_1) R \cos\theta = (N_1 + N_2) R \sin\theta$

WITH:

R: REFERENCE RADIUS OF THE PRIMITIVE SATELLITES CONE.



- BEVEL GEAR DIFFERENTIAL: FINAL EQUATIONS

- TAKING INTO ACCOUNT THE PREVIOUS DEFINITION OF THE N FORCES, THE ROTATION EQUILIBRIUM EQUATION BECOMES:

$(T_2 - T_1) R \cos\theta = (T_1 + T_2) \tan\phi R \sin\theta$

$T_2 (\cos\theta - \tan\phi \sin\theta) = T_1 (\cos\theta + \tan\phi \sin\theta)$

- IF R' IS THE REFERENCE DIAMETER OF THE PLANET WHEELS PRIMITIVE CONE WE HAVE ALSO:

$(T_1 + T_2) R' \cos\theta = M$

- FINALLY:

$$\eta_{1,1} = \frac{M_1}{M_2} = \frac{T_1}{T_2} = \frac{(1 - \tan\phi \tan\theta)}{(1 + \tan\phi \tan\theta)} = \frac{(1 - f \tan\theta)}{(1 + f \tan\theta)}$$

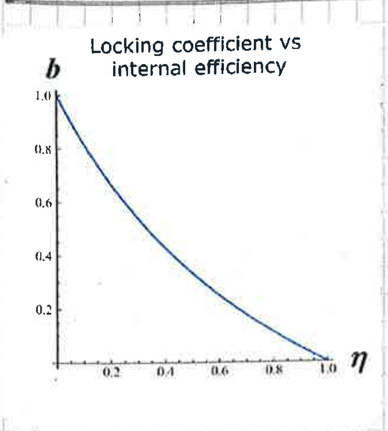
T13 DIFFERENTIALS AND FINAL DRIVES (PART 2)

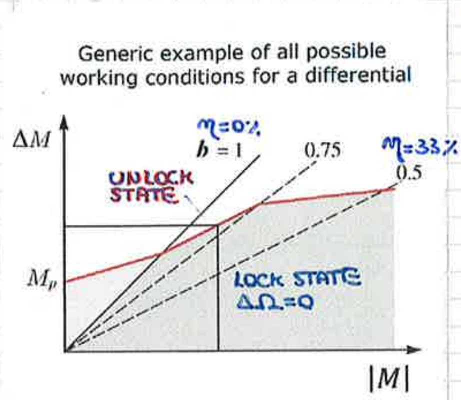
- SELF LOCKING DIFFERENTIAL
  - DEFINITION
  - LOCKING COEFFICIENT
  - GEARED VS CLUTCH-PACK DIFFERENTIALS
- TYPES OF SELF LOCKING AND CONTROLLED DIFFERENTIALS
  - ZF SYSTEM: SELF-LOCKING CLUTCH-PACK ZF DIFFERENTIAL
  - TRAC-LOK: SELF-LOCKING CLUTCH-PACK TRAC-LOK DIFFERENTIAL
  - TORSEN SYSTEM: SELF LOCKING GEARED TORSEN DIFFERENTIAL (TYPE 1)
  - FERGUSON JOINT: LIMITED-SLIP VISCOUS FERGUSON JOINT
  - HALDEX CONTROLLED DIFFERENTIAL
- DIFFERENT EFFECT ON VEHICLE DYNAMICS
  - CONTENT
  - REAR DRIVING AXLE IDEAL DIFFERENTIAL - REAL DIFFERENTIAL
  - REAR DRIVING AXLE DIFFERENTIAL COUPLED WITH VISCOUS JOINT
  - EFFECT ON UNDERSTEER CURVE OF THE CAR
  - PROPERTIES OF AN IDEAL DRIVING AXLE DIFFERENTIAL
- TRANSFER BOX DIFFERENTIAL: MAXIMUM ACCELERATION
  - CONTENT
  - MAX LONGITUDINAL ACCELERATION
  - EFFECT OF AWD ON VEHICLE HANDLING PERFORMANCE

[TEXTBOOK: Vol. I, PAR. 13.3, 13.4, 13.5]

- LOCKING COEFFICIENT (b)

$$b = \frac{M_2 - M_1}{M_2 + M_1} = \frac{1-\eta}{1+\eta} \qquad \begin{cases} \eta = 0 & b = 1 \\ \eta = 1 & b = 0 \end{cases}$$





WE PLOT STRAIGHT LINES CONSIDERING b OR η

THE RED LINE (TYPICAL OF CERTAIN TYPE OF SELF-L. DIFF.) REPRESENTS THE LIMIT OF ΔM BEFORE UNLOCKING

$$\Delta M \begin{cases} < b \cdot |M| & : \text{LOCKED } (\Delta\Omega = 0) \quad b \propto |M| \ (\text{or } b \propto \Delta\Omega \text{ or } b = \text{const}) \\ > b \cdot |M| & : \text{UN-LOCKED } (\Delta\Omega \neq 0) \text{: DIFF. WORKS EXACTLY} \end{cases}$$

IN CORRESPONDENCE OF THE RED LINE.

(POINTS OUTSIDE CANNOT BE REACHED)

- RED LINE: ALL WORKING CONDITIONS WITH UN-LOCKED DIFFERENTIAL $\Delta\Omega \neq 0$

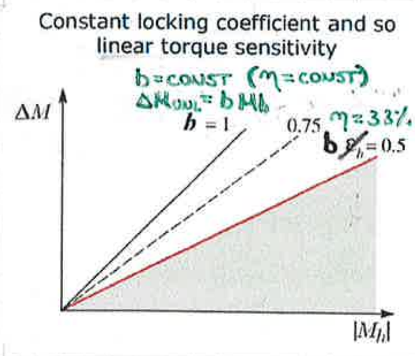
SHADED AREA: WORKING CONDITIONS WITH LOCKED DIFFERENTIAL $\Delta\Omega = 0$

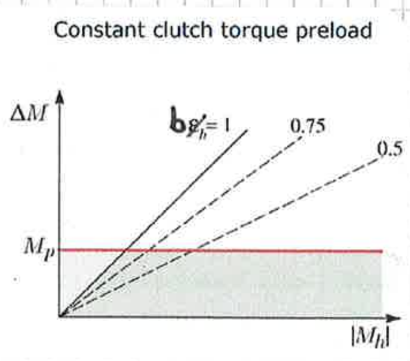
$\Delta M < b|M| = f(|M|)$

ΔM IS NOT SUFFICIENT TO OVERCOME THE INTERNAL FRICTION.

POINTS OUTSIDE THE SHADED AREA CANNOT BE REACHED.

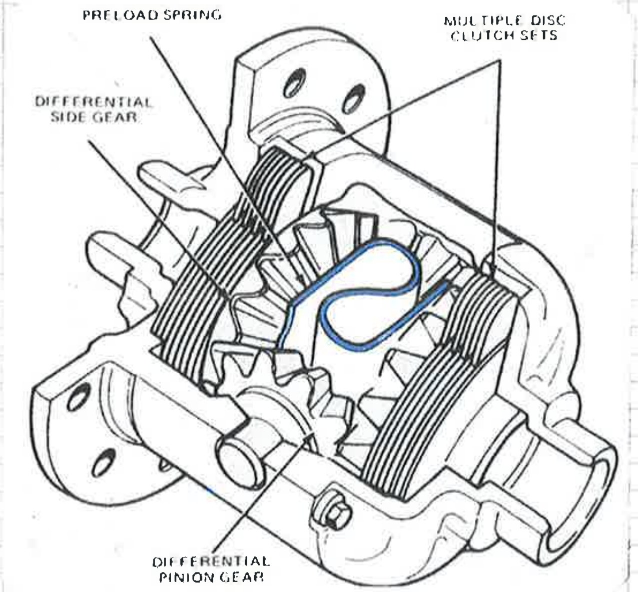
- GEARED VS CLUTCH-PACK DIFFERENTIALS





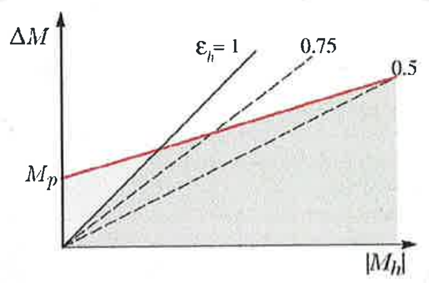
- GEARED DIFFERENTIALS: EMPLOY LOW EFFICIENCY GEAR TRAINS, RESULTING IN AN ALMOST CONSTANT INTERNAL EFFICIENCY AND LOCKING COEFFICIENT.
- CLUTCH-PACK DIFFERENTIALS: EMPLOY CLUTCHES HAVING SOME SPRING PRE-LOAD RESULTING IN A TORQUE DIFFERENCE NEEDED TO UNLOCK (M_p) THE DIFFERENTIAL EVEN WHEN NO TORQUE IS APPLIED TO THE DIFFERENTIAL HOUSING.

2 - TRAC-LOK : SELF-LOCKING CLUTCH-PACK TRAC-LOK DIFFERENTIAL



A PRELOAD SPRING IS POSITIONED BETWEEN THE TWO PLANET GEARS ; IT WORKS ALWAYS IN PARALLEL WITH THE INPUT TORQUE. THEREFORE, THE FRICTION TORQUE - THAT MUST BE OVERCOME TO UNLOCK THE DIFF. - IS THE SUM OF TWO CONTRIBUT. ONE DERIVING BY THE PRESENCE OF THE PRELOAD SPRING (M_P) AND THE OTHER PROPORTIONAL TO THE INPUT TORQUE ($k|M_h|$).

$\Rightarrow$ INPUT TORQUE SENSITIVITY BUT NO MORE DIRECT PROPORTIONALITY BETWEEN LOCKING T. (ΔM) AND INPUT T. ($|M_h|$).



$F_a = F_s + kM_h$

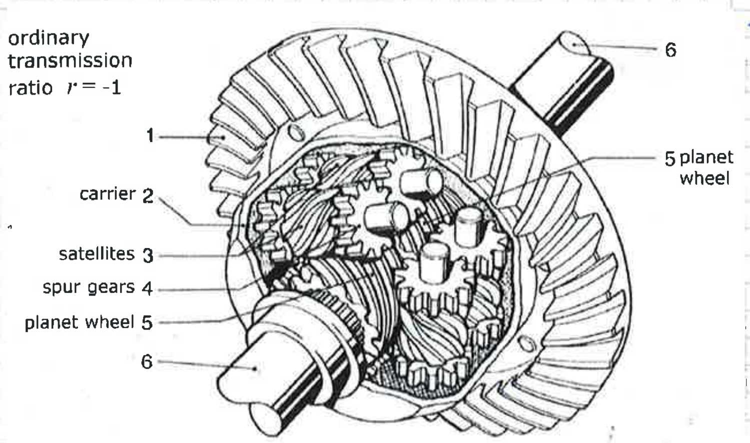
TRAC-LOCK: WHEN $\Delta M \geq \Delta M_{UNL} = M_P + k|M_{IN}| \Rightarrow$ UNLOCK

$\left(b = \frac{M_2 - M_1}{M_1 + M_2} = \frac{\Delta M}{M_{IN}} = \frac{1-\eta}{1+\eta} \neq \text{CONST} \Leftrightarrow \eta \neq \text{CONST} \right)$

$(b \propto |M_{IN}|)$

- INTERNAL EFFICIENCY AND LOCKING COEFF. ARE NEVER CONSTANT. ($b \neq$ CONST)
- TORQUE SENSITIVE DIFFERENTIALS BUT NOT LINEARLY. ($\Delta M \propto M_{IN}$)

3 - TORSEN SYSTEM: SELF-LOCKING GEARED TORSEN DIFFERENTIAL (TYPE-1)



- NO MORE BEVEL GEARS ; THE SATELLITES ARE COMPOSED BY AN HELICAL CENTRAL PART (HIGH HELIX ANGLE) AND BY TWO SPUR GEARS THEIR AXES OF ROT. ARE $\perp$ WRT DIFF. ROT A.

EACH S. ENGAGES A WORM P.G. AND ANOTHER S.

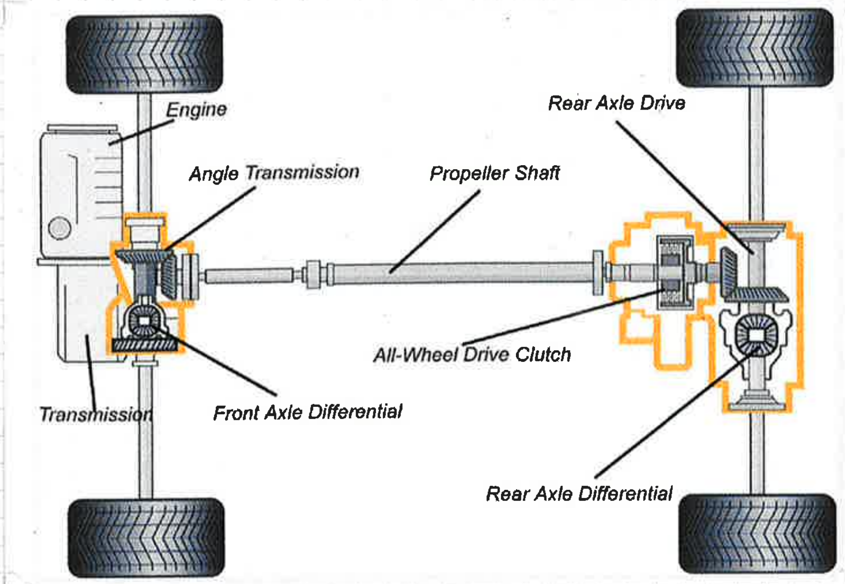
HIGH FRICTION $\Rightarrow$ VERY LOW η. $\left(TBR \uparrow = \frac{1}{\eta \downarrow} \right)$

WHEN $\Delta M \geq \Delta M_{UN} = b|M_{IN}| \Rightarrow$ DIFF. UNLOCKS

(b = CONST DEPENDING ON MECH. PARAMETERS)

- THE MECHANICAL EFFICIENCY IS VERY POOR DUE TO RELEVANT FRICTION OF WORM GEARS.
- THE LOCKING COEFFICIENT IS CONSTANT. (b = CONST)

5 HALDEX CONTROLLED DIFFERENTIAL

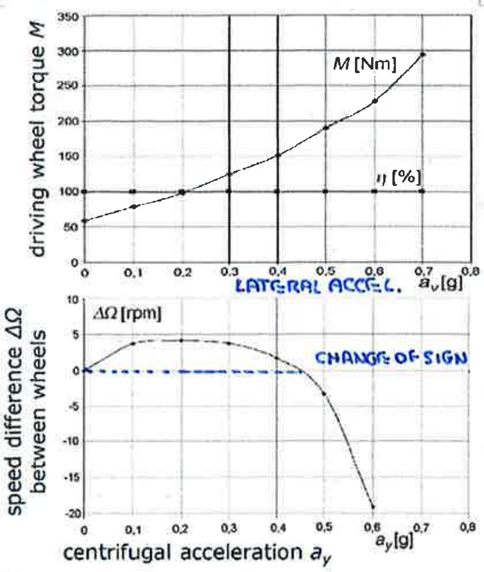


IT IS ESSENTIALLY A MULTI-DISK CLUTCH THAT COULD BE CONTROLLED AS A FUNCTION OF THE RELATIVE SPEED BETWEEN THE 2 SHAFTS.
=> CONTINUOUS CHANGE OF T TRANSFERRED BETWEEN THE AXLES DEPENDING ON THE WORKING COND.

- BASED ON A CONTROLLED MULTI-DISC CLUTCH CONNECTING THE FRONT AND REAR AXLES.

- REAR DRIVING AXLE IDEAL (OPEN) DIFFERENTIAL (200[m] RADIUS STEERING PAD TEST)

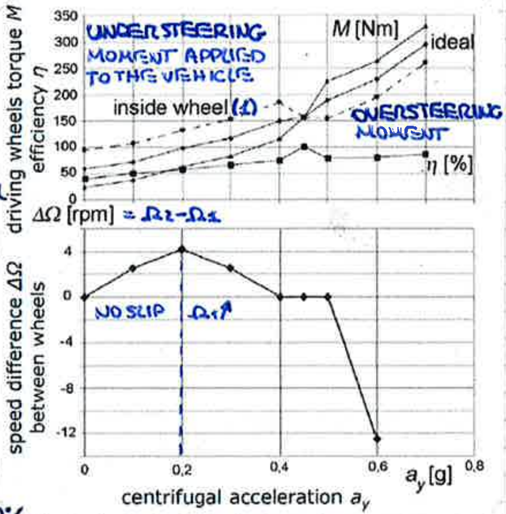
$\boxed{\text{EFFICIENCY} = 100\%}$ $[\eta = 100\%]$



- **TORQUE IS EQUAL ON BOTH THE WHEELS** AND INCREASES QUICKLY WITH SPEED BECAUSE OF THE DRIVING RESISTANCE INCREASE AND BECAUSE OF SIDE SLIP FORCES LONGITUDINAL COMPONENTS.
- VERTICAL LOAD TRANSFER ON DRIVING WHEELS INCREASES ABOUT WITH THE SQUARE OF THE SPEED, BEING PROPORTIONAL TO THE ACCELERATION a_y.
- THE **SPEED DIFFERENCE $\Delta\Omega$ INCREASES** BECAUSE OF THE **DIFFERENT WHEEL PATH**, AT FIRST, AND **DECREASES AFTER**, BECAUSE OF THE FACT THAT THE INSIDE WHEEL, INITIALLY SLOWER, WILL SEE ITS LONGITUDINAL SLIP INCREASED.

- REAR DRIVING AXLE REAL (OPEN) DIFFERENTIAL

$\boxed{\text{EFFICIENCY} = \text{CONST} \leq 100\%}$ $[\eta \leq 100\% \text{ (1 CUSP)}]$



- IN THIS CASE, ANGULAR SPEED DIFFERENCE WILL ALLOW A TORQUE DIFFERENCE TOO; THE **INSIDE WHEEL, INITIALLY SLOWER**, WILL **RECEIVE A LARGER TORQUE UNTIL SPEED DIFFERENCE** BECOMES **ZERO**; TORQUE DIFFERENCE IS PROPORTIONAL TO TOTAL DRIVING TORQUE.
- **AROUND THE POINT WHERE SPEED DIFFERENCE** IS **ZERO**, THE DRIVING WHEELS **TORQUE LINES** WILL **CROSS**.
- **EFFICIENCY DIAGRAM** SHOWS A **CUSP** WITH **VALUE 100%**, WHERE THE **SPEED DIFFERENCE** GOES TO **ZERO**.

$\frac{V_1}{R_1} = \frac{V_2}{R_2} : V_1 = V_2 \underbrace{\frac{R_1}{R_2}}_{<1} \Rightarrow \Omega_1 < \Omega_2$

- THE DRIVING WHEEL TORQUE INCREASES ($M\uparrow$) WHEN THE LATERAL ACCELERATION INCREASES ($a_y\uparrow$)
- IN THE **REAL CASE**, THE **INSIDE WHEEL (1)**, BECAUSE OF ITS **LOWER ROT. SPEED** ($\Omega_1 < \Omega_2$), **RECEIVES MORE TORQUE** ($M_1 > M_2$); WHEN THE **LATERAL ACC. INCREASES** ($a_y\uparrow$), THANKS TO THE **LOAD TRANSFER**, THE INSIDE WHEEL IS SUBJECTED TO A LOWER VERTICAL FORCE THAT **CAUSES SLIP**; SO, $\Omega_1\uparrow \Rightarrow \Delta\Omega\downarrow$; WHEN $\Omega_1 > \Omega_2 \Rightarrow M_2 > M_1$ (HIGHER T GOES TO THE SLOWER WHEEL). WHERE $\Delta\Omega = 0$ THE **DIFF.** WORKS AS **LOCKED DIFF.**

WHEN $M_{INSIDE} > M_{OUTSIDE}$: UNDERSTEERING MOMENT

WHEN $M_{OUTSIDE} > M_{INSIDE}$: OVERSTEERING MOMENT

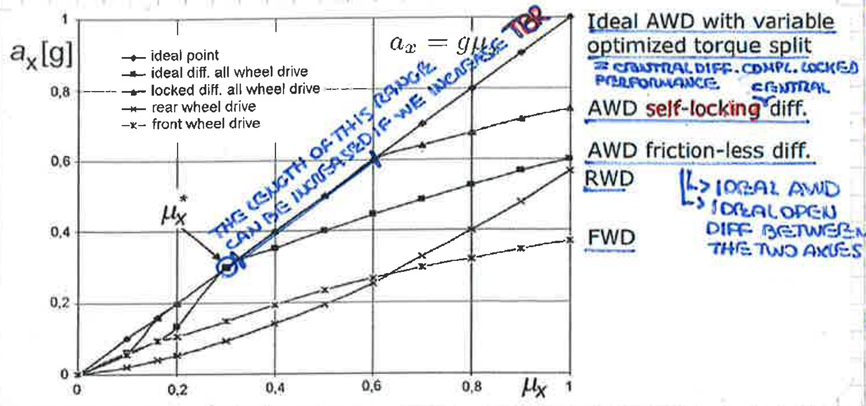
- WE CAN OUTLINE THE FOLLOWING FACTS: (UNDERSTEERING $\Omega_{INS} < \Omega_{OUT}$ $M_{INS} > M_{OUT}$)
  - THE ZERO SPEED STEERING ANGLE INCREASES BECAUSE OF THE DIFFERENTIAL FRICTION TORQUE.
  - THE **NON-SYMMETRIC TRACTION TORQUE** INDUCES A **YAW MOMENT** ("MOMENTO DI IMBARDATA") (UNDERSTEERING DIRECTION) AT A POSITIVE SPEED DIFFERENCE; THE DRIVER MUST INCREASE THE STEERING WHEEL SIDE SLIP ANGLE, IN ORDER TO EQUILIBRATE THIS TORQUE.
  - THIS ADDITIONAL YAW TORQUE GOES TO ZERO WHERE THE SPEED DIFFERENCE BECOMES ZERO AND BECOMES, AFTER THIS POINT, AN OVERSTEERING YAW TORQUE.
  - THE ACCELERATION LIMIT WHERE THE REAR WHEEL SIDE SLIP ANGLE BECOMES EXCESSIVE AND THE STEERING ANGLE GOES TO ZERO (POWER OVERSTEER) SHIFTS TOWARD HIGHER VALUES BECAUSE OF THE LOWER TRACTION FORCE ON THE INSIDE WHEEL.
  - INCREASE UNDERSTEER WHEN ENTERING THE CURVE BUT WITH AN AMPLER EXPLOITATION OF THE ROAD GRIP.

- PROPERTIES OF AN IDEAL DRIVING AXLE DIFFERENTIAL

- AN IDEAL DIFFERENTIAL LOCKED BY MEANS OF A CONTROLLED CLUTCH SHOULD SATISFY THE FOLLOWING CONDITIONS:
  - SIMULATE THE BEHAVIOUR OF A NO FRICTION DIFFERENTIAL FOR HIGH CURVATURE CURVES AT LOW SPEED, OR WHEN THE ROLLING SPEEDS DIFFERENCE IS ONLY DUE TO PRESSURE OR WHEN A HIGH GRIP IS NOT REQUIRED.
  - SIMULATE THE BEHAVIOUR OF A LOCKED DIFFERENTIAL ON HIGH ACCELERATIONS OR LOW GRIP ROADS.
  - TO ACT AS A FREE DIFFERENTIAL DURING BRAKING.

(12/01 2:52:00)

- MAX LONGITUDINAL ACCELERATION (ALL POSS. CONF.:)

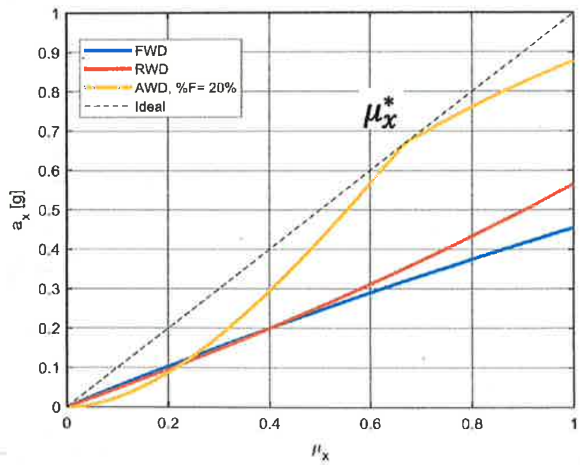


WE PLOT THE MAXIMUM POSSIBLE ACHIEVABLE ACC. (a_x) AS A FUNCTION OF THE AVAILABLE FRICTION COEFF. (μ_x) BETWEEN TIRES AND GROUND FOR DIFF. ARCHITECTURES OF THE CAR.

: THERE IS ONLY ONE VALUE OF μ_x FOR WHICH WE CAN MAXIMIZE THE CAR PERFORMANCE

IN CASE OF SELF LOCKING DIFF. THERE IS A RANGE OF μ_x IN WHICH WE ARE ABLE TO MAXIMIZE THE CAR P.
IF TBR ↗ μ_x RANGE OF MAX PERF. ↗

- FIGURE: DIAGRAM OF THE MAXIMUM LONGITUDINAL ACCELERATION (a_x) AS A FUNCTION OF THE LONGITUDINAL TRACTION COEFFICIENT (μ_x), IN THE CASE OF AN IDEAL MATERIAL POINT (WITHOUT LOAD TRANSFER), A FRONT WHEEL DRIVE (FWD), A REAR WHEEL DRIVE (RWD) AND AN ALL WHEEL DRIVE (AWD) WITH NO FRICTION DIFFERENTIAL AND FOR A LOCKED DIFFERENTIAL WITH CONSTANT LOCKING COEFFICIENT.
- THE HIGHER THE LOCKING COEFFICIENT THE WIDER THE EXTENSION WHERE THE MATERIAL POINT BEHAVIOUR IS COPIED.



- AWD WITH CONSTANT TORQUE DISTRIBUTION.
- ON THE LEFT OF μ_x^* REAR WHEELS WILL SPIN BECAUSE THE LOAD TRANSFER IS NOT SUFFICIENT TO GUARANTEE THE NECESSARY TRACTION FORCE; THE OPPOSITE OCCURS FOR HIGHER ACCELERATIONS.

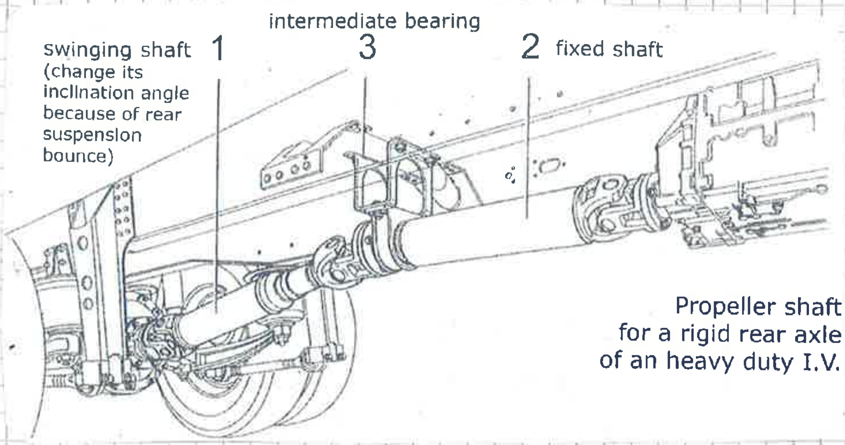
T14 SHAFTS AND JOINTS

- **SHAFTS**
  - CONTENT
- **PROPELLER SHAFTS**
  - IV APPLICATIONS
  - TORSIONAL FREQUENCY
  - BENDING NATURAL FREQUENCY
  - NATURAL FREQUENCIES COMPUTATION
- **HALF SHAFTS**
  - LAYOUT
  - DESIGN CONSIDERATION
  - HALF SHAFTS WITH DVA
  - DVA DESIGN CONSIDERATIONS
  - MULTIPLE DVA DESIGN CONSIDERATIONS
- **UNIVERSAL JOINTS**
  - PROPERTIES
  - DOUBLE UNIVERSAL JOINTS
  - UNIVERSAL HOOK JOINT FOR PROPELLER SHAFTS
  - UNIVERSAL JOINTS TRANSMISSION FOR A FRONT RIGID STEERING AXLE
- **CONSTANT SPEED JOINTS**
  - CONTENT
  - COMPLETE HALF SHAFT FOR A FRONT WHEEL DRIVEN CAR
  - CONSTANT SPEED RZEPPA JOINT: DESCRIPTION
  - CONSTANT SPEED JOINTS CHARACTERISTICS
  - CONSTANT SPEED RZEPPA JOINT: COMPONENTS
  - CONSTANT SPEED RZEPPA JOINT: SCHEME

[TEXTBOOK: VOL I, CHAPTER 14]

• PROPELLER SHAFTS

- I.V APPLICATIONS



- THE TWO PIECES DESIGN ALLOWS TO INCREASE THE NATURAL BENDING FREQUENCIES OF THE SHAFT, FAR FROM THE NATURAL BENDING FREQUENCY OF THE CLASSIS TO AVOID ANNOYING BOOMS.
FOR SHORTER VEHICLES, AS LIGHT DUTY INDUSTRIAL VEHICLES, A SINGLE PIECE PROPELLER SHAFT CAN BE USED.

- TORSIONAL FREQUENCY

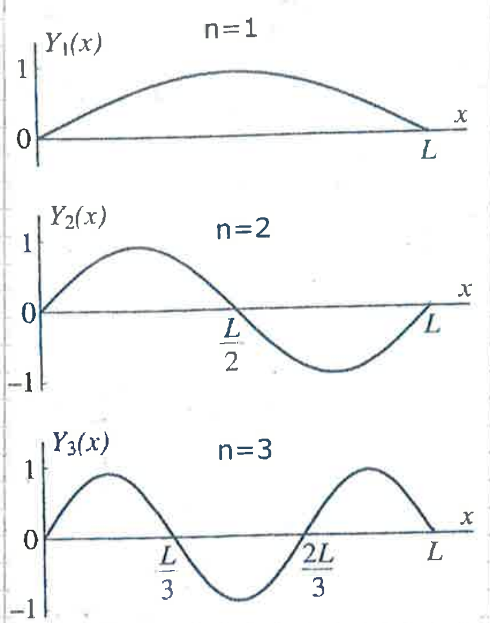
- ALSO THE NATURAL TORSIONAL FREQUENCY OF THE SYSTEM, INCLUDING POWERTRAIN, TRANSMISSION SHAFT AND VEHICLE, HAS A FUNDAMENTAL IMPORTANCE ON VEHICLE COMFORT AND DRIVEABILITY; THIS NATURAL FREQUENCY IS DIRECTLY INFLUENCED BY THE PROPELLER SHAFT.
- A VERY SIMPLE MODEL OF THE SYSTEM COULD INCLUDE TWO FLYWHEELS OF INERTIA J_m AND J_v REPRESENTING RESPECTIVELY THE POWERTRAIN INERTIA (ENGINE AND GEARBOX) AT A GIVEN SPEED) AND THE INERTIA OF THE VEHICLE. THEESE FLYWHEELS ARE CONNECTED THROUGH A TRANSMISSION SHAFT, WHOSE LENGHT IS L.
- IF G IS THE SHEAR MODULE OF THE SHAFT MATERIAL, I_P THE POLAR INERTIA OF ITS CROSS SECTION, WE WILL HAVE, AT RESONANCE CONDITION:

$$\omega = \sqrt{\frac{G I_P}{J_m L} + \frac{G I_P}{J_v L}}$$

: TORSIONAL NATURAL FREQUENCY OF THE TRANSMISSION SHAFT [Rad/S].

NATURAL MODES FOR PINNED-PINNED EULER-BERNOULLI BEAM:

$\omega_m = m^2 \cdot \omega_c$, m IS THE MODE NUMBER.

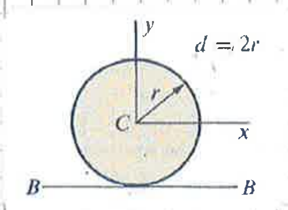


HOW TO INCREASE THIS FREQUENCY? $\rightarrow$ $\omega_c = \frac{\pi^2}{L^2}\sqrt{\frac{E}{\rho}\frac{I_s}{S}}$

LET CONSIDER AN EXAMPLE:

SOLID VS HOLLOW SHAFT

CIRCLE:



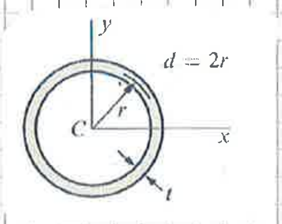
$A = \pi R^2 = \frac{\pi d^2}{4}$

$I_x = I_y = \frac{\pi R^4}{4} = \frac{\pi d^4}{64}$; $I_{xy} = 0$

$I_P = \frac{\pi R^4}{2} = \frac{\pi d^4}{32}$; $I_{BB} = \frac{5\pi R^4}{4} = \frac{5\pi d^4}{64}$

$\frac{I_s}{S} = \frac{I_x}{A} = \frac{\frac{\pi R^4}{4}}{\pi R^2} = \frac{R^2}{4}$

CIRCULAR RING: (APPROX. FORMULA, FOR SMALL t)



$A = 2\pi R t = \pi d t$

$I_x = I_y = \pi R^3 t = \frac{\pi d^3 t}{8}$; $I_{xy} = 0$

$I_P = 2\pi R^3 t = \frac{\pi d^3 t}{4}$

$\frac{I_s}{S} = \frac{I_x}{A} = \frac{\pi R^3 t}{2\pi R t} = \frac{R^2}{2}$

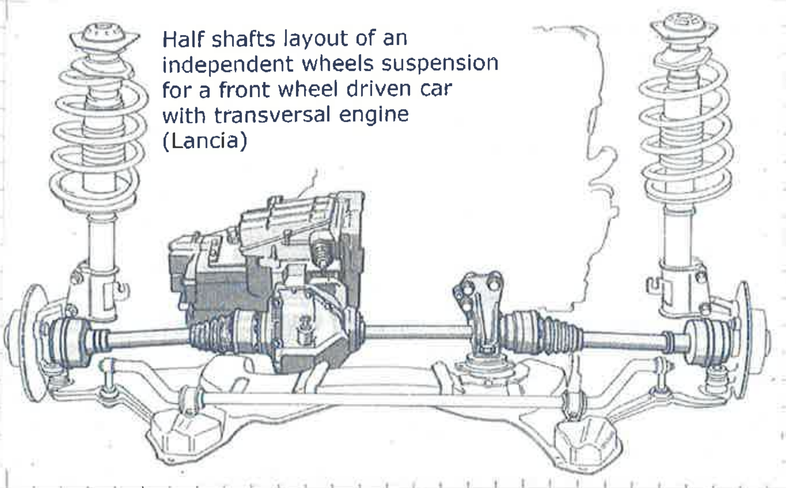
$$\frac{\omega_{c,\,HOLLOW}}{\omega_{c,\,SOLID}} = \sqrt{2}\,\underbrace{\frac{R_{HOLLOW}}{R_{SOLID}}}_{>1}$$

- THEN IT IS IMPORTANT TO SIZE TRANSMISSION SHAFTS WITH VERY RIGID CROSS SECTION AND VERY LIGHT DESIGN (TUBULAR SECTION)

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 187 di 260

• HALF SHAFTS

- LAYOUT



Half shafts layout of an independent wheels suspension for a front wheel driven car with transversal engine (Lancia)

• UNIVERSAL JOINTS ARE NOT APPLIED TO HALF SHAFTS: CONSTANT VELOCITY (CV) JOINTS ARE APPLIED BECAUSE OF THE LIMITED SPACE AND WIDE ANGLES.

- DESIGN CONSIDERATIONS

• TRANSVERSALLY MOUNTED POWERTRAIN → HALF-SHAFT WITH DIFFERENT LENGTHS → DIFFERENT TORSIONAL STIFFNESS IF THE SECTION IS THE SAME.

$k_s = \frac{G I_p}{L}$: TORSIONAL STIFFNESS

- PROBLEM: SELF STEERING MOMENT BECAUSE STIFFER HALF-SHAFT TRANSMITS HIGHER TORQUE DURING TRANSIENTS
- SOLUTION 1: BIGGER CROSS SECTION ON THE LONGER HALF-SHAFT (INCREASE I_p)
- PROBLEM DUE TO SOLUTION 1: NATURAL BENDING FREQUENCY OF THE LONGER HALF SHAFT COULD BE TOO LOW.

$$\omega_c = \frac{\pi^2}{L^2}\sqrt{\frac{E I_s}{\rho S}}$$

- SOLUTION 2: THE LONGER HALF-SHAFT DIVIDED IN TWO PIECES, THE FIRST FIXED TO THE POWERTRAIN ASSEMBLY.

• IN EQUAL LENGTH HALF SHAFT APPLICATIONS LENGTH OF HALF SHAFT IS SHORTER, HENCE ITS FIRST BENDING MODE IS HIGHER AND ABOVE ENGINE MAIN FIRING ORDER EXCITATION FREQUENCY.

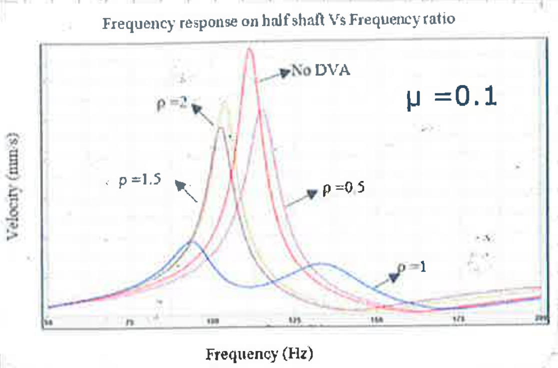
• THE APPLICATION OF TUBOLAR SECTION HERE MAY PROVE IMPOSSIBLE BECAUSE OF THE LIMITED SPACE.

- DVA DESIGN CONSIDERATIONS

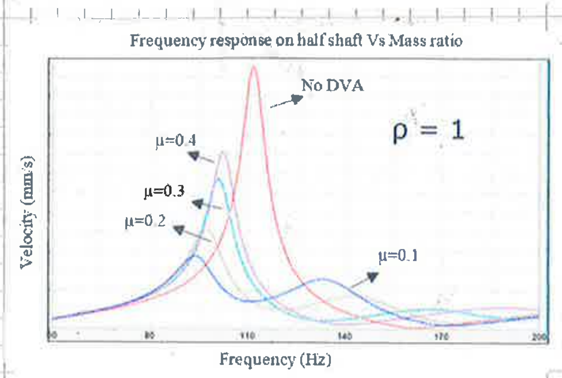
- MASS RATIO (μ), FREQUENCY RATIO (ρ), DAMPING RATIO (η) AND LOCATION OF DAMPER ARE TYPICAL TUNABLE PARAMETERS OF DVA TO OBTAIN MAXIMUM VIBRATION ABSORPTION EFFECT.

$\mu = \frac{m}{M}$: MASS RATIO (m: MASS OF DVA ; M: MASS OF HALF SHAFT)

$\rho = \frac{f}{F}$: FREQUENCY RATIO (f: NATURAL FREQ. OF DVA ; F: 1st MODE OF H.S.)



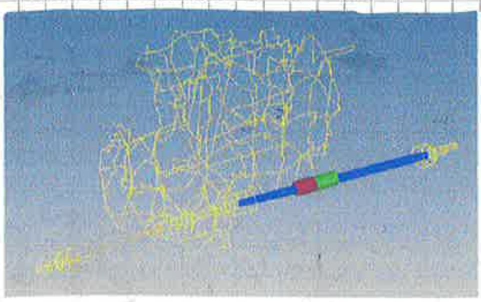
- FREQUENCY RATIO OF 1 GIVES A GOOD REDUCTION IN RESPONSE AMPLITUDE OF THE HALF SHAFT.

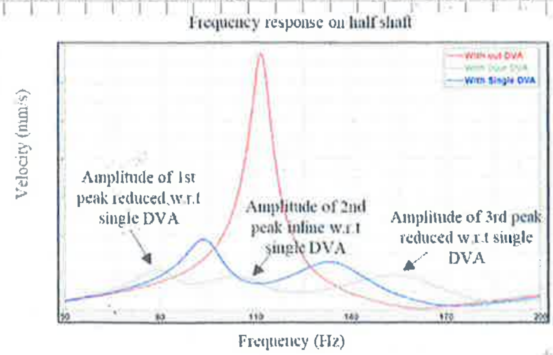


- IT IS EVIDENT FROM THESE RESULTS THAT MASS RATIO (μ) 0.1 GIVES A SIGNIFICANT REDUCTION IN RESPONSE AMPLITUDE OF THE HALF SHAFT.

- MULTIPLE DVA DESIGN CONSIDERATIONS

- MULTIPLE TUNED MASS DAMPERS (MTMD) FREQUENCIES TUNED IN NEIGHBORHOOD OF THE NATURAL FREQ. OF A SDOF PRIMARY SYSTEM IMPROVES VIBR. ABSORBING EFFECT.
- OPTIMIZATION TOOL MAY BE EMPLOYED TO PROGRAM THE AUTOMATIC PROCESSING OF ALL DESIRED ITERATIONS (PARAMETERS COMBINATION OF THE 2 DVA).
- THE OBJECTIVE OF OPTIMIZATION IS TO FIND MULTIPLE DVA TUNABLE PARAMETERS TO GIVE MINIMUM HALF SHAFT RESPONSE AT PEAKS WITHIN 50÷200 [Hz] RANGE AFTER INSTALLING DVA.

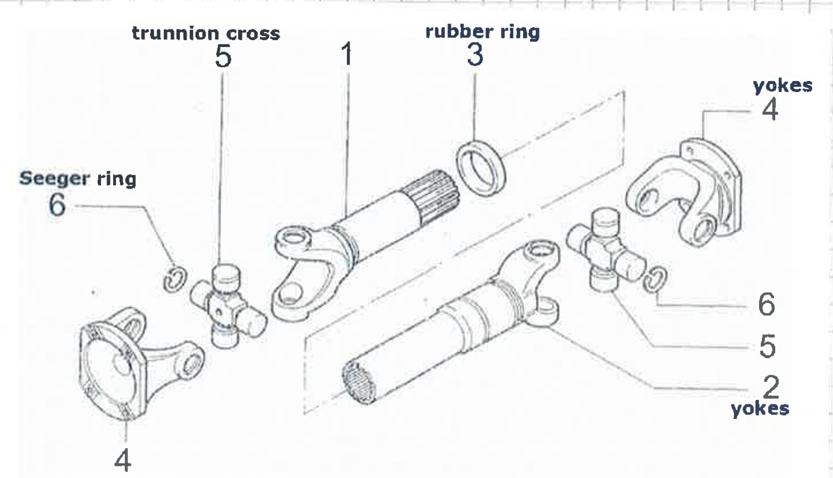




- FRF ANALYSIS ON THE HALF SHAFT SHOWS THAT TWO PEAKS WITH A SINGLE DVA WERE FURTHER SPLIT INTO 3 PEAKS WITH OPTIMIZED DUAL DVA.

- IF WE INSTALL ON THE SECOND SHAFT A SECOND UNIVERSAL JOINT, CONNECTING THE THIRD SHAFT (3), WE CAN OBTAIN A CONSTANT SPEED TRANSMISSION IF:
  a) THE SECOND JOINT HASE THE SAME WORKING ANGLE α;
  b) ITS INPUT YOKE IS ON THE SAME PLANE AS THE OUTPUT YOKE OF THE FIRST JOINT (THIS SENTENCE IS ONLY TRUE IN THE IDEAL CASE WHERE THE ROTATING MASS OF THE SECOND SHAFT (2) IS NEGLIGIBLE IN COMPARISON WITH THE TOTAL ROTARY MASS OF THE SYSTEM).
- THIS TASK CAN BE ACCOMPLISHED IN TWO WAYS:
  1) THE CONFIGURATION APPLIED TO PROPELLER SHAFTS IS THAT SHOWN IN THE UPPER PART OF PREVIOUS FIGURE (SHAFTS 1 AND 3 ALWAYS PARALLEL); THIS CONDITION MUST BE OBTAINED BY A SUITABLE ELASTO-KINEMATIC BEHAVIOUR OF THE REAR AXLE SUSPENSION.
  2) THE SECOND LAYOUT IN THE LOWER PART OF THE FIGURE IS USED PARTICULARLY FOR HALF SHAFTS OF FRONT DRIVING AXLE OF SOME OFF-ROAD VEHICLES, FEATURING RIGID AXLE SUSPENSION, WHERE THE HIGH VALUE OF THE TORQUE DOESN'T ALLOW TO USE CONSTANT SPEED JOINTS.

- UNIVERSAL HOOKE JOINT FOR PROPELLER SHAFTS



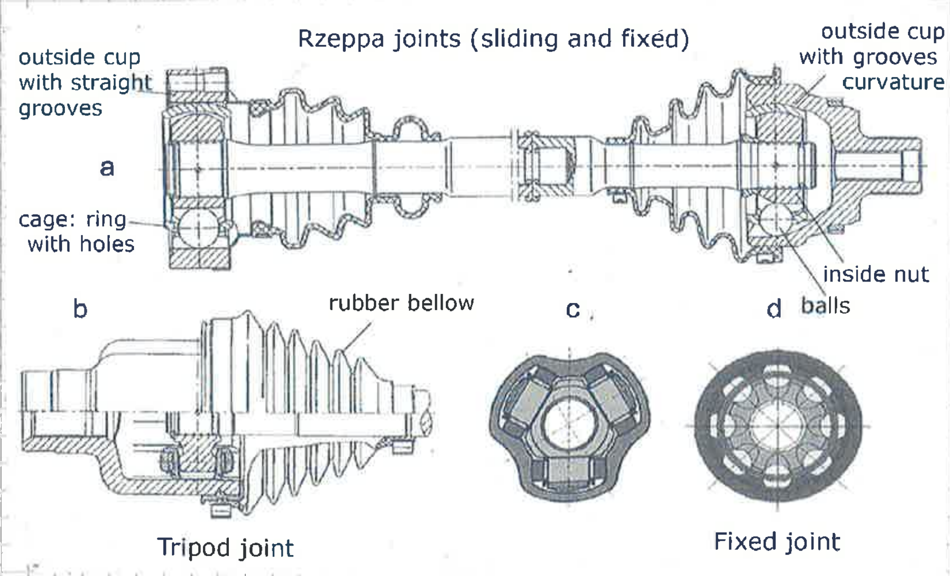
- TWO TRUNNION CROSSES (5) CONNECT THE YOKES (2 AND 4);
- THE CONNECTION BETWEEN CROSSES AND YOKES IS MADE USUALLY BY SEALED NEEDLE BEARINGS, KEPT IN PLACE BY SEEGER RINGS.

- CONSTANT SPEED JOINTS

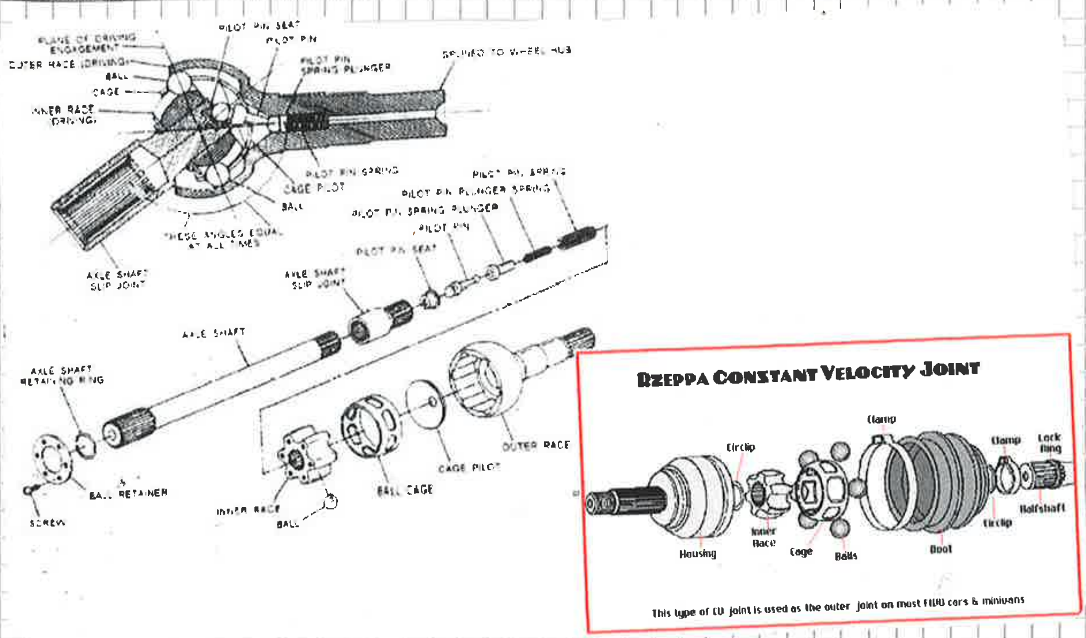
- CONTENT

- THE DOUBLE UNIVERSAL JOINT CANNOT BE USED IN THE TRANSMISSION TO THE STEERING WHEELS OF AN INDEPENDENT SUSPENSION.
- THIS SOLUTION HAS BEEN APPLIED IN THE PAST FOR NON STEERING REAR WHEELS: ALSO THIS SOLUTION DOES NOT APPEAR TODAY ACCEPTABLE BECAUSE OF THE LIMITATIONS ON CAMBER RECOVERY, IN ORDER TO KEEP PARALLEL THE FIRST AND THIRD SHAFTS OF THE TRANSMISSION.
- IN INDEPENDENT WHEEL SUSPENSIONS ONLY CONSTANT VELOCITY JOINTS OF THE RZEPPA TYPE ARE APPLIED, AS THOSE SHOWN IN THE UPPER PART OF NEXT FIGURE; THE SCHEME REPRESENTS A SECTION OF A COMPLETE HALF SHAFT FOR A FRONT WHEEL DRIVEN CAR (THE WHEEL, NOT REPRESENTED, IS ON THE RIGTH OF THE SCHEME).

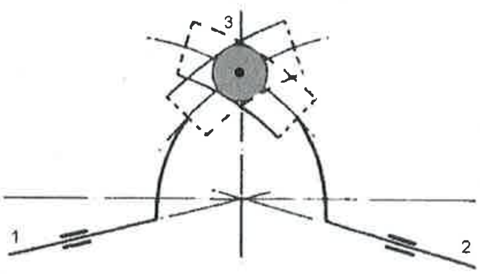
- COMPLETE HALF SHAFT FOR A FRONT WHEEL DRIVEN CAR.

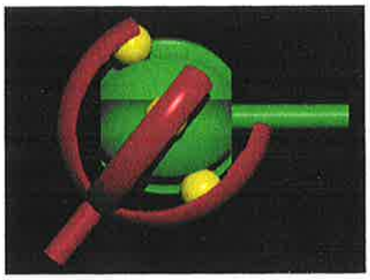


- CONSTANT SPEED RZEPPA JOINT : COMPONENTS



- CONSTANT SPEED RZEPPA JOINT : SCHEME





- IN THE LEFT FIGURE, ONLY ONE BALL (3) IS REPRESENTED, ENGAGED WITH TWO GROOVES IN THE SAME TIME; THESE GROOVES ARE RESPECTIVELY FIXED TO THE TWO PARTS OF THE JOINT (1) AND (2); ONLY THE PART OF THESE GROOVES REPRESENTED WITH FULL LINE IS REALLY EXISTING ON THE NUT AND THE CUP.
- DURING JOINT ROTATION THE BALLS WILL BE SUBJECTED TO DISPLACEMENTS ALONG THEIR GROOVES PROPORTIONAL TO THE JOINT WORKING ANGLE.
- THIS DISPLACEMENT CANNOT BE DESCRIBED BY PURE ROLLING OF THE BALLS IN THEIR GROOVES.
- FRICTION OF THE BALL JOINT MUST BE CAREFULLY TAKEN INTO ACCOUNT: ENERGY LOSS, EXCHANGE OF FORCE IN TRANSVERSAL DIRECTION, INCREASES THE FORCE TO HAVE THE JOINT SLIDING UNDER TORQUE.
- TRIPOD JOINT SOLVE THIS PROBLEM WITH A DIFFERENT ARCHITECTURE: THE BALLS ARE REPLACED BY THREE ROLLERS, MOUNTED ON PINS WITH NEEDLE BEARINGS AND CAN ROLL IN THE STRAIGHT GROOVES OF THE CUP.

• SHIFTING MECHANISM

- FUNCTIONS

• MECHANICAL DEVICES THAT ENABLE THE DRIVER ENGAGING GEARS TO OBTAIN THE DESIRED TRANSMISSION RATIO ARE CALLED : SHIFTING MECHANISMS.

• MAIN FUNCTIONS:

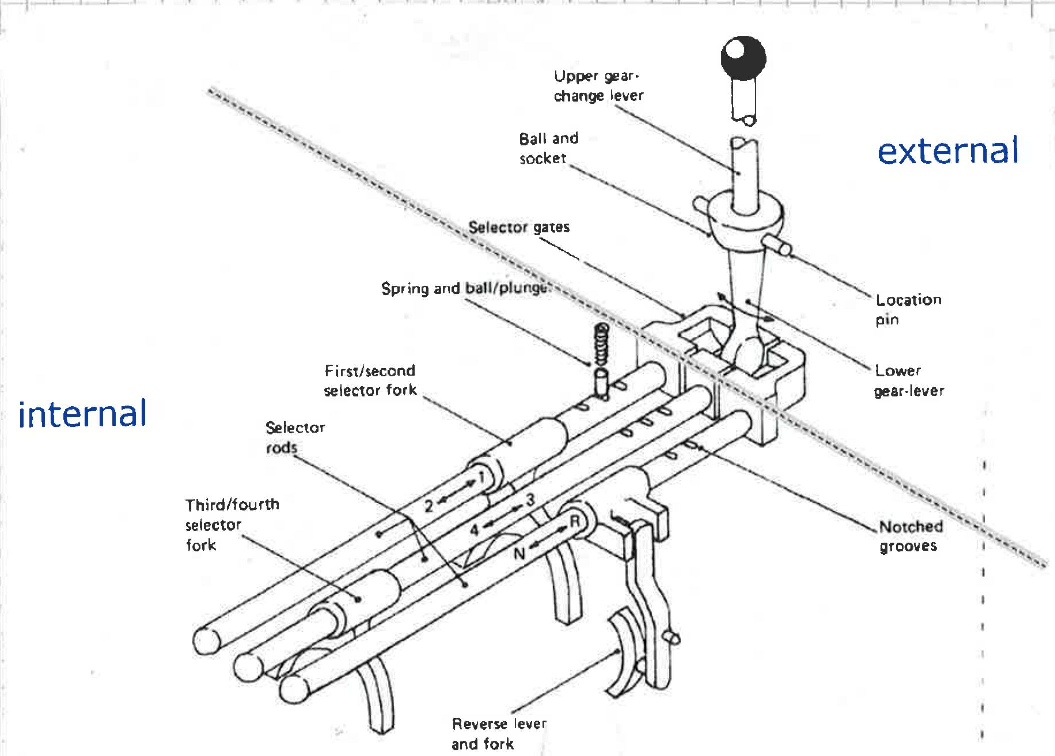
- SELECTING THE APPROPRIATE GEAR
- ACTIVATING SLEEVE AND SYNCHRONIZERS TO ENGAGE SELECTED GEAR WITHOUT PROBLEMS ON GEARS MECHANICAL INTEGRITY, ERGONOMIC ASPECTS AND INTERIOR NOISE.

• THE PART OF THESE MECHANISMS CONTAINED IN THE GEARBOX CASING ARE CALLED: INTERNAL MECHANISMS.

• THE SHIFTING MECHANISMS MOUNTED PARTLY ON THE OUTSIDE OF THE GEARBOX CASING, PARTLY ON THE VEHICLE BODY ARE CALLED: EXTERNAL MECHANISMS; THEY CONNECT THE GEARSHIFT LEVER WITH THE INTERNAL SHIFTING MECHANISMS.

• SHIFTING MECHANISMS ARE RESPONSIBLE FOR THE DRIVER FEELING FROM THE SHIFT STICK. AN IDEAL FEELING SHOULD BE : POSITIVE AND PRECISE.

- SIMPLIFIED SCHEME



© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 199 di 260

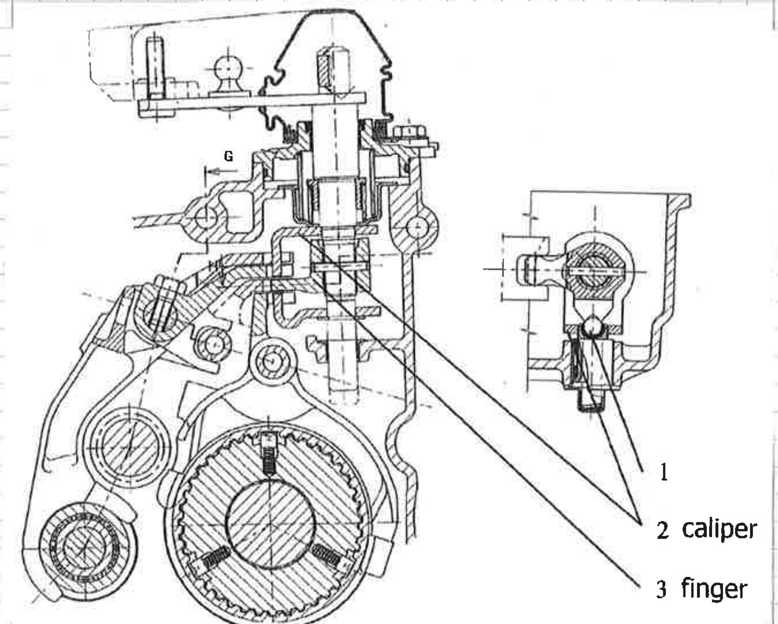
- CONTENT

- THE MECHANISM SHOWN IN THE PREVIOUS FIGURE IS MADE BY THREE SELECTOR BARS (4,5,6), EACH OF THEM WITH A FORK THAT ENGAGES WITH A GROOVE ON THE SHIFTING SLEEVE; BY MOVING ONE BAR AT A TIME, THE DESIRED SPEED CAN BE SHIFTED. EACH BAR IS MOVED BY A GATE 8, EXISTING ON EACH FORK.
- A FINGER LEVER 3 ON A SHAFT 2 CAN EITHER ROTATE AROUND THE SHAFT CENTER LINE OR SHIFT ALONG THIS CENTER LINE: WHEN SHIFTED IT SELECTS THE CORRECT SELECTOR BAR; WHEN ROTATED, IN ONE OF THE THREE GATES, IT SHIFTS THE DESIRED SPEED. THE FINGER MATCHES WITH A SUITABLE LOW CLEARANCE WITH THE GATES ON THE FORKS.
- AUXILIARY DEVICES ARE NEEDED, SUITABLE FOR:
  - MAINTAINING THE MECHANISMS IN THE SELECTED POSITION, EVEN IF THE DRIVER ABANDONS THE SHIFT STICK.
  - AVOIDING THAT TWO OR MORE GEARS ARE SELECTED IN THE MEAN TIME.

IN ORDER TO AVOID PROBLEMS ON DRIVER AND PASSENGERS AND ON THE GEARBOX MECHANICAL INTEGRITY.

- SHIFTING MECHANISM WITH CALIPER INTERLOCKING DEVICE

- TO IMPROVE SHIFTING QUALITY (AFFECTED BY CLEARANCES AND TERANCES, SHAPE ERRORS, WEAR) A BETTER INTERLOCKING DEVICE HAS BEEN DESIGNED, WITHOUT PLUNGERS AND GROOVES.
- THE FINGER THAT MOVES THE BARS SHOWS ON HIS VIEW ON THE RIGHT OF THE DRAWING THREE SPHERICAL SEATS THAT CAN MATCH A BALL PRESSED BY A SPRING (ONLY ONE IS SHOWN). THE THREE CORRESPONDING POSITIONS FIT THE NEUTRAL POSITION OF THE THREE BARS.
- A U SHAPED BRACKET ("STAFFA"), CALLED CALIPER, WITH TWO PRONGS ("PUNTE"), MATCHES THE CARTRIDGE ("CARTUCCIA") OF THE BALL; THE CALIPER CAN SLIDE UP AND DOWN IN THE NEUTRAL POSITION, BUT CAN'T ROTATE. THE TWO PRONGS MATCH THE END OF THE FINGER WITH NO CLEARANCE.
- THE CALIPER ALLOWS THE MOTION OF ONE BAR AT A TIME, WHILE LOCKING THE REMAINING BARS IN NEUTRAL POSITION.
- IF THE FINGER IS MOVED ON AN INTERMEDIATE POSITION, THE CALIPER WOULD BLOCK IN NEUTRAL POSITION ALL THE MECHANISM.

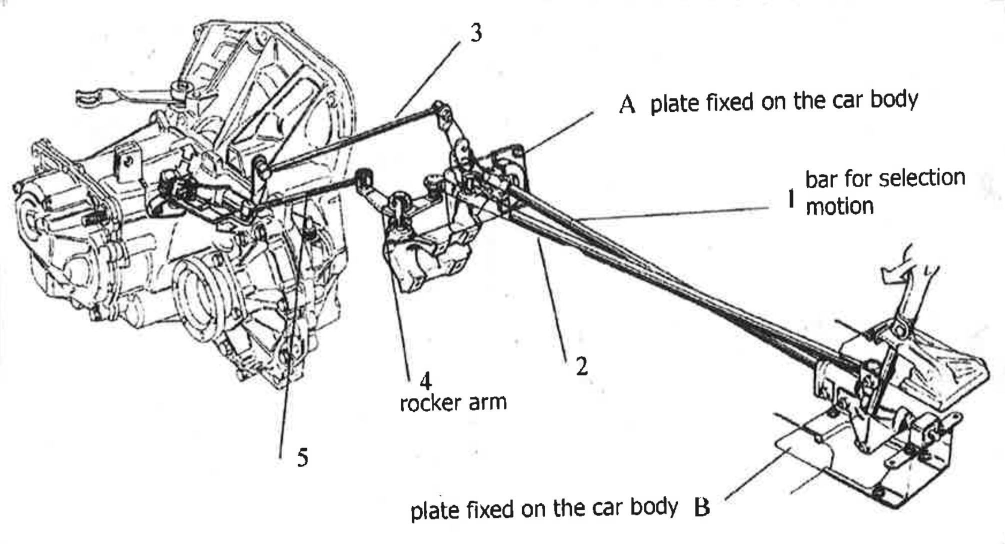


• EXTERNAL SHIFTING MECHANISM

- FUNCTIONS

- MAIN FUNCTIONS OF THE EXTERNAL SHIFTING MECHANISMS ARE:
  - TO CONVEY THE MOTION OF THE LEVER IN A DIFFERENT DIRECTION; FOR EXAMPLE, IN A FRONT WHEEL DRIVEN CAR WITH TRANSVERSAL ENGINE THE ENGAGEMENT MOTION IN THE GEARBOX IS TRANSVERSAL, WHILE THE CORRESPONDING MOTION OF THE SHIFT STIFT IS LONGITUDINAL, WHEN THE LEVER IS ON THE TUNNEL OR ON THE DASHBOARD, OR ALMOST VERTICAL, WHEN THE LEVER IS ON THE STEERING WHEEL SHAFT;
  - TO MAINTAIN THE ENGAGEMENT POSITION OF THE SHIFT STICK UNCHANGED, WITH REFERENCE TO THE NEUTRAL, EVEN IF THE POWERTRAIN IS MOVING, BECAUSE OF THE VEHICLE VERTICAL ACCELERATION OR BECAUSE THE TORQUE VARIATION; THE PROBLEM IS PARTICULARLY IMPORTANT, BECAUSE ON THE FRONT WHEEL DRIVEN CARS THE POWERTRAIN IS REACTING ALSO TO THE WHEEL TRACTION FORCE.

- BAR SHIFTING MECHANISM



- **SHIFTING MECHANISM ATTRIBUTES**

- CONTENT

- THE SHIFTING MECHANISM MUST GUARANTEE A PRECISE AND POSITIVE FEELING ON THE LEVER WITH A SMOOTH SHIFTING EFFORT.
  - PRECISION : CAPACITY TO MANTAIN UNCHANGED THE ENGAGEMENT POSITIONS OF THE SHIFT STICK, IN ANY WORKING CONDITION OR, AT LEAST, UNCHANGED WITH REFERENCE TO THE NEUTRAL POSITION.
  - POSITIVITY : CAPACITY TO REACT TO THE DRIVERS HAND IN A CONSISTENT WAY; DRIVERS APPRECIATE LIMITED EFFORTS FOR THE SELECTING MOTION AND FOR THE FIRST PART OF THE ENGAGEMENT STROKE; HIGHER EFFORTS ARE ACCEPTED AND EXPECTED AT THE END OF THE ENGAGEMENT STROKE, BUT THEY MUST QUICKLY VANISH, WHEN THE GEAR HAS BEEN ENGAGED.
  - SMOOTHNESS : CAPACITY TO LIMIT THE VARIATION OF THE REACTION FORCE WITH REFERENCE TO AN IDEAL ARCHETYPE; THE REACTION FORCE MUST BE NOT ONLY SMALL BUT MUST ALSO SHOW SMALL VARIATION IN DIFFERENT MANEUVERS; THE OPPOSITE OF SMOOTH IS A SHIFT STICK STICKING AND SLIPPING. ("CHE SI ATTACCA E SCIVOLA").

• INTRODUCTION

- CONTENT

1 • GEARBOX AUTOMATION REQUIRES USING DEVICES THAT CAN START-UP THE VEHICLE AND CHANGE SPEED SMOOTHLY, WITHOUT THE HELP OF DRIVER FEELING; FOR THIS REASON ALTERNATIVE DEVICES TO THE CONVENTIONAL CLUTCH WERE DEVELOPED IN THE PAST FOR AUTOMATIC TRANSMISSIONS, ALSO BY USING INTERMEDIATE NON MECHANIC ENERGY, SUCH AS ELECTRIC OR HYDRAULIC ONE.

2 • THE HYDRAULIC TORQUE CONVERTER AND THE ELECTROMAGNETIC POWDER CLUTCH (THIS LAST ONE ON VERY FEW APPLICATIONS) ARE USED WITH TRADITIONAL AUTOMATIC GEARBOXES (AT), FOR THEIR INTRINSIC COMFORT CHARACTERISTICS.
THEY CAN ONLY BE MATCHED WITH GEARBOXES WHERE SPEEDS CAN BE CHANGED WITHOUT TORQUE INTERRUPTION; THE VERY HIGH ROTATING MASSES OF THESE DEVICES WILL CAUSE A CONVENTIONAL SYNCHRONIZER TO WORK WITH A VERY LONG ENGAGEMENT TIME, SO THAT* (INSTEAD OF C. SYNC.) MULTI-DISC WET CLUTCHES ARE USED. (HIGH $T_{cl,t}$) (THE PROBL. IS NOT TO SYNCH. A VERY HIGH INERTIA WITH CONV. S. BUT THE REQ. t)

3 • THE TORQUE CONVERTER IS A PARTICULAR HYDRAULIC MACHINE THAT ALLOWS CONNECTING TWO SHAFTS WITH A CONTINUOUSLY VARIABLE TRANSMISSION RATIO.

4 • OPPOSITE TO FRICTION CLUTCHES, THE INPUT (ENGINE) AND THE OUTPUT (TRANSMISSION) TORQUE VALUES OF THE TORQUE CONVERTER ARE NOT BOUND TO BE EQUAL, BUT THEY ARE DETERMINED BY MORE COMPLEX RELATIONSHIPS DEPENDING ON THE TRANSMISSION RATIO.

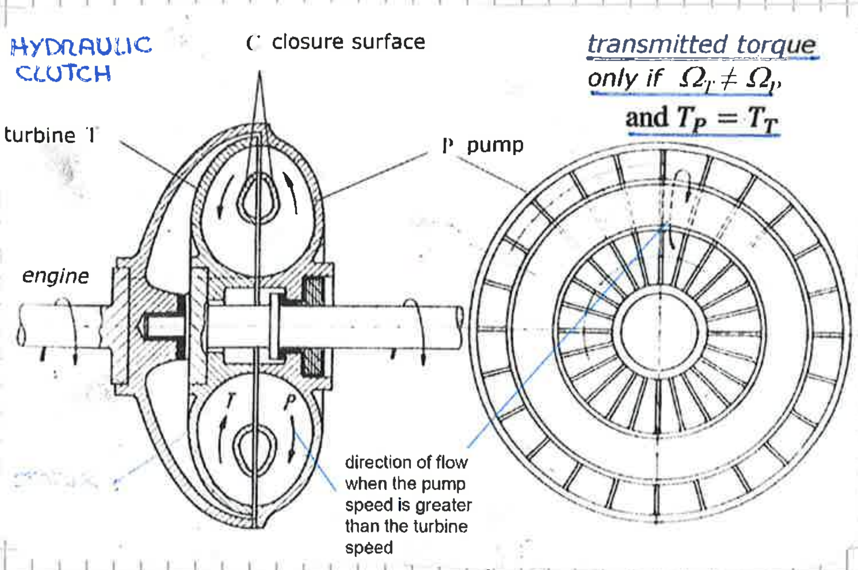
• TODAY OTHER NEW SOLUTIONS, BASED ON MECHANICAL CLUTCHES AND ELECTRONICALLY CONTROLLED ELECTRO-HYDRAULIC (OR ELECTRIC) ACTUATORS, HAVE BEEN DEVELOPED, LIKE IN THE CASE OF DCTs. (DCT NOT USED WITH H. T. CONVERTER)

• THE MOST IMPORTANT SOLUTIONS RECENTLY DEVELOPED ARE:
- AUTOMATED CLUTCHES
- AUTOMATED CLUTCHES TO BE COUPLED WITH AUTOMATED (ROBOTISED) MANUAL T. (AMT)
- DOUBLE CLUTCH UNITS (FOR DCTs)

• THE TWO FIRST TYPES OF CLUTCHES ARE MECHANICALLY SIMILAR TO A CONVENTIONAL CLUTCH, BUT WITH AN ELECTRONIC CONTROL AND AN ELECTRO-HYDRAULIC (OR ELECTRIC) ACTUATION; THE LAST TYPE IS ALSO CHARACTERISED BY A SPECIFIC ARCHITECTURE OF THE MECHANICAL CLUTCH.

IN MT THE AIM OF THE CLUTCH OPENING DURING THE SYNCHR. PHASE IS THE REDUCTION OF THE EQUIVALENT INERTIA THAT MUST BE SYNCHRONIZED.

- HYDRAULIC CLUTCH : SCHEME



1 - HERMANN FÖTTINGER HAD THE IDEA OF INTEGRATING THE PUMP AND THE TURBINE IN A SINGLE COMPACT MACHINE, AVOIDING CONNECTING PIPES.

2 - TURBINE AND PUMP ARE ALMOST IDENTICAL (THEY ARE MIRRORED)

3 - RADIAL CIRCULAR FLOW FROM THE CENTER TO THE PERIPHERY OF THE PUMP (THANKS TO CENTRIFUGAL FORCE) THEN IT ENTERS IN THE TURBINE

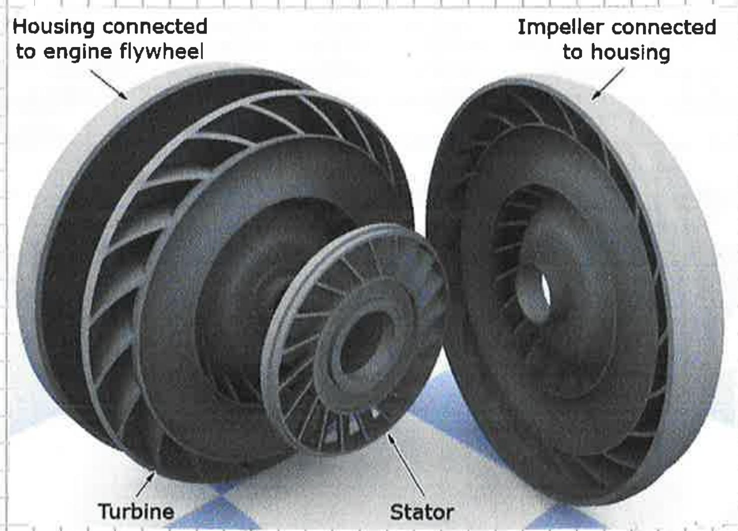
- TORQUE CONVERTER : MAIN COMPONENTS

IMPELLER / PUMP → ENGINE

TURBINE → TRANSMISSION INPUT SHAFT

STATOR → TRANSMISSION HOUSING

IN STEADY-STATE : $T_P + T_S = T_T$ → T_T MAY BECOME MUCH GREATER THAN T_P (MAIN ADV. OF ADDING THE STATOR!)

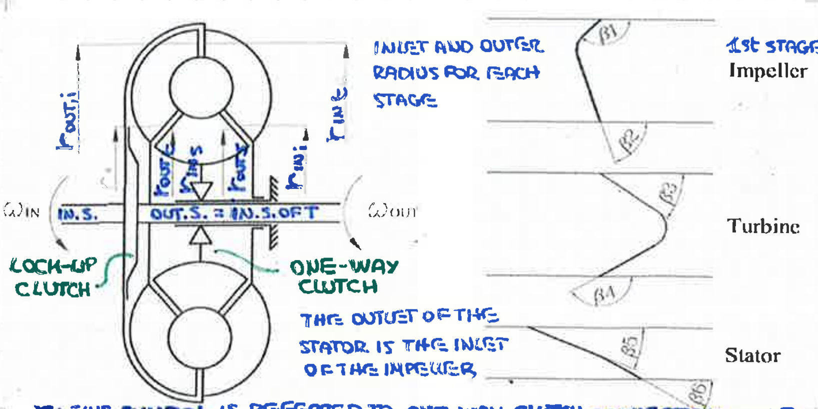


4 - THE TORQUE CONVERTER, WRT. H. CLUTCH, ADD A 3rd BLADED WHEEL: STATOR/REACTOR

5 - THE STATOR IS CONNECTED TO A STANDING EL, SUCH AS THE G. HOUSING.

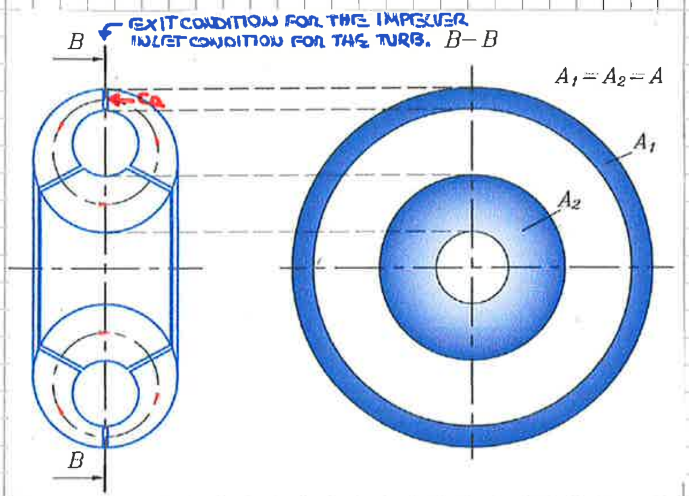
6 - TC IS AN HYDRAULIC TURBOMACHINE COMPOSED BY 3 STAGES : IMPELLER, TURBINE, STATOR

- TORQUE CONVERTER : BASICS



T : THIS SYMBOL IS REFERRED TO ONE-WAY CLUTCH CONNECTION CONF.

- IF WE WANT TO ANALYSE THE BEHAVIOUR OF T.C. THE PARAMETERS TO TAKE INTO ACCOUNT TO BETTER MODEL THE SYSTEM ARE R_{IN}, R_{OUT} AND BLADE ANGLES $\beta_{in,out}$ OF IMP, TURB, STATOR (T.R. IS AN HYDRAULIC TURBOMACHINE)
- A ONE-WAY CLUTCH IS USED TO LOCK THE ANGULAR POSITION OF THE STATOR (3rd STAGE)
- WHEN IT IS OPEN $(SR \geq 0.9)$: T.C. → H.C. (MIN. INT.)



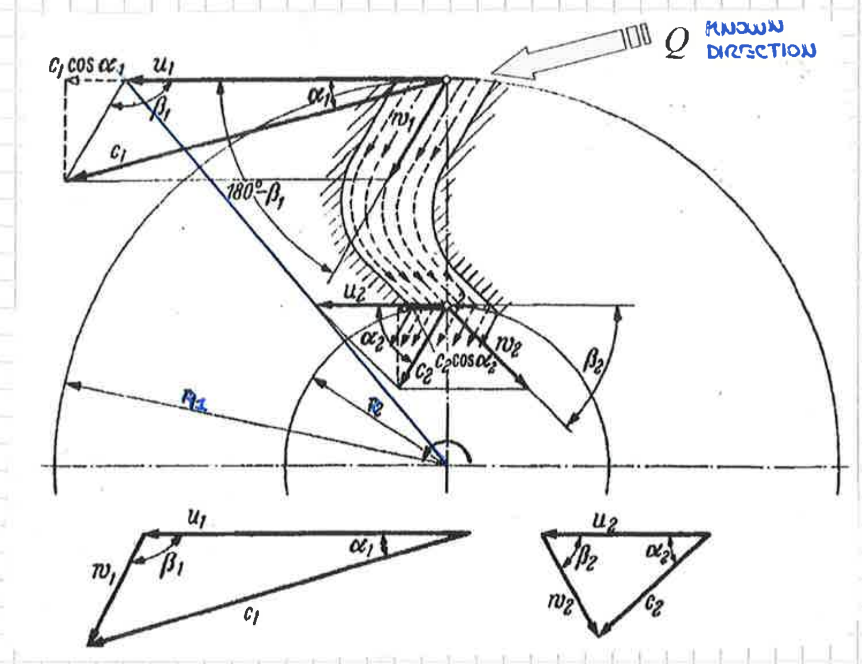
TORQUE CONVERTER GEOM. PARAMETERS:

$\dot{m} = \rho_1 A_1 C_{a1} = \rho_2 A_2 C_{a2}$: FLOW RATE OF FLUID

WITH $C_{a2} \cong C_{a1} \cong C_a$ $(C_a \perp A)$

- ANNULAR VELOCITY OF THE FLUID IS CONSTANT IN THE TORQUE CONVERTER (C_a)

- SPEED TRIANGLES ON A BLADED WHEEL INVESTED BY A NOZZLE



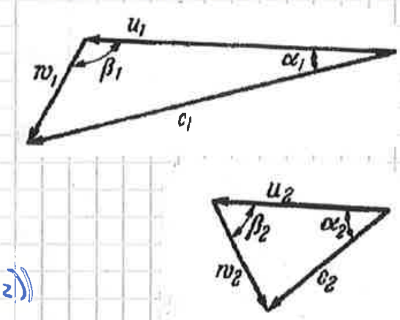
Q : MASS FLOW RATE ; β : BLADE INCLINATION ANGLE ; α : INCLINATION ANGLE OF ABS. V. WRT TANG. DIR.

ASSUMPTION : SINGLE, ALMOST RADIAL, CHANNEL

$W_2 A_2 = W_1 A_1 \Rightarrow W_2 = W_1 A_1 / A_2$ (FROM THE CONTINUITY LAW OF MASS FLOW RATE)

$U_2/R_2 = U_1/R_1 \Rightarrow U_2 = U_1 R_2 / R_1$ (BECAUSE THE CHANNEL IS A RIGID BODY)

WE CAN DRAW THE SPEED TRIANGLES



- ROTATING BLADED WHEEL EQUATIONS

- ANGULAR MOMENTUM AT WHEEL INTAKE (M_1)

$M_1 = Q\, C_1 \cos\alpha_1 R_1$ ($M_1 = \dot{m}\, C_{1u} R_1$)

- ANGULAR MOMENTUM AT WHEEL EXIT (M_2)

$M_2 = Q\, C_2 \cos\alpha_2 R_2$ ($M_2 = \dot{m}\, C_{2u} R_2$)

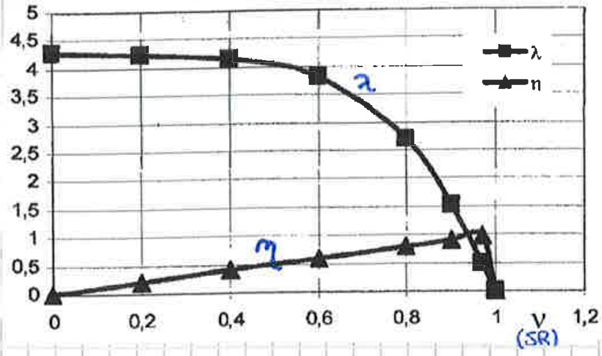
- TORQUE ON THE WHEEL (M) ($M = \dot{m}(C_{1u} R_1 - C_{2u} R_2)$)

$M = M_1 - M_2 = Q (C_1 \cos\alpha_1 R_1 - C_2 \cos\alpha_2 R_2)$

C_{1u}, C_{2u} : PROJECTION ALONG THE TANGENTIAL DIRECTION OF THE ABSOLUTE SPEED (C) AT INLET (C_1) AND OUTLET (C_2).

- CHARACTERISTIC CURVES

- HYDRAULIC CLUTCH (THE STATOR IS FREE TO ROTATE; TR = 1)



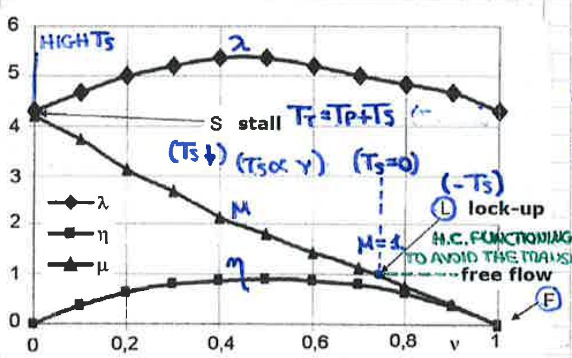
(λ IS MULTIPLIED BY 10.000)

$\mu = TR = 1$ $\quad \eta = \mu \nu \Rightarrow \eta \uparrow \propto \nu \uparrow$

THEN, WHEN WE ARE REACHING $\nu \sim 1$, $\eta \downarrow$ BECAUSE WE ARE ABLE TO GENERATE T ONLY WHEN THERE IS RELATIVE SPEED ($\nu = \omega_{OUT}/\omega_{IN} \neq 1$); HIGHER PERF. COEFF λ AT LOW ν (HIGH $\Delta\omega$) ($\nu \uparrow$ $\lambda \downarrow$)

- THE PERFORMANCE COEFFICIENT (λ) IS DECREASING WITH THE SPEED TRANSM. RATIO (ν) BECAUSE THE PROCESSED TORQUE IS DEPENDING UPON THE FLOW, DETERMINED BY THE SPEED DIFFERENCE OF THE WHEELS; WHEN THE WHEELS ARE SYNCHRONOUS, $\nu = 1$ AND THE TORQUE IS ZERO. THE EFFICIENCY SHOULD BE ALWAYS EQUAL TO THE SPEED TRANSMISSION RATIO BUT AT $\nu = 1$. AT THIS POINT η GOES ALSO TO ZERO.

- TORQUE CONVERTER (THE STATOR IS LOCKED)



STALL: TURBINE STILL, PUMPING ROTATING ($\nu = 0$)

LOCK-UP: PUMP AND TURBINE TORQUE EQUAL ($\mu = 1$)

FREE FLOW: NO TURBINE TORQUE ($\mu = 0$)

(λ IS MULTIPLIED BY 10.000) λ, η SYMM. BEHAV. WRT THEIR MAX

$\eta = \nu\mu$ $\quad$ DUE TO THE PRESENCE OF BLOCKED STATOR: $\mu \neq CONST$

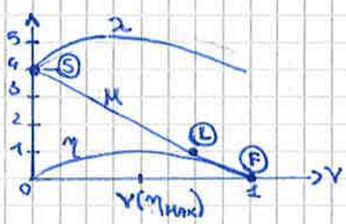
$\nu \downarrow : \mu \uparrow$; η: PARABOLA TREND; η_{MAX} FOR MID VALUE OF ν ($\simeq 0.5$)

- SINCE THERE IS A REACTION ELEMENT, TORQUE TRANSMISSION RATIO ISN'T CONSTANT; ($TR = \mu \neq M_2/M_1 = T_T/T_i \neq CONST$) THE TORQUE ABSORBED BY THE PUMP DOESN'T CHANGE VERY MUCH AS THE SPEED TRANSMISSION RATIO ($SR = \nu$) INCREASES, BECAUSE OF THE REACTOR WHEEL. TR STARTS FROM A VALUE OF $3 \div 4$, WHEN $SR = 0$ AND THE OUTPUT SHAFT IS STALLING, AND GOES TO ZERO, WHEN THE INPUT AND OUTPUT SHAFTS ARE SYNCHRONOUS ($SR = 1$). AS A CONSEQUENCE, THE EFFICIENCY $\eta = \nu\mu$ WILL BE ZERO AT $\nu = 0$ AND $\nu = 1$ WHERE $\mu = 0$.

S $\nu = 0 : \mu = 4$; $\eta = 0$: STALL CONDITION; T AMPLIFICATION (HIGH T_S)

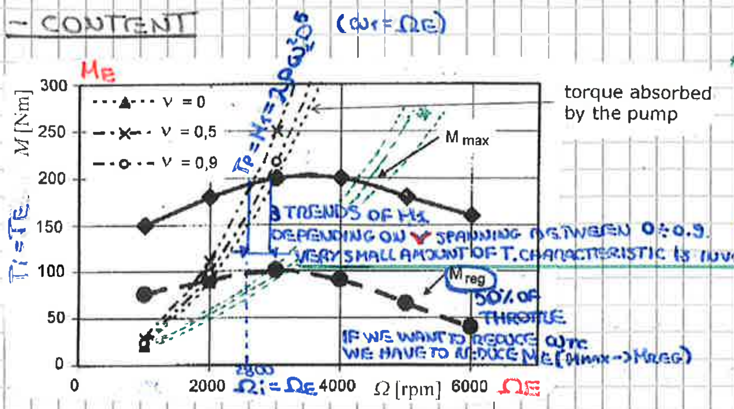
L $\nu \to 1 : \mu = 1$; $\eta \simeq 0{,}8$; LOCK-UP CONDITION ($\omega_{OUT} < \omega_{IN}$)

F $\nu = 1 : \mu = 0$; $\eta = 0$: FREE-FLOW CONDITION ($\omega_{OUT} = \omega_{IN}$)



- TORQUE CONVERTER PERFORMANCE ON A VEHICLE

– CONTENT ($\omega_i = \Omega_E$)



*IF WE WANT TO MAKE T.C. ROTATING FASTER WE CAN USE A TC CHARACTERIZED BY A SMALLER DIMENSION ($M_1 = \lambda \rho \omega^2 D^5 \Rightarrow \omega = \sqrt{M_1/\lambda\rho D^5}$)

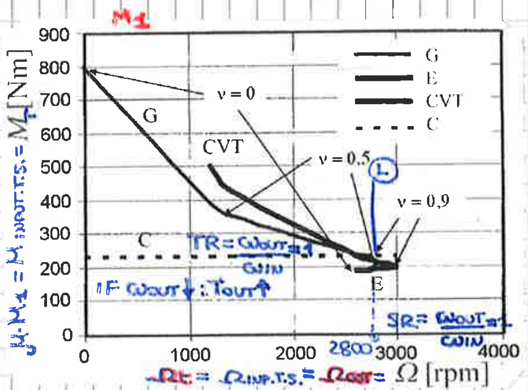
$\Rightarrow$ WE CAN CHANGE OPERATING POINT OF THE ENGINE CONSIDERING ALSO THE T.C. SIZING. (1st OBSERVATION)

WE CAN CHANGE OPERATING POINT ADJUSTING THE TORQUE DELIVERED BY THE ENGINE (2nd OBSERVATION)

- AS A VALUE FOR THE SPEED TRANSMISSION RATIO IS ASSUMED, FOR THE EQUILIBRIUM OF THE PUMP AND ENGINE SYSTEM, ENGINE SPEED WILL BE FORCED AT THE VALUE AT THE INTERSECTION POINT OF THE ENGINE T (M_E) AND OF THE PUMP T (M_1) DIAGRAMS.

IN ORDER TO CHANGE THE MATCHING POINT, ENGINE T (M_E) MUST BE CHANGED.

CALCULATION OF THE GEARBOX INPUT TORQUE (CURVE G) STARTING FROM THE ENGINE TORQUE (CURVE E):



IF WE TAKE THE INTERSECTION BETWEEN M_1 ($\propto \nu$) AND ME TRENDS WE FIND M_1 (THE TORQUE ABSORBED BY THE IMP. THEN IF WE ENTER IN THE CHARACT. DIAGRAM OF T.C. WE CAN EVALUATE THE TORQUE AMPLIFICATION THROUGH THE SPEED REG OF T.C.: IF WE REDUCE THE TURBINE SPEED, THE DELIVERED (OUTPUT) TORQUE INCREASES.

WE CHOOSE A CERTAIN TYPE OF T.C.

THE ENGINE W.P.: $\omega_E, T_E \Rightarrow \omega_i = \omega_E ; T_i = T_E$ | $\mu = \mu(\nu)$

STALL COND: $\nu = \omega_t/\omega_i = 0 \Rightarrow \omega_t = 0$, $\mu = \mu_{MAX} \Rightarrow T_t = T_{t\,MAX}$ | $\omega_t \uparrow : \nu \uparrow \; \mu \downarrow : T_t \downarrow$

THERE ARE PRESENT ALSO THE CURVES OF E, CVT, G.

E: DIRECT CONNECTION WITH THE INPUT TRANSMISSION SHAFT.

CVT: TRASMISSION ABLE TO CONTINUOUSLY CHANGE $\tau = \omega_{OUT}/\omega_{IN}$; IT IS LIMITED BETWEEN 2 VALUES WE CANNOT REACH ZERO OUTPUT SPEED AS THE TC (BECAUSE TC HAS THE CAPABILITY TO MANAGE ALSO $\nu = 0$); IT HAS A MINIMUM SPEED RATIO τ BELOW WHICH IT IS NOT POSSIBLE TO GO.

C: CLUTCH $T_{CL} = 1.15\, T_{E\,MAX}$ (HERE $T_{E\,MAX} = 200$ [Nm]); IT IS ABLE, THANKS TO THE CLUTCH SLIP, TO MANAGE ALSO THE ZERO OUTPUT SPEED OF THE CLUTCH IN ORDER TO MANAGE THE START-UP OF THE VEHICLE.

T17 AUTOMATIC TRANSMISSIONS (PART 1)

- **GENERAL ISSUES**
  - CONTENT
  - AUTOMATION LEVEL
  - GEARSHIFT MODE
  - STEPPED AND CONTINUOUSLY VARIABLE TRANSMISSIONS
  - AUTOMATIC GEARBOX CLASSIFICATION
- **CAR TRANSMISSIONS WITH FIXED ROTATION AXIS**
  - CONTENT
- **ROBOTIZED OR AUTOMATED TRANSMISSIONS (AMT)**
  - CONTENT
  - ELECTRO-HYDRAULICALLY ACTUATED SHIFTING MECHANISM
  - HYDRAULIC ACTUATION GROUP
  - ACTUATOR PISTON WITH THREE STABLE POSITIONS
  - HYDRAULIC REGULATION OF THE ACTUATION SYSTEM
  - S-CAM ACTUATOR
  - WORKING DETAILS OF AN S-CAM ACTUATOR
- **SEMIAUTOMATIC TRANSMISSIONS**
  - CONTENT
  - OLD EXAMPLE
- **MULTI DISC CLUTCH GEARBOXES**
  - CONTENT

[TEXTBOOK: VOL. I, PAR. 15.1, 15.2]

1st - AUTOMATION LEVEL (ONE WAY TO CLASSIFY T.)

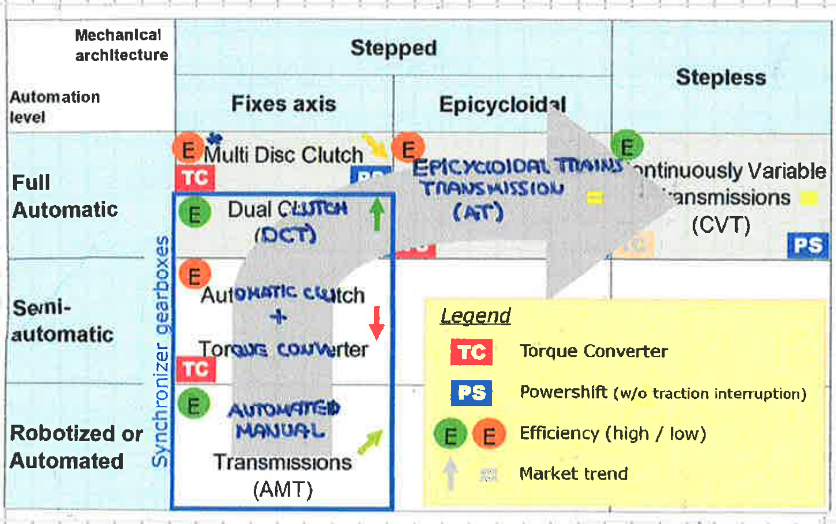
- FULL AUTOMATIC: THE GEARBOX CONTROL SYSTEM HAS FULL AUTHORITY ON SPEED SELECTION, SHIFTING SEQUENCE AND START-UP FUNCTIONS; THEY ALSO HAVE A SEMIAUTOMATIC MODE (MANUAL SELECTION). (AT
- SEMI AUTOMATIC: SPEED SELECTION AND/OR START-UP ARE MANUAL; THIS KIND OF GEARBOX HAS LOST INTEREST ON CARS, BECAUSE IT DOESN'T REPRESENT A FAVORABLE TRADE-OFF BETWEEN COST AND BENEFIT. IT IS ADOPTED ON SOME INDUSTRIAL VEHICLE, BECAUSE OF THE RELEVANT EFFORT OF MOVING MANUAL COMMANDS. (DCT, CVT ?
- AUTOMATED OR ROBOTIZED: MECHANISMS ARE DERIVED FROM MANUAL GEARBOXES, INCLUDING CLUTCH, BY THE APPLICATION OF CONTROLLED ACTUATORS; THEY HAVE AUTOMATIC AND SEMI AUTOMATIC MODES. THE TARGET OF THESE GEARBOXES IS TO REDUCE PRODUCTION AND OPERATION COST, IF COMPARED TO CLASSICAL AT, BUT THEY HAVE ALSO A POSITIVE IMPACT ON IMPROVING SPORT PERFORMANCE (FOR THIS REASON THEY WERE ADOPTED ON RACE CARS). (AMT

2nd - GEARSHIFT MODE (OTHER WAY TO CLASSIFY T.)

- A VERY IMPORTANT ASPECT CONDITIONING AUTOMATIC GEARBOX CONFIGURATION IS THE GEARSHIFT MODE; WE USUALLY DISTINGUISH BETWEEN GEAR SHIFTS WITH POWER INTERRUPTION AND GEAR SHIFTS WITHOUT POWER INTERRUPTION (OR POWERSHIFT) (
- IN THE FIRST CASE GEARS ARE SHIFTED AS IN A MANUAL GEARBOX USING SYNCHRONIZED OR NON SYNCHRONIZED DOG CLUTCHES; DURING GEAR SHIFT THE POWER FLOW GOING THROUGH THE GEARBOX MUST BE INTERRUPTED TO ALLOW THE DISENGAGEMENT OF THE EXISTING SPEED AND THE ENGAGEMENT OF THE NEXT ONE.
- THE POWER AVAILABLE TO MOVE THE VEHICLE GOES TO ZERO DURING THIS MANEUVRE AND THE VEHICLE SLOWS DOWN: THE JERK IS SIGNIFICANT AND IS PERCEIVED VERY WELL OVER CERTAIN DURATION VALUES, BUT ALSO THE VEHICLE PERFORMANCES ARE REDUCED.
- AMT ARE NO-POWERSHIFT GEARBOXES. (AMT, AT ?
- ON CVT, SHIFTING MANEUVRES ARE, BY DEFINITION, POWERSHIFT TYPE, IN CONSIDERATION OF THE VERY SMALL DIFFERENCE BETWEEN PREVIOUS AND NEXT TRANSMISSION RATIOS. (CVT, DCT ?

- AUTOMATIC GEARBOX CLASSIFICATION

4th
- FROM THE ARCHITECTURE POINT OF VIEW, GEAR TRAINS USED ON **STEPPED AUTOMATIC GEARBOXES** CAN BE CLASSIFIED ACCORDING TO TWO CATEGORIES:
  - **GEAR WHEELS** WITH **FIXED ROTATION AXIS**, SIMILAR TO THOSE ADOPTED ON MANUAL GEARBOX; IT IS THE SOLUTION ADOPTED FOR DCT (
  - **EPICYCLOIDAL GEAR TRAINS** OF DIFFERENT CONFIGURATION THAT IN ASSOCIATION **WITH BAND BRAKES** OR **MULTI DISK CLUTCHES** CAN REALIZE DIFFERENT RATIOS INCLUDING REVERSE SPEED, **FEATURING ALWAYS COAXIAL INPUT** AND **OUTPUT SHAFTS**. (AT
- IN GENERAL, AUTOMATIC TRANSMISSIONS CAN BE THEN CLASSIFIED BY:
  - 1st - **AUTOMATION LEVEL**: FULL AUTOMATIC, SEMIAUTOMATIC, AUTOMATED.
  - 2nd - **GEARSHIFT MODE**: WITH TORQUE INTERRUPTION OR POWERSHIFT.
  - 3rd - **TRANSMISSION RATIOS AVAILABILITY**: STEPPED OR STEPLESS (CVT).
  - 4th - **GEAR TRAINS TYPE**: FIXED ROTATION AXIS OR EPICYCLOIDAL.
  - 5th - **START-UP DEVICE**: FRICTION CLUTCH OR TORQUE CONVERTER



SEMI-AUTOMATIC:
AUTOMATIC CLUTCH + TC (SEMI-A; NON-P; STEPPED; FIX R. AXIS; TC; REDUCED; SYN)

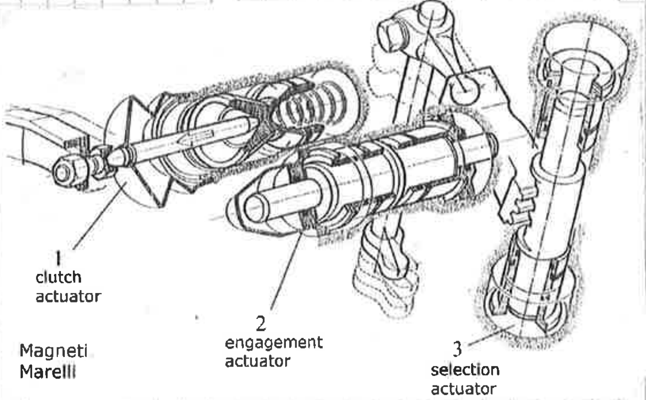
MULTI DISK CLUTCH:
ORDINARY FIX R. AXIS GEAR TRAIN + MULTI DISK CLUTCHES INSTEAD OF SYNCHRONIZERS: IN THIS WAY WE INCREASE THE TORQUE CAPACITY OF THE SYNCHR. COMPONENTS
(FULL AUT; POWERSHIFT; STEPPED; FIX RA)
(TORQUE CONVERTER; REDUCED; H.D.C.)

ACCORDING TO THE MARKET TREND:

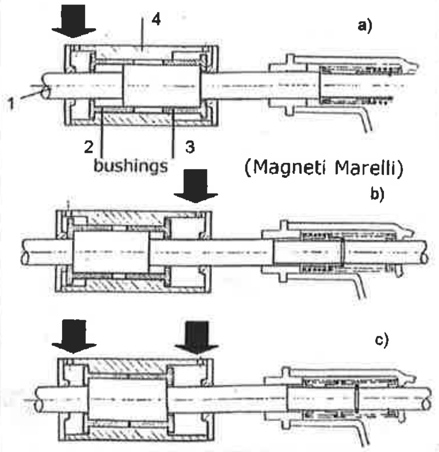
	1st AUTOMATION LEVEL	2nd GEARSHIFT M.	3rd n OF τ	4th GEARS	5th START-UP DEVICE	η	G. ENGAG.
AMT	AUTOMATED / ROB.	NON-POWERSHIFT	STEPPED	FIX. R. AXES	FRICTION CLUTCH	H	SYNCH.
DCT	FULL AUTOMATIC	POWERSHIFT	STEPPED	FIX. R. AXES	FRICTION CLUTCH	H	SYNCH.
AT	FULL AUTOMATIC	POWERSHIFT	STEPPED	/FIX R. AXES EPIC. G. TRAINS	TORQUE CONVERTER	R	MULTI-D. CL.
CVT	FULL AUTOMATIC	POWERSHIFT	STEPLESS	FIX. R. AXES	F. CL. OR T.C.	H	E/H. ACTUAT.

AMT: ARCHITECTURE DERIVED FROM MT; CONTROL SYSTEM → ELECTRIC / ELECTRO-HYDRAULIC SERVOS.

- HYDRAULIC ACTUATION GROUP



- ACTUATOR PISTON WITH THREE STABLE POSITIONS



RIGHT STABLE POSITION

(DESIGN FEATURE WHICH ALLOWS AN INTERMEDIATE STABLE PISTON POSITION TO BE OBTAINED)

LEFT STABLE POSITION

INTERMEDIATE STABLE POSITION

- HYDRAULIC REGULATION OF THE ACTUATION SYSTEM

- IN THE GENERATOR, REGULATOR AND DISTRIBUTOR GROUP THERE IS A PRESSURE ACCUMULATOR THAT REGULATES PRESSURE AT A VALUE ALMOST INDEPENDENT ON THE ACTUAL FLOW RATE REQUIRED BY ACTUATORS. TO OBTAIN THIS, THE PUMP IS DRIVEN ON-OFF BY AN ELECTRIC MOTOR, CONTROLLED BY A PRESSURE SENSOR. IN THE ACCUMULATOR, OIL PRESSURE IS AVAILABLE ALSO WHEN MOTOR IS STOPPED; IT IS POSSIBLE TO DESIGN THE PUMP ACCORDING TO THE AVERAGED REQUIRED FLOW, INSTEAD THAN ON PEAK VALUES.
- THREE DIFFERENT ELECTROVALVES ARE ALSO PRESENT IN THE GROUP:
  - ONE, PROPORTIONAL TYPE, FOR CLUTCH ACTUATION;
  - TWO, PROPORTIONAL TYPE, FOR ENGAGEMENT MECHANISM;
  - TWO, ON-OFF TYPE, FOR SELECTION MECHANISM.
- VALVE TYPE CHOICE FOR ENGAGEMENT MECHANISM IS DECIDED BY THE NEED OF CONTROLLING FORCE IN ORDER TO OBTAIN HIGH ACTUATION SPEED, BUT IN THE MEAN TIME NOT TO DAMAGE SYNCHRONIZERS.
- THE SHIFT CONTROL SYSTEM ALSO INCLUDES: AN AXIAL POSITION SENSOR (ENGAGEMENT STROKE) AND AN ANGULAR POSITION SENSOR (SELECTION STROKE). SELECTED GEARS CAN BE IDENTIFIED THROUGH THESE SENSORS.

• SEMIAUTOMATIC TRANSMISSIONS

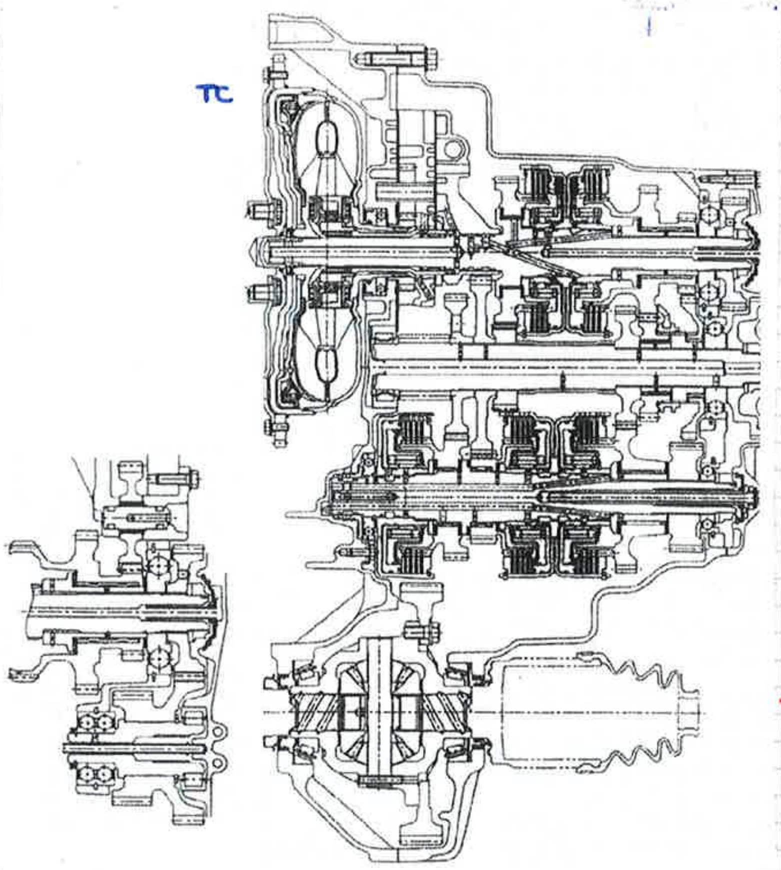
- CONTENT

- THESE GEARBOXES WITH SYNCHRONIZERS WERE NORMALLY DERIVED FROM MANUAL GEARBOXES, LIKE THE ROBOTIZED ONES.
- MANUAL GEARBOXES WITH TORQUE CONVERTER WERE DEVELOPED: THE TORQUE CONVERTER WAS COUPLED TO A CLUTCH, WORKING AS AN AUTOMATIC CVT FROM STALL TO LOCK-UP.
- THE CLUTCH, WITH ELECTROMAGNETIC ACTUATOR, WAS OPEN AUTOMATICALLY AT ANY SPEED CHANGE. SELECTION AND ENGAGEMENT MANOEUVRES WERE MANUAL, ACTING ON THE SHIFT STICK; THE ELECTROMAGNET OF THE CLUTCH WAS SWITCHED BY A SENSOR, ABLE TO MEASURE THE EFFORT APPLIED TO THE SHIFT STICK KNOB.
- THIS KIND OF SEMIAUTOMATIC GEARBOXES ELIMINATED THE CLUTCH PEDAL ONLY, BEING SHIFT STICK STILL MANUAL. THEY HAVE A VERY LIMITED DIFFUSION ON TODAY CARS, HAVING: LOW AUTOMATION LEVEL, LOW EFFICIENCY AND RELATIVELY HIGH COST.

• MULTI DISC CLUTCH GEARBOXES

- CONTENT

- THE CONFIGURATION OF THIS AUTOMATIC GEARBOX KIND IS STILL SIMILAR TO ONE OF A MANUAL GEARBOX.
- SYNCHRONIZERS ONLY ARE SUBSTITUTED BY MULTI DISC WET CLUTCHES AND SPEED SHIFT CAN BE POWER-SHIFT TYPE.
- REVERSE SPEED TOO IS MADE WITH AN IDLER, IN A SIMILAR WAY TO A MANUAL GEARBOX.
- WITH THIS KIND OF GEARBOX THERE ARE NO LIMITS, AS WITH EPICYCLOIDAL ONES, TO CHOOSE THE DESIRED TRANSMISSION RATIOS.



- FULL AUTOMATIC
- POWERSHIFT
- STEPPED
- FIXED ROTATIONAL AXES (CYL. HEL. GEARS)
- TC (AS START-UP DEV.)
- REDUCED EFF.
- MULTI-DISK WET CLUTCHES (HIGH TORQUE CAP.)
  - ↳ WE DON'T NEED TO UNLOAD THE INPUT SHAFT DURING SYNCH. PROCESS
- NOT SO USED ARCHITECTURE

COUNTERSHAFT POWERSHIFT GEARBOX WITH MULTI DISC WET CLUTCHES (HONDA).

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 231 di 260

• CAR EPICYCLOIDAL (EPICYCLIC / PLANETARY) CAR TRANSMISSIONS

– CONTENT

- IN THIS MODULE SOME OF THE MOST IMPORTANT GEAR TRAINS USED ON AUTOMATIC GEARBOXES ARE EXAMINED, BUT THE CONFIGURATIONS IN USE FOR THE PRODUCTION AUTOMATIC TRANSMISSIONS ARE MANY MORE.
- THESE **TRAINS** ARE **ALWAYS MADE** WITH **CYLINDRICAL GEARS**; **IN THE PAST SOME EXAMPLE** WITH **BEVEL GEARS** WAS AVAILABLE. (USUALLY CYL. GEARS)
- IN THE NEXT SLIDES EPICYCLOIDAL GEAR TRAINS OF INCREASING COMPLEXITY WILL BE EXAMINED (BEFORE SHOWING SOME EXAMPLE OF CAR APPLICATIONS):

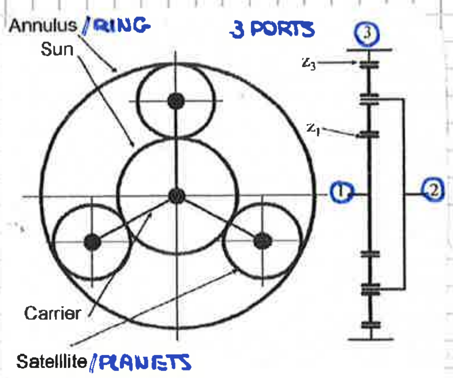
A BASIC EPICYCLOIDAL GEAR TRAIN (BASIC / SIMPLE) (EPIC. / PLANETARY) (TRAIN / SET) (IN SOME CASE DIFF.)

B COMPOUND EPICYCLOIDAL GEAR TRAINS { WILSON (WITHOUT OR WITH REVERSE; 3 F.S.) (2 IDENTICAL E.G.T IN SERIES)
RAVIGNEAUX

C COMPOSITION OF GEAR TRAINS

- THE SIMPLEST CONFIGURATION FOR AN EPICYCLOIDAL GEAR TRAIN IS THE SAME ALREADY PRESENTED FOR GEAR REDUCERS AND TRANSFER BOX DIFFERENTIALS.
- THIS SYSTEM HAS **3 MECHANICAL PORTS** AND SHOWS **2 D.O.F**: IN ORDER TO TRANSMIT A TORQUE, **ONE ELEMENT** MUST BE **FIXED** TO THE **GEARBOX CASING (STATOR)**. (2 D.O.F. WE HAVE TO IMPOSE 2/3 SPEEDS IN ORDER TO COMPLETELY DEFINE KINEMATIC
⇒ 3rd SPEED IS AUTOMATICALLY COMPUTED (ONE OF SPEEDS $\omega = 0 \rightarrow$ STATOR)

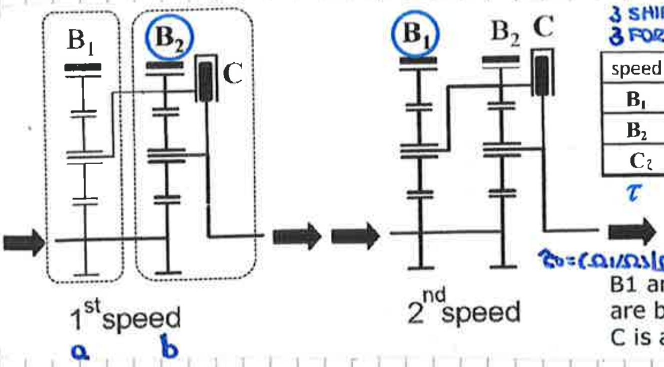
– SCHEME OF A SIMPLE EPICYCLOIDAL GEAR TRAIN



- THE WHEEL TRAIN IS MADE BY AN **INTERNAL TOOTH GEAR (ANNULUS)** AND BY AN **EXTERNAL TOOTH GEAR (SUN)** WITH **CONCENTRIC CENTRE LINES**; THE TWO WHEELS MESH WITH **OTHER WHEELS (SATELLITES)** WHOSE **HUBS** ARE **SUPPORTED BY SHAFTS MOUNTED ON A ROTATING STRUCTURE COAXIAL TO THE SUN**, CALLED **CARRIER**.
- **SATELLITES** ARE **NORMALLY THREE** AND ARE **SET** ON THE CARRIER **AT CONSTANT ANGLES**; **RADIAL COMPONENTS** OF GEARING **FORCES** ARE THEREFORE **SELF EQUILIBRATED**.

E.G.T. : EPICYCLOIDAL GEAR TRAIN
P.G.S. : PLANETARY GEAR SET

- SCHEME OF A WILSON COMPOUND EPICYCLOIDAL GEARTRAIN $\boxed{B}$



3 SHIFTING ELEMENTS
3 FORWARD SPEEDS

speed	I	II	DD
B_1		X	
B_2	X		
C_2			X
τ	3	1.8	1

$\tau_0 = -2$

$\tau_0 = (\Omega_1/\Omega_3|_{\Omega_2=0}) = -\frac{z_3}{z_1} = -2$

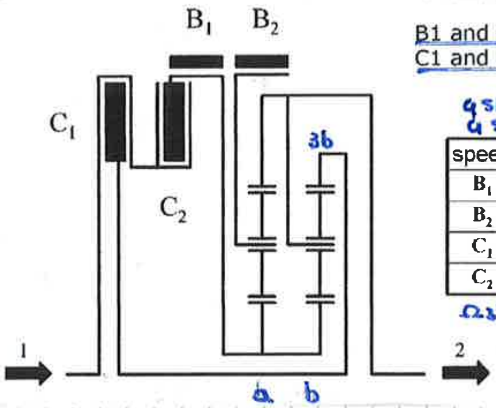
B1 and B2 are band brakes, C is a clutch

- SUN GEARS (1_a, 1_b) ARE FIXED TOGETHER
- CARRIER (2_a) IS FIXED TO RING (3_b)
- A CLUTCH (C) IS USED TO FIX (2_a, 3_b, TO CARRIER (2_b) → DIRECT DRIVE
- 2 BAND BRAKES (B_2, B_1) (ACTUATED BY A CONTR.S) ARE USED TO LOCK ONE OF 2 RINGS (3_b OR 3_a) TO REALIZE ONE SPEED (I, II)

- IN THE WILSON GEAR TRAIN, TWO SIMPLE TRAINS ARE MATCHED IN SUCH A WAY AS THE CARRIER OF THE FIRST TRAIN IS FIXED TO THE ANNULUS OF THE SECOND ONE: BY STOPPING ONE OF THE ANNULUS WHEELS AT A TIME (B1 AND B2), TWO DIFFERENT NOT INVERTED TRANSMISSION RATIOS CAN BE OBTAINED; BY FIXING THE TWO CARRIER TOGETHER (C), A DIRECT DRIVE IS GENERATED.

IF B1 IS OPEN ⟺ IT DOES NOT IMPOSE TORQUE ($T_{Ra}=0$) ⇒ THE WHOLE PL. GEAR SET IS UNABLE TO ABSORB OR TRANSMIT POWER THROUGH IT ⇒ IDLE STATE
(B2 CLOSED → I) (SUN $1a$ CANNOT ABSORB TORQUE)

- SCHEME OF A SECOND WILSON EPICYCLOIDAL GEAR TRAIN $\boxed{B}$ +R



B1 and B2 are band brakes,
C1 and C2 are multi disc clutches

4 SHIFTING ELEMENTS (2 B.B.; 2 CL)
4 SPEEDS (3+1)

speed	I	II	III	R
B_1		X		
B_2	X			X
C_1	X	X	X	
C_2			X	X

B2, R, C2

$\Omega_{3b}=\Omega_{2a} \rightarrow \Omega_{3a}=\Omega_{2b}$

IF B2 IS OPEN: (B1 CLOSED → II)
$\begin{cases} \text{WILLIS F. FOR } a: \Omega_1=(1+6)\Omega_2-6\Omega_3 \\ \text{WILLIS F. FOR } b \\ \Omega_{1a}=\Omega_{1b} \\ \Omega_{2a}=\Omega_{3b} \end{cases} \Rightarrow \tau_{II}=\frac{\Omega_{IN}}{\Omega_{OUT}}=\frac{\Omega_1}{\Omega_{2b}}$

IF C IS CLOSED: WE IMPOSE $\Omega_{2b}=\Omega_{3b}$
(B1, B2 OPEN) THEN $\Omega_{2b} (=\Omega_{3b}) = \Omega_1$ (D.D.)

← IF WE WANT THE REVERSE SPEED WE CHANGE C.

C_1 CONNECTS THE INPUT SHAFTS TO THE RING GEAR OF SECOND SET (3b)
C2 CONNECTS THE INPUT SHAFTS TO THE SUN GEARS OF THE TWO SETS (1a, 1b)
IN THIS CONFIGURATION CARRIER (2b) IS FIXED TO (3a)

IF C2 & B2 ARE CLOSED ⇒ REVERSE SPEED

- THREE FORWARD AND ONE REVERSE SPEED
TWO BAND BRAKES; TWO MULTI DISC WET CLUTCHES.

- THEREFORE, THE **GEAR TRANSMISSION RATIOS** ARE:

$$\tau_1 = 2 + \frac{z_S}{z_A} \quad ; \quad \tau_2 = 1 + \frac{z_S}{z_A} \quad ; \quad \tau_3 = 1 \quad ; \quad \tau_R = -\frac{z_A}{z_S}$$

- IF, FOR EXAMPLE, $\frac{z_S}{z_A} = \frac{1}{3}$ $\Leftrightarrow z_1/z_3 = 1/3 = \sigma^{-1} \Rightarrow \sigma = 3$, THE FIRST SPEED RATIO WILL BE 2.333, THE SECOND SPEED RATIO 1.333, THE THIRD SPEED RATIO 1 AND THE REVERSE -3;
- IF, INSTEAD, $\frac{z_S}{z_A} = \frac{2}{5}$, THE FIRST SPEED RATIO WILL BE 2.4, THE SECOND 1.4, AGAIN 1 FOR THE THIRD AND -2.5 FOR THE REVERSE SPEED.
- RECALLING WHAT WE SAID ABOUT TORQUE CONVERTERS, **OBTAINABLE RATIOS** ARE **SIMILAR TO** THOSE OF A REFERENCE **MANUAL FOUR SPEED GEARBOX** (THE STALL TORQUE CONVERTER TORQUE RATIO CAN BE ABOUT 2.5);
  THE **POSSIBILITY OF OBTAINING A DESIRED SPEED STEP OR A SUITABLE REVERSE SPEED RATIO** IS UNFORTUNATELY **COMPROMISED** $\Rightarrow$ (NON-GEOMETRIC PROGRESSION)

W.F. $\Omega_1 = (1+\sigma)\Omega_2 - \sigma\Omega_3 \Leftrightarrow \frac{\Omega_1 - \Omega_2}{\Omega_3 - \Omega_2} = -\sigma \Leftrightarrow \frac{\Omega - \Omega_{OUTPUT}}{\Omega_{INPUT} - \Omega_{OUTPUT}} = -\frac{z_3}{z_1} \Leftrightarrow \frac{\Omega_{INPUT} - \Omega_{OUTPUT}}{\Omega - \Omega_{OUTPUT}} = -\frac{z_1}{z_3}$

PROPERLY:

$$i = \frac{\Omega_{IN}}{\Omega_{OUT}} \quad ; \quad \sigma = \frac{z_3}{z_1} = 3$$

(NON-GEOMETRIC PROGRESSION) (I $\rightarrow$ II $\rightarrow$ III : $\tau \nearrow$)

I: C_1, B_2 : $i_I = \frac{\Omega_{IN}}{\Omega_{OUT}} = \frac{\Omega_{3b}}{\Omega_{2b}} = 2 + \frac{1}{\sigma} = 2 + \frac{z_1}{z_3} = 2 + \frac{1}{3} = \frac{7}{3} = 2.33 \Rightarrow \tau_I = \frac{\Omega_{OUT}}{\Omega_{IN}} = \frac{1}{i} = \frac{1}{2.33} = 0.43$

II: C_1, B_1 : $i_{II} = \frac{\Omega_{IN}}{\Omega_{OUT}} = \frac{\Omega_{3b}}{\Omega_{2b}} = 1 + \frac{1}{\sigma} = 1 + \frac{z_1}{z_3} = 1 + \frac{1}{3} = \frac{4}{3} = 1.33 \Rightarrow \tau_{II} = \frac{\Omega_{OUT}}{\Omega_{IN}} = \frac{1}{i} = \frac{1}{1.33} \approx 0.75$

III: C_1, C_2 : $i_{III} = \frac{\Omega_{IN}}{\Omega_{OUT}} = \frac{\Omega_{3b}}{\Omega_{2b}} = 1$ (D.D.) $= 1 \Rightarrow \tau = \frac{\Omega_{OUT}}{\Omega_{IN}} = \frac{1}{i} = 1 \Rightarrow \Omega_{OUT} = \Omega_{INPUT}$

R: B_2, C_2 : $i_R = \frac{\Omega_{IN}}{\Omega_{OUT}} = \frac{\Omega_{1b}}{\Omega_{2b}} = -\sigma = -\frac{z_3}{z_1} = -3 \Rightarrow \tau = \frac{\Omega_{OUT}}{\Omega_{IN}} = \frac{1}{i_R} = -\frac{1}{3} = -0.33$: $\Omega_{OUT} = -0.33\,\Omega_{IN}$

NOTE: $|\Omega_{OUT_R}| < |\Omega_{OUT_I}|$

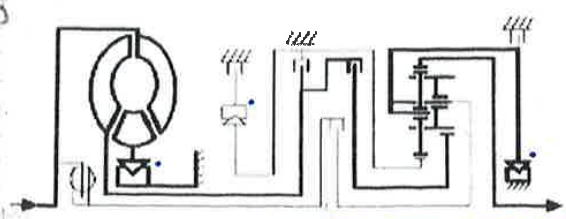
- IT IS **NOT** POSSIBLE TO OBTAIN A GEOMETRIC PROGRESSION BECAUSE:
  ALL TRANSMISSION RATIOS ARE LINKED EACH OTHER, DUE TO THE PRESENCE OF PARAMETER σ, CHARACTERISTIC OF THE PLANETARY GEAR SET.

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 237 di 260

- RAVIGNEAUX EPICYCLOIDAL GEAR TRAIN RATIOS [B]

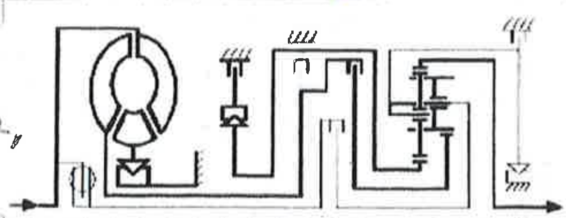
- IN **FIRST SPEED**: THE RIGHT SUN GEAR FIXED TO THE INPUT SHAFT AND THE ANNULUS GEAR FIXED TO THE OUTPUT SHAFT, WHILE THE CARRIER IS THE STATOR ELEMENT:

$\tau_1 = \frac{z_A}{z_{Sd}}$ $\qquad \tau_1 = \frac{z_3}{z_{1dx}}$



3 ONE-WAY CLUTCHES WORK { AS FREE WHEELS IN ONE DIREC. / AS FIXED JOINTS IN THE OTHER D.

- IN **SECOND SPEED** THE ANNULUS WHEEL IS FIXED TO THE OUTPUT SHAFT, WHILE THE INPUT SHAFT IS FIXED TO THE RIGHT SUN GEAR, BEING THE LEFT ONE LOCKED.



- FOR THE LEFT GEAR TRAIN, WE CAN WRITE:

$$\frac{\Omega_{Ss} - \Omega_P}{\Omega_A - \Omega_P} = -\frac{z_A}{z_{Ss}}$$

BECAUSE THE LEFT SUN GEAR IS LOCKED AND THE ANNULUS IS FIXED TO THE OUTPUT SHAFT ($\Omega_{Ss} = 0$, $\Omega_A = \Omega_2$):

$$\frac{-\Omega_P}{\Omega_2 - \Omega_P} = -\frac{z_A}{z_{Ss}}$$

- FOR THE RIGHT EPICYCLOIDAL GEAR TRAIN, WE CAN WRITE:

$$\frac{\Omega_{Sd} - \Omega_P}{\Omega_2 - \Omega_P} = -\frac{z_A}{z_{Sd}}$$

IF WE REMEMBER THAT THE RIGHT SUN GEAR IS FIXED TO THE INPUT SHAFT, WE CAN ALSO WRITE:

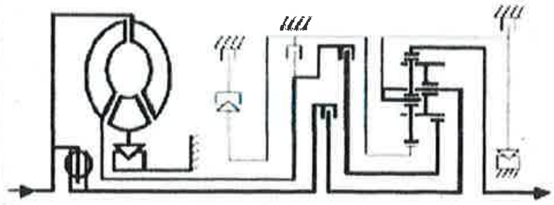
$$\frac{\Omega_1 - \Omega_P}{\Omega_2 - \Omega_P} = \frac{z_A}{z_{Sd}}$$

IF WE COMPARE THE EQUATIONS, WE OBTAIN:

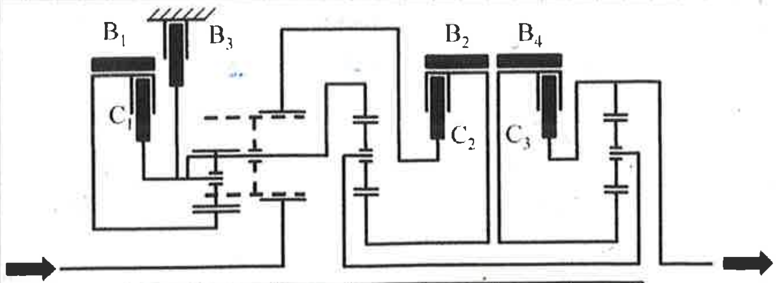
$$\tau_2 = \frac{1 + \frac{z_{Ss}}{z_{Sd}}}{1 + \frac{z_{Ss}}{z_A}} \qquad \tau_2 = \frac{1 + \frac{z_{1sx}}{z_{1dx}}}{1 + \frac{z_{1sx}}{z_3}}$$

- IN **THIRD SPEED** THE CARRIER AND THE RIGHT SUN GEAR ARE FIXED, THEREFORE THE GEARBOX IS IN DIRECT DRIVE, AND:

$\tau_3 = 1$ $\qquad \tau_3 = 1$



- SCHEME OF AN AUTOMATIC POWERSHIFT GEARBOX (5 SPEEDS) $\boxed{C}$



Speed	I	II	III	IV	V	RM
C_1			X	X	X	
C_2	X			X	X	X
C_3	X	X	X	X		X
B_1		X				
B_2	X	X	X			
B_3						X
B_4					X	

1 RAVIGNEAUX COMPOUND E.G.T ON LEFT
+
2 SIMPLE E.G.T CONNECTED TOGETHER
(3 E.G.T IN TOTAL)

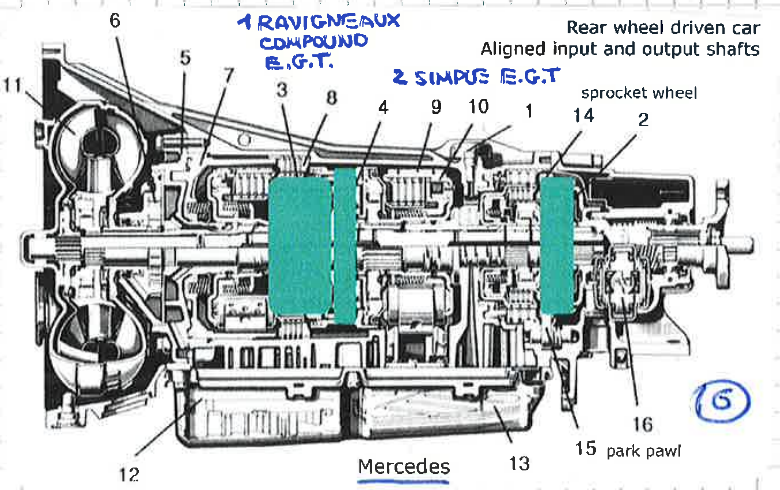
3 MULTIPLE-DISK WET CLUTCHES
4 BRAKES (MULTIPLE-DISK CLUTCH BRAKES)

5 FORWARD SPEEDS [MERCEDES]
1 REVERSE SPEED

- SCHEME WITH 3 EPICYCLOIDAL GEAR TRAINS:

1 RAVIGNEAUX JOINED TO 2 SIMPLE ONES, WITH 3 CLUTCHES AND 4 BRAKES

5 DIFFERENT FORWARD SPEEDS ARE OBTAINED.

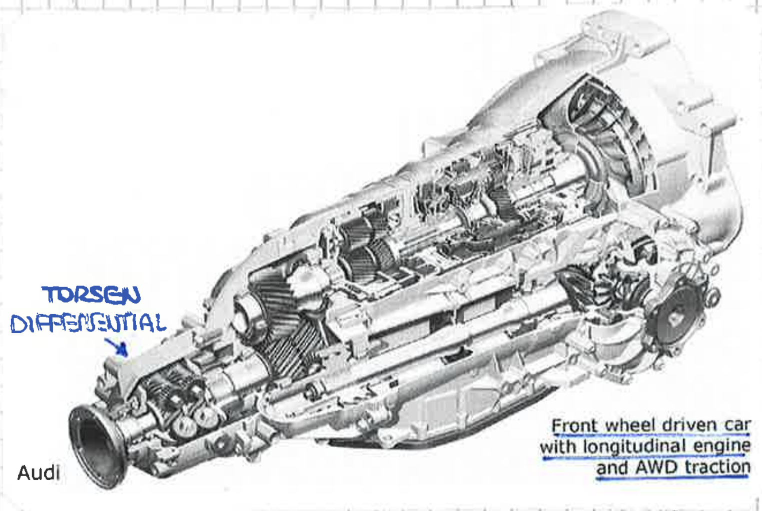


Mercedes

IN THIS CONFIGURATION WE USE
MULTIPLE-DISK CLUTCH BRAKES
(INSTEAD OF THE BAND BRAKES)
BECAUSE THEY ARE EASIER TO CONTROL

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 241 di 260

- AUTOMATIC POWERSHIFT GEARBOX (6 SPEEDS) [C]



1 RAVIGNEAUX E.G.T.
1 SIMPLE E.G.T
(2 E.G.T. IN TOTAL)

3 MULTIPLE-DISK WET CLUTCHES
2 BRAKES (MULTIPLE-DISK CLUTCH BRAKES)

6 FORWARD SPEEDS [AUDI]
1 REVERSE SPEEDS (I THINK)

- THE SCHEME INCLUDES A **SIMPLE** AND **RAVIGNEAUX GEAR TRAINS** THAT IN CONJUCTION WITH **3 CLUTCHES** AND **2 BRAKES** MAKE **6 SPEEDS AVAILABLE**.
- THE RAVIGNEAUX GEAR TRAIN OUTPUT SHAFT MOVES THE DIFFERENTIAL FINAL DRIVE THROUGH AN AUXILIARY SHAFT.
- A **TORSEN DIFFERENTIAL** IS VISIBLE ON THE LEFT.

- AUTOMATIC POWERSHIFT GEARBOX ARCHITECTURES

- THE **POSSIBILITIES** AVAILABLE TO **FRONT WHEEL DRIVEN CARS** WITH **TRANSVERSAL ENGINE** ARE **TWO**:
  - A **FIRST SOLUTION** PROVIDES FOR **TORQUE CONVERTER** AND OIL PUMP **IN LINE** WITH THE **ENGINE**; THE **TORQUE CONVERTER OUTPUT SHAFT** IS FIXED TO A **SPROCKET WHEEL** THAT, THROUGH A **SILENT CHAIN**, MOVES THE **INPUT SHAFT OF** THE **REAL GEARBOX** ON A **PARALLEL CENTRE LINE**. EVERY GEOMETRICAL PROBLEM ON THE TRANSVERSAL DIRECTION OF THE CAR IS ELIMINATED; ON THE CONTRARY THERE ARE CONSEQUENCES ON THE **FRONT OVERHANG** OF THE CAR THAT **BECOMES RELEVANT**. (d)ss
  - A **SECOND SOLUTION** PROVIDES **DIVIDING** THE **EPICYCLOIDAL GEARBOX** IN **TWO SECTION**. THE FIRST SECTION COULD INCLUDE A **RAVIGNEAUX GEAR TRAIN COAXIAL TO THE ENGINE**; ITS **OUTPUT SHAFT** MOVES THROUGH A SINGLE STAGE TRAIN A **SIMPLE EPICYCLOIDAL GEAR TRAIN COAXIAL TO THE DIFFERENTIAL PINION**. IT IS THEREFORE POSSIBLE TO HAVE FIVE SPEEDS WITHIN THE SPACE AVAILABLE ON A FRONT WHEEL DRIVE WITH TRANSVERSAL ENGINE.

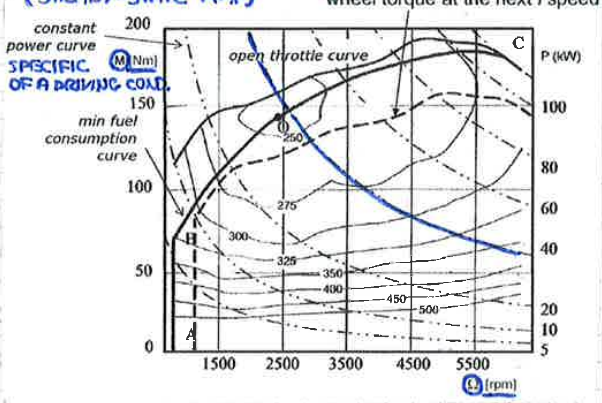
- **CAR CVTs**

- CONTENT

- CONTINUOUSLY VARIABLE TRANSMISSIONS (CVTs) ARE MOSTLY DIFFUSED ON FRONT WHEEL DRIVES.
- THEY ARE PARTICULARLY APPRECIATED FOR MAKING AVAILABLE AN UNLIMITED NUMBER OF TRANSMISSION RATIOS, WITH BENEFITS FOR COMFORT, PERFORMANCE AND FUEL ECONOMY. (∞ τ LIMITED BETWEEN MAX, MIN) (NO SHOCKS, NO JERK)

ENGINE SPECIFIC CONSUMPTION MAP: (IMPROVED TRACT. CAPABILITY) (ABLE TO FOLLOW MIN F.C. LINE)

(STEADY-STATE MAP)



THE MINIMUM FUEL CONSUMPTION LINE ABC AND THE AREA OF PRACTICAL USE FOR A STEPPER GEARBOX ARE SHOWN

IF WE REQUIRE POWER CHARACTERIZED BY ONE OF THESE HYPERBOLAE WE CAN MOVE ALONG IT IN ALL ITS POINT IF WE HAVE THE POSSIBILITY TO CHANGE CONTINUOUSLY THE τ BETWEEN THE ENGINE AND THE WHEEL. WE SELECT τ THAT IS ABLE KEEP THE OPERATING POINT OF THE ENGINE AT THE BEST FUEL CONSUMPTION VALUE.

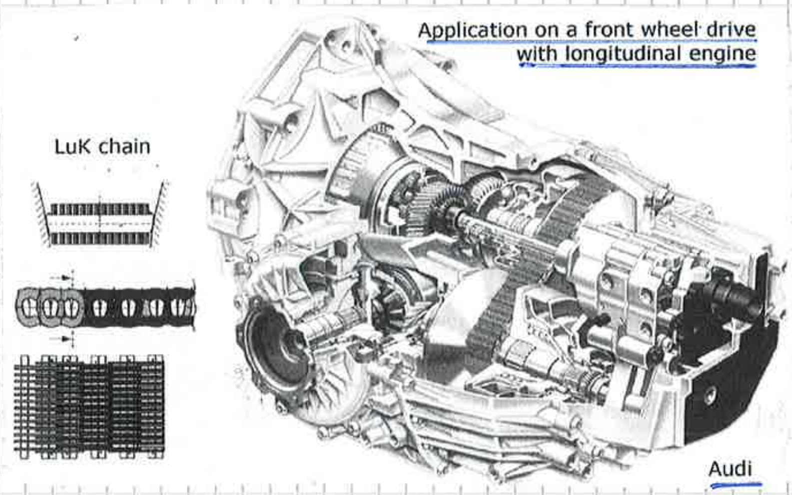
- SYSTEMS

- POSSIBLE SYSTEMS SUITABLE FOR A CVT INCLUDE IN PRINCIPLE: (OTHER EX: E-CVT TOYOTA)
  1. ELECTRIC TRANSMISSIONS; (SERIES HEV: ICE → GENERATOR → EM; ENERGY CONVERSION)
  2. HYDRODYNAMIC TRANSMISSIONS (AMONG THEM THE ALREADY KNOWN TORQUE CONVERTER)
  3. HYDROSTATIC TRANSMISSIONS; (FOR SPECIAL PURPOSE VEHICLES; IN CAR APP. NOT USED AT ALL)
  4. VARIABLE GEOMETRY MECHANICAL TRANSMISSIONS. (V.D. STEEL BELT, LUK CHAIN, R.B. TOROIDAL)
- THE FIRST THREE CATEGORIES HAVE BEEN CONSIDERED MANY TIMES IN THE PAST, BUT HAVE BEEN ABANDONED ON CONVENTIONAL VEHICLES FOR THEIR POOR TRANSMISSION EFFICIENCY, CANCELLING THE THEORETICAL ADVANTAGE AVAILABLE BY A CVT; THEY CAN BE RECONSIDERED FOR HYBRID VEHICLES.
- VARIABLE GEOMETRY MECHANICAL TRANSMISSIONS CAN SHOW SIMILAR EFFICIENCY AS THAT OF GEAR TRANSMISSIONS; AT THIS TIME BELT OR CHAIN DRIVES ARE AVAILABLE.
- ON SPECIAL PURPOSE VEHICLES, SUCH AS TRACTORS, EARTH MOVING MACHINES, AIRPORT BOARDING BUSSES, ETC. SOME HYDROSTATIC CVT IS ALSO USED.

- VAN DOORN STEEL BELT CVT: FRICTION FORCES AND POWER LOSS

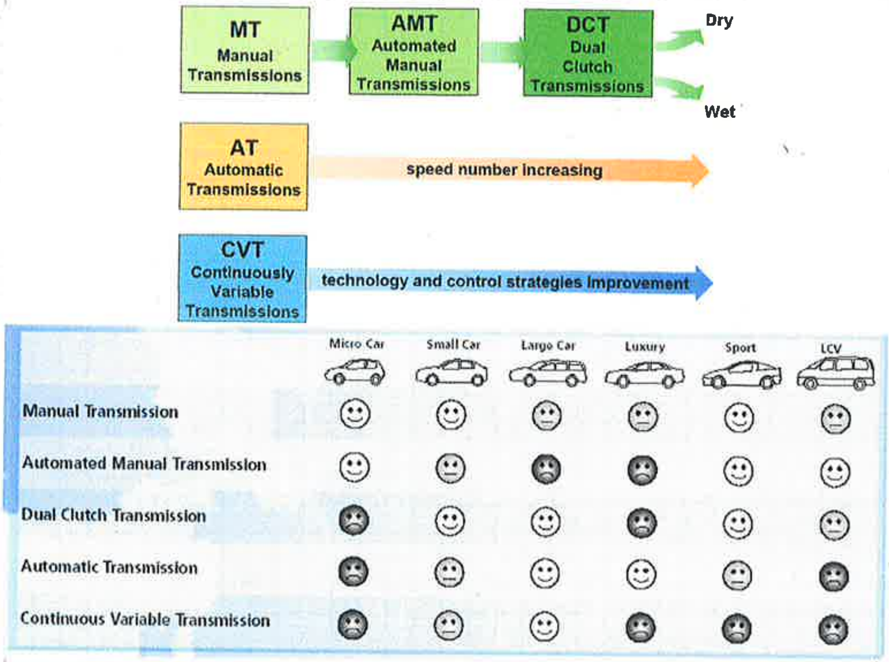
- FRICTION FORCES BETWEEN ELEMENTS AND PULLEYS WOULD NOT WASTE ENERGY IF THERE WERE NO RELATIVE MOTIONS. THEY ARE VERY LIMITED BECAUSE THERE IS NO MACROSCOPIC SLIP BETWEEN PARTS; NEVERTHELESS THERE IS LOCAL MICROSCOPIC RELATIVE MOTION BECAUSE OF TWO FACTS:
  1) THRUST ELEMENT CONTACT AREA MUST HAVE A CERTAIN RADIAL EXTENSION TO LIMIT CONTACT PRESSURE; ON EACH CONTACT AREA THERE WILL BE A SMALL SLIP ON POINTS OUTSIDE PRIMITIVE RADIUS.
  2) THRUST ELEMENTS ENTERING AND LEAVING PULLEYS MUST SLIDE BECAUSE THEY CHANGE THEIR PATH FROM STRAIGHT TO CIRCULAR.
- THE POWER WASTED BY THESE SMALL RELATIVE MOTIONS WILL BE IN ANY CASE DETERMINED BY PRESSURE ACTING BETWEEN BELT SIDES AND PULLEYS; THIS PRESSURE MUST BE LIMITED TO THE MINIMUM NECESSARY TO AVOID MACROSCOPIC SLIP: CONTACT PRESSURE MUST BE THEREFORE CAREFULLY ADJUSTED BY THE CONTROL SYSTEM AS A FUNCTION OF TRANSMITTED TORQUE.

- CHAIN CVT



LUK CHAIN USED INSTEAD OF STEEL BELT.
WORKING PRINCIPLE VERY SIMILAR
TWO PULLEYS;
LUK CHAIN PERMITS LOWER CURVATURE R.

- NON-MANUAL TRANSMISSIONS EVOLUTION AND APPLICATION



- SEMIAUTOMATIC GEARBOXES

- SEMIAUTOMATIC AND AUTOMATED GEARBOXES ARE ON THE MARKET WITH CLUTCH AND TORQUE CONVERTER IN SERIES.
- TORQUE CONVERTER IS SET AS AT FIRST WITH THE PURPOSE TO INCREASE START-UP TORQUE AND TO SMOOTH TRANSMISSION OUTPUT TORQUE; THE TORQUE CONVERTER FEATURES A LOCKUP CLUTCH TO INCREASE THE TRANSMISSION EFFICIENCY IN CRUISE DRIVE. * CONTRARY OF ONE WAY CLUTCH
- TORQUE CONVERTERS INCLUDE AN ADDITIONAL FREE WHEEL BETWEEN PUMP AND TURBINE, ABLE TO TRANSMIT ONLY NEGATIVE TORQUE: (SO WE AMPLIFY E.BRAKING) IN THIS WAY THE DESIRED ENGINE BRAKING EFFECT IS GRANTED.
- AFTER TORQUE CONVERTER, A RETARDER IS FIT, TO HELP THE VEHICLE BRAKES ON LONG DOWNHILL DRIVES.
- RETARDERS CAN BE ASSIMILATED TO HIGH DIAMETER HYDRODYNAMIC CLUTCHES, WITH A LIMITED RADIAL BLADE DIMENSION. ↳ (COMPOSED BY PUMP & T.)
- THE PUMP IS FIXED TO THE TORQUE CONVERTER OUTPUT SHAFT; THE TURBINE BLADES ARE DIRECTLY CUT ON GEARBOX CASING AND ARE STILL. (THIS BRAKING HYDRODYNAMIC CLUTCH CAN BE MODULATED BY CHANGING OIL QUANTITY)

- AUTOMATIC GEARBOX

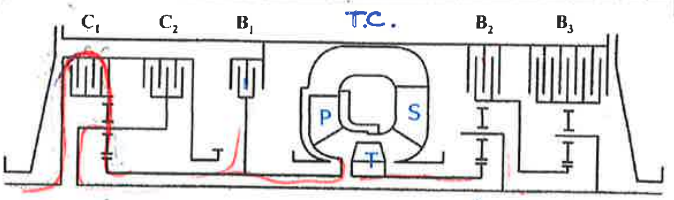
- TOTALLY AUTOMATIC GEARBOXES ARE ALSO AVAILABLE TO INDUSTRIAL VEHICLES AND THEY ARE ALMOST SIMILAR TO THOSE OF CONVENTIONAL DRIVE CARS WITH TORQUE CONVERTER AND EPICYCLOIDAL GEAR TRAINS.
- SPEED NUMBER IS LIMITED TO 5 OR 6; FOR THIS FACT THIS KIND OF GEARBOX IS SUITABLE TO BUSES OR LIMITED SPEED TRUCKS, SUBJECTED TO FREQUENT STOPS. (URBAN BUSES)
- A VERY PARTICULAR AUTOMATIC GEARBOX, SUITABLE FOR URBAN BUSES, WAS DEVELOPED TO OBTAIN DIFFERENT RETARDED SPEEDS USEFUL IN HILLY TOWNS: IN THIS GEARBOX THE TORQUE CONVERTER, INSTALLED IN A CENTRAL POSITION, INSTEAD OF IN THE JOINT FLANGE WITH THE ENGINE, IS USED AS A START-UP DEVICE AS WELL AS A RETARDER.
- EXAMPLES OF THESE TWO DIFFERENT APPLICATIONS ARE SHOWN IN THE FOLLOWING SLIDES.

*
ONE-WAY CLUTCH IS LOCKED DURING ACCELERATION (⇒ TORQUE AMPLIFICATION)
AND IT IS FREE WHEEL WHEN WE ARE IN THE COASTING PHASE

LOCKUP CLUTCH IS LOCKED WHEN WE ARE IN COASTING PHASE
AND IT IS FREE WHEEL WHEN IT IS IN ACCELERATION PHASE.

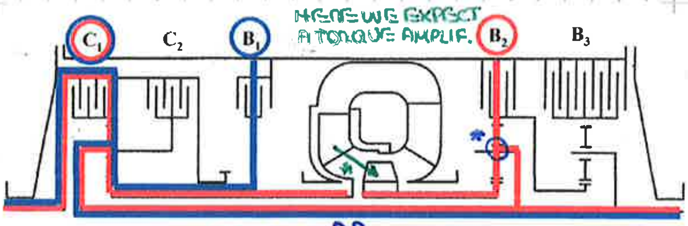
[14/01 : 2:29:00]

– WORKING SCHEME OF THE AT FOR URBAN BUS



Speed	I	II	III	I R	II R	III R	RM
C_1	X	X			X		X
C_2			X			X	
B_1		X	X	X		X	
B_2	X						
B_3				X	X	X	X

• 3 EPICYCLOIDAL GEAR TRAINS TOGETHER WITH 2 CLUTCHES AND 3 BRAKES ALLOW TO OBTAIN 3 NORMAL FORWARD SPEEDS, 3 RETARDED FORWARD SPEEDS (I R, II R, III R) AND 1 REVERSE SPEED.



Speed	I	II	III (D.D.)	I R	II R	III R	RM
C_1	X	X			X		X
C_2			X			X	
B_1		X	X	X		X	
B_2	X						
B_3				X	X	X	X

* SUM OF POWER PASSING THROUGH TC WITH THE POWER THAT GOES DIRECTLY TO THE OUTPUTS.

CONSIDERING SPEED II : B1 BLOCKS THE PUMP OF THE T.C. (WE DON'T USE IT); THEREFORE, THE POWER FLOW, THANKS TO C1, GOES TO THE OUTPUT, THROUGH THE FIRST (LEFT) E.G.T.

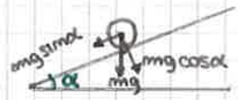
FOR THE RETARDER FORWARD SPEEDS WE ACTUATE BRAKE B3 => COUNTER ROTATION OF THE TURBINE SPEED, THAT IN THIS CASE ACTS AS A PUMP; SINCE THIS PUMP-TURBINE IS COUNTER-ROTATING WITH RESPECT TO THE PUMP, IT APPLIES A NEGATIVE TORQUE => BRAKING EFF.

T2 GEAR RATIOS DEFINITION

RESISTANCE TO MOTION

ROLLING RESISTANCE (R_R):

$$R_R = f \sum F_{zi} = (f_0 + KV^2)\sum F_{zi} = (f_0 + KV^2)[mg\cos(\alpha) - \frac{1}{2}\rho V^2 S C_z]$$



AERODYNAMIC RESISTANCE (R_A):

$$R_A = \frac{1}{2}\rho V_R^2 S C_x$$

GRADE RESISTANCE (R_W):

$$R_W = mg\sin(\alpha)$$

TOTAL RESISTANCE TO MOTION (R):

$$R = R_R + R_A + R_W = (f_0 + KV^2)[mg\cos(\alpha) - \frac{1}{2}\rho V^2 S C_z] + \frac{1}{2}\rho V_R^2 S C_x + mg\sin(\alpha)$$

ROAD LOAD (R):

$$R = A + BV^2 + CV^4$$

$A = mg[f_0\cos(\alpha) + \sin(\alpha)] \rightarrow A = mg[f_0 + i]$

$B = Kmg\cos(\alpha) + \frac{1}{2}\rho S(C_x - C_z f_0) \rightarrow B = Kmg + \frac{1}{2}\rho S(C_x - C_z f_0)$

$C = -\frac{1}{2}K\rho S C_z$

hp_1: $V_R = V$: STILL AIR | hp_3: $\cos(\alpha) \approx 1$; $\sin(\alpha) \approx \tan(\alpha) \approx i$: GRADE OF THE ROAD

hp_2: CV^4 IS USUALLY NEGLECTED EXCEPT IN RACING CARS.

REQUIRED/NEEDED (OR AVAILABLE) POWER TO THE WHEELS (V = CONST, STRAIGHT R.):

$$P = R \cdot V = AV + BV^3 + CV^5$$

GEAR RATIO

$\tau = \frac{\omega_{OUT}}{\omega_{IN}} = \frac{z_{IN}}{z_{OUT}}$: TRANSMISSION RATIO

$i = \frac{1}{\tau} = \frac{\omega_{IN}}{\omega_{OUT}}$: REDUCTION RATIO

TOP GEAR RATIO

$\eta_t = \frac{P_A}{P_E}$: TRANSMISSION EFFICIENCY

$\tau_t = \frac{V}{R_R \Omega_E}$: OVERALL GEAR RATIO ($\log(V) = \log(P) + \log(\tau_t)$)

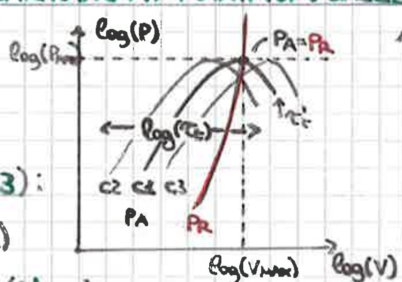
$\tau_t' = \frac{V_{MAX}}{R_R(\Omega_E)_{P_{MAX}}}$: OVERALL GEAR RATIO AT MAXIMUM SPEED

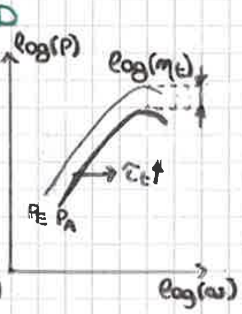
"FAST GEAR RATIO" (C1):

$\tau_{TOP} \approx \tau_t'$

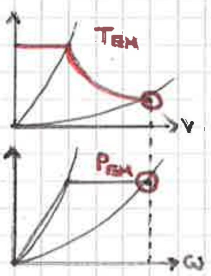
"ECONOMY GEAR RATIO" (C3):

$\tau_{TOP} > \tau_t'$ (OVERGEARED S.)





ELECTRIC MOTOR:



MAXIMUM VEHICLE SPEED (V_{MAX}):

$\eta_t P_{E\,MAX} = P_{A\,MAX} = AV_{MAX} + BV_{MAX}^3 + CV_{MAX}^5$ $\quad P_{E\,MAX} \leftrightarrow \tau_t' = \tau(\Omega_E(P_{MAX}))$ $\begin{cases} = \tau_t^m & \text{IF (C1)} \\ = \tau_t^{m-1} & \text{IF (C3)} \end{cases}$

OVERALL TRANSMISSION RATIO (τ_t):

$\tau_t = \tau_g \tau_f$ | $\tau_g^m = 1$: COUNTERSHAFT T. | $\tau_f \ll 1$: GEAR RATIO AT FINAL DRIVE

$\tau_g^m > 1$: SINGLE STAGE T. ($\tau_g^m \approx 1.5$ FOR EX.)

© Proprietà riservata dell'autore - Digitalizzazione e distribuzione a cura del CENTRO APPUNTI - Corso Luigi Einaudi, 55/B - Torino / Pagina 255 di 260

INTERMEDIATE GEAR RATIOS

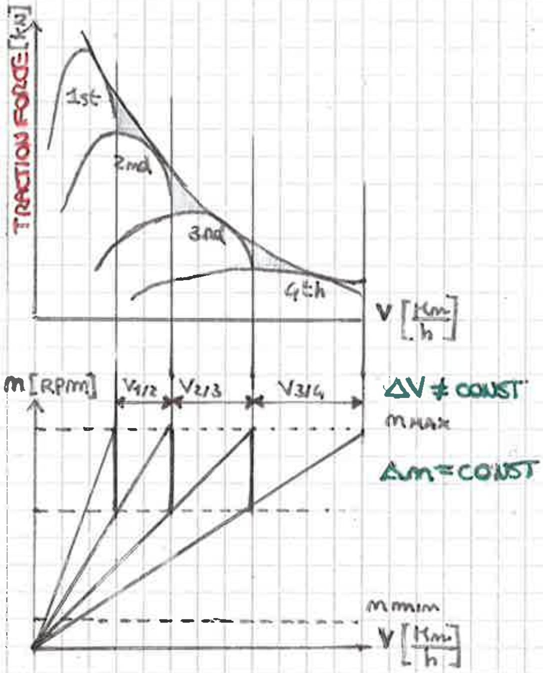
FIRST CRITERION

$\varphi = \left(\frac{\tau_{gm}}{\tau_{g1}}\right)^{\frac{1}{m-1}}$: RATIO OF GEOM. PROGR = RATIO BETWEEN TWO SUBSEQUENT G. RATIO

$\tau_1/\tau_2 = \tau_2/\tau_3 = \tau_3/\tau_4 = \varphi_{1/2} = \varphi_{2/3} = \varphi_{3/4} = \varphi$

1 - THIS CONDITION MAKES BIGGEST THE AREA BETWEEN THE CURVES OF P_A AND P_N, WITH THE ASSUMPTION $\eta_t = CONST$ FOR ALL THE GEAR RATIOS.

2 - IN THIS WAY ALL THE PA-CURVES ARE EQUISPACED.



3 - IF WE DEFINE A THRESHOLD (m_{MAX}) FOR THE GEARSHIFT, THE ENGINE ACCELERATES UNTIL IT IS REACHED; THEN THE GEARSHIFT IS TRIGGERED; THE ENGINE SLOWS DOWN OF THE SAME AMOUNT (Δm) FOR ALL GEARS.

$\Delta m = CONST$

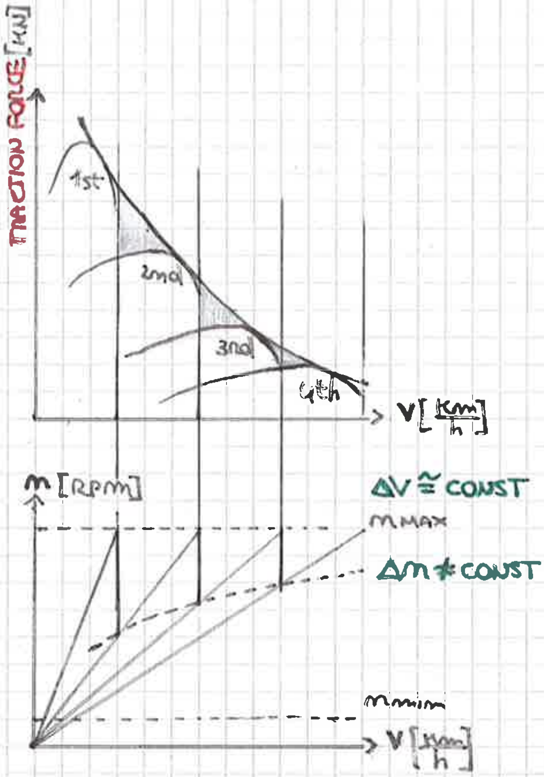
$\Delta V \neq CONST$

SECOND CRITERION

IN SOME CASES WE HAVE ADV. IN HAVING THE CURVES A BIT CLOSER TO EACH OTHER IN THE HIGH SPEED RANGE. $\tau_1/\tau_2 = \varphi_{1/2} \neq \tau_2/\tau_3 = \varphi_{2/3} \neq$

$\varphi_{1/2} > \varphi_{2/3} > \varphi_{3/4}$ PROGRESSIVE REDUCTION OF φ

1 - THIS CAN GIVE A FEELING OF SPORT DRIVING, SINCE THE GEAR RATIOS ARE MORE CROWDED IN THE ZONE OF MOST COMMON USE.



2 - IN THIS CASE WE HAVE A PROGRESSIVE REDUCTION OF Δm FROM THE 1st GEARSHIFT TO THE LAST ONE BUT SAME ΔV.

$\Delta m \neq CONST$

$\Delta V = CONST$

- THE CHOICE DEPENDS ON TYPE OF VEHICLE MISSION AND ON THE BALANCE BETWEEN ACCELERATION PERFORMANCE AND FUEL CONSUMPTION.

FUEL CONSUMPTION (AT CONSTANT SPEED)

REQUIRED ENERGY (AT WHEELS):

$$e = P_N \cdot t = \frac{P_N \cdot d}{V}$$

FUEL CONSUMPTION (PER UNIT DISTANCE) [l/100km] (Q)

THEORETICALLY:

$$Q = \frac{R}{\eta \rho_f H_i} = \frac{A + BV^2 + CV^4}{\eta_E \eta_t \rho_f H_i}$$: STRONG hp: DENOMINATOR = CONST

PRACTICAL PROCEDURE:

$P_E = \frac{P_N}{\eta_t}$; $q(T_E, \omega_E) = \frac{1}{\eta_E H_i}$: SPECIFIC FUEL CONSUMPTION

$$Q = \frac{q P_E}{\rho_f V} = \frac{q P_N}{\eta_t \rho_f V} = \frac{P_N}{\eta_t \eta_E \rho_f H_i V} = \frac{P_E}{\eta_E \rho_f H_i V} \quad ; \quad P_E \eta_t - P_N = m_E V \frac{dV}{dt} \Rightarrow P_E = \frac{1}{\eta_t}\left(P_N + m_E V \frac{dV}{dt}\right)$$

! THE BEST POINT WHERE TO PERFORM THE GEARSHIFT, IN ORDER TO MINIMIZE THE FUEL CONSUMPTION, IS AT THE BEGINNING OF THE NEXT GEAR.

FUEL CONSUMPTION DURING A CYCLE $(Q = \sum_i \Delta e_i \; ; \; \Delta e_i = Q_i (t_{i+1} - t_i))$

$$\begin{cases} \eta_t = \frac{P_A}{P_E} \Rightarrow P_E = \frac{1}{\eta_t}(P_A) \\ P_A - P_N = m_E V \frac{dV}{dt} \Rightarrow P_A = P_N + \underbrace{m_E V \frac{dV}{dt}}_{\text{INERTIAL POWER}} \end{cases}$$

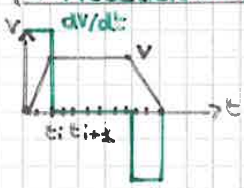
$$P_E = \frac{1}{\eta_t}\left(P_N + m_E V \frac{dV}{dt}\right) = \frac{1}{\eta_t}\left(AV + BV^3 + CV^5 + m_E V \frac{dV}{dt}\right)$$: POWER REQUIRED TO E. TO MOVE THE V.

$$Q = \frac{q P_E}{\rho_f V} = \frac{P_E}{\eta_E \rho_f H_i V} = \left[\frac{1}{\eta_t}\left(AV + BV^3 + CV^5 + m_E V \frac{dV}{dt}\right)\right] \frac{1}{\eta_E \rho_f H_i V}$$

$$\Delta e_i = Q_i (t_{i+1} - t_i) = \left[\frac{1}{\eta_t}\left(A + BV_{m_i}^2 + CV_{m_i}^4 - m_E \frac{V_{i+1} - V_i}{t_{i+1} - t_i}\right)\right] \frac{1}{\eta_E \rho_f H_i}(t_{i+1} - t_i)$$ (+ ACCELERATION)

WITH $V_{m_i} = \frac{V_{i+1} + V_i}{2}$

$\Delta e_i = 0$ (− ACCELERATION)



$Q = \sum_i \Delta e_i$: TOTAL FUEL CONSUMPTION

- FOR A SPECIFIC POINT OF THE MISSION, TO REDUCE q, WE CAN CHANGE THE GEARSHIFT SCHEDULE (WORKING ON A ISO-POWER LINE): A → B
- IF WE HAD INFINITE AVAILABILITY OF GEAR RATIOS WE COULD MOVE SMOOTHLY (IN A GIVEN RANGE) IN ALL THE POINTS ON THE ISO-POWER HYPERBOLE.